GLASGOW CALEDONIAN UNIVERSITY MSc/MEng
Department of Engineering

CONDITION MONITORING

Lecture Slides for Mechanical Condition Monitoring

Mechanical Condition Monitoring

Module Content

This module aims to provide an understanding of both **Mechanical** and **Electrical Condition Monitoring** and **associated instrumentation requirements** for successful condition monitoring.

The main focus in **Mechanical Condition Monitoring** is vibration monitoring since this is the most popular method of determining the condition and diagnosing faults in rotational machines, although other techniques used in condition monitoring are also covered.

The module also includes a review of relevant sensors, data acquisition/ analysis and the essential **instrumentation** required in condition monitoring.

Electrical Condition Monitoring will develop an understanding of the need for, and challenges in, measuring electrical signals in machinery. The application of standard and non-standard electrical condition monitoring systems to a range of electrical plant will be explained. The students will learn to use condition monitoring tools and then to evaluate the data provided by them.

Mechanical Condition Monitoring

- Develop an understanding of the principles of condition monitoring and its application areas.

- Gain a theoretical insight into vibration theory and a detailed understanding of vibration analysis techniques to be able to critically analyse collected data from various vibration monitoring equipment.

- Develop an understanding of other condition monitoring methods such as thermography and oil/debris analysis.

Electrical Condition Monitoring

- Develop an understanding of the various stresses which exist in electrical plant and how these lead to degradation of the system performance;
- Develop an understanding of the range of techniques that can be applied to determine the presence of electrical faults;
- Learn the application of standard diagnostic techniques to data from electrical condition monitoring systems;
- Develop an appreciation of the shortcomings of the analysis of the data presented by the techniques for electrical condition monitoring;

Sensors, Data acquisition and Analysis for Condition Monitoring

- Understand the operation of a range of sensing techniques used for the measurement of the motion of rotating and reciprocating machines.
- Understand the sensing techniques used for the measurement of mechanical vibration.
- Understand a range of techniques used for the measurement of temperature, both contact sensors and radiation sensors.
- Be able to specify the basic requirements of a data acquisition system intended to perform measurements relevant to a condition monitoring application.

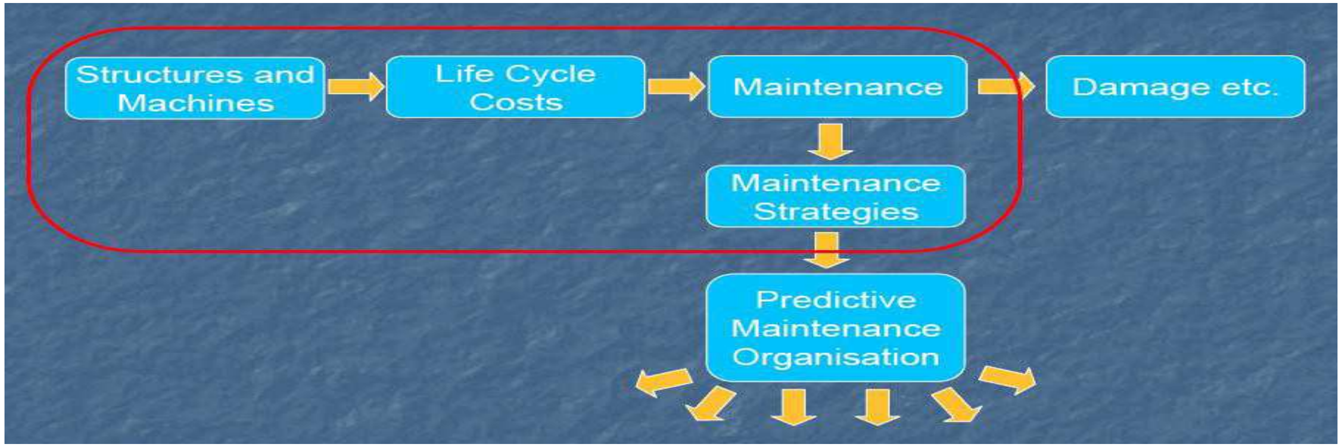
Mechanical Condition Monitoring

- Maintenance strategies, concept of condition monitoring and main methods involved (vibration monitoring, visual / performance monitoring, Oil & debris analysis etc.)
- Basic vibration theory, vibration measurement and analysis, machine vibration; Rotational machine faults and vibration characteristics.
- Applications of vibration monitoring to rotating machines. Vibration monitoring in practice
- Overall vibration monitoring and experience based spectrum analysis to detect machine condition and faults in bearings and gears. Current diagnostic techniques/tools commercially available.
- Thermal Monitoring :Introduction to thermal monitoring; thermal monitoring techniques, application of thermal monitoring to manufacturing processes. Thermal imaging camera, and its application as a condition monitoring tool.
- Lubricant analysis/monitoring : Introduction to tribology - lubricant types and their properties. Introduction to wear debris monitoring; collecting and quantifying wear debris; wear debris and oil analysis in practice.

Maintenance

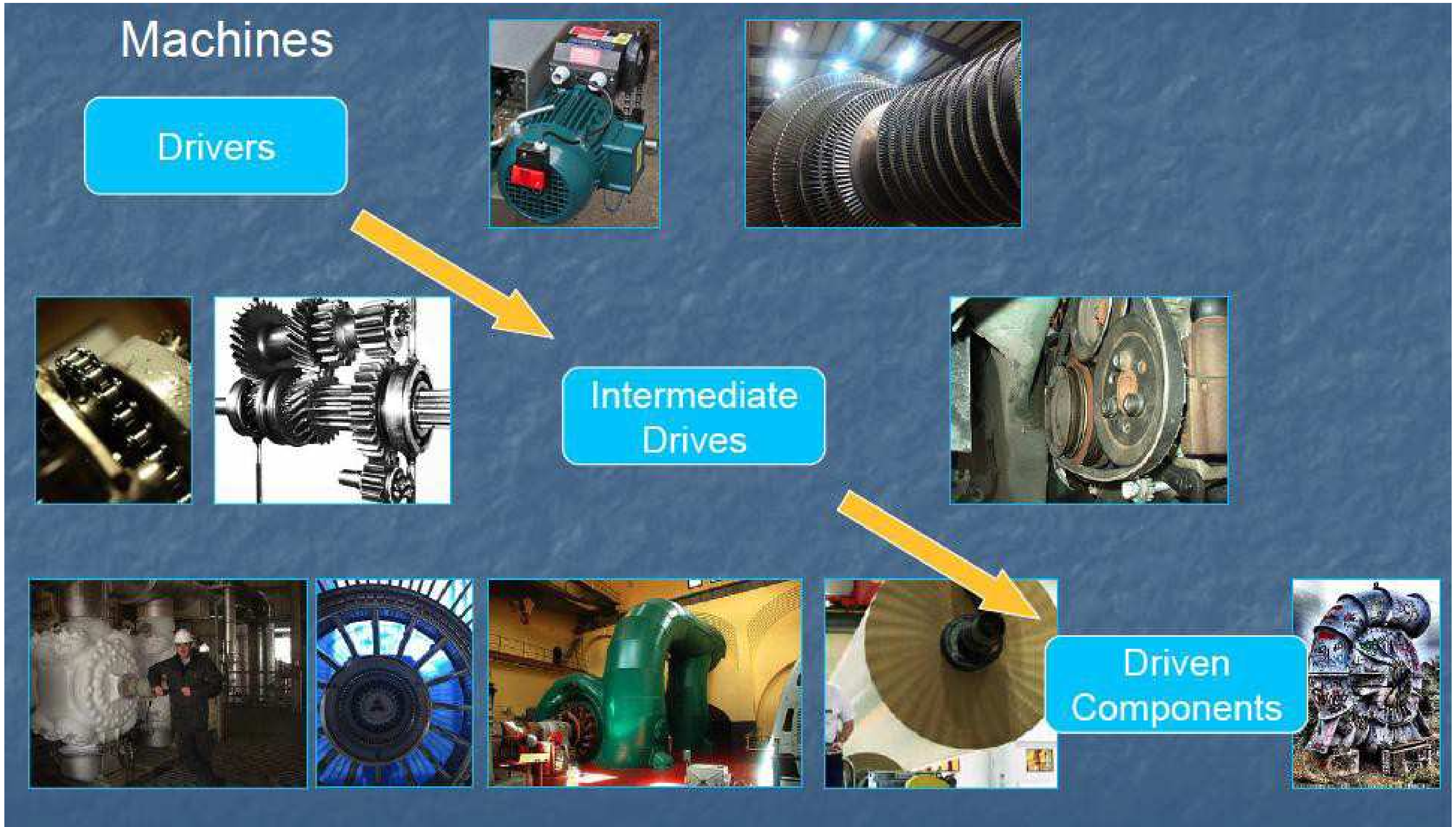
- What is the importance of maintenance on the life-cycle costs of machines and structures?
- What are the factors to be considered when organising a maintenance strategy?
- What are the cutting-edge techniques for the early identification of damage in a variety of situations?

Importance of Maintenance



- Depending on industry, maintenance costs can represent between 15 and 60%
  of production.
- Estimated that one-third of all maintenance costs is wasted due to unnecessary or improperly carried out maintenance. (~$60bn out of $200bn).
- Difficult to compete with countries like Japan who have more advanced maintenance strategies.

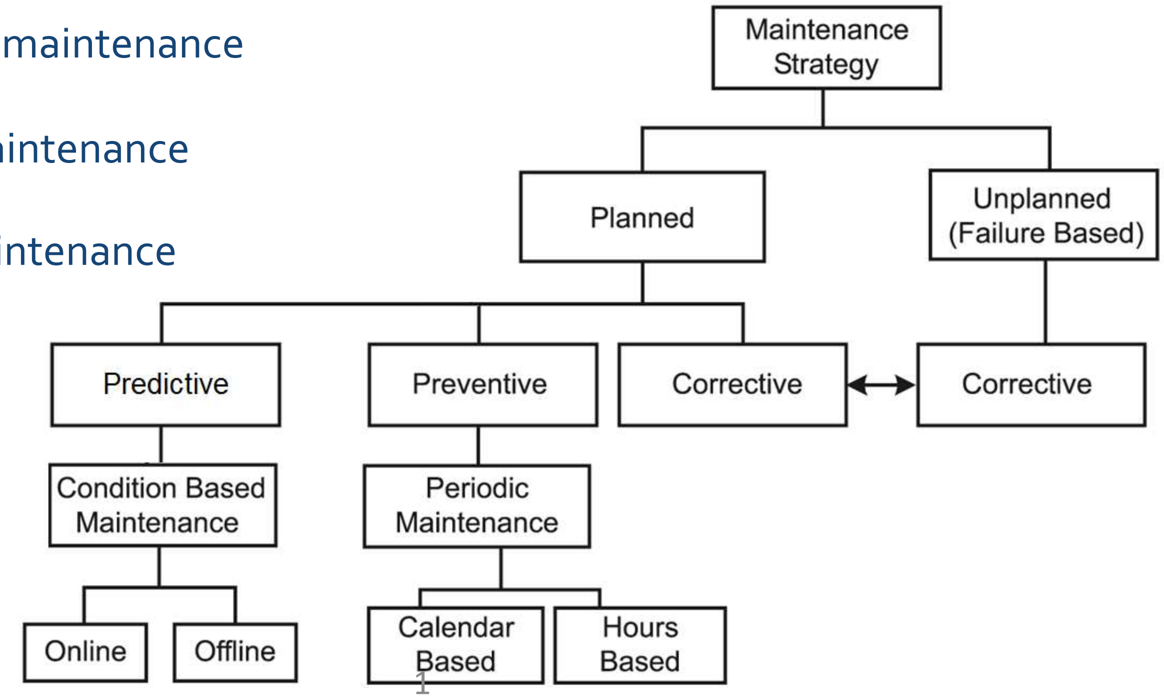
Machines Considered



Maintenance Strategies

There are essentially three main approaches to maintenance of structures and machines.

- Run-to failure maintenance
- Preventive maintenance
- Predictive maintenance



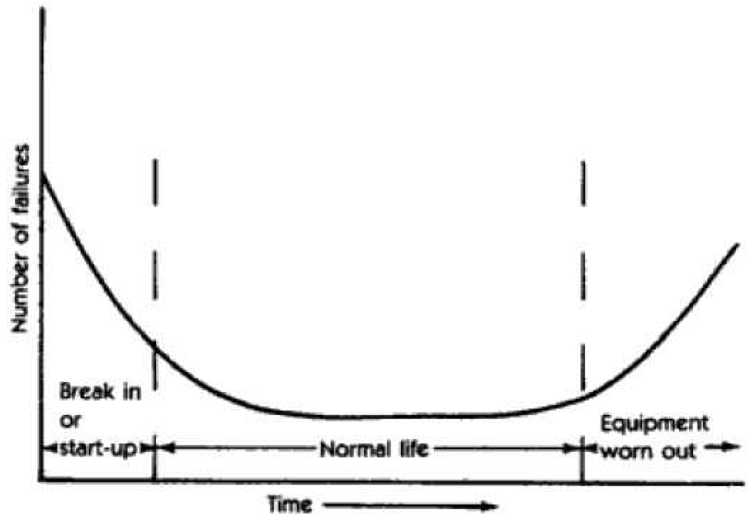
Run-to-failure Maintenance

- If it ain't broke, don't fix it – sounds reasonable
- No money spent on maintenance until machine or structure stops working
- Also known as reactive maintenance
- Most expensive maintenance method.
  - High spare parts inventory
  - High overtime costs
  - Long machine downtime
  - Low production availability
  - Spare machines required
  - Knock-on effects on other machines and overall loss of production

Preventive Maintenance

- Many definitions –all maintenance is time-driven
- Based on elapsed time or hours of operation
- Time between maintenance decided on statistical data
- Generally based on bathtub curve

  –not reliable in many cases
- Treats all similar machines as same.
- Scheduled maintenance costs are around one-third of run-to-failure costs



Predictive Maintenance

- Involves the regular monitoring of actual mechanical condition of machine or structure and other indicators of operating condition provide data for maximum interval between repairs.

- Involves Non destructive Techniques (NDT) which are only part of the predictive maintenance strategy.

- The actual operating condition of the machine is used to optimise total plant or structure operation.

Predictive Maintenance

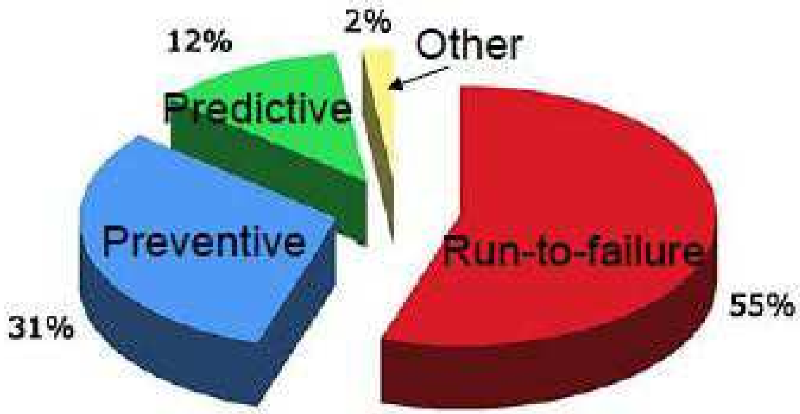
- **Costs**
  - monitoring equipment
  - staff training
  - labour costs for measurement and analysis

- **Savings**
  - elimination of unexpected breakdowns and secondary breakdowns
  - increased time between services
  - reduction of spare part stock
  - reduction in insurance premium

- **Benefits**
  - increased reliability
  - increased quality
  - increased profitability
  - increased productivity

*** NDT tools will vary depending upon machine and types of likely damage.**

Maintenance strategy in average facility



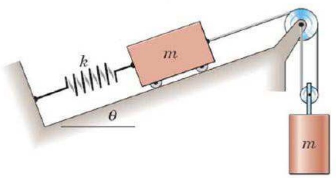
Condition Based Maintenance (CBM)

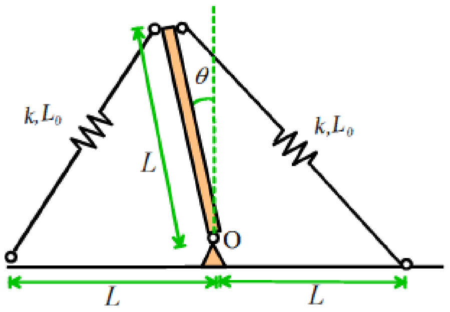
The objective of CBM is not just **the prediction of time to failure**, it is also to **maximize the operating time** for all components and optimize maintenance practices, as well as **Operational Readiness**

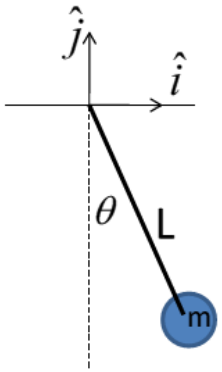
Vibration Basics

Natural frequency for 1 DOF undamped system

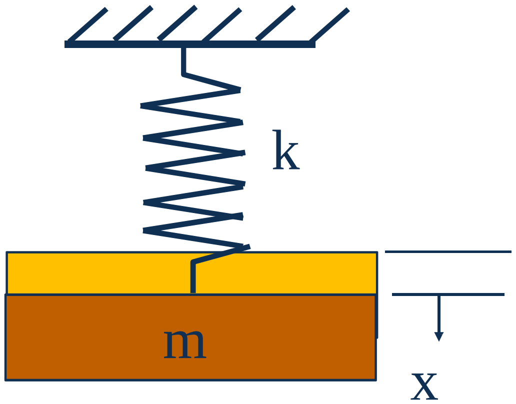
Information on the ′natural frequency′ , ′Vibration mode′ , and dissipation of a system





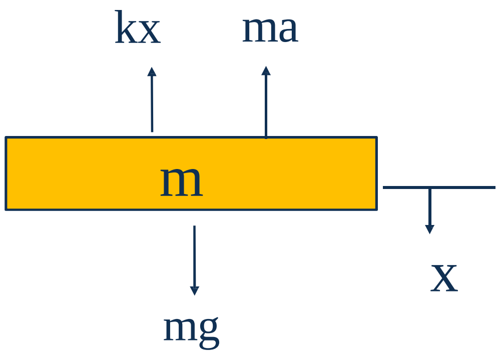


Natural frequency for 1 DOF undamped system



Static Equilibrium

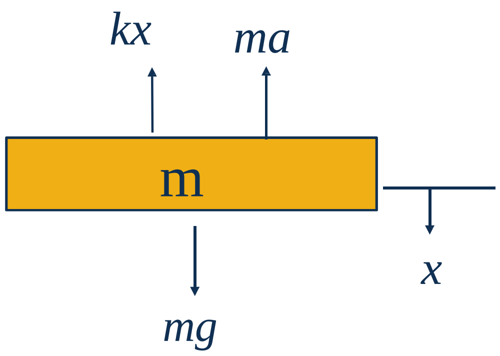
$F = kx$ $\qquad$ F = mg



k = spring rate = force/deflection

x = displacement from static position

Natural frequency for 1 DOF undamped system



EOM for small vibration of any 1DOF undamped system has form

$$m\frac{d^2x}{dt^2} + kx - mg = 0$$

$$m\ddot{x} + kx = 0$$

Natural frequency for 1 DOF undamped system

$$m\ddot{x} + kx = 0$$

$$m\left(\ddot{x} + \frac{k}{m}x\right) = 0 \qquad \omega_n^2 = \frac{k}{m} \qquad \omega_n = \sqrt{\frac{k}{m}}$$

$$\ddot{x} + \omega_n^2 x = 0$$

ω_n is the natural frequency

Solution is of the form of

$$x = e^{st} \qquad \dot{x} = se^{st} \qquad \ddot{x} = s^2 e^{st}$$

Therefore, by substitution

$$s^2 e^{st} + \omega_n^2 e^{st} = 0$$

Natural frequency for 1 DOF undamped system

By substitution

$$s^2 e^{st} + \omega_n^2 \, e^{st} = 0$$

$$s^2 + \omega_n^2 = 0$$

$$s = \sqrt{-\omega_n^2} = +/- i\omega_n$$

$$x = C_1 e^{i\omega_n t} + C_2 e^{-i\omega_n t}$$

$$x = C_1 \left(\cos\omega_n t + i\sin\omega_n t\right) + C_2 \left(\cos\omega_n t + i\sin\omega_n t\right)$$

Grouping terms

$$x = \left(C_1 + C_2\right)\cos\omega_n t + \left(C_1 - C_2\right) i\sin\omega_n t$$

Natural frequency for 1 DOF undamped system

$$x = (C_1 + C_2)\cos\omega_n t + (C_1 - C_2) i \sin\omega_n t$$

$$\text{Let} \quad (C_1 + C_2) = A \qquad (C_1 - C_2) i = B$$

$$x = A\cos\omega_n t + B\sin\omega_n t$$

A & B are now found from initial conditions

$$@\ t = 0 \qquad x(0) = A = X_0 \qquad \text{(initial displacement)}$$

$$x(0) = A(1) + B(0)$$

$$\dot{x} = -X_0\omega \sin\omega_n t + B\omega_n \cos\omega_n t$$

$$\dot{x}(0) = -X_0\omega_n (0) + B\omega_n (1) = V_0$$

Natural frequency for 1 DOF undamped system

$$\dot{x}(0) = -X_0\omega_n(0) + B\omega_n(1) = V_0$$

$$B = \frac{V_0}{\omega}$$

Therefore, for a free vibrating single DOF system

$$x(t) = X_0 \cos\omega_n t + \left(\frac{V_0}{\omega_n}\right)\sin\omega_n t$$

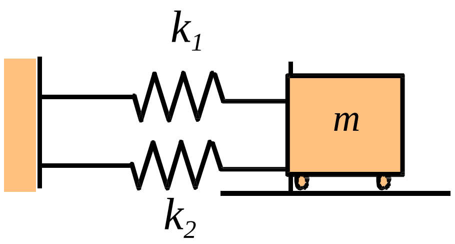
$$\omega_n = \sqrt{\frac{k}{m}}$$

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}}$$

Natural frequency for 1 DOF undamped system

Parallel: stiffness

$k = k_1 + k_2$



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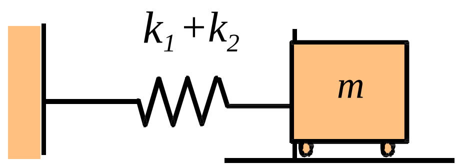
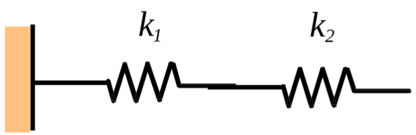
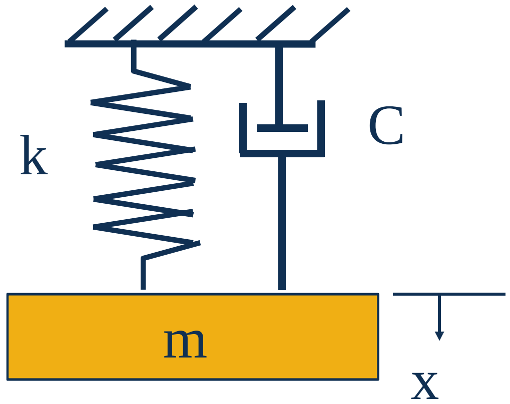


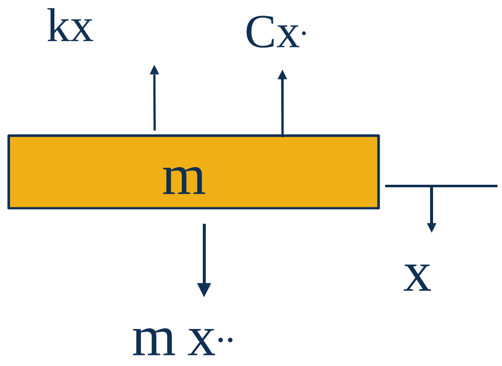
Series: stiffness



$$\frac{1}{k} = \frac{1}{k_1} + \frac{1}{k_2}$$

Natural frequency for 1 DOF Damped system





$$\sum F_x = 0$$

$$m\frac{d^2x}{dt^2} + C\frac{dx}{dt} + kx = 0$$

Natural frequency for 1 DOF Damped system

$$\sum F_x = 0$$

$$m\frac{d^2x}{dt^2} + C\frac{dx}{dt} + kx = 0$$

$$\frac{d^2x}{dt^2} + \left(\frac{C}{m}\right)\frac{dx}{dt} + \left(\frac{k}{m}\right)x = 0$$

Utilizing the quadratic equation

$$\lambda = \frac{-\left(\frac{C}{m}\right) +/- \sqrt{\left(\frac{C}{m}\right)^2 - 4\frac{k}{m}}}{2} \quad \text{or} \quad -\left(\frac{C}{2m}\right) +/- \sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}$$

Natural frequency for 1 DOF Damped system

$$s_1, s_2 = -\left(\frac{C}{2m}\right) +/- \sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}$$

$\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}$ can be $= 0$, $+$ and real, - and imaginary

The solution is of the form

$$x(t) = C_1 e^{s_1 t} + C_2 e^{s_2 t}$$

$$s_1, s_2 = -\left(\frac{C}{2m}\right) +/- \sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}$$

Natural frequency for 1 DOF Damped system

Expanding

$$x(t) = C_1 e^{\left[-\left(\frac{C}{2m}\right)+\sqrt{\left(\frac{C}{2m}\right)^2-\frac{k}{m}}\right]t} + C_2 e^{\left[-\left(\frac{C}{2m}\right)-\sqrt{\left(\frac{C}{2m}\right)^2-\frac{k}{m}}\right]t}$$

or

$$x(t) = e^{-\left(\frac{C}{2m}\right)t}\left\{C_1 e^{\left[+\sqrt{\left(\frac{C}{2m}\right)^2-\frac{k}{m}}\right]t} + C_2 e^{\left[-\sqrt{\left(\frac{C}{2m}\right)^2-\frac{k}{m}}\right]t}\right\}$$

Note that three conditions can occur

$\sqrt{\left(\frac{C}{2m}\right)^2-\frac{k}{m}}$ can be $= 0$, $+$ and real, - and imaginary

Natural frequency for 1 DOF Damped system

Condition 1, CRITICAL DAMPING

$$\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}} = 0$$

$$\left(\frac{C}{m}\right)^2 = 4\frac{k}{m} \quad \text{or} \quad C = 2\sqrt{km} = C_c \quad \text{(critical damping)}$$

$$x(t) = e^{-\left(\frac{C}{2m}\right)t}\left\{C_1(1) + C_2(t)\right\}$$

Recall $\quad \omega_n = \sqrt{\frac{k}{m}} \qquad \omega_n^{\ 2} = \frac{k}{m}$

By substitution $\quad C_c = 2\sqrt{km} = 2m\omega_n$

Natural frequency for 1 DOF Damped system

Condition 2, UNDER DAMPING

$$\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}} < 0 \quad \{\text{imaginary}\}$$

$$x(t) = e^{-\left(\frac{C}{2m}\right)t}\left\{C_1\, Cos\left(\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}\right)t + C_2\, Sin\left(\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}\right)t\right\}$$

Note: The system now has damped oscillatory behavior

Natural frequency for 1 DOF Damped system

Condition 3, OVER DAMPING

$$\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}} > 0 \quad \{\text{All Real}\}$$

$$x(t) = e^{-\left(\frac{C}{2m}\right)t}\left\{C_1 e^{\left[+\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}\right]t} + C_2 e^{\left[-\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}}\right]t}\right\}$$

Note: The system now has no oscillatory behavior

Natural frequency for 1 DOF Damped system

By definition

$$C = 2\sqrt{km} = C_c \quad \text{(critical damping)}$$

$$\frac{C}{C_c} = \xi = \text{(damping ratio)}$$

Also

$$\xi = \frac{C}{C_c} = \frac{C}{2\sqrt{km}} \qquad C = \xi\, 2\sqrt{km}$$

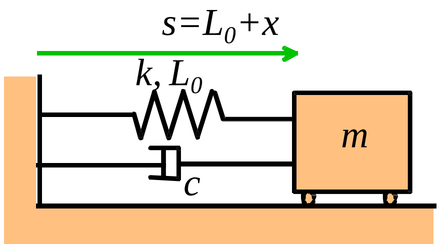
$$\sqrt{\left(\frac{C}{2m}\right)^2 - \frac{k}{m}} = \sqrt{\xi^2\omega_n^2 - \omega_n^2} = \omega_n\sqrt{\xi^2 - 1}$$

$$\frac{C}{2m} = \xi\,\frac{2\sqrt{km}}{2m} = \xi\sqrt{\frac{k}{m}} = \xi\,\omega_n$$

Therefore

$$\omega_d = \omega_n\sqrt{\xi^2 - 1}$$

Natural frequency for 1 DOF Damped system



$$\mathbf{F} = m\mathbf{a} \Rightarrow \frac{d^2x}{dt^2} + \frac{c}{m}\frac{dx}{dt} + \frac{kx}{m} = 0$$

$$\frac{d^2x}{dt^2} + 2\zeta\omega_n\frac{dx}{dt} + \omega_n^2 x = 0 \qquad \omega_n = \sqrt{\frac{k}{m}} \qquad \zeta = \frac{c}{2\sqrt{km}}$$

Natural frequency for 1 DOF Damped system

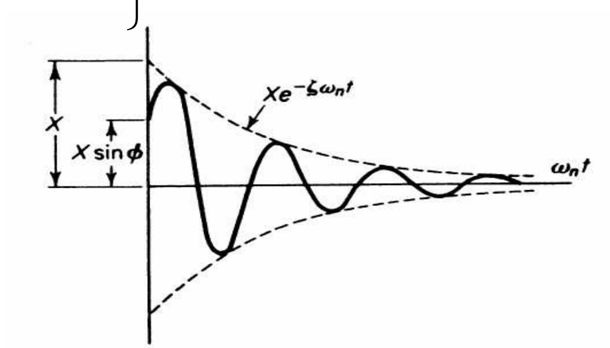
$$\frac{d^2x}{dt^2} + 2\zeta\omega_n \frac{dx}{dt} + \omega_n^2 x = 0 \qquad \omega_n = \sqrt{\frac{k}{m}} \qquad \zeta = \frac{c}{2\sqrt{km}}$$

Initial conditions: $x = x_0 \quad \dfrac{dx}{dt} = v_0 \quad t = 0$

Underdamped:

$$\varsigma < 1$$

$$x(t) = \exp(-\varsigma\omega_n t)\left\{ x_0 \cos\omega_d t + \frac{v_0 + \varsigma\omega_n x_0}{\omega_d} \sin\omega_d t \right\}$$



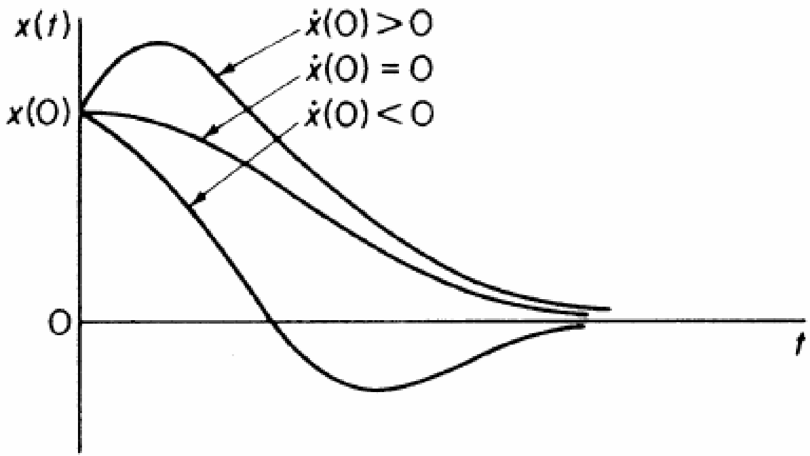
Natural frequency for 1 DOF Damped system

$$\frac{d^2x}{dt^2} + 2\zeta\omega_n \frac{dx}{dt} + \omega_n^2 x = 0 \qquad \omega_n = \sqrt{\frac{k}{m}} \qquad \zeta = \frac{c}{2\sqrt{km}}$$

Initial conditions: $x = x_0 \quad \frac{dx}{dt} = v_0 \quad t = 0$

Critically damped:

$\varsigma = 1$



Critically damped gives fastest return to equilibrium

Natural frequency for 1 DOF Damped system

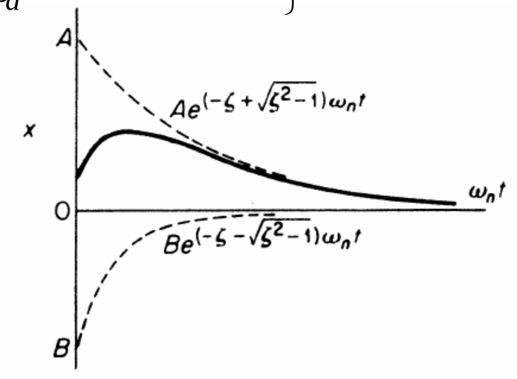
$$\frac{d^2x}{dt^2} + 2\zeta\omega_n \frac{dx}{dt} + \omega_n^2 x = 0 \qquad \omega_n = \sqrt{\frac{k}{m}} \qquad \zeta = \frac{c}{2\sqrt{km}}$$

Initial conditions: $x = x_0 \quad \frac{dx}{dt} = v_0 \quad t = 0$

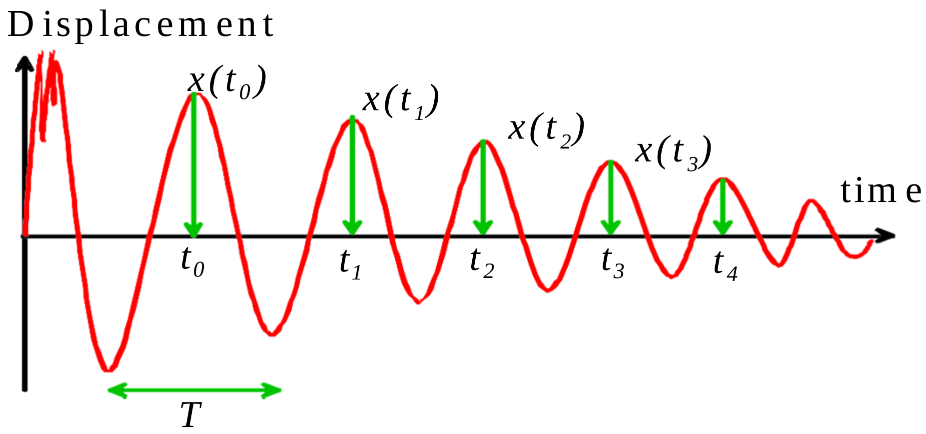
Overdamped:

$$\varsigma > 1$$

$$x(t) = \exp(-\varsigma\omega_n t)\left\{\frac{v_0 + (\varsigma\omega_n + \omega_d)x_0}{2\omega_d}\exp(\omega_d t) - \frac{v_0 + (\varsigma\omega_n - \omega_d)x_0}{2\omega_d}\exp(-\omega_d t)\right\}$$



Natural frequency for 1 DOF Damped system



Log decrement: $$\delta = \frac{1}{n}\log\left(\frac{x(t_0)}{x(t_n)}\right)$$

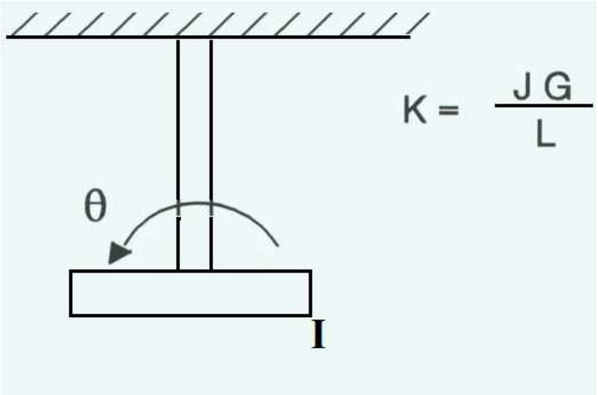
Period: T

Then $$\varsigma = \frac{\delta}{\sqrt{4\pi^2 + \delta^2}} \qquad \omega_n = \frac{\sqrt{4\pi^2 + \delta^2}}{T}$$

Torsional Vibration

$$k_{torsion} = \frac{JG}{L} \; (N.m/rad)$$

$$\omega_n = \sqrt{\frac{k_{torsion}}{I}} \; (rad/s)$$



G is modulus of rigidity (N/m^2)

J=polar moment

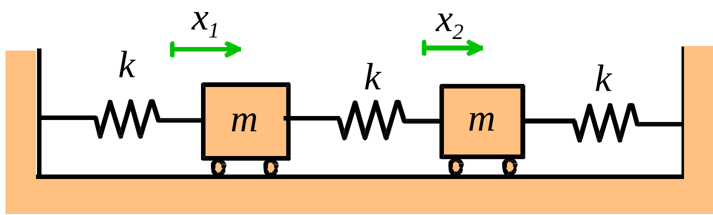
$$J = \frac{\pi d^4}{32}$$

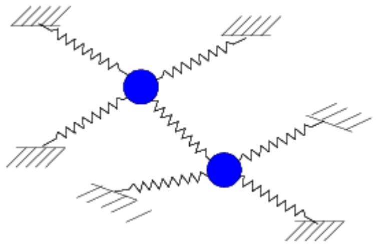
L= length of shaft (m)
D = diameter of shaft (m)
I = disk moment of inertia (kgm^2)

Vibration Principles

If masses are particles

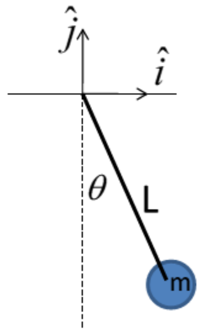
Expected No. of vibration modes = No. of masses x No. of directions masses can move independently





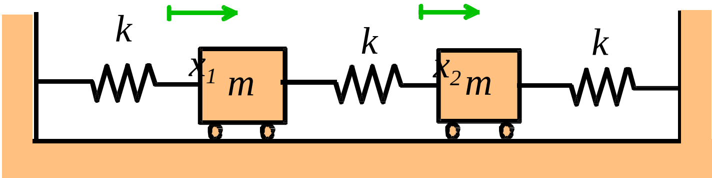
If masses are rigid bodies (can rotate and have inertia)

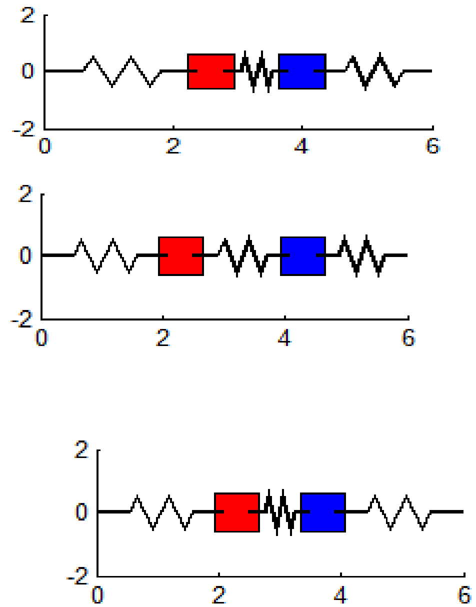
Expected No. of vibration modes = No. of masses x (No. of directions masses can move + No. possible axes of rotation)



Vibration modes and natural frequencies

- A system usually has the same number of natural frequencies as the degrees of freedom
- **Vibration modes are the** special initial deflections that cause entire system to vibrate harmonically
- **Natural Frequencies** are the corresponding vibration frequencies

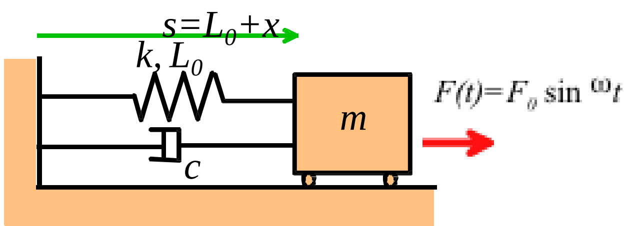




Forced Vibration

Important information include:

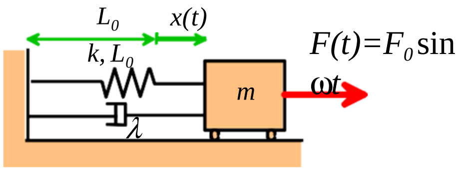
- 'Amplitude' and 'phase' of steady-state response of a forced vibration system
- amplitude-v-frequency formulas (or graphs), resonance, high and low frequency response for 3 systems

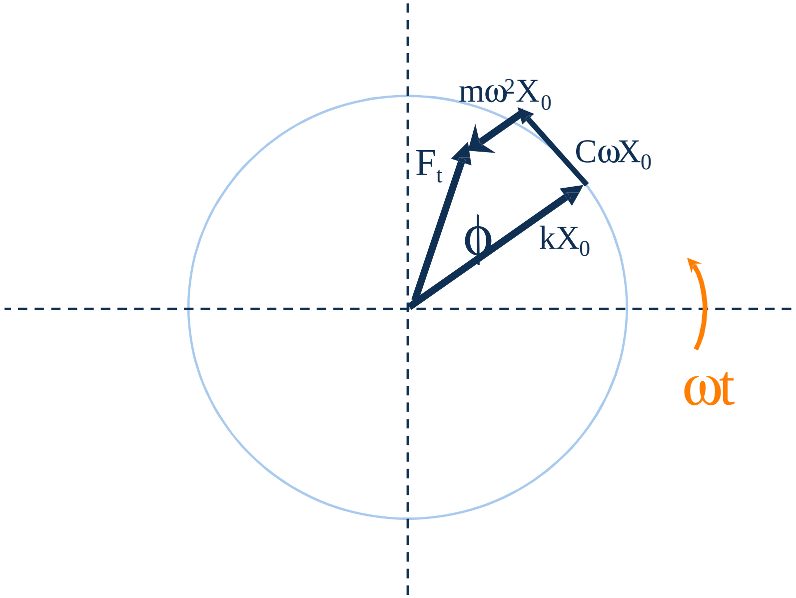


$$\frac{1}{\omega_n^2}\frac{d^2x}{dt^2}+\frac{2\varsigma}{\omega_n}\frac{dx}{dt}+x=KF_0\sin\omega t$$

$$\omega_n=\sqrt{\frac{k}{m}},\qquad \varsigma=\frac{\lambda}{2\sqrt{km}},\qquad K=\frac{1}{k}$$

Forced Vibration





Force Balance

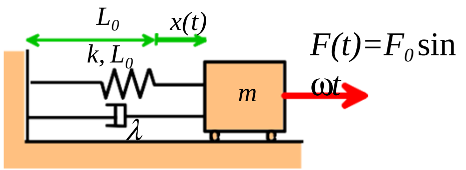
$$F_o = \sqrt{(X_0 k - m\omega^2 X_0)^2 + (C\omega)^2}$$

$$F_o = X_0 k \sqrt{\left(1 - \frac{m\omega^2}{k}\right)^2 + \left(\frac{C\omega}{k}\right)^2}$$

$$F_o = X_0 k \sqrt{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + \left(\frac{C\omega}{k}\right)^2}$$

$$F_o = X_0 k \sqrt{\left(1 - \frac{\omega^2}{\omega_n^2}\right)^2 + \left(2\xi\frac{\omega}{\omega_n}\right)^2}$$

Forced Vibration



$$\frac{1}{\omega_n^2}\frac{d^2x}{dt^2}+\frac{2\varsigma}{\omega_n}\frac{dx}{dt}+x=KF(t)$$

$$\omega_n=\sqrt{\frac{k}{m}}, \qquad \varsigma=\frac{\lambda}{2\sqrt{km}}, \qquad K=\frac{1}{k}$$

$$x(t)=X_0\sin(\omega t+\phi)$$

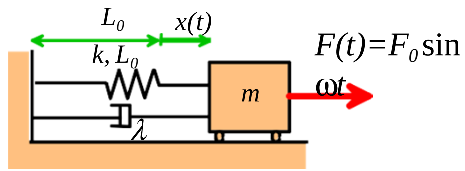
$$X_0=\frac{KF_0}{\left\{\left(1-\omega^2/\omega_n^2\right)^2+\left(2\varsigma\omega/\omega_n\right)^2\right\}^{1/2}} \qquad \phi=\tan^{-1}\frac{-2\varsigma\omega/\omega_n}{1-\omega^2/\omega_n^2}$$

Forced Vibration

$$X_{st} = \frac{F_o}{k} \quad \text{static response}$$

$$\frac{X_0}{X_{st}} = \frac{1}{\sqrt{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\left(\frac{C}{C_c}\right)\frac{\omega}{\omega_n}\right)^2}}$$

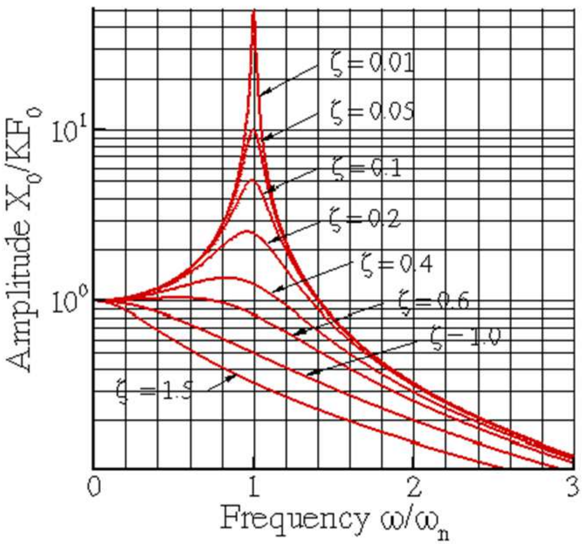
$$X_0 = \frac{\frac{F_o}{k}}{\sqrt{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\left(\frac{C}{C_c}\right)\frac{\omega}{\omega_n}\right)^2}}$$

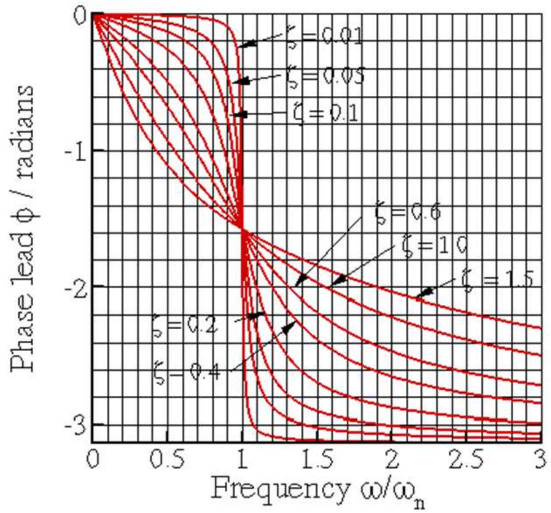


Frequency Response

$$x(t) = X_0 \sin(\omega t + \phi)$$

$$X_0 = \frac{KF_0}{\left\{\left(1-\omega^2/\omega_n^2\right)^2 + \left(2\varsigma\omega/\omega_n\right)^2\right\}^{1/2}} \qquad \phi = \tan^{-1}\frac{-2\varsigma\omega/\omega_n}{1-\omega^2/\omega_n^2}$$

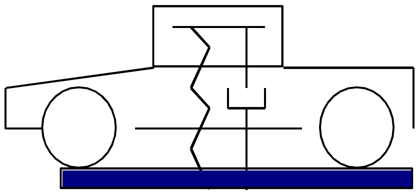




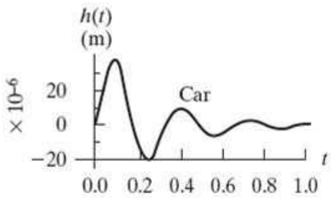
System vibrates at same frequency as force
Amplitude depends on forcing frequency, nat frequency, and damping coeft.

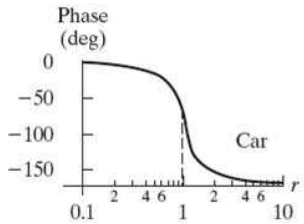
Ex.: Different devices with the same frequency

car

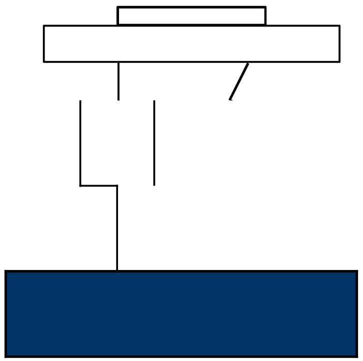


$$m = 1000\,\text{kg}$$
$$k = 400{,}000\,\text{N/m}$$
$$C = 8000\,\text{Ns/m}$$
$$\omega_n = \sqrt{\frac{400{,}000}{1000}} = 20 \text{ rad/s}$$
$$\zeta = \frac{8000}{2(1000)(20)} = 0.2$$
$$\omega_d = 20\sqrt{1-0.2^2} = 19.5959\text{rad/s}$$

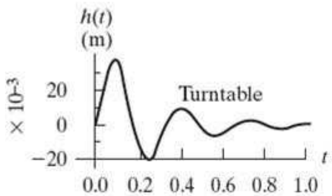




CD drive

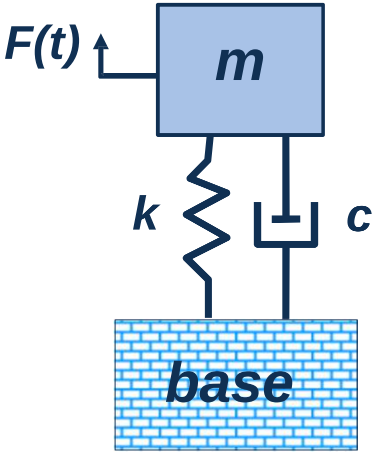


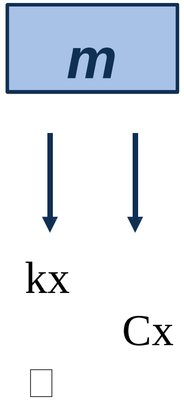
$$m = 1\,\text{kg}$$
$$k = 400\,\text{N/m}$$
$$C = 8\,\text{Ns/m}$$
$$\omega_n = \sqrt{\frac{400}{1}} = 20 \text{ rad/s}$$
$$\zeta = \frac{8}{2(1)(20)} = 0.2$$
$$\omega_d = 20\sqrt{1-0.2^2} = 19.5959\text{rad/s}$$



Magnitude plot will have the same shape
Time responses will have the same form for similar (but scaled) disturbances BUT WITH DIFFERENT MAGNITUDES

Base Excitation

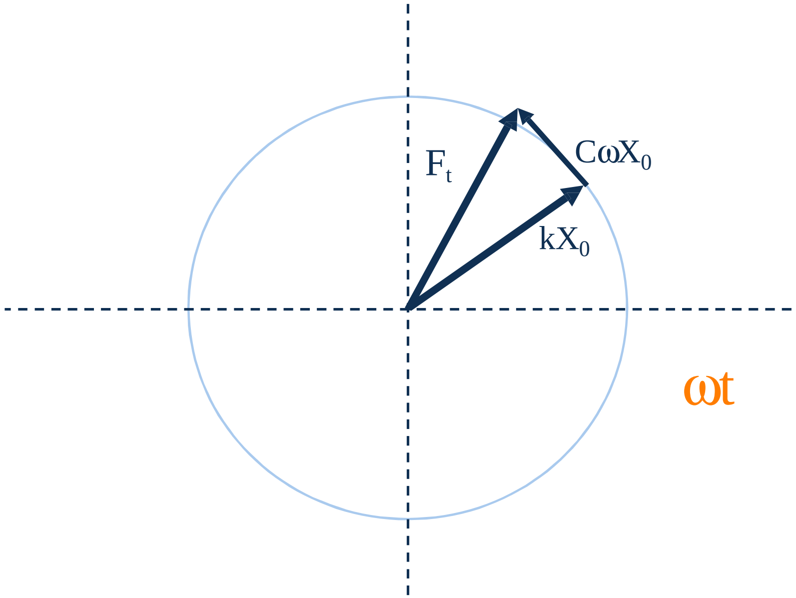




Force Transmitted to the base

$$F_t = C\frac{dx}{dt} + kx$$

Transmissibility



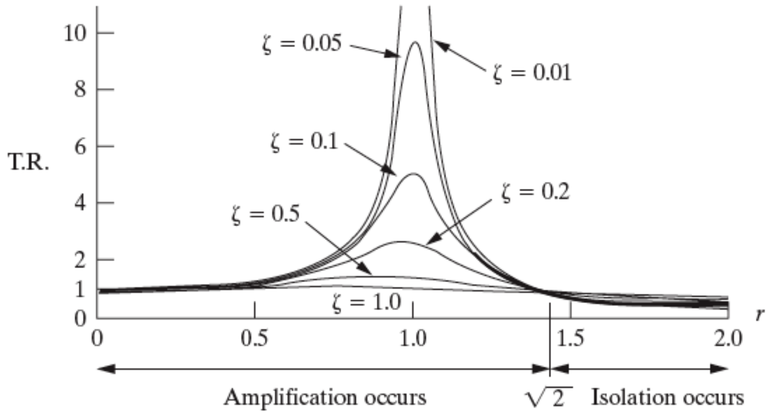
$$F_t = \sqrt{(X_0 k)^2 + (C\omega X_0)^2}$$

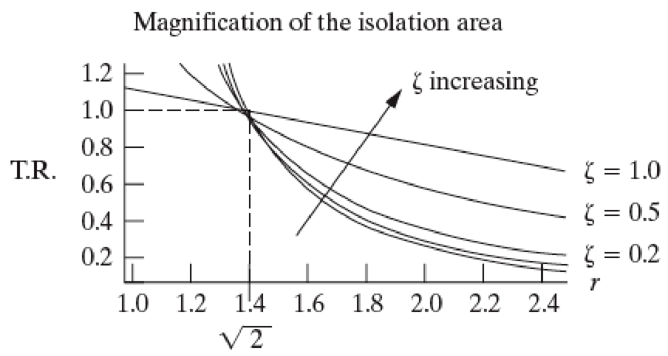
$$F_t = X_0 k \sqrt{1 + \left(\frac{C\omega}{k}\right)^2}$$

$$F_t = X_0 k \sqrt{1 + \left(2\xi \frac{\omega}{\omega_n}\right)^2}$$

Transmissibility- isolation as a function of stiffness

- For stiffness such that the frequency ratio is larger than root 2, isolation occurs, but increased damping reduces the effect
- For less than root 2, increased damping reduces the magnitude.





Transmissibility

$$\left|\frac{F_t}{F_0}\right| = \frac{\sqrt{k^2+(C\omega)^2}}{\sqrt{(k-m\omega^2)^2+(C\omega)^2}} = \frac{\sqrt{1+(2\zeta r)^2}}{\sqrt{(1-r^2)^2+(2\zeta r)^2}} = |TR|$$

$$\tan\Phi = (\frac{2\zeta r}{1-r^2})$$

$$X = \frac{F_0}{\sqrt{(k-m\omega^2)^2+(C\omega)^2}}$$

$$\frac{X}{X_0} = \frac{1}{\sqrt{(1-r^2)^2+(2\zeta\omega)^2}}$$

r = $\frac{\omega}{\omega_n}$ $\qquad$ $\zeta = \frac{C}{C_c}$ $\qquad$ $\omega_n = \sqrt{\frac{k}{m}}$

for $\zeta = 0$ and $r > \sqrt{2}$; $|TR| = \frac{1}{r^2-1}$, $\qquad$ $C_c = 2m\omega_n$

Mechanical Condition Monitoring
Vibration Analysis
Lecture 2

Outline

- Maintenance Strategies
- Condition Monitoring Techniques
- Characteristics of a Vibration Signal
- Overall Vibration Criteria
- Fault Diagnostics of rotating components

Objectives

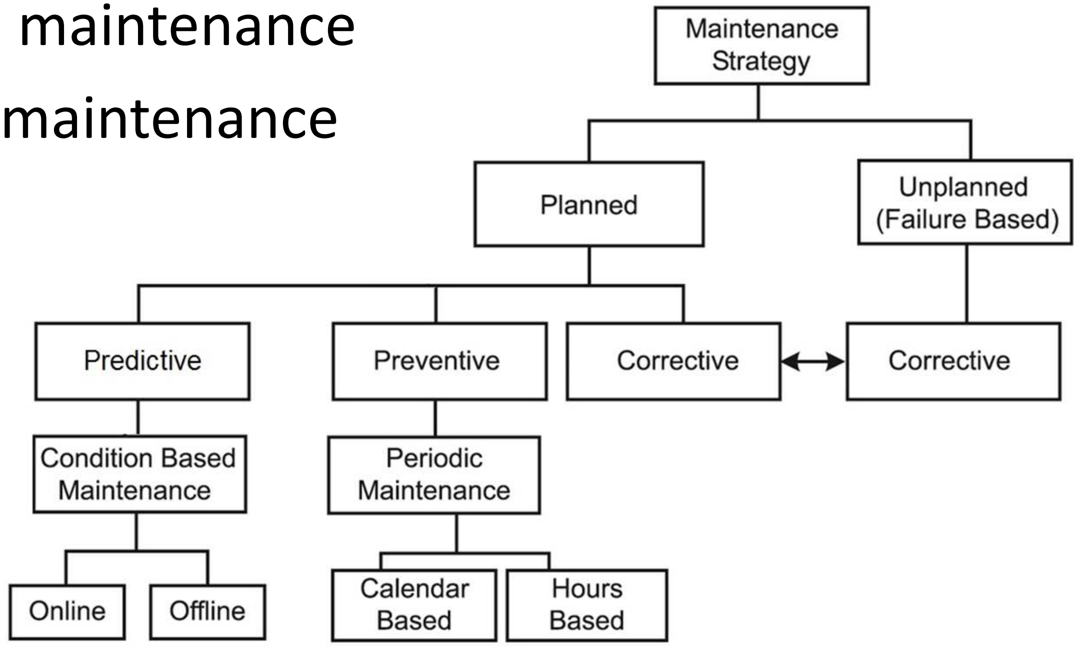
- Maintenance is the management, control, execution and quality of those activities which will ensure that optimum levels of availability and overall performance of plant are achieved, in order to meet business objectives - The British Department of Trade & Industry (DTI) (Rao, B.K.N.).
- Maintenance activities can be characterised as:

  a) general purpose, b) essential and c) critical

Maintenance Strategies

There are essentially three main approaches to maintenance of structures and machines:

- Run-to failure maintenance
- Preventive maintenance
- Predictive maintenance



General Purpose Machines

- Failure does not affect plant safety
- Not critical to plant production
- Machine has an installed spare or can operate on demand
- These machines require low to moderate expenditure, expertise and time to repair
- Secondary damage does not occur or is minimal

Essential Equipment

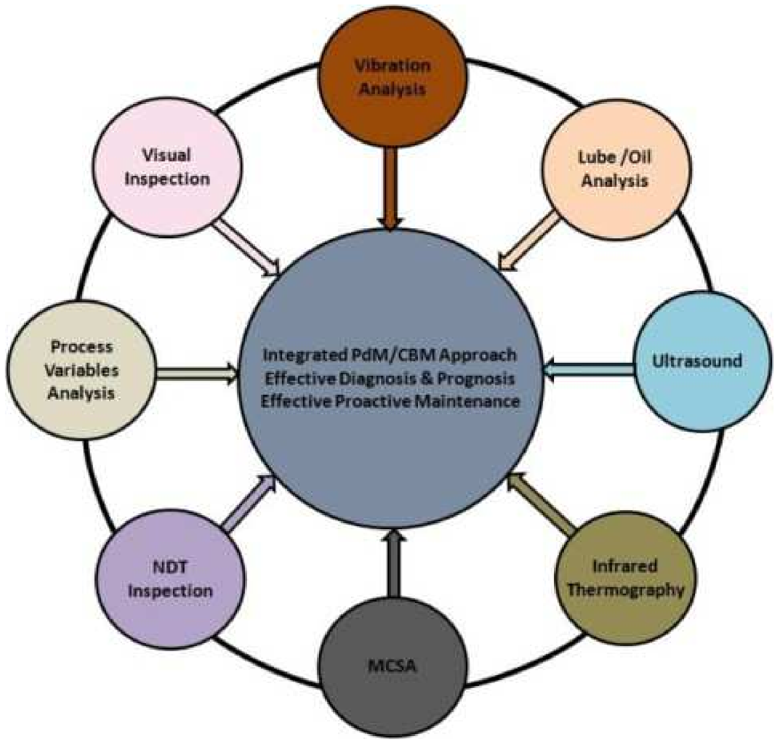
- Machines whose failure can affect plant safety
- Machines that are essential for plant operation and where shutdown will curtail a unit operation or part of the process
- Machines that may or may not have an installed spare available
- Shut-down is possible but may affect production process
- High power and speed might not be running continuously
- Some machines that demand time-based maintenance
- These machines require moderate expenditure, expertise and time to repair

Critical Equipment

- Machines whose failure can affect plant safety
- Machines that are essential for plant operation and where a shut-down will curtail the production process
- Machines which do not have spare parts
- Machines that have high capital cost, are very expensive to repair, or take a long time to repair

Condition Monitoring Techniques

- Condition monitoring attempts to detect symptoms of eminent failure and approximates time of a functional failure.
- It utilises a combination of techniques to obtain the actual operating condition of the machines based on collected data.
- It can operate online or offline



Condition Monitoring Techniques

The specific techniques used depend on the type and operation of the machines:

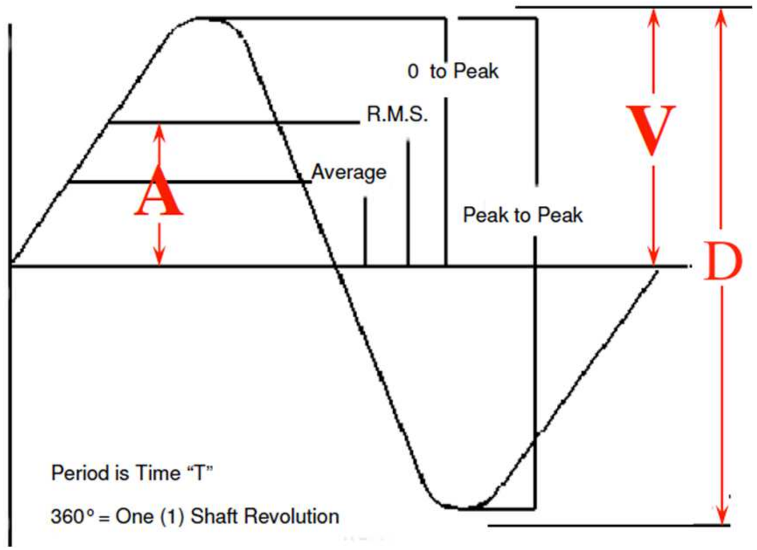
- **Vibration monitoring** – this is the most commonly used and effective technique to detect internal defects in rotating machinery.
- **Acoustic emission monitoring** – this involves detection and location of cracks in bearings, structures, pressure vessels and pipelines.
- **Oil analysis** – lubrication oil is analysed and the occurrence of certain microscopic particles in it can be connected to the condition of bearings and gears.
- **Particle analysis** – worn machinery components, whether in reciprocating machinery, gearboxes or hydraulic systems, release debris. Collection and analysis of this debris provides vital information on the deterioration of these components.
- **Ultrasonic monitoring** – this is used to measure thickness of corrosion or crack on pipelines, offshore structures, pressure vessels.

Vibration Monitoring

- All rotating machines produce vibrations that are a function of the machine **operating conditions** and **machine dynamics**.
- When a machine has a defect, the energy level of the specific component increases

Characteristics of a Measured Vibration Signal

- Frequency: its relation to the **natural frequency**, **rotation frequency**, and **defect frequencies**
- RMS Velocity
- P to P velocity
- Displacement
- Acceleration
- Phase
- Bandwidth



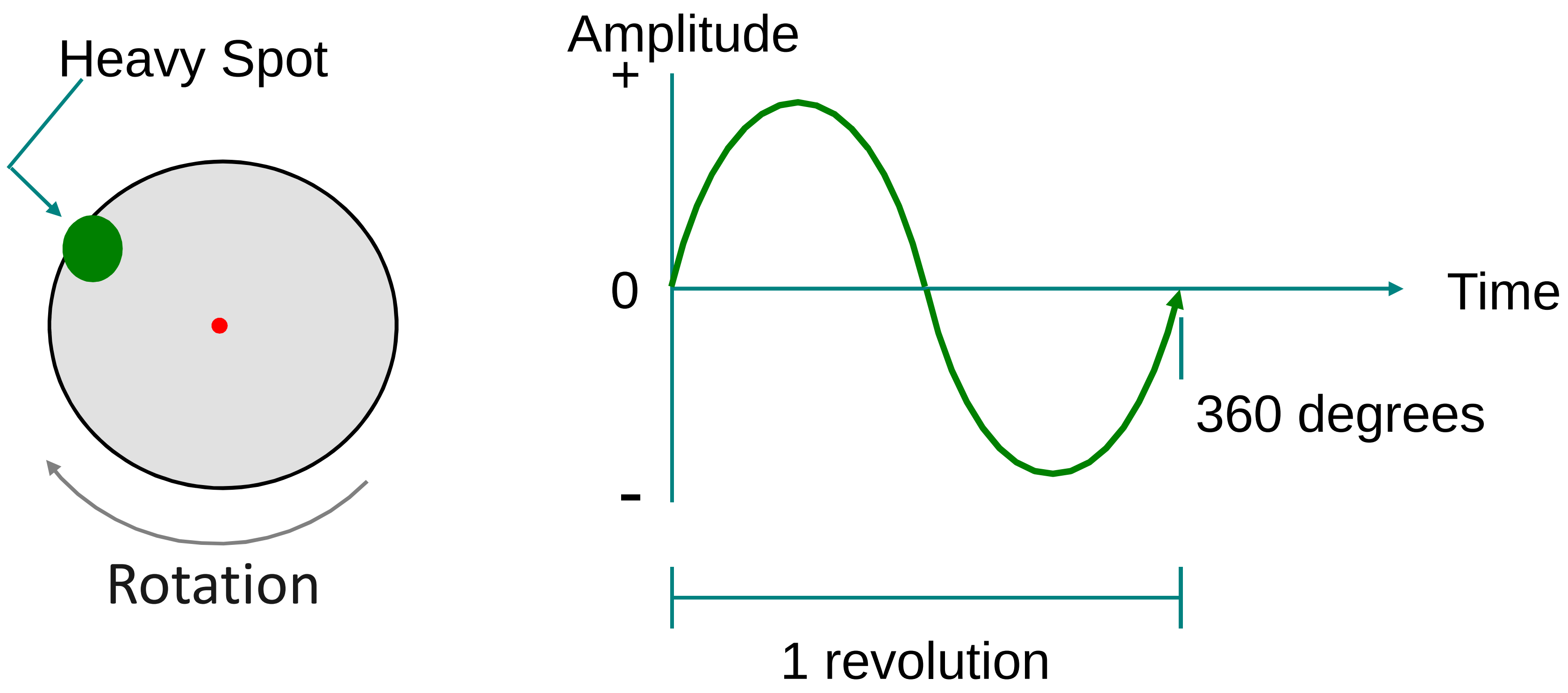
= Frequency Span / Analyzer Lines

Characteristics of a Measured Vibration Signal

Some useful parameters characterizing vibration:

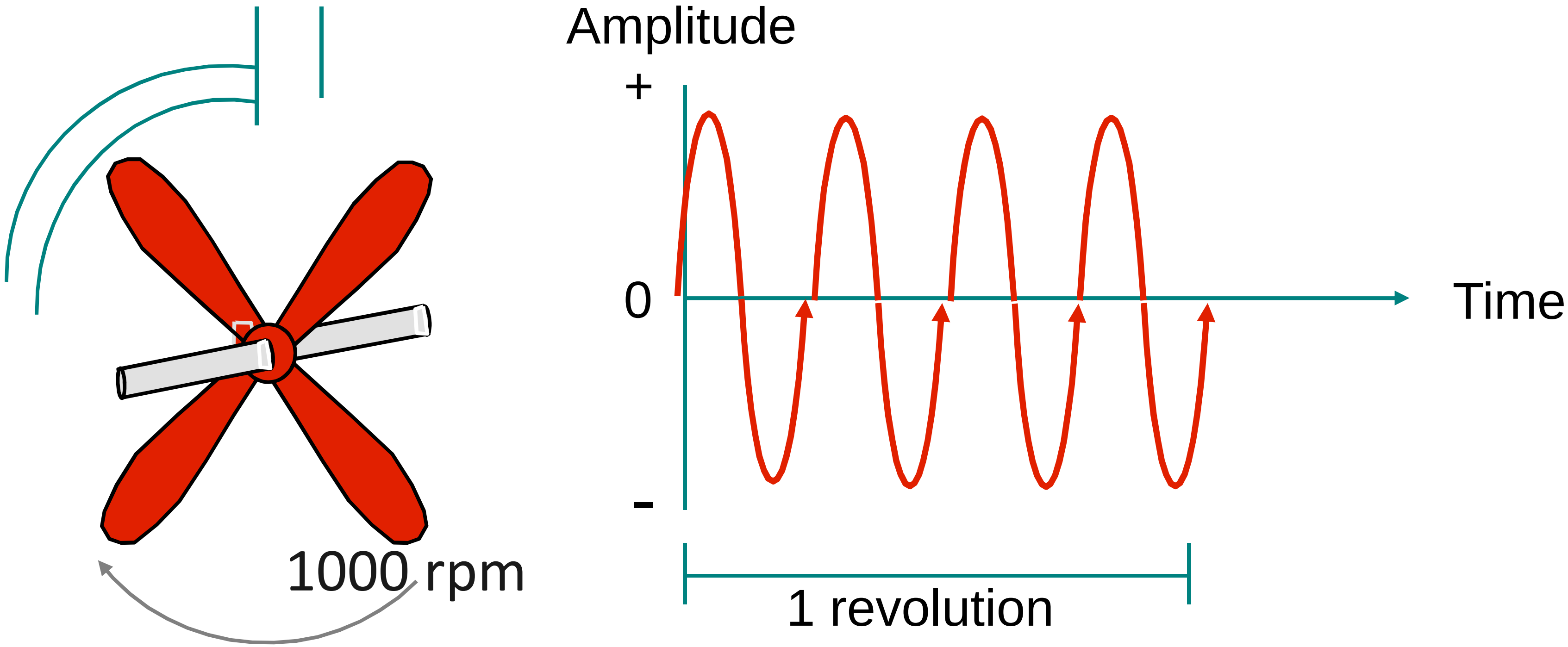
Displacement (m)	Velocity (m/s)	Acceleration (m/s^2)
Frequency (Hz)	Bandwidth (Hz)	Spike Energy (gSE)
Power Spectral Density	Peak Value	Root mean square (RMS)
Crest factor (CF)	Arithmetic mean (AM)	Geometric mean (GM)
Standard deviation (SD)	Kurtosis (K)	Skewness
Phase (deg)		

Information contained in a time waveform



3600 rpm = 3600 cycles per minute
60 Hz = 60 cycles per second
1 order = one times turning speed

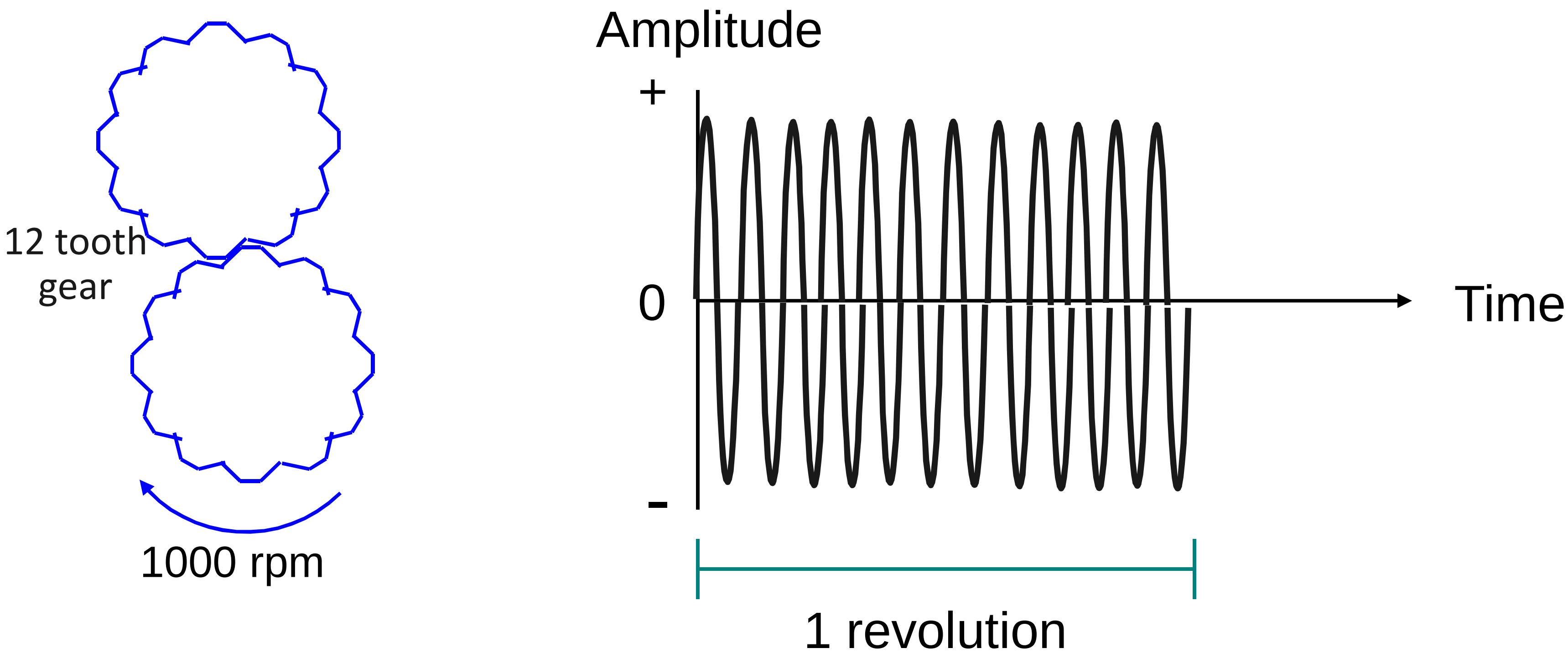
Information contained in a time waveform



4 blades = vibration occurs 4 times per revolution

4 x 1000 rpm = vibration occurs at 4000 cycles per minute

Information contained in a time waveform



12 teeth are meshing every revolution of the gear

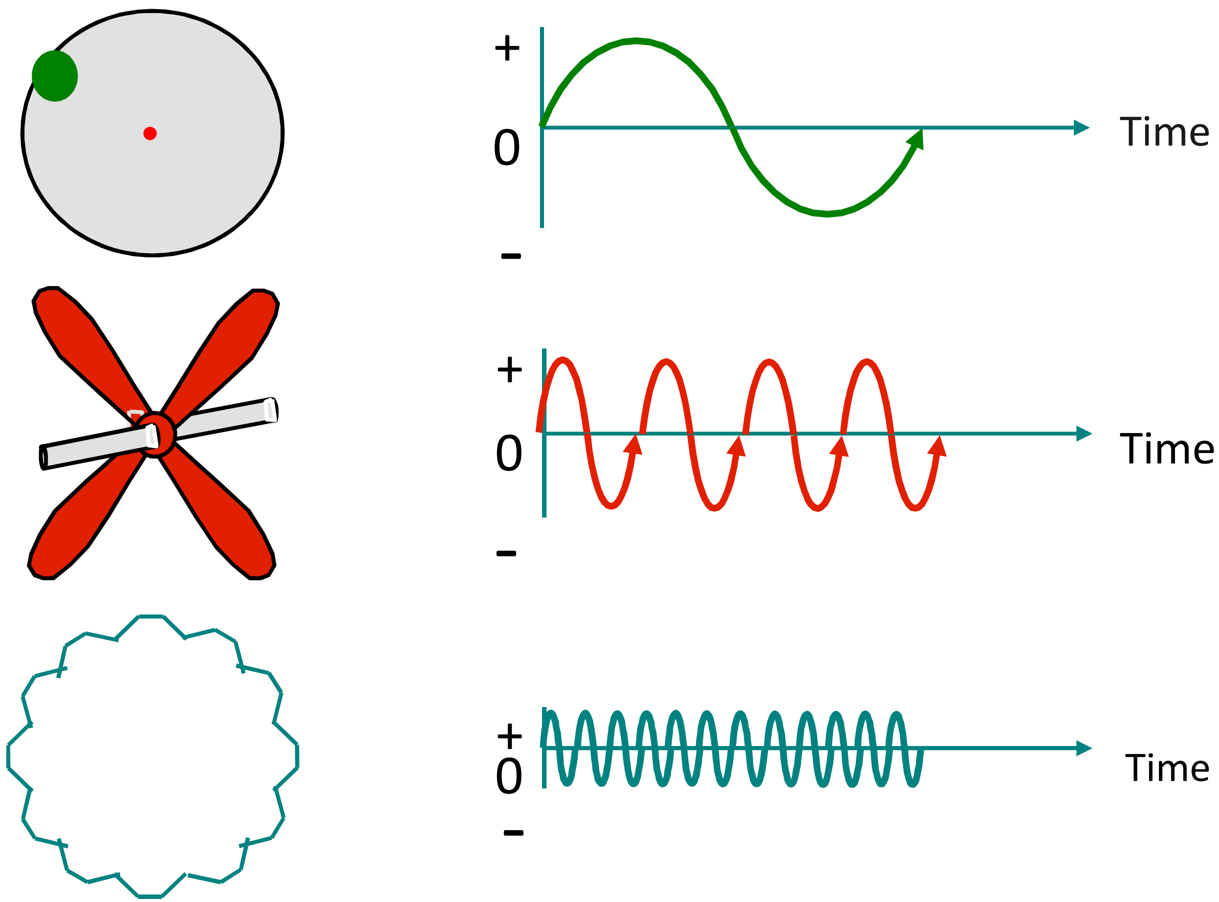
12 x 1000 rpm = vibration occurs at 12,000 cycles per minute

frequency= 12,000 cpm = 200 Hz

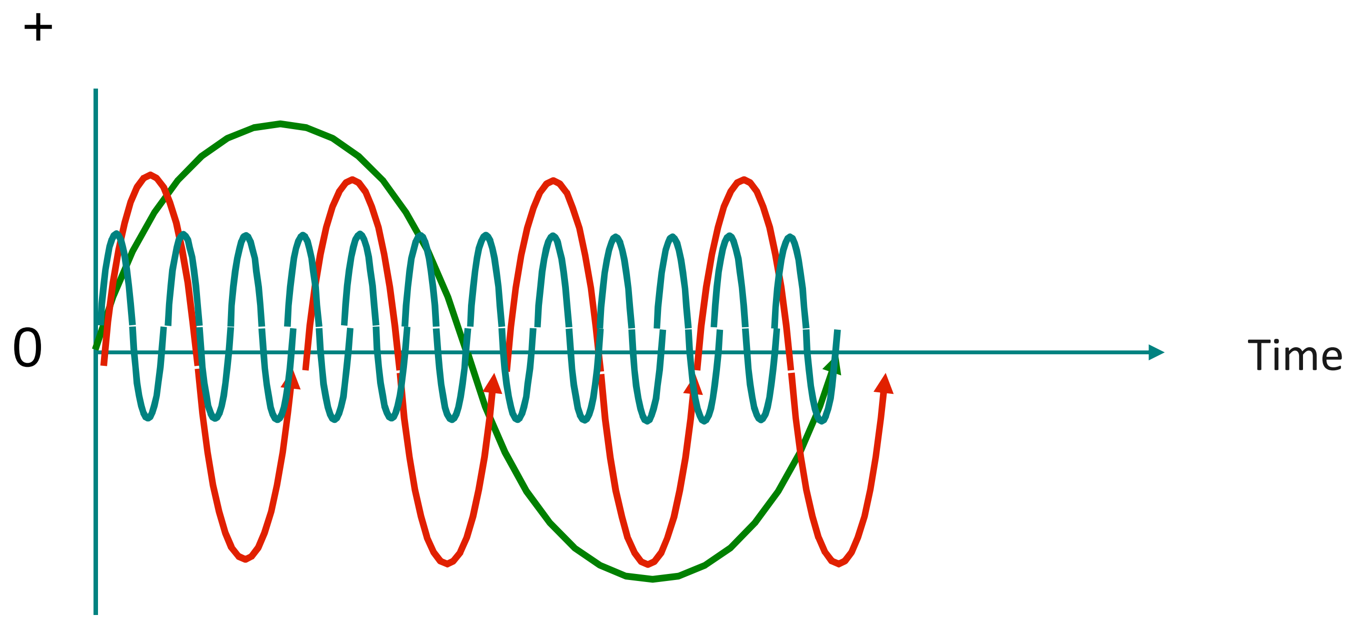
Vibration Profiles

- Vibration data recorded from machines is usually extremely complex as there are many sources of vibration.
- Each source generates its own profile which will essentially be *added* together to give the composite profile.
- Time-domain plots are useful in the overall analysis of machine trains to study operating condition changes but time-domain data are difficult to use.

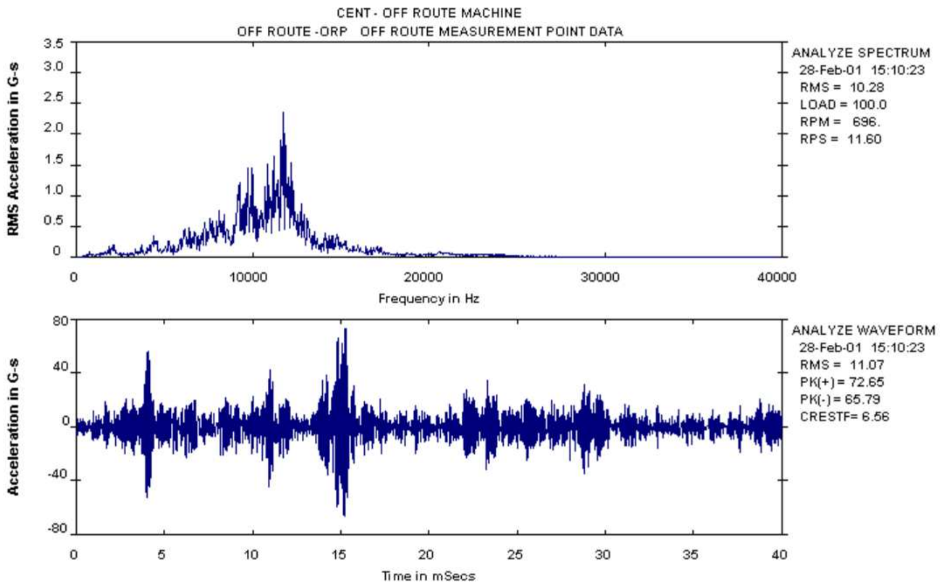
Information contained in a time waveform



Information contained in a system time waveform

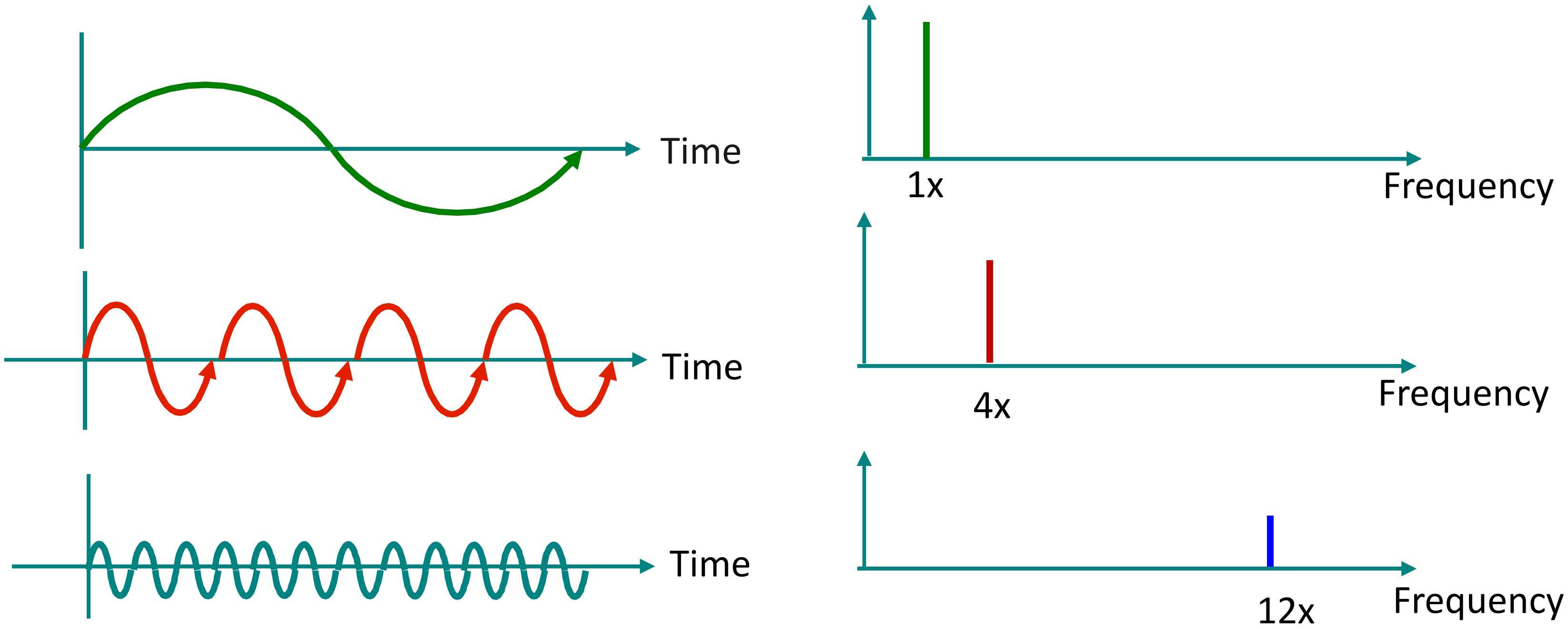


Complex/system time waveform

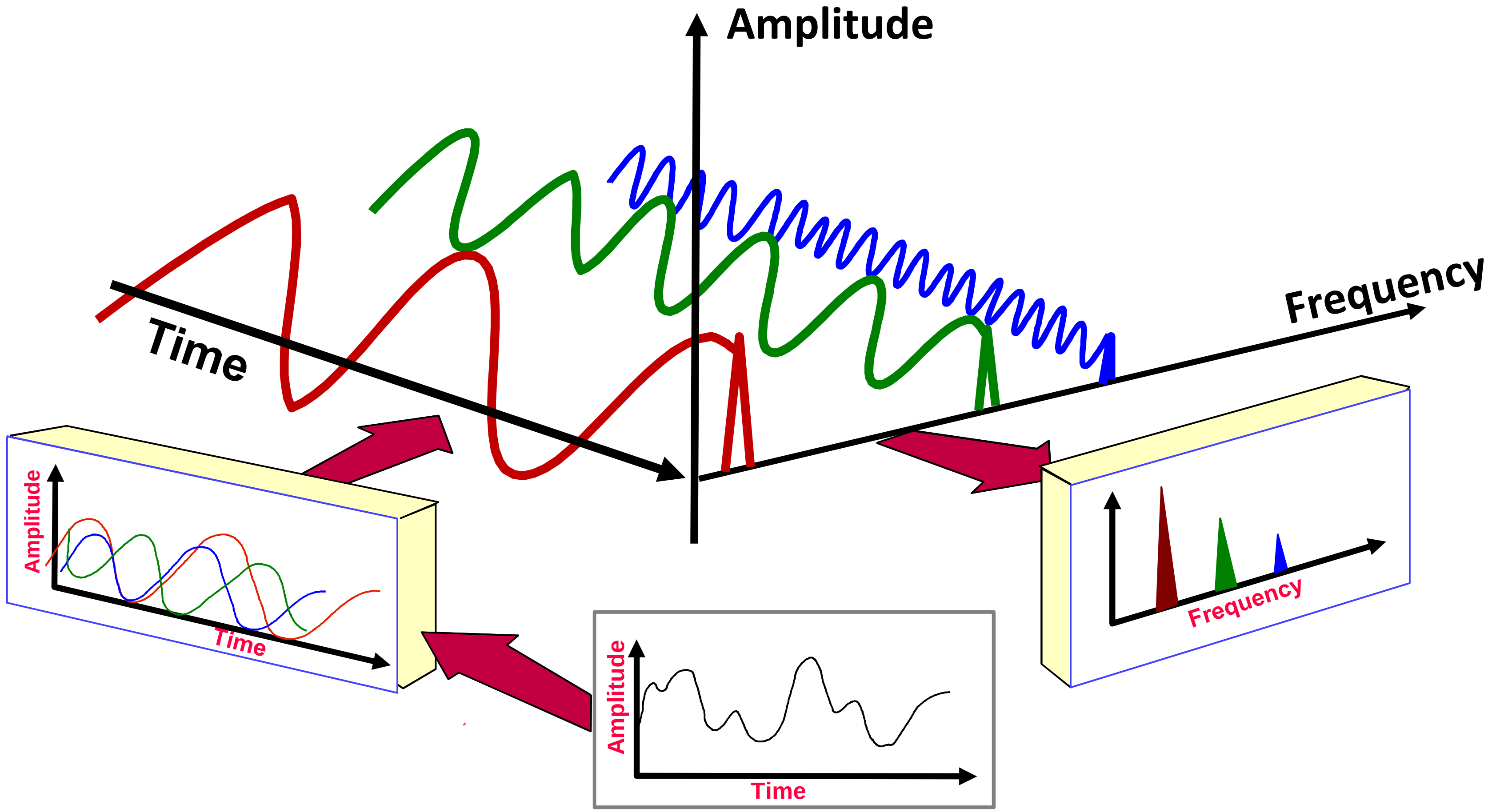


Time domain to frequency domain

- Time-domain data may be broken down into their frequency components using a Fast Fourier Transform (FFT).
- Frequency-domain data are required for equipment operating at more than one running speed and all rotating applications.



Time domain to frequency domain



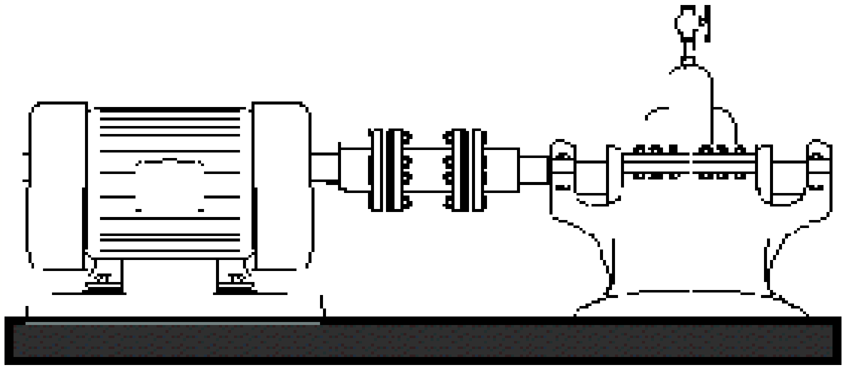
Vibration Analysis Technique

- **Comparative analysis** directly compares two or more data sets in order to detect changes in operating condition of machine.
- It is limited to direct comparison of time-or frequency-domain signature generated by machine.
- Comparison may be to machine baseline or industry standards.
- **Baseline** must be updated after maintenance.
- Key to using vibration analysis for predictive maintenance is the ability to differentiate between **normal** and **abnormal** vibration profiles.
- Many vibrations are normal for rotating or moving machinery, e.g. normal rotation of shafts and other rotors, contact with bearings, gear-mesh etc.
- Specific problems with machinery generate abnormal, yet identifiable, vibrations, e.g. loose bolts, misaligned shafts, worn bearings, leaks and incipient metal fatigue.

Vibration Profiles of Rotating Machines

- A rotating machine has one or more machine elements that turn with a shaft –e.g. rolling-element bearings, impellers and other rotors.
- In a perfectly balanced machine, all rotors run on their true centreline and forces are equal.
- In industrial machinery, rotors imbalance will generally be present due to uneven weight distribution or due to the imbalance between generated lift and gravity.
- Pumps, fans, compressors will be subject to imbalance caused by turbulent or unbalanced media flow.
- Combination of these forces with stiffness of rotor-support system will determine the vibration level.

Machinery Fault Detection Using Vibration



Unbalance of rotating parts	Misalignment of couplings and bearings	Bend or bow shafts
Worn or damage gears and bearings	Bad drive belts and chains	Torque variations
Electromagnetic forces	Aerodynamic forces	Hydraulic forces
Looseness	Rubbing	Resonance

Machine Failure Mode Analysis

- General Failure Modes
- Machine-Train Component Failure Modes

  -Bearing Failures

  -Gear Failures

  -Shaft Failure

Overall Vibration Criteria

- The overall level is a single number
- Calculation of the unfiltered amplitude of a vibration waveform.
- In the analysis of the majority of machinery vibration signatures, the absolute level of spectrum components is not as valid an indicator of machine problems as is the rate of increase in level of the components.
- The most common amplitude unit is Velocity.
- Displacement may be used when relative motion or slow speed is a consideration.
- Acceleration is often used in gearbox and high speed machinery as well as bearing troubleshooting.

ISO Standard 2372

ISO 2372 – ISO Guideline for Machinery Vibration Severity (10-1000 Hz)					
Ranges of Vibration severity		Examples of quality judgment for separate classes of machines			
Velocity – in/s – Peak	Velocity – mm/s – rms	Class I	Class II	Class III	Class IV
0.015	0.28				
0.025	0.45				
0.039	0.71				
0.062	1.12				
0.099	1.8				
0.154	2.8				
0.248	4.5				
0.392	7.1				
0.617	11.2				
0.993	18				
1.54	28				
2.48	45				
3.94	71				

Good
Acceptable
Monitor Closely
Not Acceptable

- The ISO standard 2372 provides vibration amplitude acceptance guidelines for rotating machinery operating between (10-1000 Hz)
- Limited in the high frequency range

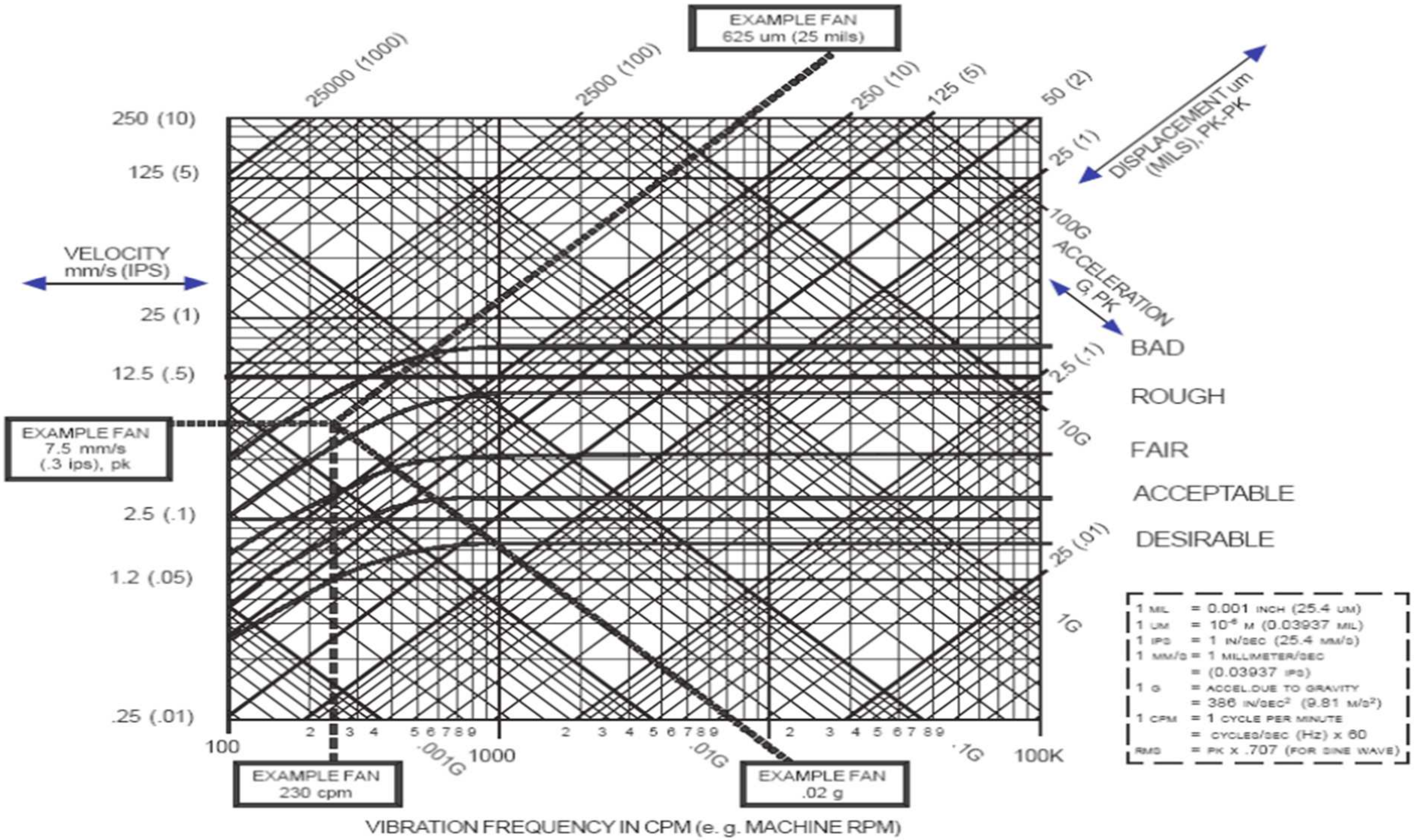
Class I Individual parts of engines and machines integrally connected with a complete machine in its normal operating condition (production electrical motors of up to 15 kW are typical examples of machines in this category).

Class II Medium-sized machines (typically electrical motors with 15–75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations.

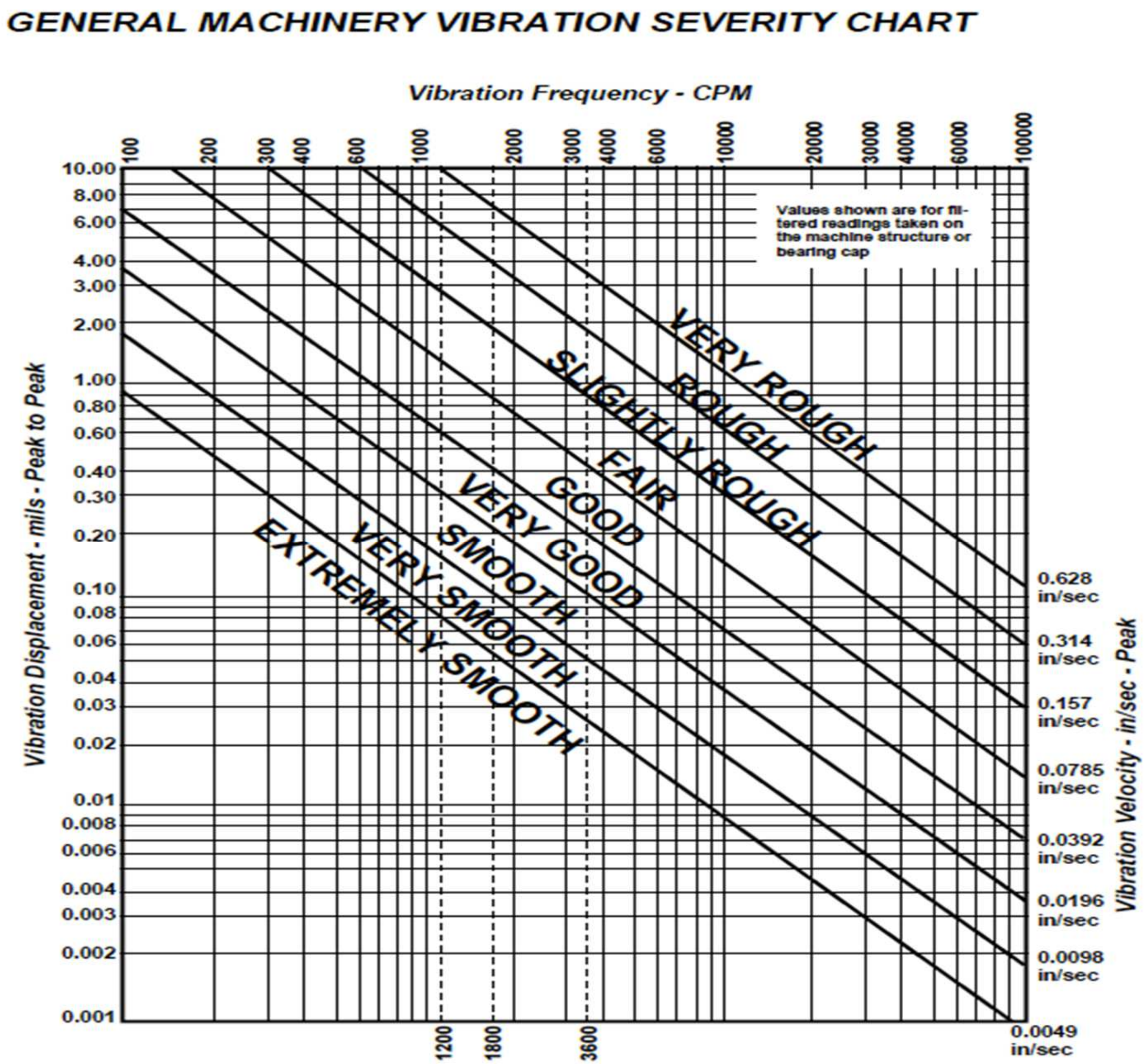
Class III Large prime movers and other large machines with rotating masses mounted on rigid and heavy foundations, which are relatively stiff in the direction of vibration.

Class IV Large prime movers and other large machines with rotating masses mounted on foundations, which are relatively soft in the direction of vibration measurement (for example – turbogenerator sets, especially those with lightweight substructures).

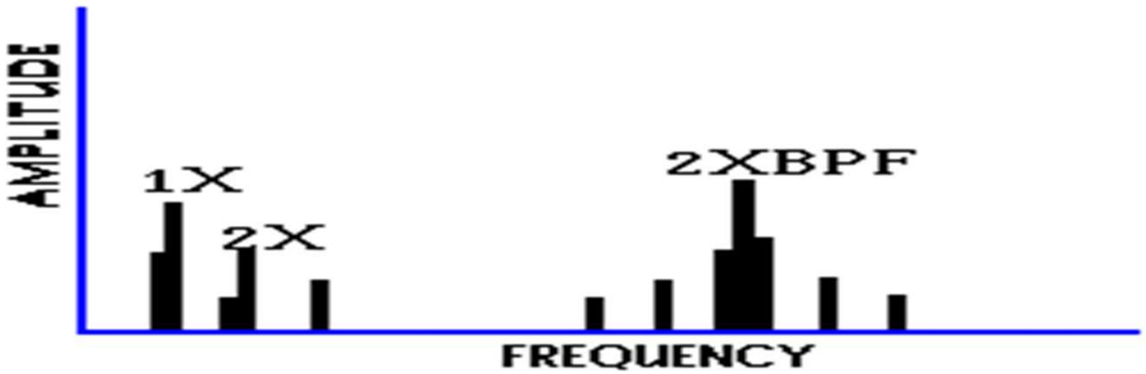
Commercial Standards
DLI Machinery Severity Chart



Overall Vibration Criteria



Machine Failure Mode Analysis



- Which frequencies exist?
- what are their relationships to the fundamental exciting frequencies?
- What are the amplitudes of each peak?
- How do the peaks relate to each other?
- If there are significant peaks, what are their source?

Machine Failure Mode Analysis

Frequency Regions

- Synchronous: rotational frequency and its harmonics

N x RPM

where N is an integer

- Sub synchronous: <1 x RPM
- Non-synchronous:

F x RPM

where F is >1 x RPM but not an integer

Machine Failure Mode Analysis

Causes of Sub-synchronous Frequencies

- Another component in the machine
- Another machine
- Belt drives
- Hydraulic instability
- Oil whirl, oil whip
- Rubs
- rotor, shaft, wheel
- Cage
- fundamental train - rolling element bearings.

Machine Failure Mode Analysis

Synchronous Frequency Causes

- Imbalance
- Pitch line run-out
- Misalignment
- Bent shaft
- Looseness
- Blade / vane pass
- Recips
- Gears
- Slot / Rotor Bar pass

Machine Failure Mode Analysis

Non – synchronous Frequency Causes

- Another machine
- Belt multiples
- Bearings.
- Resonance
- Electrical
- Chains

- Compressor surge
- Detonation
- Sliding surfaces
- Lube pumps
- Centrifugal clutches
- U-joints

Machine Failure Mode Analysis

Critical Speeds

- Critical speeds result due to the natural vibrating frequencies of the machine-train they are functions of the mass and stiffness of the machine.
- When the running speed coincides with one of the critical speeds excessive vibration occurs which is generally undesirable.
- Best way to confirm a critical-speed problem is to change running speed - amplitude of vibration components (1x, 2x, 3x running speed) will immediately drop if problem is due to critical-speed.

Machine Failure Mode Analysis

Looseness

- Balance means that all forces generated by rotating element of machine-train are in equilibrium. Any change in state of equilibrium creates an imbalance.
- *Imbalance* is one of most common condition monitoring problems. All machines exhibit some level of imbalance.
- Dominant frequency component is at running speed (1x) of shaft. Harmonics (2x, 3x, etc...) may be observed in multi-plane imbalance.

Machine Failure Mode Analysis

Looseness

- *Mechanical looseness* (e.g. poor bolting to foundations) can be present in vertical and horizontal planes and can create a variety of patterns in vibration signature.
- In some cases only 1x frequency is excited but generally full and half multiples of the running speed are present in spectra (0.5x, 1x, 1.5x, 2x etc.)

Machine Failure Mode Analysis

Misalignment

- *Misalignment* is virtually always present in machine trains.
- Three types of misalignment: internal, offset and angular.
- All three types excite 1x frequency as they create an apparent imbalance in machine.
- Internal and offset also excite 2x frequency as shaft creates two high-spots.

Machine Failure Mode Analysis

Modulations

- These are **frequency components** that appear in vibration signal but cannot be attributed to any specific physical cause or forcing function.
- They can be viewed as “ghost” or artificial frequencies
- They can result in significant machine-train damage.
- Ghosts are caused when two or more frequencies combine to produce another frequency component.
- Not an absolute indication of problem within machine-train but increased amplitude can amplify defects.

Machine Failure Mode Analysis

Modulation example

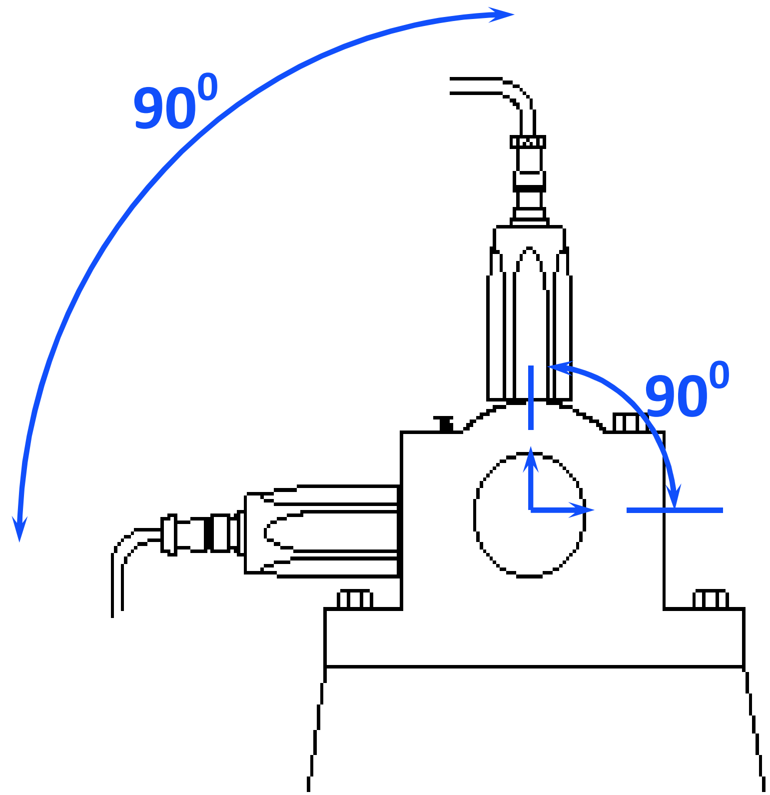
- Consider 10-tooth pinion gear rotating at 10 rpm whilst driving 20-tooth bullgear with an output speed of 5 rpm.
- Gear generates frequency components at 5, 10 and 100rpm (i.e. 10 teeth x 10rpm).
- This set can generate ghost frequencies at 15rpm (10+5), 110rpm (100+10), 95rpm (100-5) etc.

Machine Failure Mode Analysis

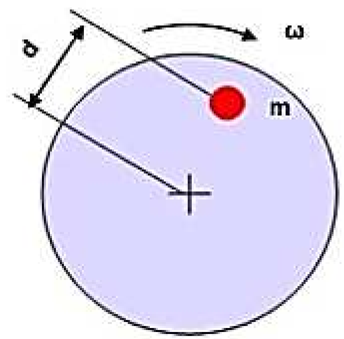
- *Process instability* is normally associated with bladed or vaned machinery such as fans and pumps.
- Process instability creates an unbalanced condition within the machine which generally excites the fundamental (1x) frequency and the blade-pass/vane-pass frequency components.

Diagnosing Unbalance

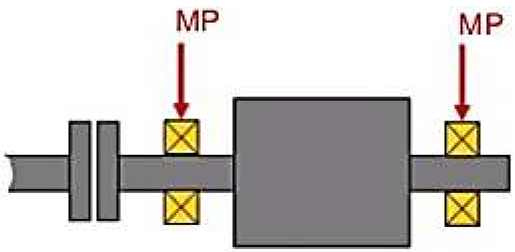
- Vibration frequency equals rotor speed.
- Vibration predominantly RADIAL in direction.
- Stable vibration phase measurement.
- Vibration increases as square of speed.
- Vibration phase shifts in direct proportion to measurement direction.

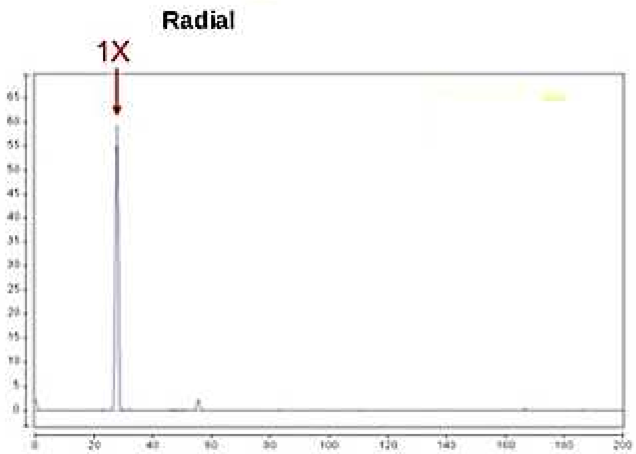


Unbalance



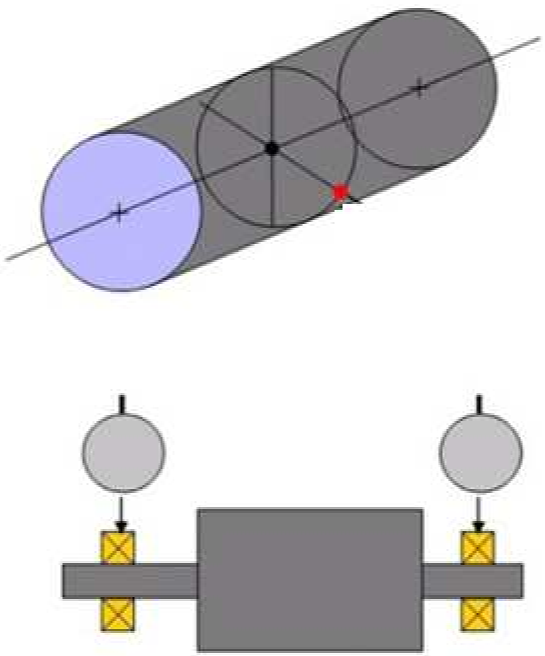
$F_{unbalance} = m\ d\ \omega^2$





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

Static Unbalance

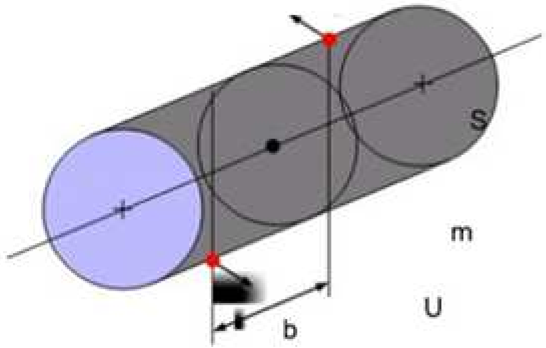


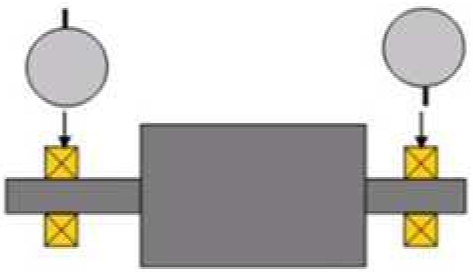
Static unbalance is caused by an unbalance mass out of the gravity centerline.

The static unbalance is seen when the machine is not in operation, the rotor will turn so the unbalance mass 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.

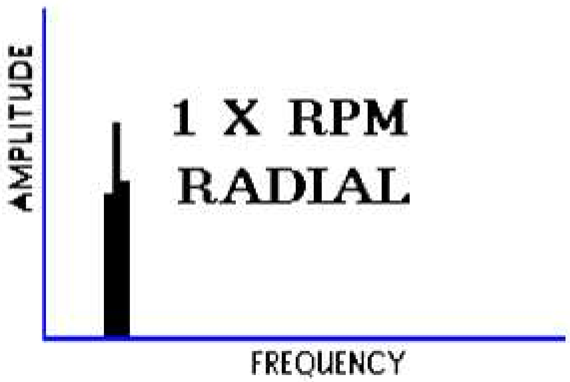
Pure Couple Unbalance





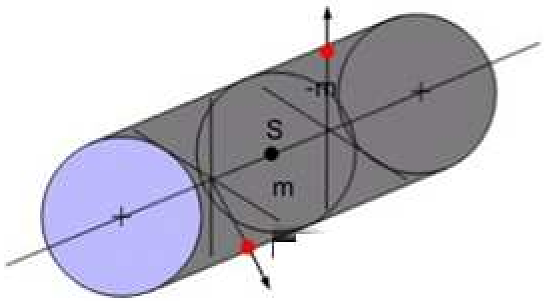
Pure couple unbalance is caused by two identical unbalance masses located at 180° in the transverse shaft area

Pure couple unbalance may be statically balanced



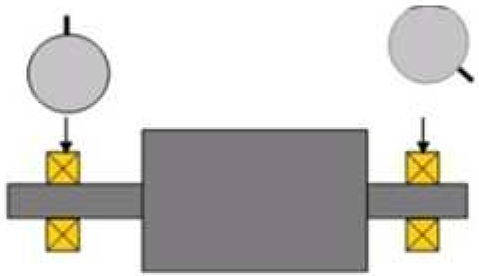
- 1X RPM always present and normally dominates
- Amplitude varies with square of increasing speed
- Can cause high axial as well as radial amplitudes
- Balancing requires Correction in two planes at 180°

Dynamic Unbalance



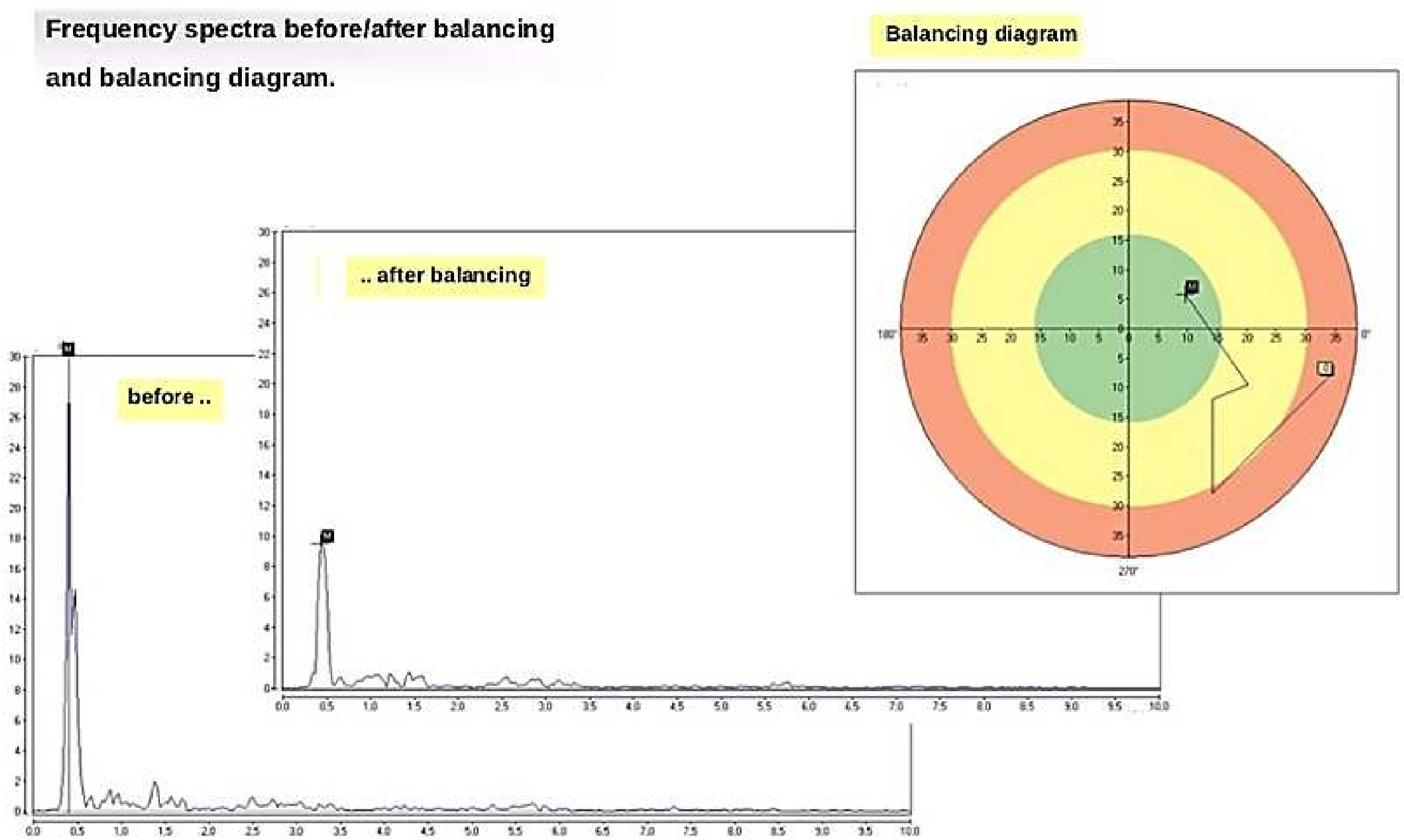
Dynamic unbalance is static and couple unbalance at the same time.

In practice, dynamic unbalance is the most common form of unbalance found.

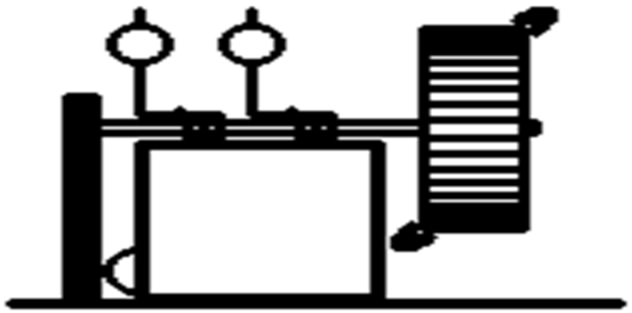


When rotating the dynamic unbalance produces a vibration signal at 1X, radial predominant and the phase will depend on the mass distribution along the axis.

Balancing

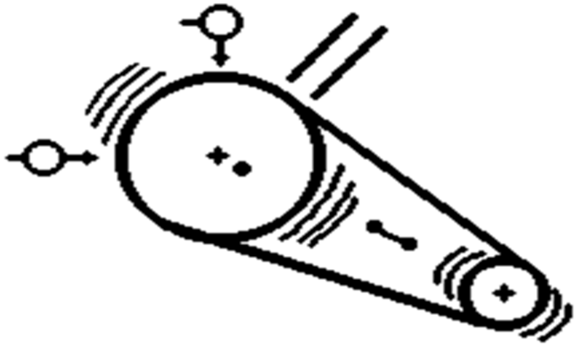

Frequency spectra before/after balancing
and balancing diagram.
Balancing diagram
.. after balancing
before ..

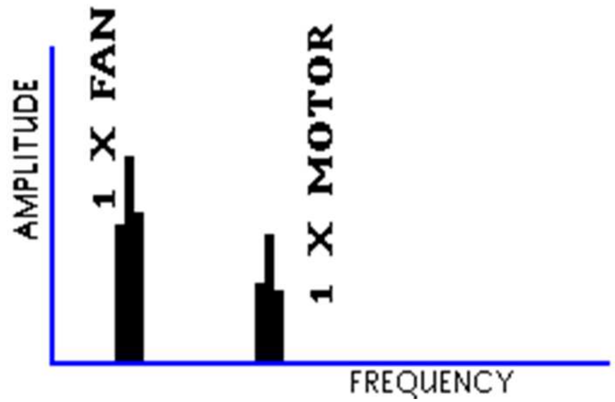
Overhung Rotor Unbalance



- 1X RPM present in radial and axial directions
- Axial readings tend to be in-phase but radial readings might be unsteady
- Overhung rotors often have both force and couple unbalance each of which may require correction

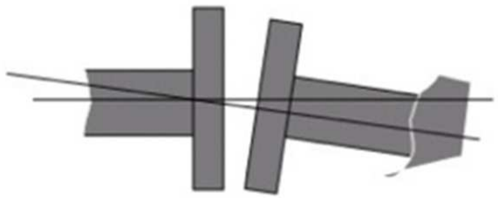
Eccentric Rotor



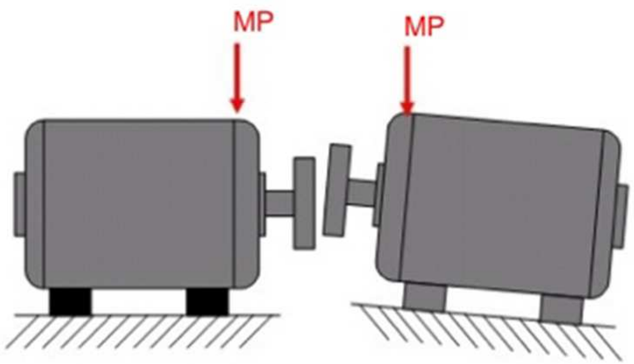


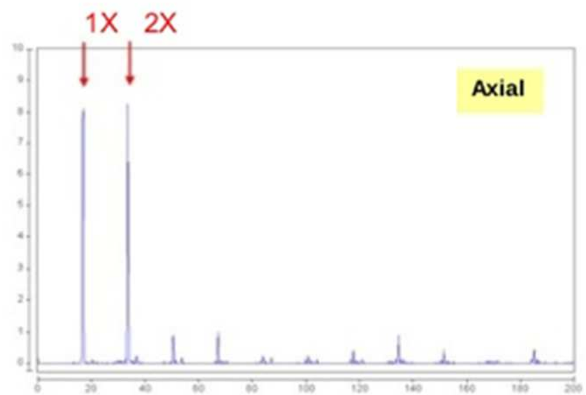
- Largest vibration at 1X RPM in the direction of the centerline of the rotors
- Comparative phase readings differ by 0^0 or 180^0
- Attempts to balance will cause a decrease in amplitude in one direction but an increase may occur in the other direction

Misalignment



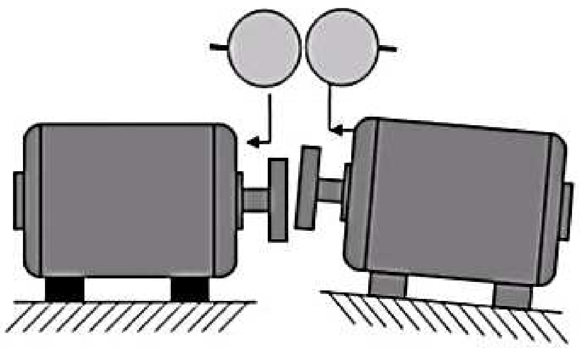
Misalignment is the condition when the geometric centerline of two coupled shafts are not co-linear along the rotation axis of both shafts at operating condition.





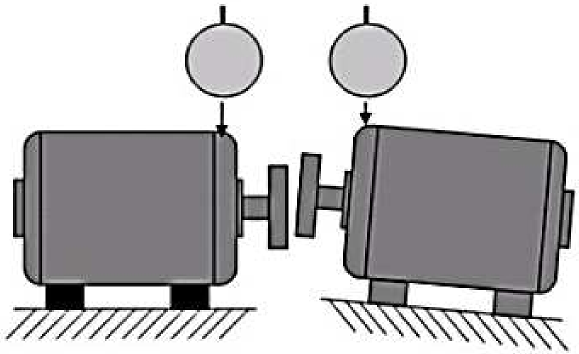
A 1X and 2X vibration signal predominant in the axial direction is generally the indicator of a misalignment between two coupled shafts.

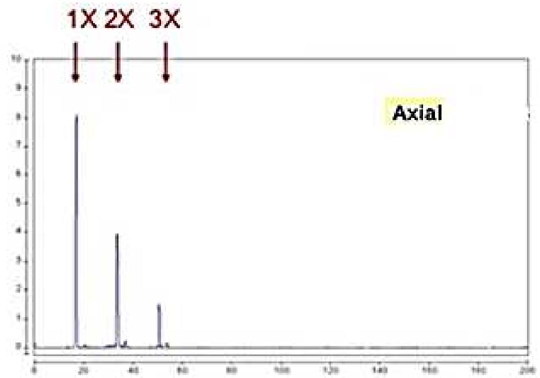
Angular Misalignment



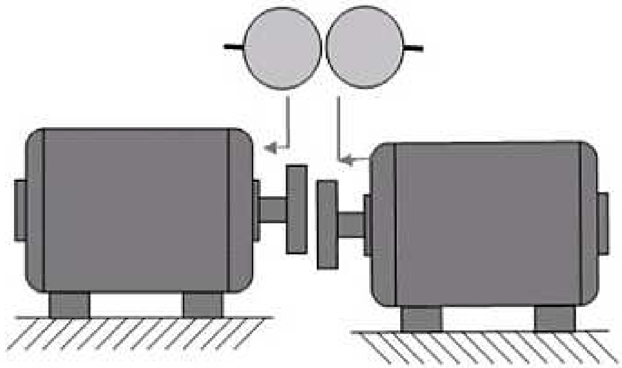
Angular misalignment is seen when the shaft centerlines coincide at one point a long the projected axis of both shafts.

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





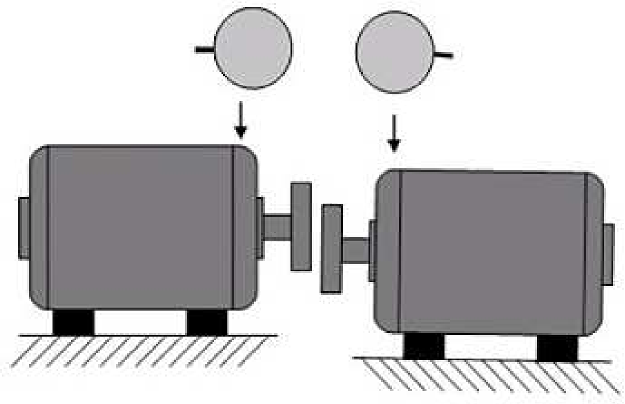
Parallel Misalignment

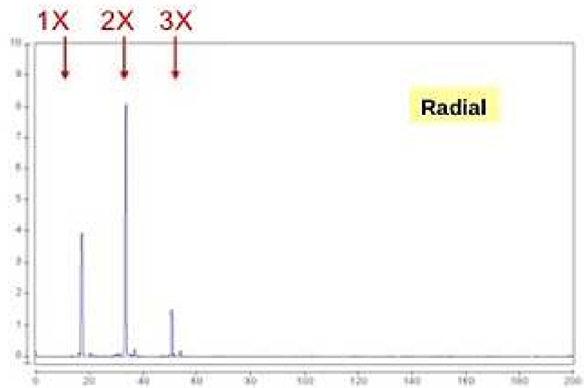


Parallel misalignment is produced when the centerlines are parallel but offset .

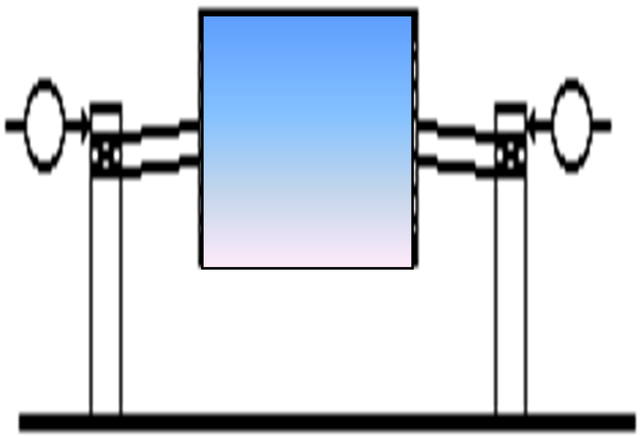
The spectrum shows high radial vibration at 2X and lower 1X with 180° phase difference across the coupling in the radial direction.

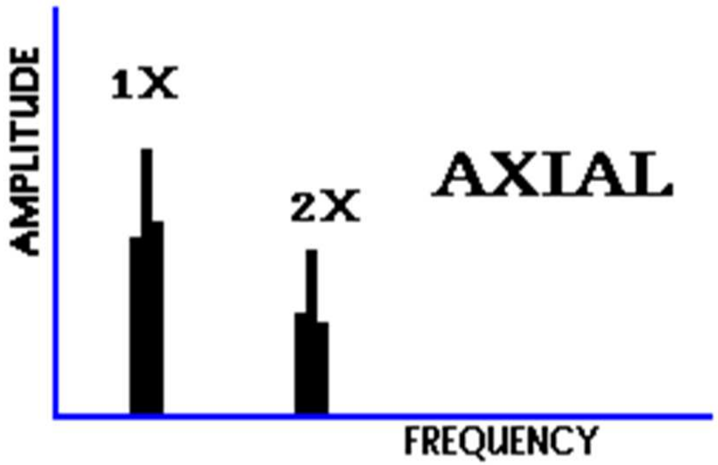
These signals may be also visible in the axial direction. in a lower amplitude and 180° phase difference across the coupling in the axial direction.





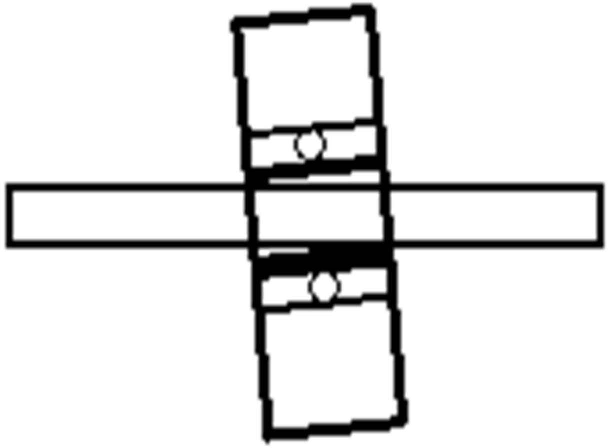
Bent Shaft

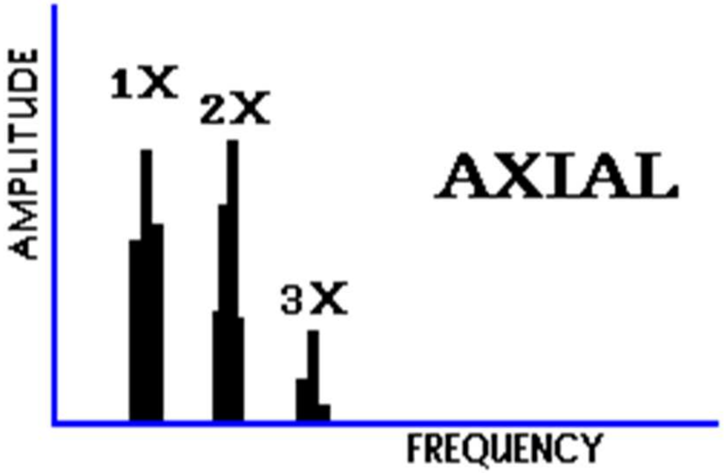




- Bent shaft problems cause high axial vibration
- 1X RPM dominant if bend is near shaft center
- 2X RPM dominant if bend is near shaft ends
- Phase difference in the axial direction will tend towards 180^0 difference

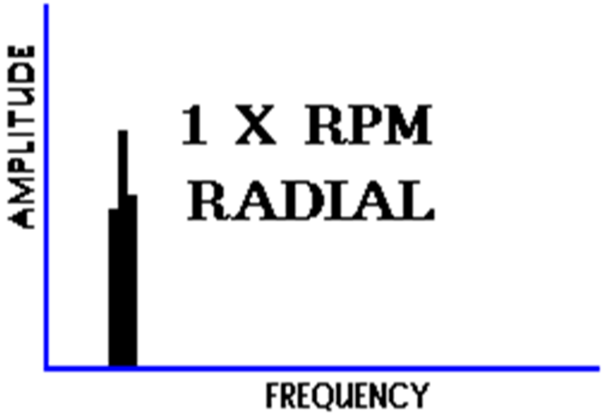
Misaligned Bearing

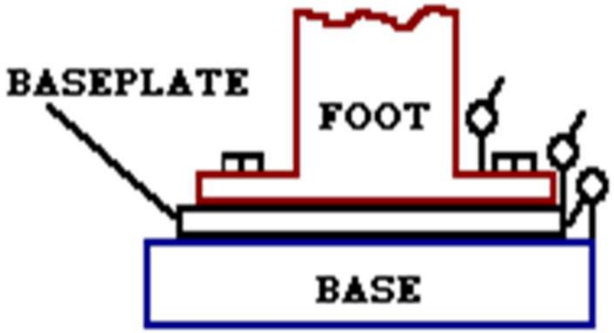




- Vibration symptoms similar to angular misalignment
- Attempts to realign coupling or balance the rotor will not alleviate the problem.
- Will cause a twisting motion with approximately 180^0 phase shift side to side or top to bottom

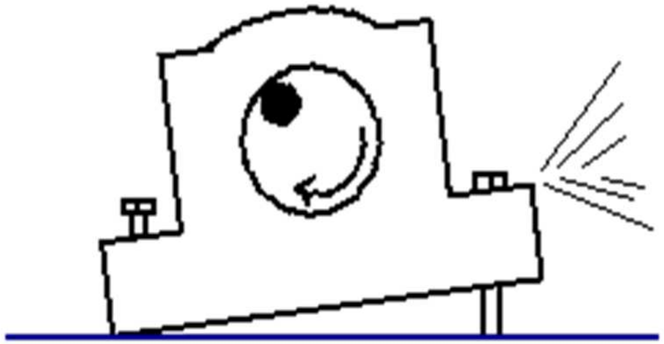
Looseness – Example 1

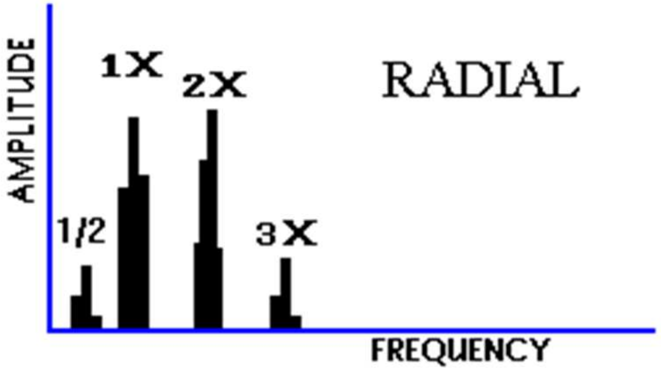




- Caused by structural looseness of machine feet
- Distortion of the base will cause “soft foot” problems
- Phase analysis will reveal approximately 180^0 phase shift in the vertical direction between the baseplate components of the machine

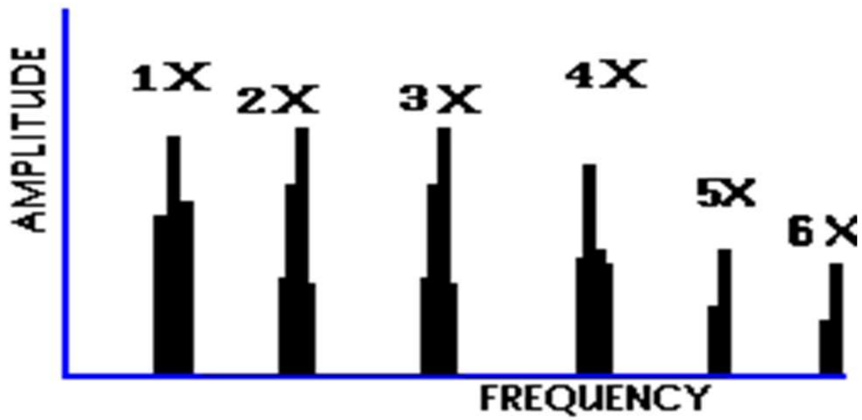
Looseness – Example 2





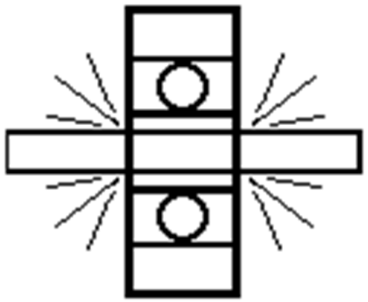
- Caused by loose pillowblock bolts
- Can cause 0.5, 1, 2 and 3X RPM
- Sometimes caused by cracked frame structure or bearing block

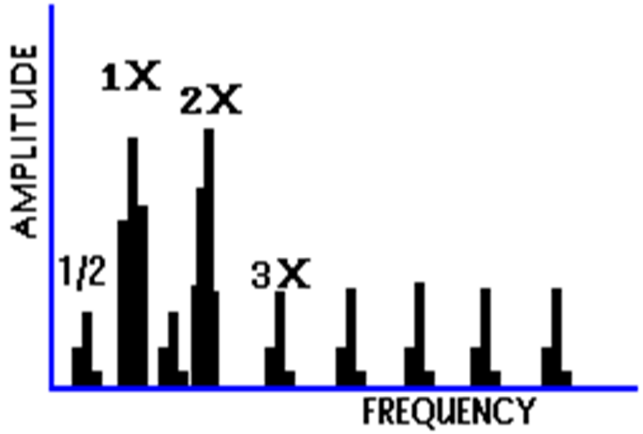
SLEEVE BEARING- WEAR / CLEARANCE



- Later stages of sleeve bearing wear will give a large family of harmonics of running speed
- A minor unbalance or misalignment will cause high amplitudes when excessive bearing clearances are present

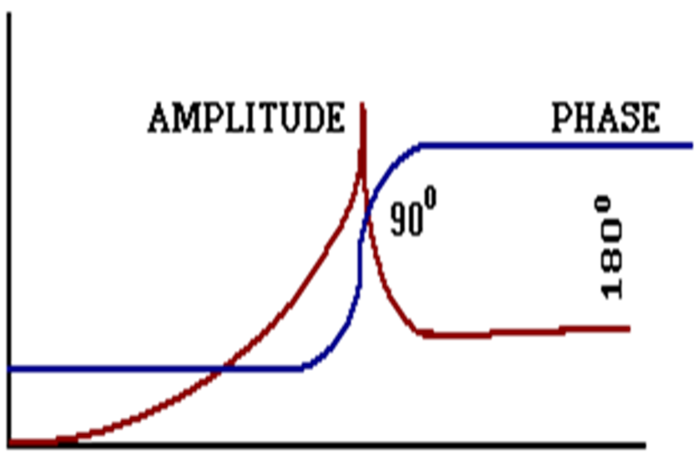
Looseness

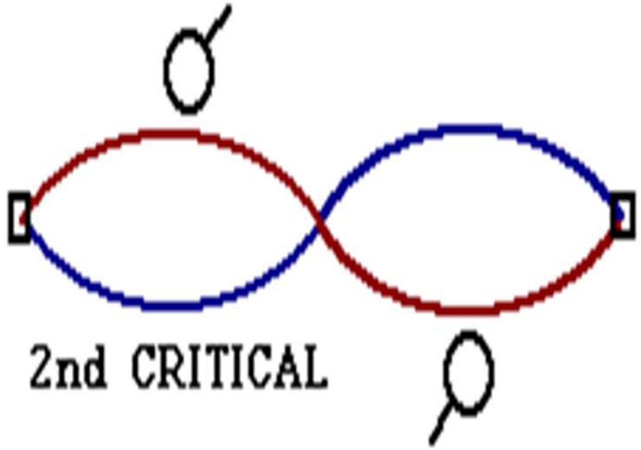




- Phase is often unstable
- Will have many harmonics
- Can be caused by a loose bearing liner, excessive bearing clearance or a loose impeller on a shaft

RESONANCE

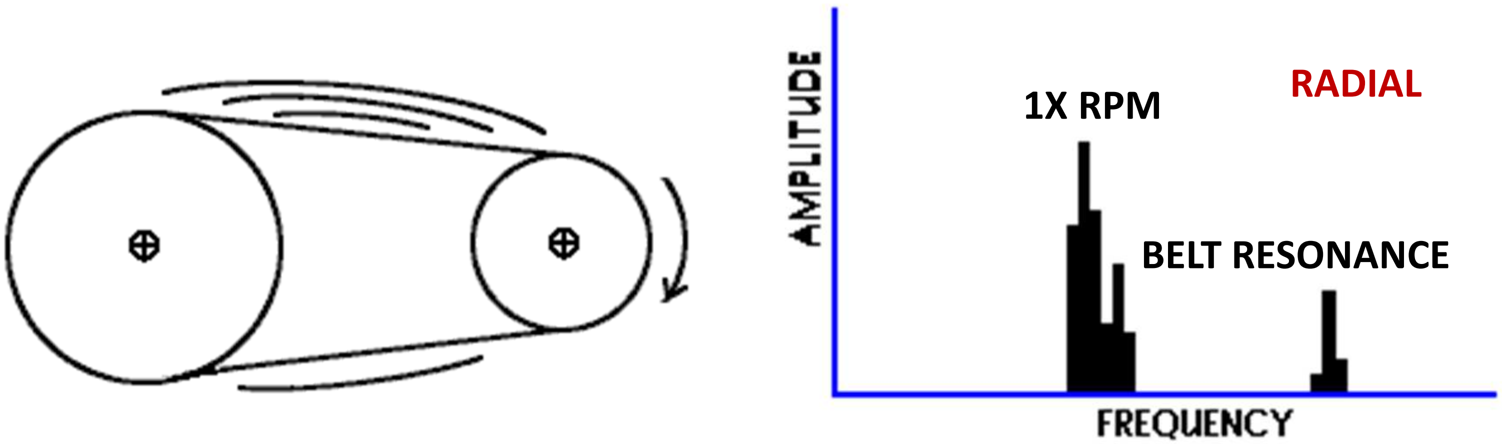




- Resonance occurs when the Forcing Frequency coincides with a Natural Frequency
- 180^0 phase change occurs when shaft speed passes through resonance
- High amplitudes of vibration will be present when a system is in resonance

Belt Problems

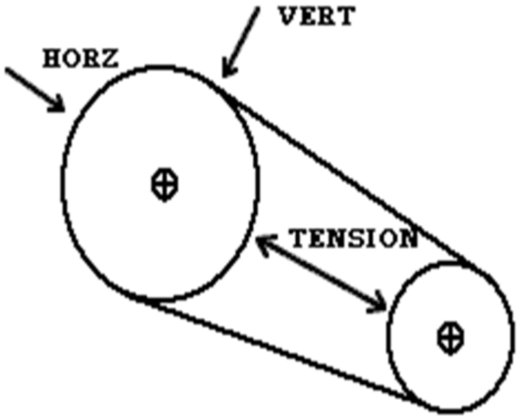
RESONANCE

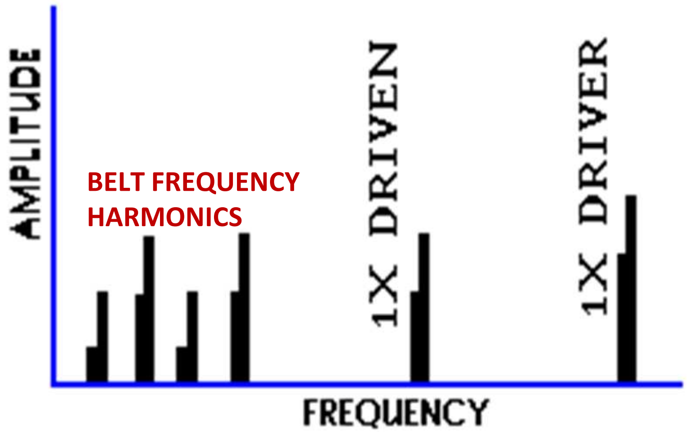


- High amplitudes can be present if the belt natural frequency coincides with driver or driven RPM
- Belt natural frequency can be changed by altering the belt tension

Belt Problems

WORN, LOOSE, OR MISMATCHED BELTS

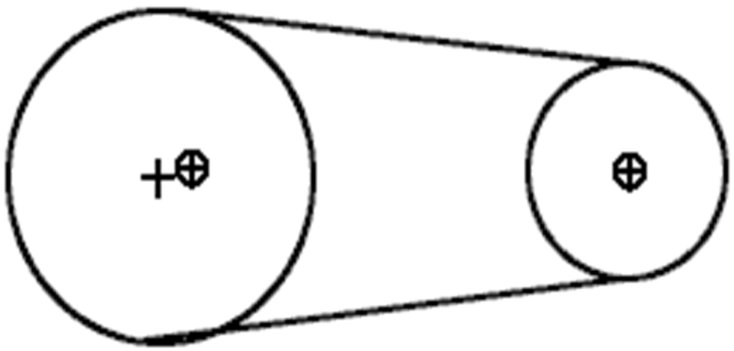


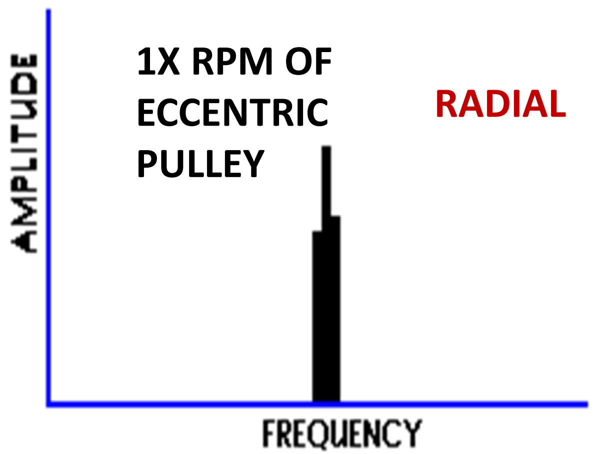


- Often 2X RPM is dominant
- Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM
- Wear or misalignment in timing belt drives will give high amplitudes at the timing belt frequency
- Belt frequencies are below the RPM of either the driver or the driven

Belt Problems

ECCENTRIC PULLEYS

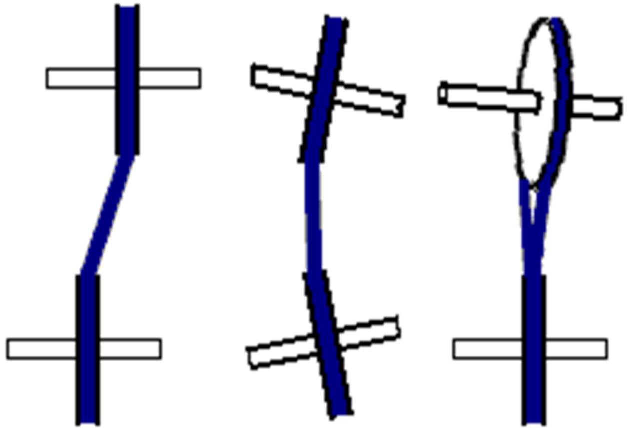


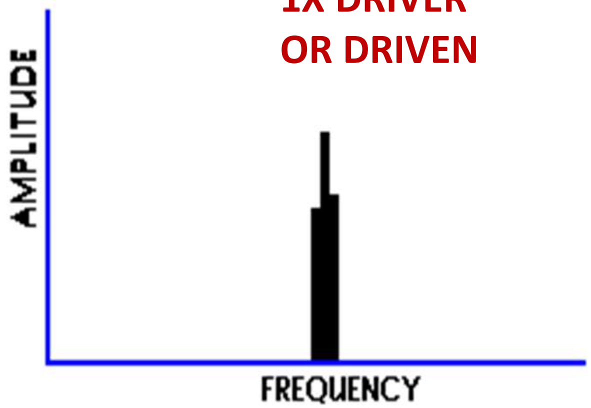


- Eccentric or unbalanced pulleys will give a high 1X RPM of the pulley
- The amplitude will be highest in line with the belts
- Beware of trying to balance eccentric pulleys

Belt Problems

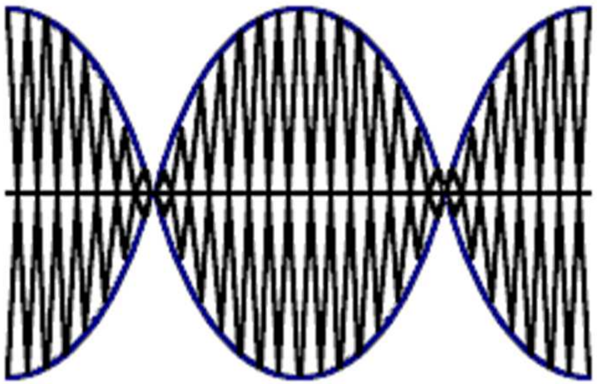
BELT/PULLEY MISALIGNMENT

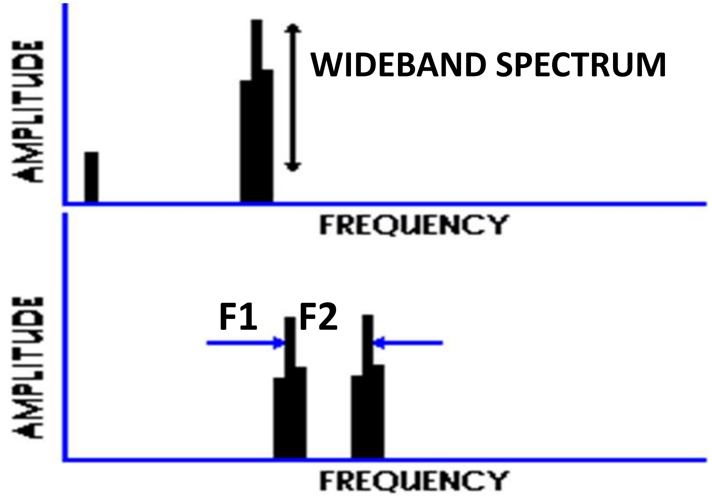




- Pulley misalignment will produce high axial vibration at 1X RPM
- Often the highest amplitude on the motor will be at the fan RPM

Beat Vibration

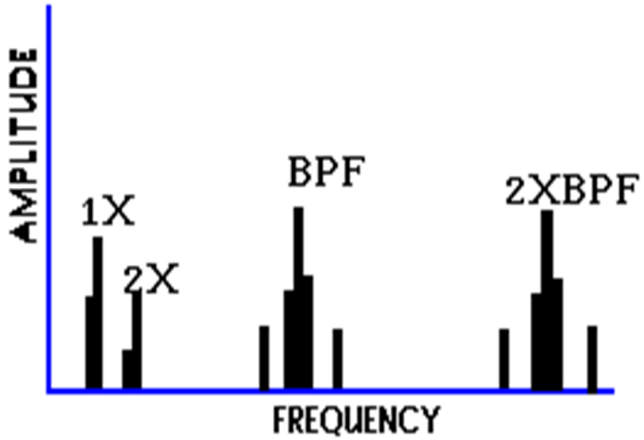




- A beat is the result of two closely spaced frequencies going into and out of phase
- The wideband spectrum will show one peak pulsating up and down
- The difference between the peaks is the beat frequency which itself will be present in the wideband spectrum

Hydraulic & Aerodynamic Forces

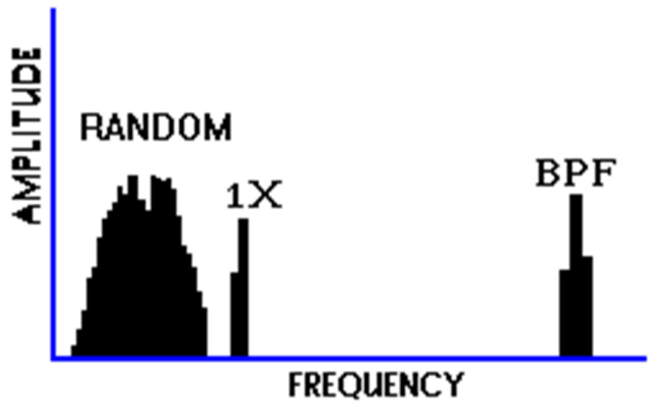
BPF = BLADE PASS FREQUENCY



- 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

Hydraulic & Aerodynamic Forces

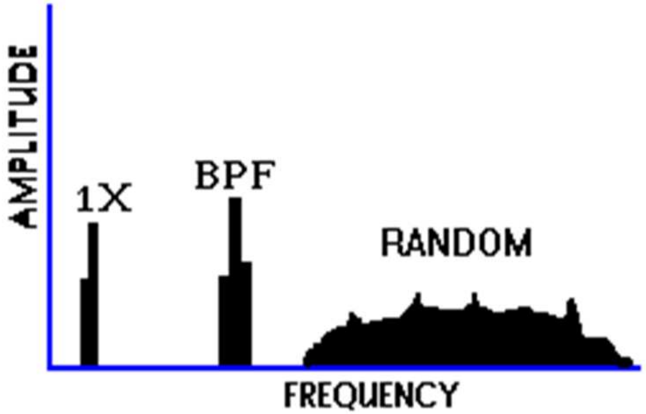
FLOW TURBULENCE



- Flow turbulence often occurs in blowers due to variations in pressure or velocity of air in ducts
- Random low frequency vibration will be generated, possibly in the 50 - 2000 CPM range

Hydraulic & Aerodynamic Forces

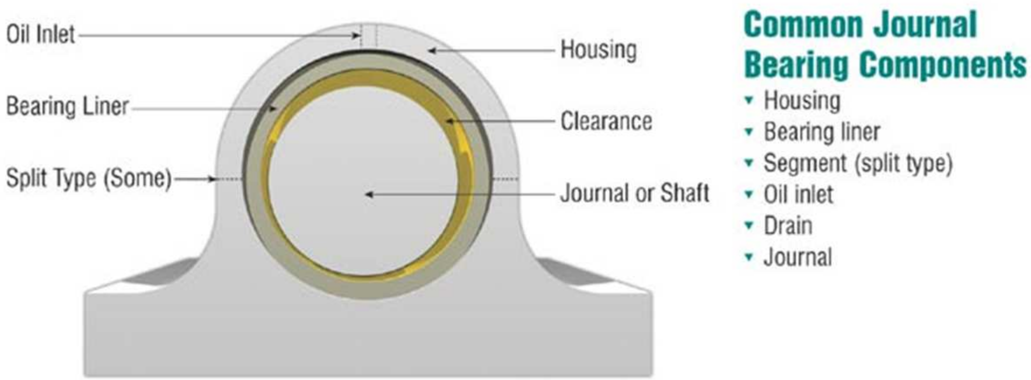
CAVITATION



- Cavitation will generate random, high frequency broadband energy superimposed with BPF harmonics
- Normally indicates inadequate suction pressure
- Erosion of impeller vanes and pump casings may occur if left unchecked
- Sounds like gravel passing through pump

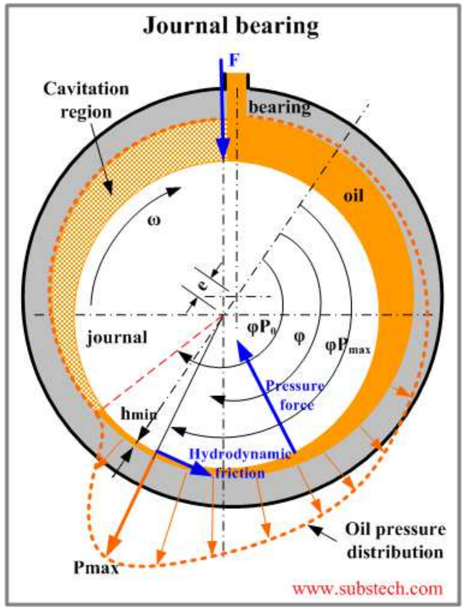
Fluid Bearings

- Fluid bearings are bearings which solely support the bearing's loads on a thin layer of liquid or gas.
- Frequently used in high load, high speed or high precision applications where rolling-element bearings have short life or high noise and vibration. Also used increasingly to reduce cost.
- Bearing rotation sucks the fluid on to the inner surface of the bearing, forming a lubricating wedge under or around the shaft.



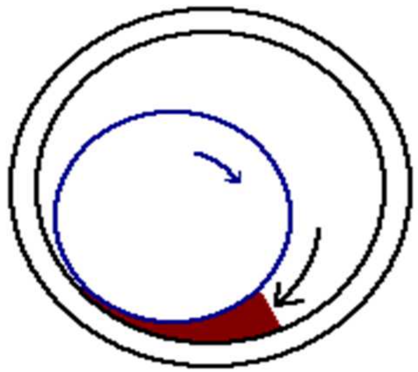
Fluid Bearings

- **Lubricating-film instability (OIL WHIRL)** is dominant failure mode for fluid bearings.
- Typically caused by eccentric rotation of shaft resulting from imbalance, misalignment or other machine-related problem or improperly designed bearing.
- Rotor is prevented from creating a stable lubricating wedge on which to ride. Oil film can drive the shaft ahead of it in whirling path with the bearing clearance.

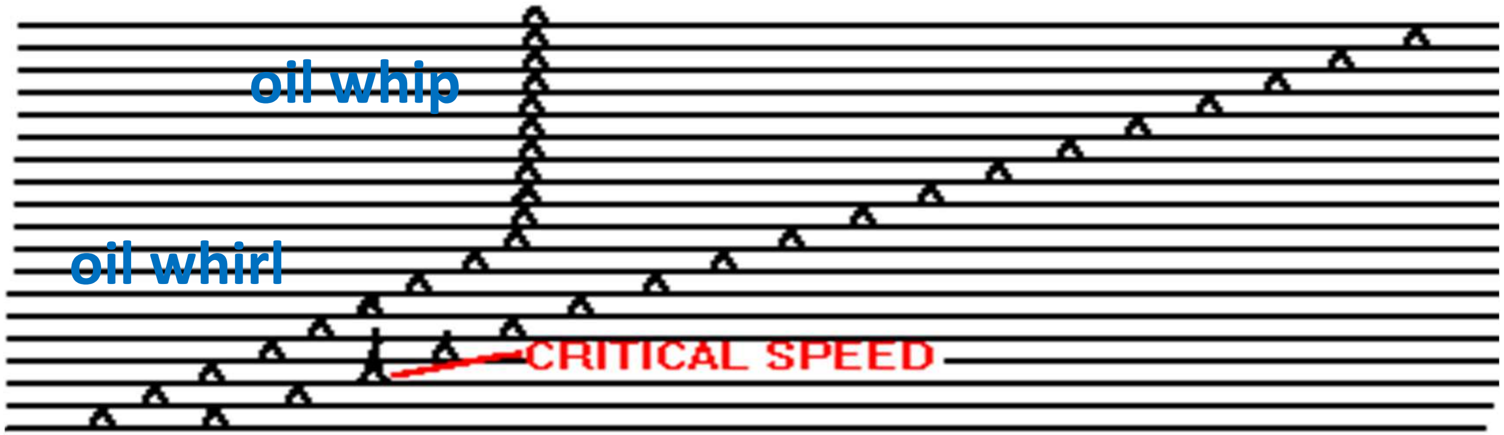


Fluid Bearings

- **Oil whirl** is easy to recognise by its unusual vibration frequency of between 40% and 48% of shaft speed.
- Vibration amplitudes are sometimes severe
- Whirl is inherently unstable, since it increases centrifugal forces therefore increasing whirl forces
- **Oil whip** can occur when oil whirl frequency coincides with and becomes locked to a natural frequency of system.
- Left uncorrected, oil whip may cause destructive vibration resulting in catastrophic failure - often in a relatively short period of time.



Oil Whip Instability



- Oil whip may occur if a machine is operated at 2X the rotor critical frequency.
- When the rotor drives up to 2X critical, whirl is close to critical and excessive vibration will stop the oil film from supporting the shaft.
- Whirl speed will lock onto rotor critical. If the speed is increased the whip frequency will not increase.

Failure Mode Analysis - Conclusion

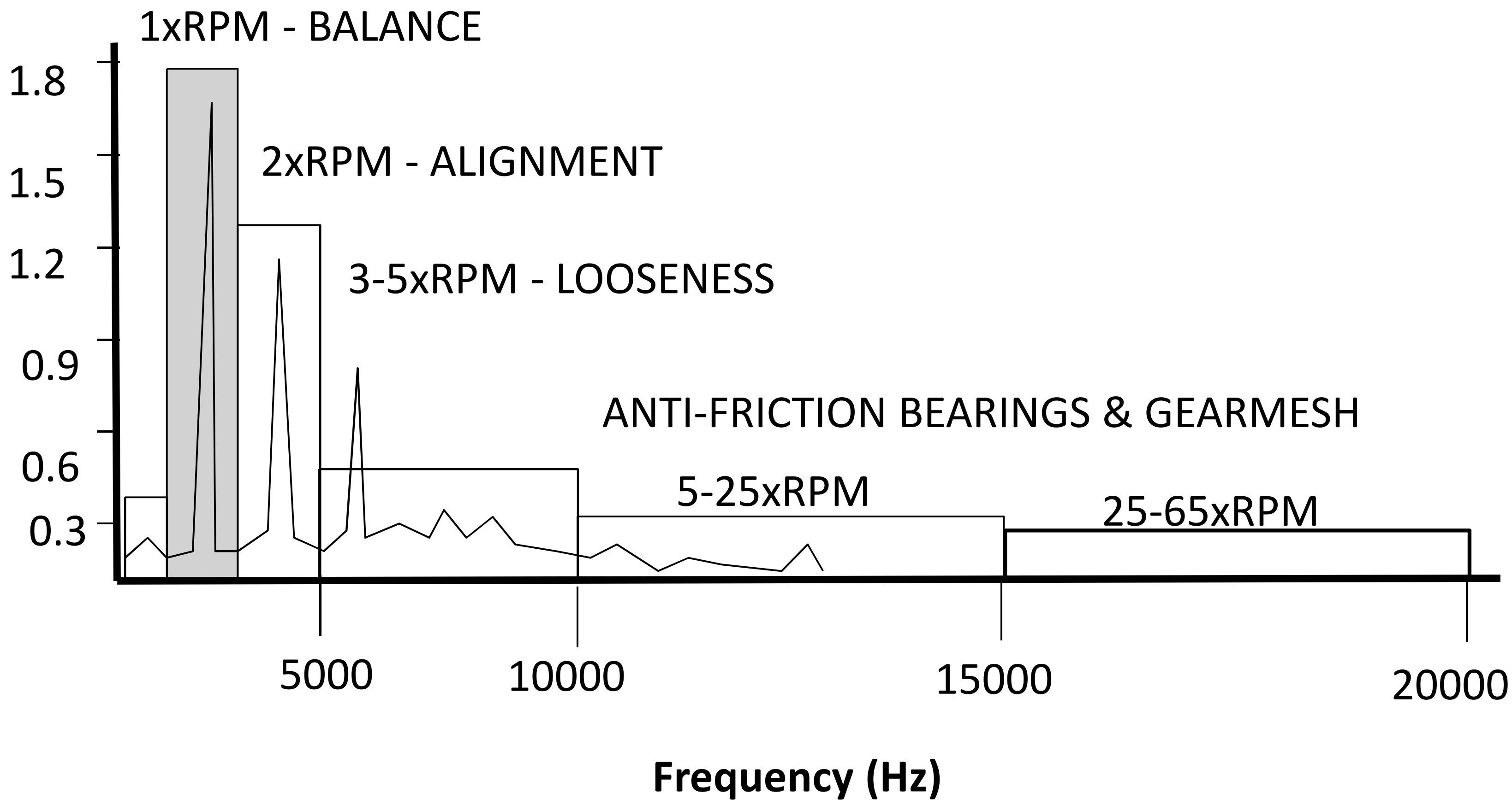
Predictive maintenance using vibration analysis is based on the following:

- All common machinery problems and failure modes have distinct vibration frequency components that can be isolated and identified.
- Frequency-domain signature is generally used because it contains **discrete peaks**, each representing specific vibration source.
- There is a cause for each frequency component.
- When the machine signature is compared over time, it will repeat until some event changes the vibration pattern.

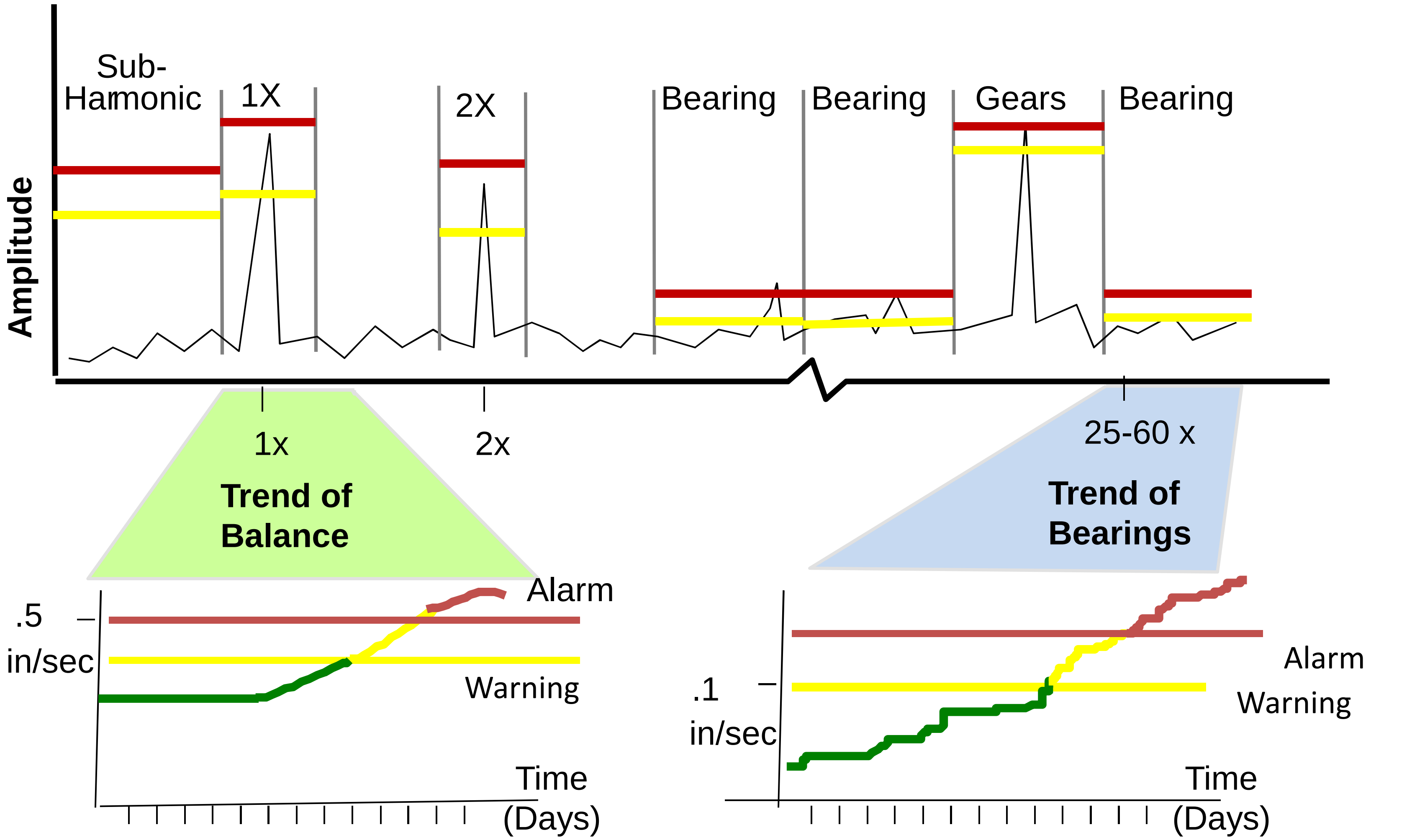
Failure Mode Analysis - Conclusion

- Several failure-mode charts available but 60 to 70% of the total vibration energy is contained in the frequency component corresponding to the running speed of the machine.
- Many common causes of failure in machinery components can be identified by understanding relationship to running speed of shaft.
- Common machine-train failure modes include critical speeds, imbalance, mechanical looseness, misalignment, modulations and process instability.

Predefined Spectrum Analysis Bands



Frequency Band Alarming and Trending



Mechanical Condition Monitoring
Rolling Element Bearing
Lecture 3

Outline

- Bearing Basics
- Bearing types
- Bearing failure causes
- Bearing life expectancy
- Vibration Profile
- Demodulation
- Examples

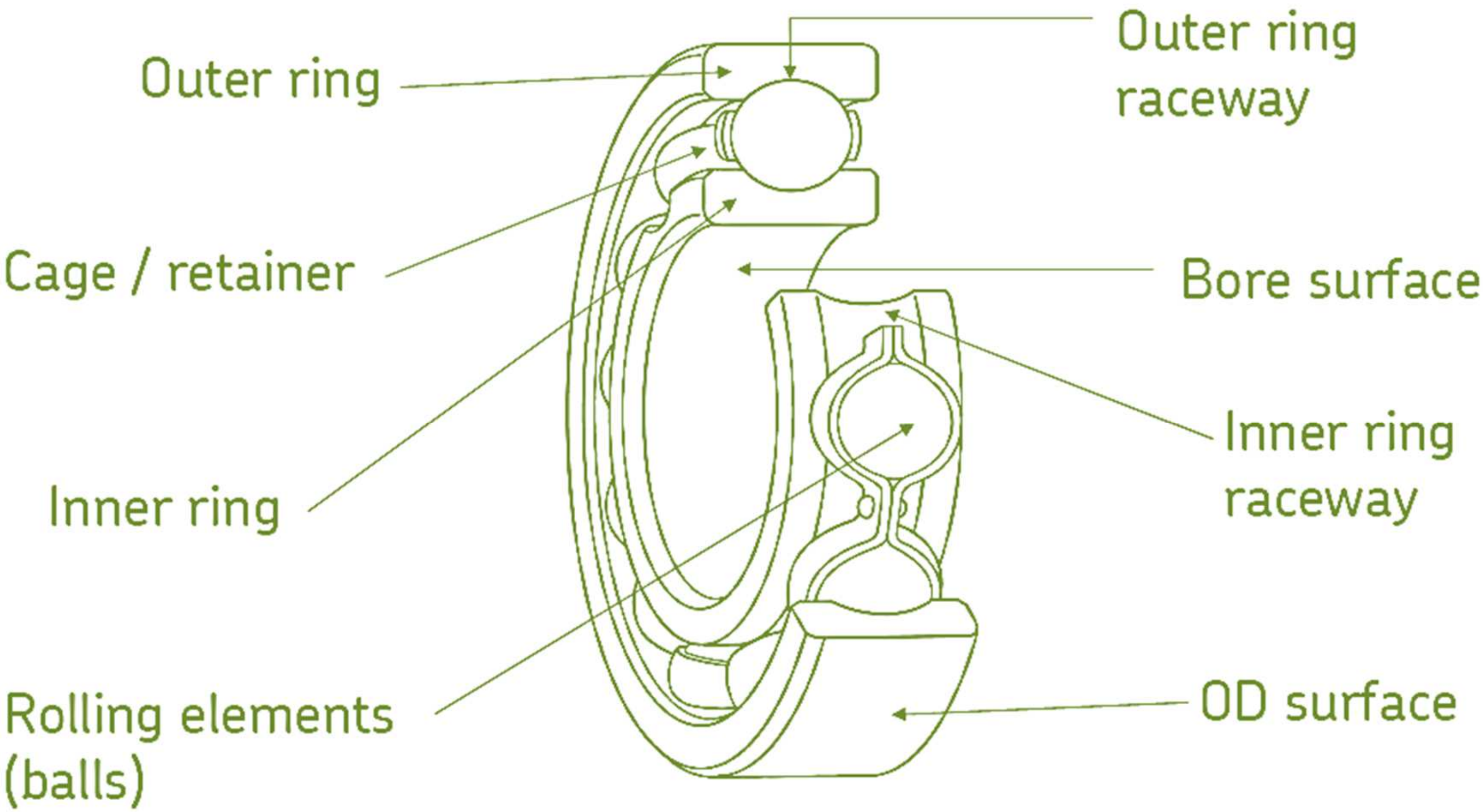
Purpose of a bearing

- To provide low friction rotation of machine parts
- To support and locate rotating equipment
- Resistance to motion which occurs when one object slides or rubs against another object.
- If not controlled, friction will result in:
  - Heat generation
  - Increased wear
  - Increased noise
  - Loss of power

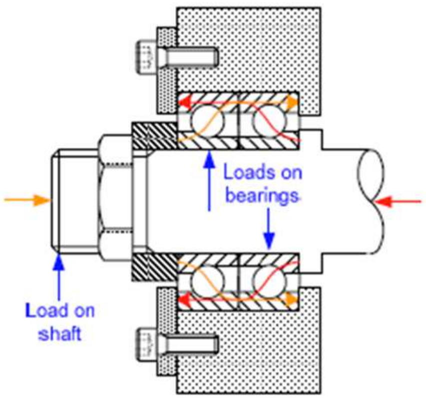
Roles of a bearing

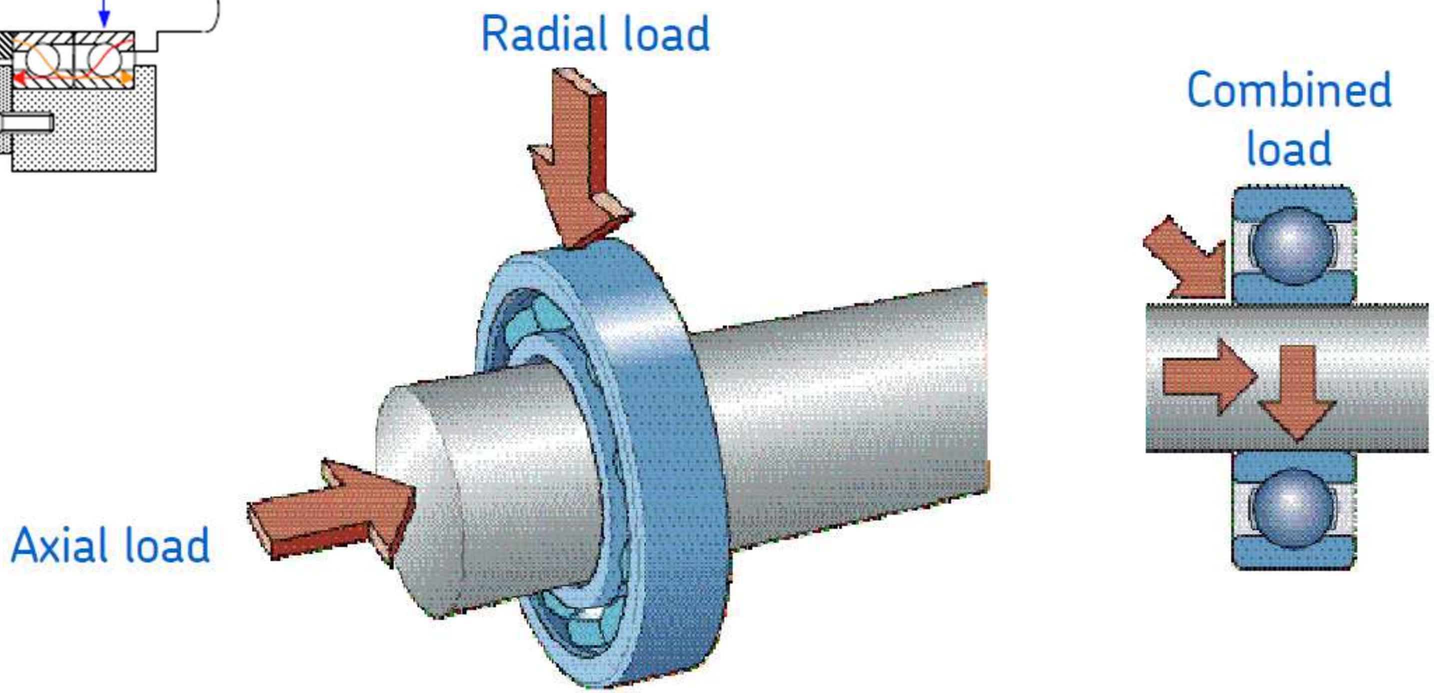
- Reduce friction
- Transmit loads
- Support the shaft
- Locate the shaft

Bearing components

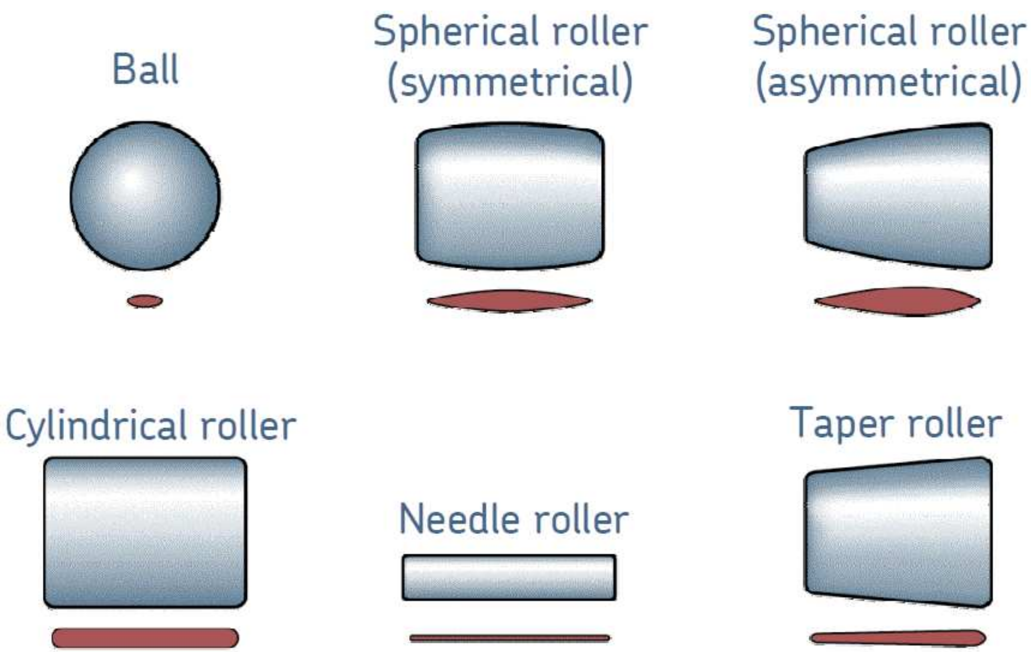


Types of static bearing loads

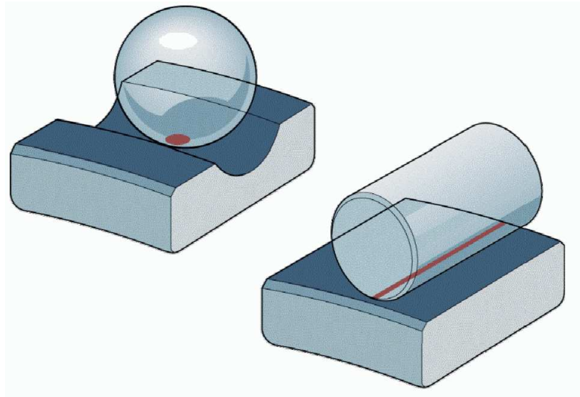




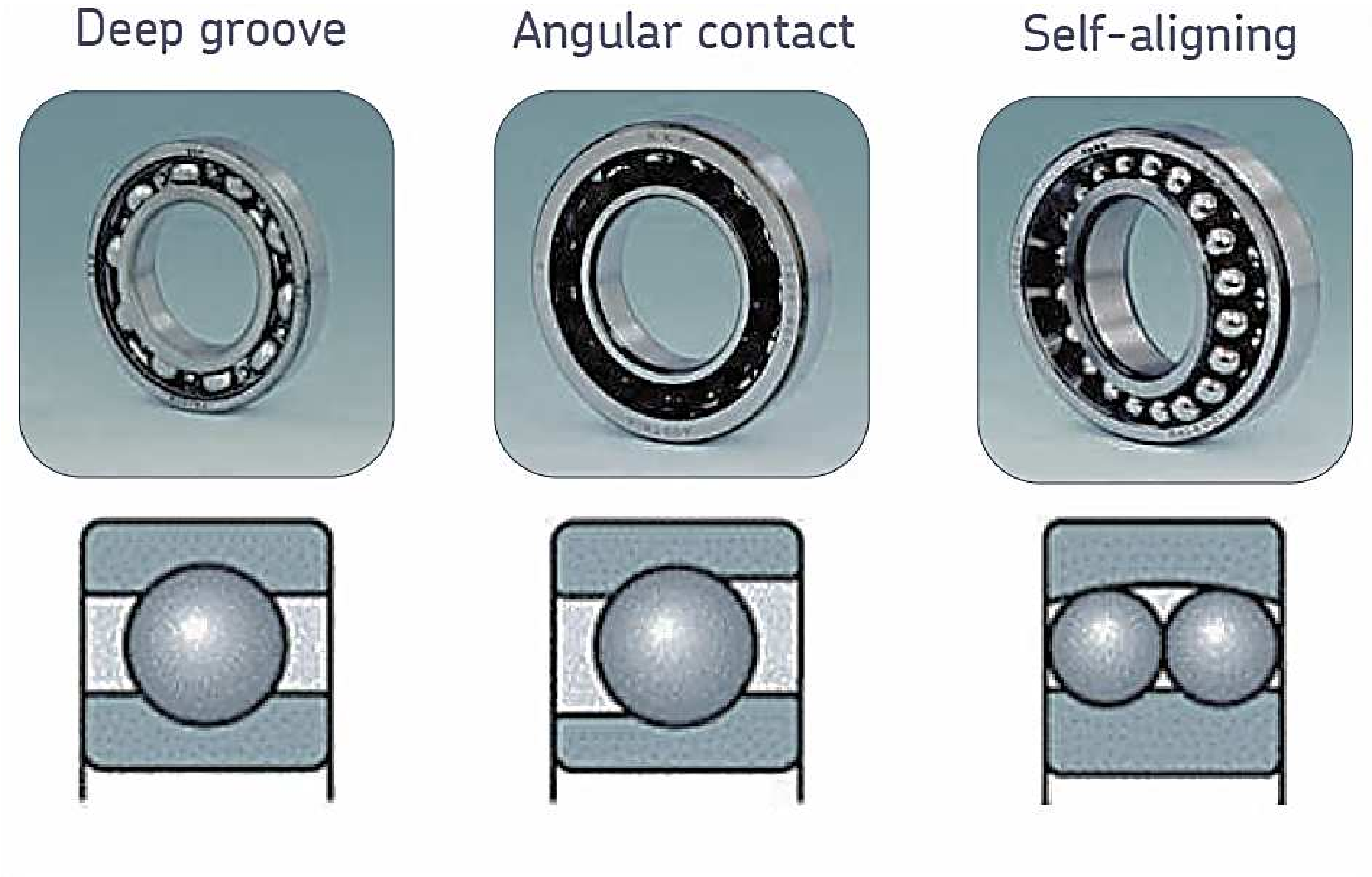
Types of rolling elements



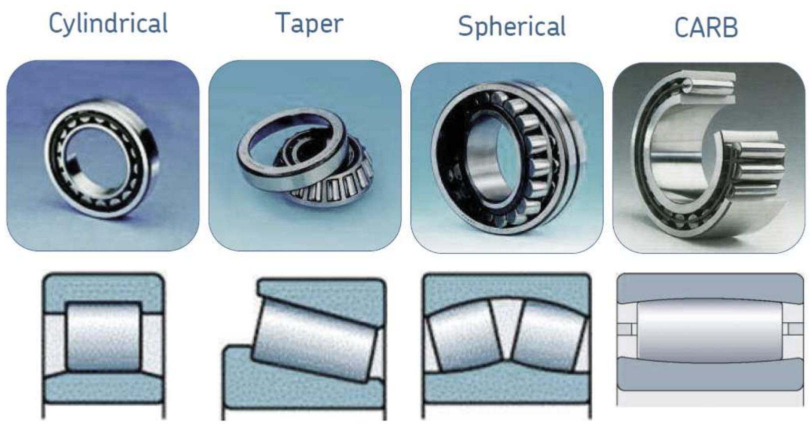
Point and line contact

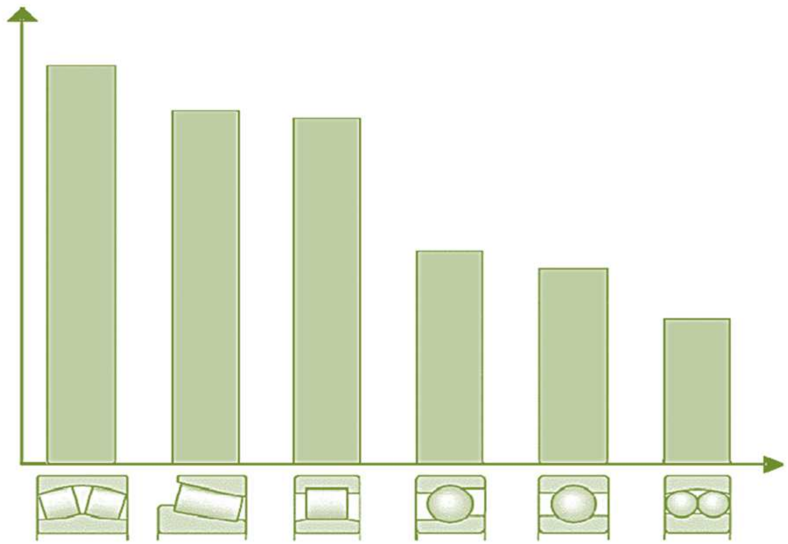


Types of ball bearings



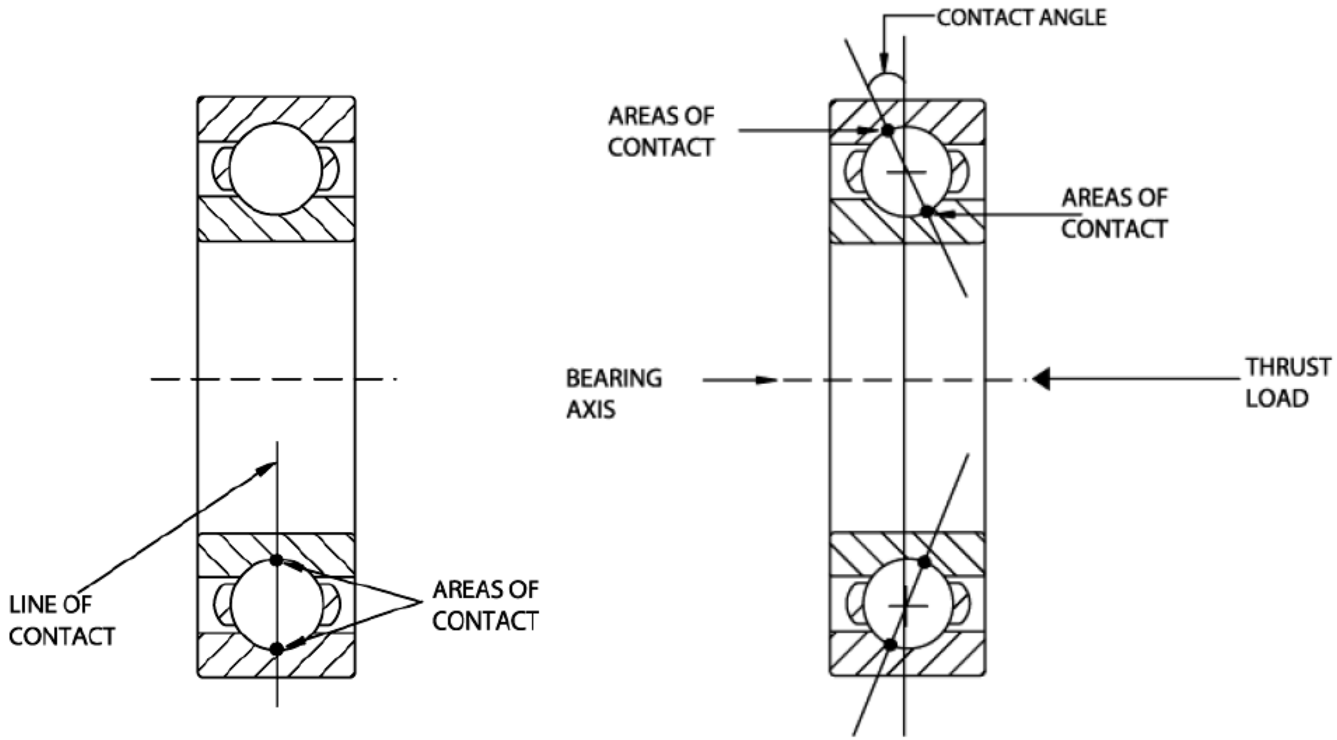
Types of roller bearings





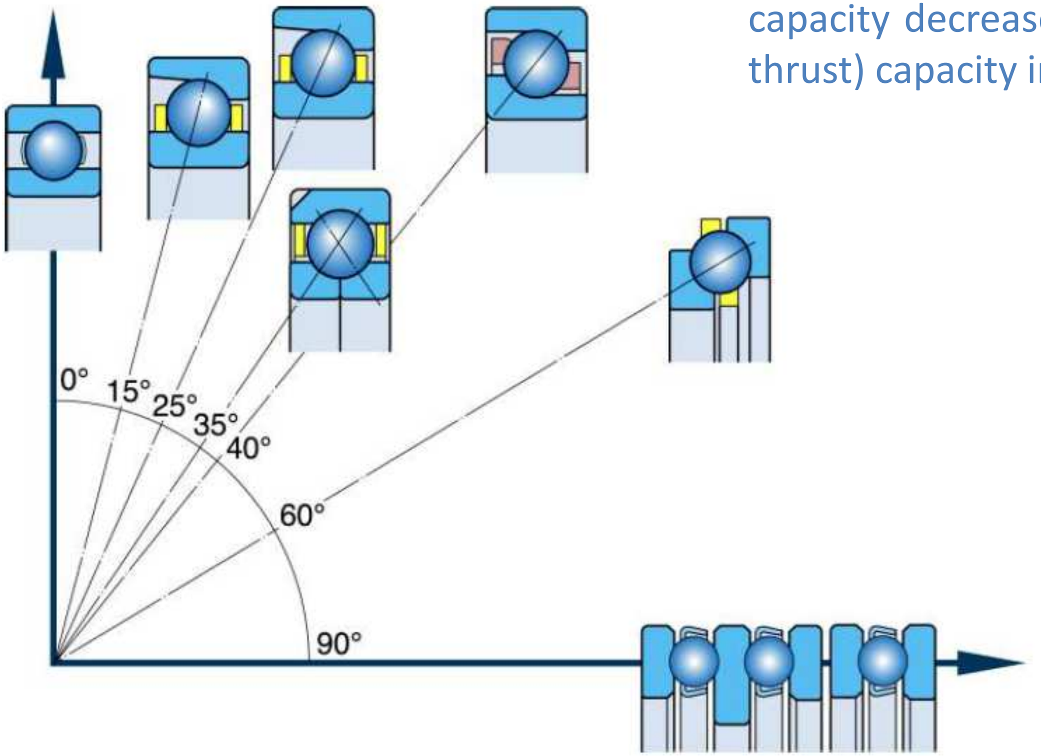
Load carrying capacity is expressed as the 'basic dynamic load rating' or 'C' in catalogues

Contact angle



- The lower the contact angle, the higher the radial load capacity
- The higher the contact angle, the higher the thrust load capacity

Bearings and contact angles



As the contact angle increases, radial load capacity decreases; while the axial load (i.e. thrust) capacity increases.

Bearing life expectancy

Based upon five assumptions:

- The bearing is defect free.
- The correct bearing type and size is selected for the application.
- Dimensions of the bearing mating parts are correct.
- The bearing will be mounted without damage.
- Good lubrication in the correct quantity will always be available to the bearing.

Causes of bearings' failure

Four predominant causes of premature bearing failure:

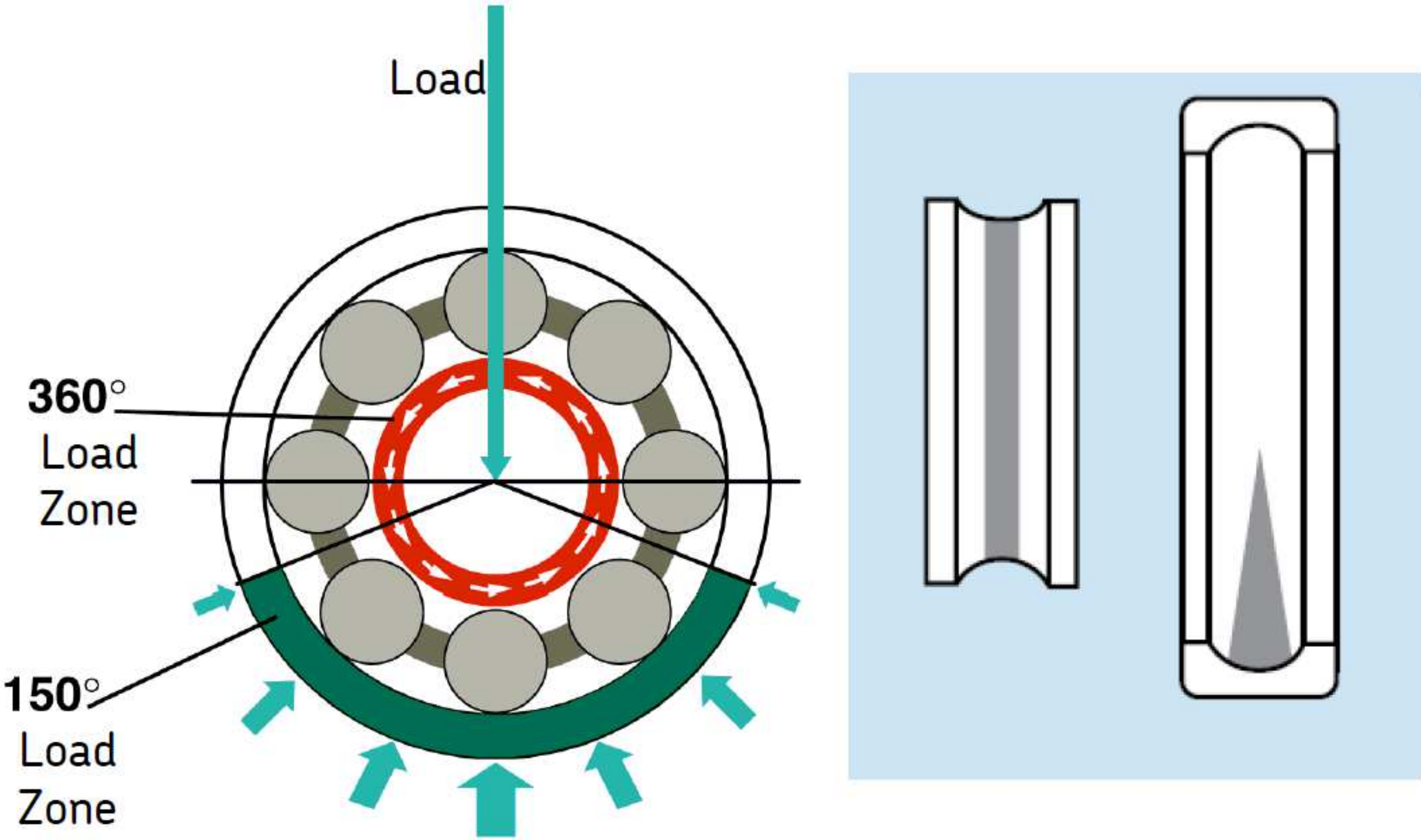
- Improper loading
- 16% Poor Installation
- 36% Poor Lubrication
- 14% Contamination
- 34% Fatigue



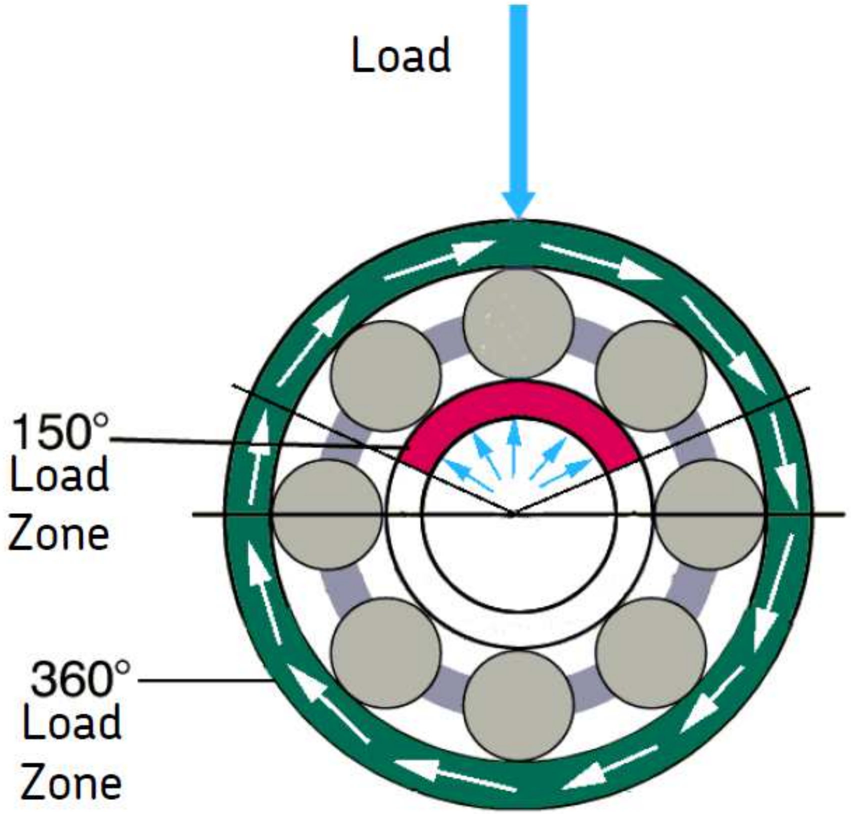

Operational damage mode causes

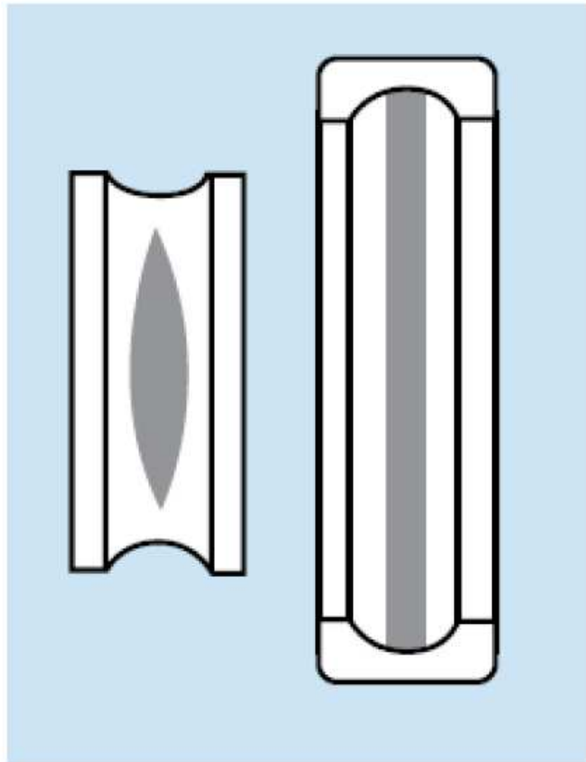
- Static vibration
- Operational misalignment
- Ineffective sealing
- Ineffective or inadequate lubrication
- Passage of electric current through the bearing
- Excessive loading

Loading patterns: Inner Ring Rotation

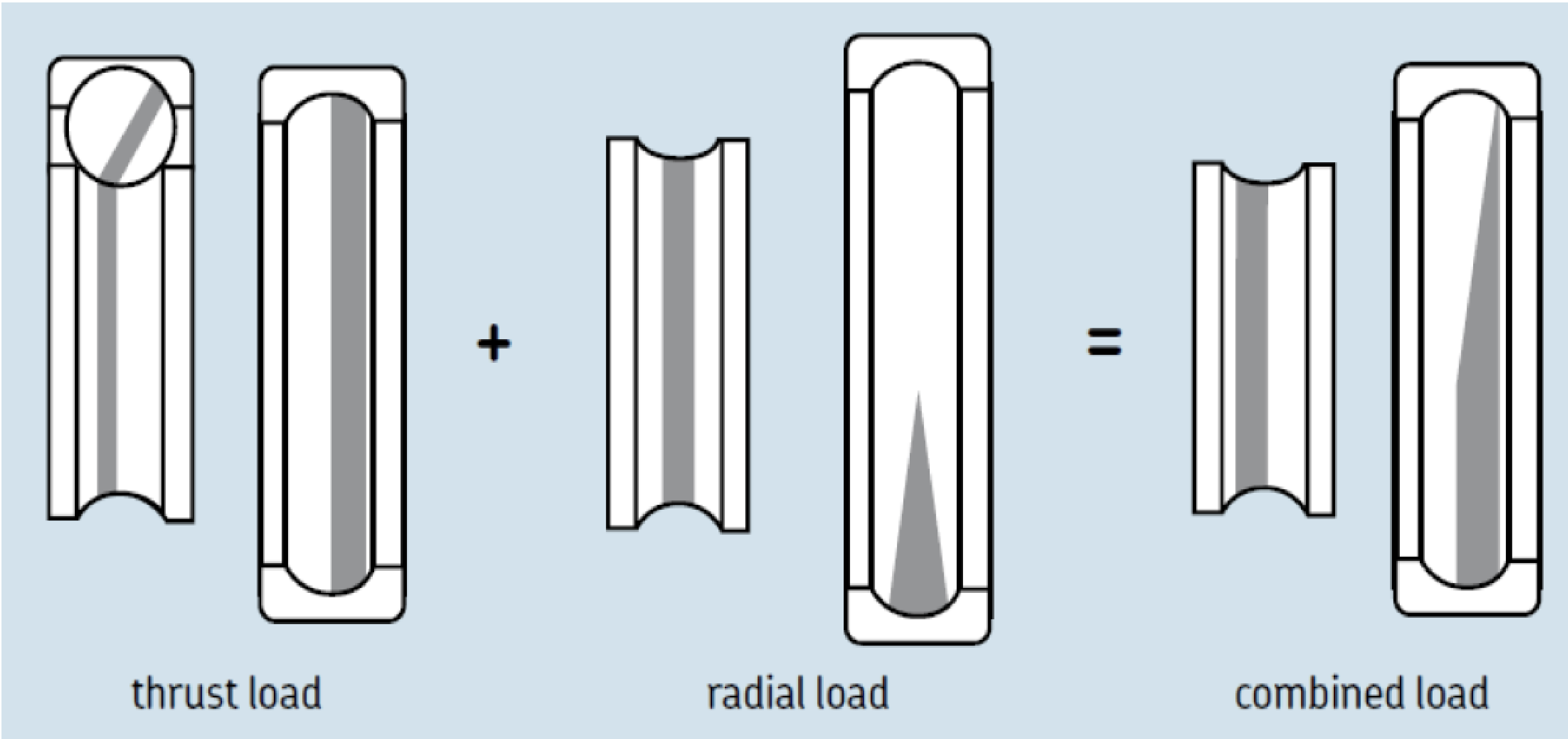


Loading patterns: Outer Ring Rotation

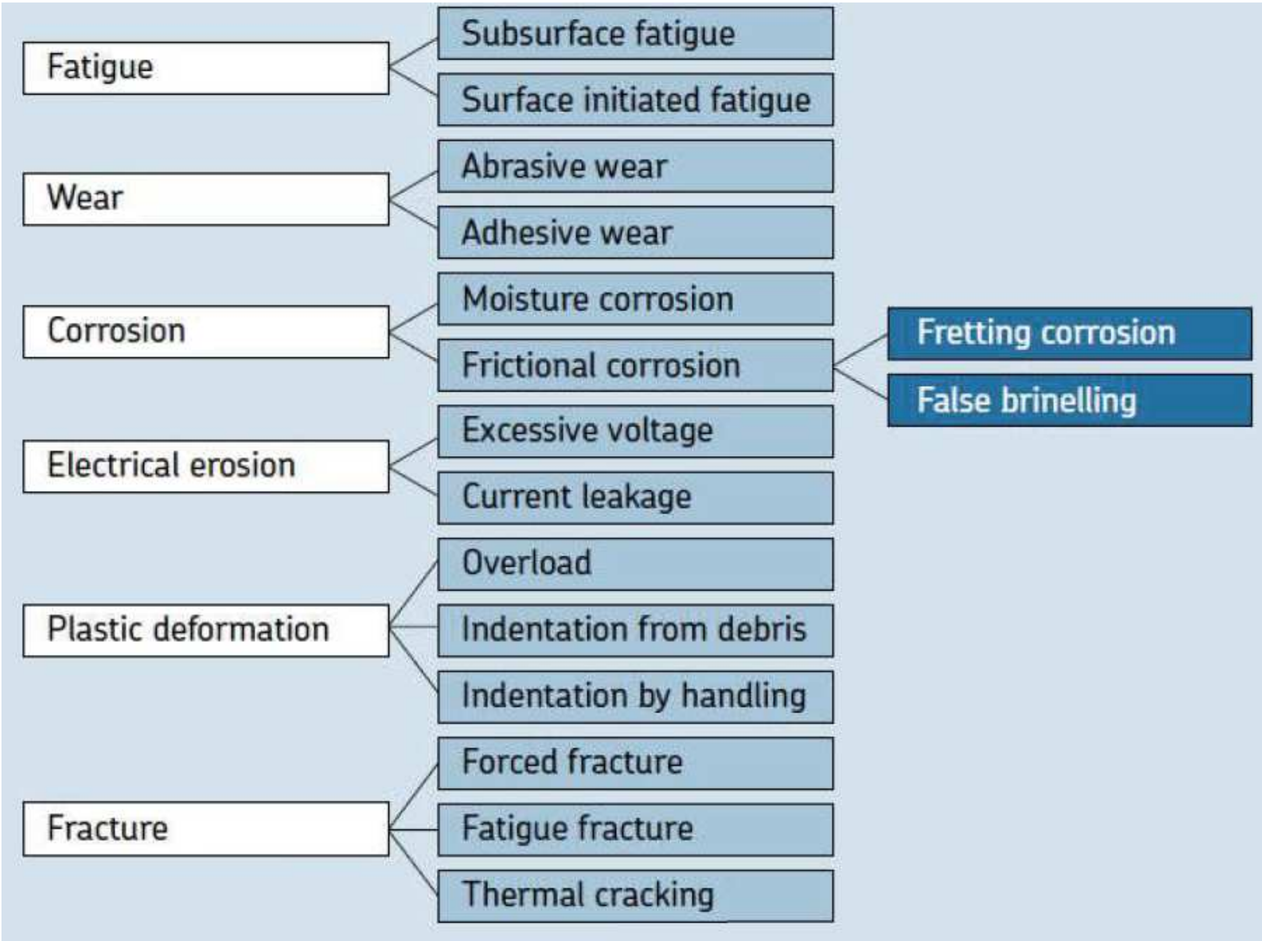




Load Zones:
Thrust load, Radial load, & combined loads

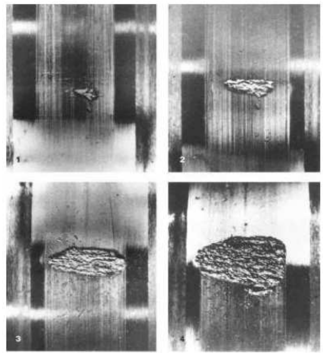


Bearing damage analysis



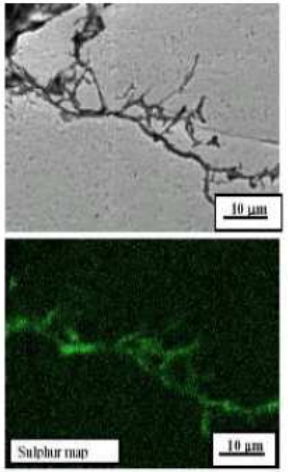
Fatigue: subsurface fatigue

- Repeated stress changes
- Material structural changes
- Micro-cracks under the surface
- Crack propagation
- Flaking, spalling, and peeling



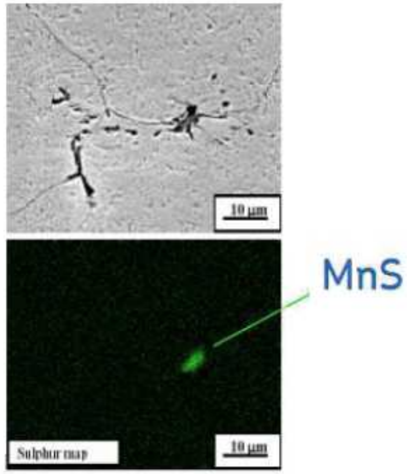
Fatigue: subsurface fatigue

Hydrogen Embrittlement



Surface cracks
with sulphur (green) penetration
—
Sulphide Stress Cracking (SSC)

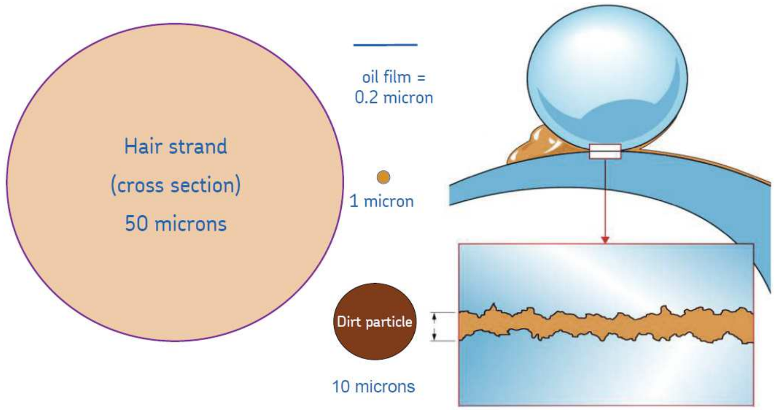
H_2S + liquid water

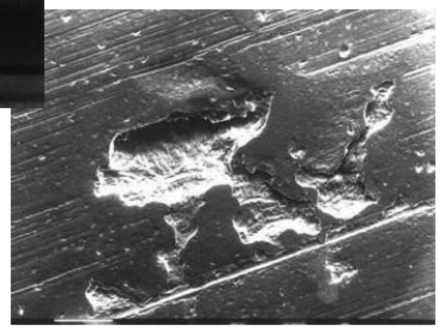


Sub-surface cracks
in hydrogen embrittled volume
—
Stress Corrosion Cracking (SCC)
& Hydrogen Induced Cracking (HIC)

Fatigue: surface initiated fatigue

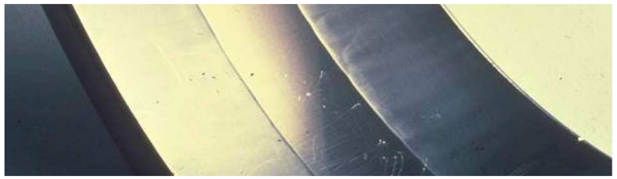
- Surface distress
- Reduced lubrication regi
- Sliding motion
- Burnishing, glazing
- Asperity micro-cracks
- Asperity micro-spalls



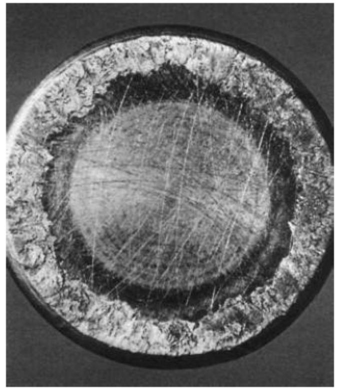


Wear: abrasive wear

- Progressive removal of material
- Ingress of dirt particles
- Accelerating process
- Dull surfaces



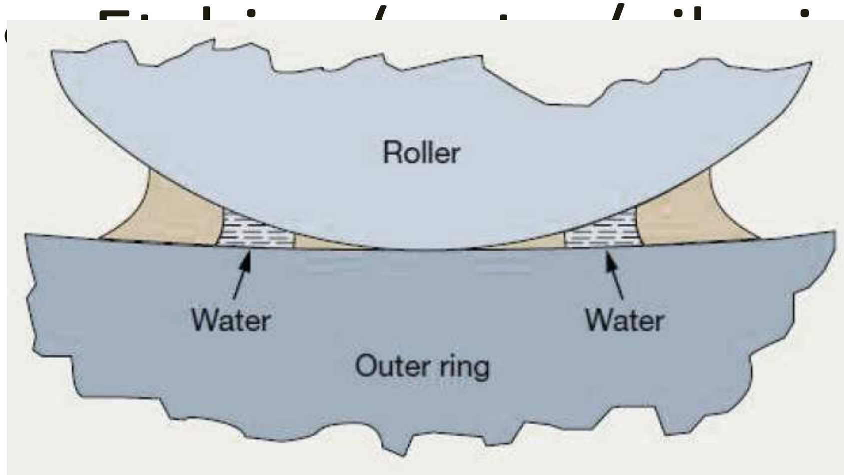
Wear: adhesive wear

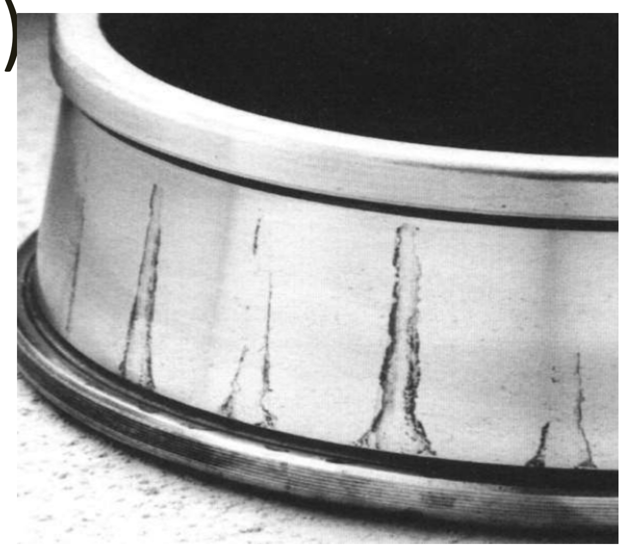


- Low loads
- Accelerations
- Smearing / skidding / galling
- Material transfer / friction heat
- Tempering / re-hardening
- With stress concentrations and cracking or flaking

Corrosion: moisture corrosion

- Oxidation / rust
- Chemical reaction
- Corrosion pits / flaking
- Etching (water / oil mixture)



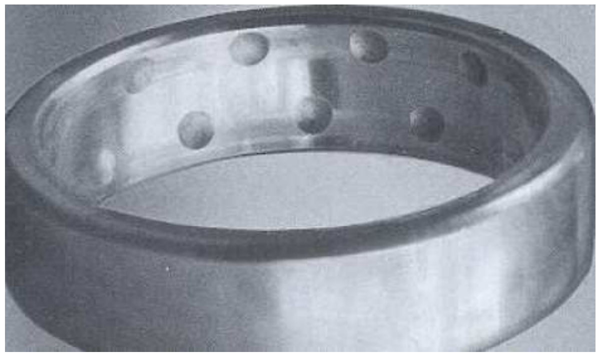


Corrosion: frictional corrosion - fretting

- Micro-movement between mating surfaces
- Oxidation of asperities
- Powdery rust / loss of material
- Occurs in fit interfaces
- Transmitting loads

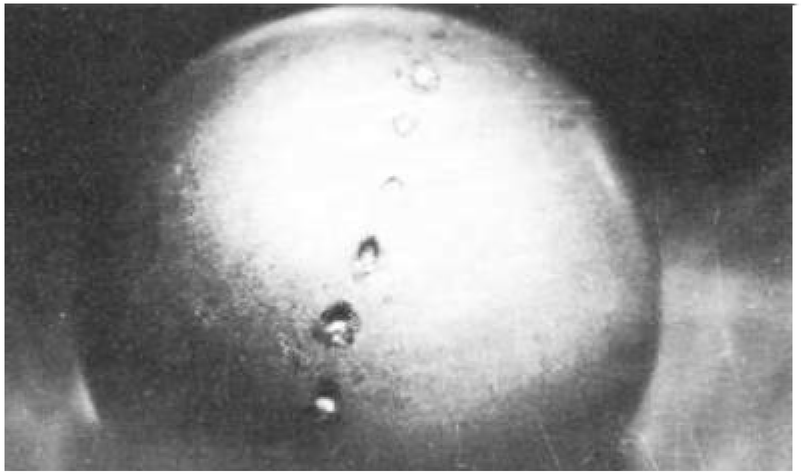
Corrosion: frictional corrosion - false brinelling

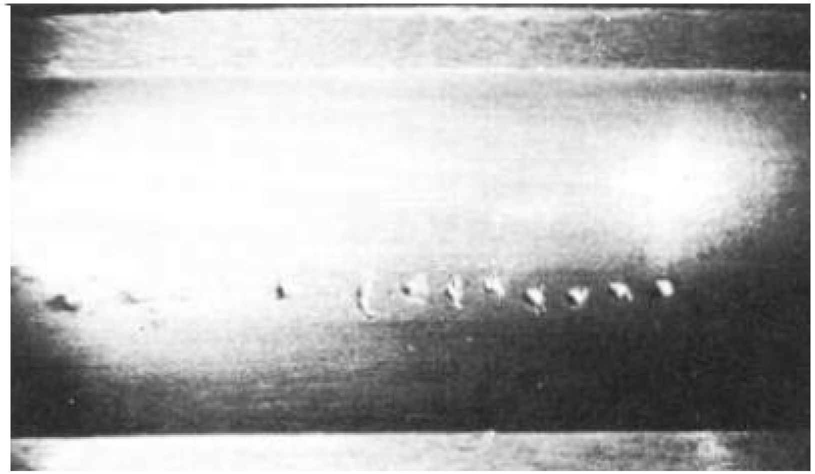
- Rolling element / raceway
- Micro movements / elastic deformations
- Vibrations
- Corrosion / wear / shiny / red depressions
- Stationary: rolling element pitch
- Rotating: parallel flutes



Electrical erosion: excessive voltage

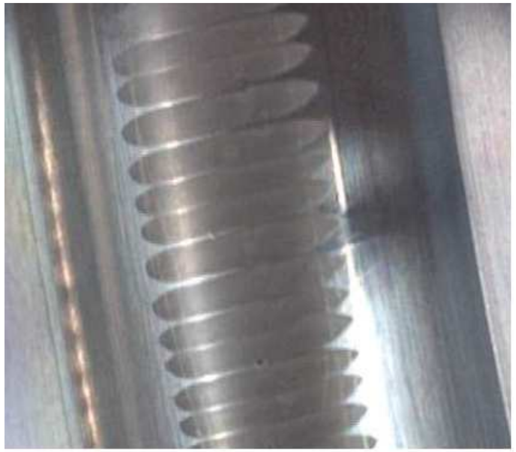
- High current / sparking
- Localized heating in very short Interval / melting / welding
- Craters up to 100 µm

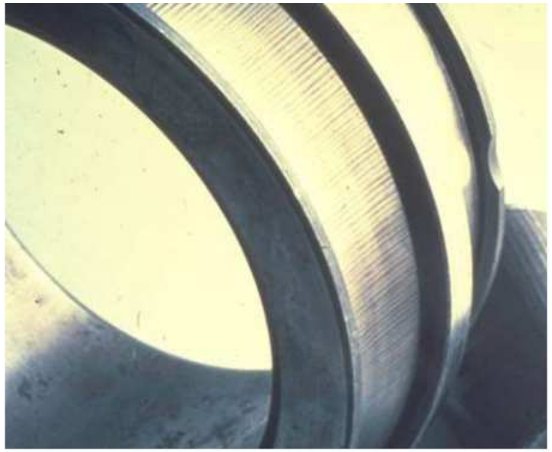




Electrical erosion: current leakage

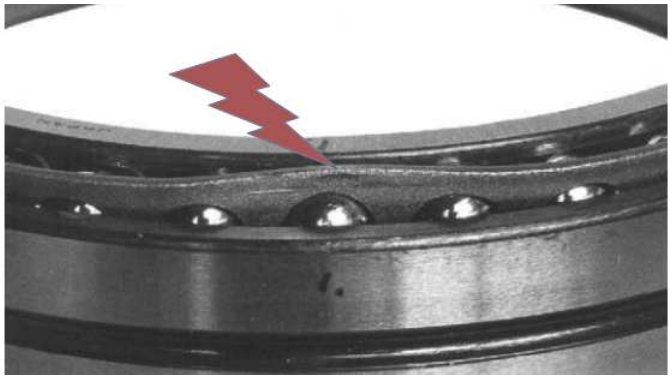
- Low current intensity
- Shallow craters closely positioned
- Development of flutes on raceways & rollers, parallel to rolling axis
- Dark gray discoloration





Plastic deformation: overload

- Static or shock loads
- Plastic deformations
- Depressions in rolling element distance
- Handling damages

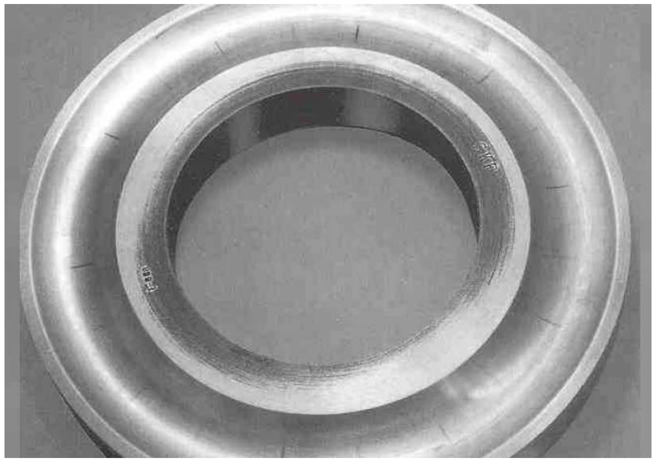


Plastic deformation: indentation from debris

- Localized overloading
- Over-rolling of particles dents
- Soft / hardened steel / hard mineral

Plastic deformation: indentation from handling

- Localized overloading
- Over-rolling of particles dents
- Soft / hardened steel / hard mineral

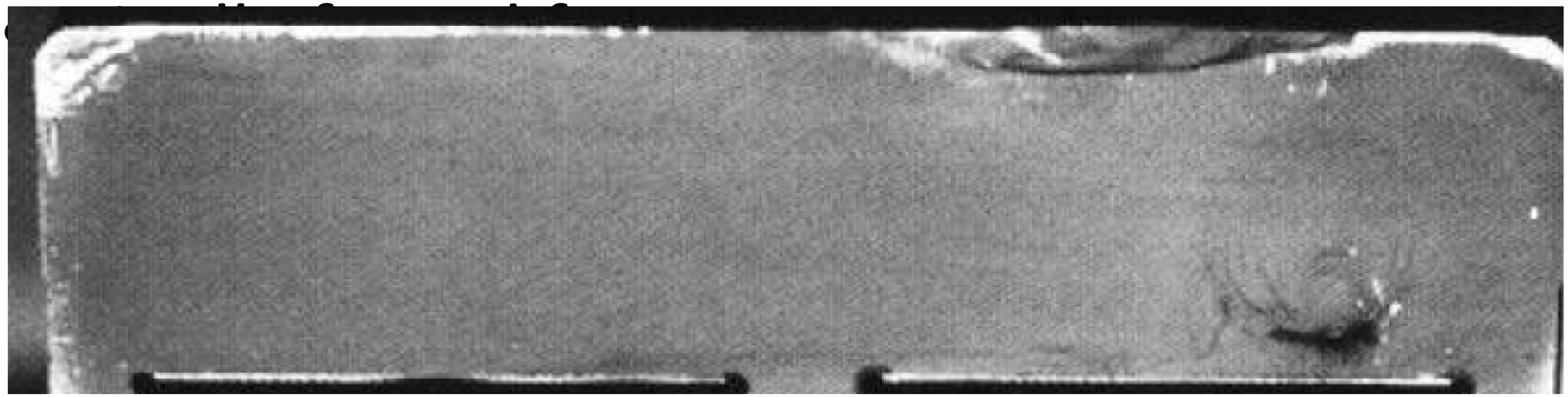


Fracture: forced fracture

- Stress concentration > tensile strength
- Impact / overstressing

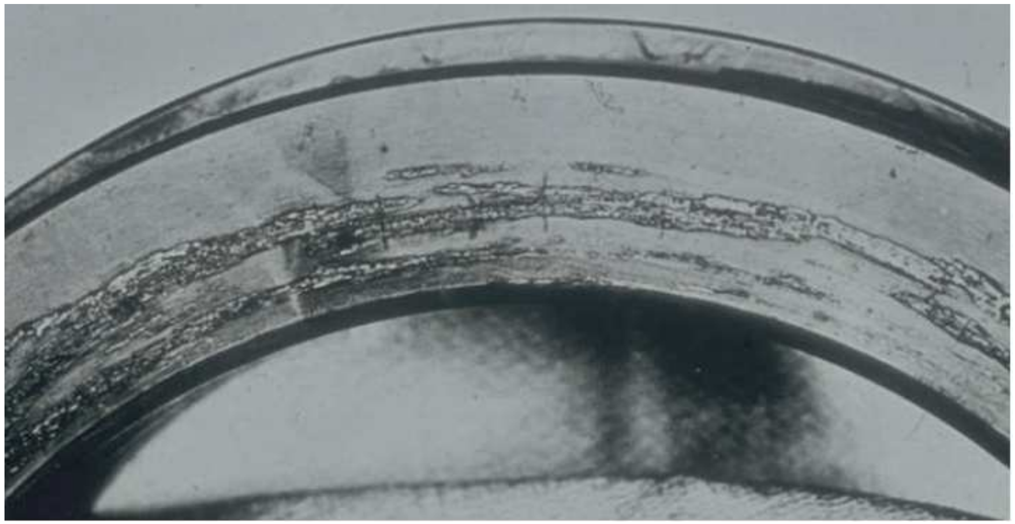
Fracture: fatigue fracture

- Rings and cages - Crack initiation / propagation
- Exceeding fatigue strength under bending



Fracture: thermal cracking

- High sliding and /or insufficient lubrication
- High friction heat
- Cracks at right angle to sliding direction



Bearing Life

- Any extra loading (e.g. misalignment, unbalance, resonance) reduces life by a cubed function:

$$\mathrm{L}_{10} = \left(\frac{16{,}667}{\mathrm{RPM}}\right) \times \left(\frac{\text{Rated Load}}{\text{Actual Load}}\right)^{3}$$

- 10% extra loading cuts life by 1/3
- 20% extra loading cuts life by half

Bearing Life, L10

- It is the life expectancy for 90% of the population
- Full load life is estimated at 1,000,000 revolutions at 3600 RPM, this is only 4.6 hours
- Guidelines:
  – Under a light load,   L10< 6%
  – Under a normal load, 6% <L10< 12%
  – Under a heavy load, L10 >12%

The Detection Technologies

- **Vibration analysis** and acoustic emission
- Oil and wear particle analysis
- Infrared thermography
- Each technology has its applications and should be used where appropriate. Under many circumstances, they are complementary.

Vibration Sources

Vibration can be due to 4 sources:

- Forced vibration due to unbalance, misalignment, blade and vane pass, gear mesh, looseness, impacts, resonance, etc…
- Resonance response due to impacts
- Stress waves or shock pulses
- Frictional vibration

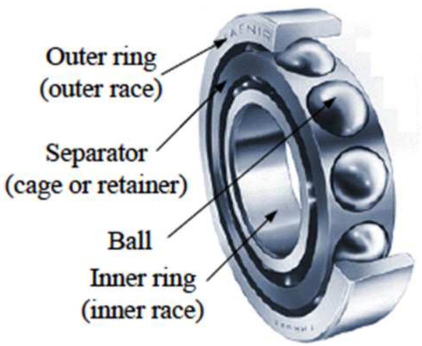
Vibration Profile

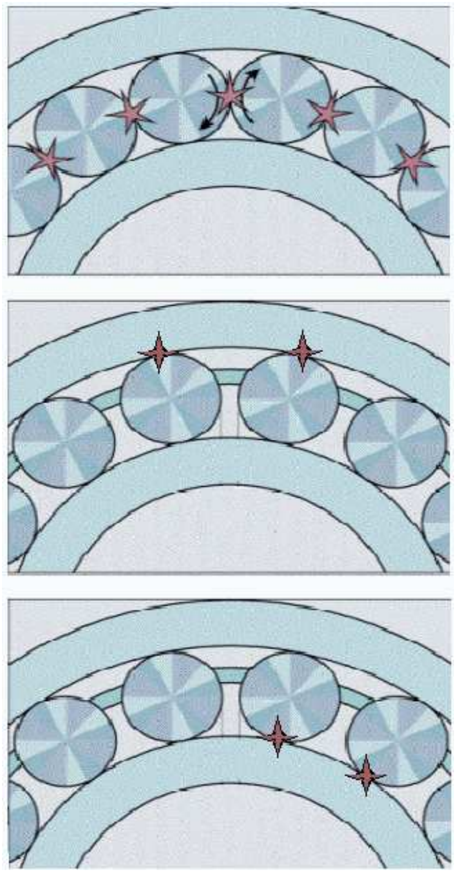
- In the vibration profile of a rolling element bearing three distinct frequencies can be found: natural, rotational, and defect
- Natural frequencies (resonance) are generated by impacts of internal parts of rolling element bearing. They are present in a new bearing.
- For a proper design, the natural frequencies are well above maximum frequency range and so rarely observed in predictive maintenance.

Rotational Frequencies

Four rotational frequencies are associated with rolling element bearings

- Fundamental train frequency
- Ball/roller spin
- Ball-pass outer-race
- Ball-pass inner-race



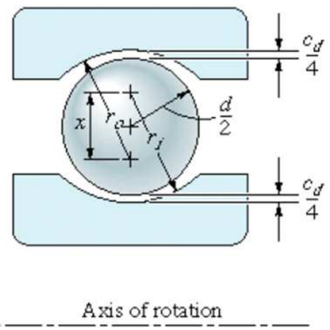


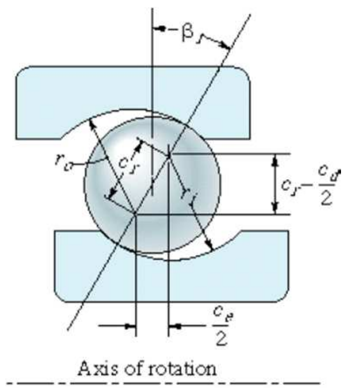
Rotational Frequencies
Fundamental train frequency (FTF)

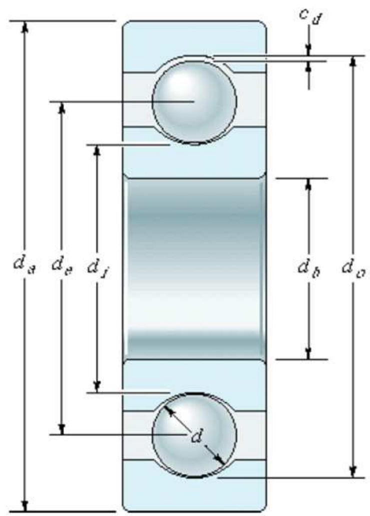
The bearing cage generates FTF as it rotates around races. Some friction exists between rolling elements and races, even with perfect lubrication.

Where f_t is the relative speed between outer and inner race.

$$FTF = \frac{1}{2} f_r \left[1 - \frac{d}{d_e} \cos\beta \right]$$







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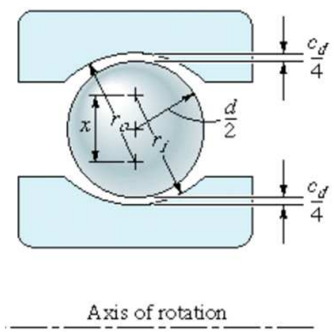
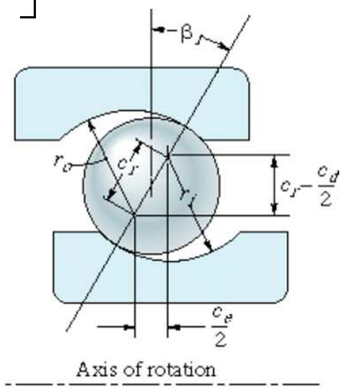
Rotational Frequencies
Ball-pass outer race (BPFO)

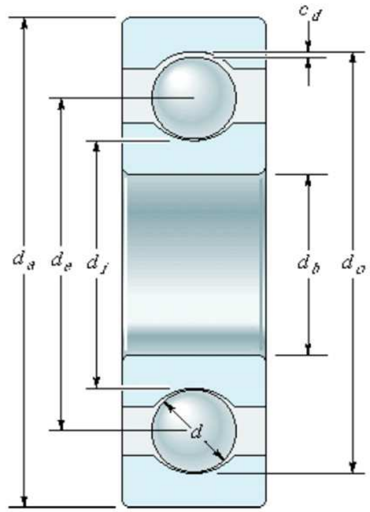
Balls or rollers passing outer race generate ball-pass outer-race frequency (BPFO):

Where n is the number of rollers

$$BPFO = \frac{1}{2} n f_r \left[1 - \frac{d}{d_e} \cos\beta \right]$$







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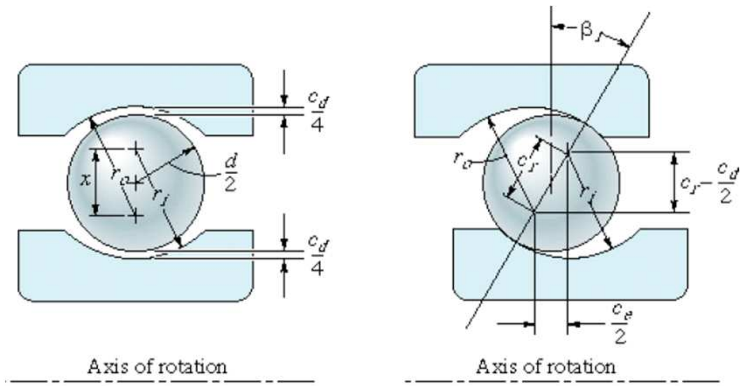
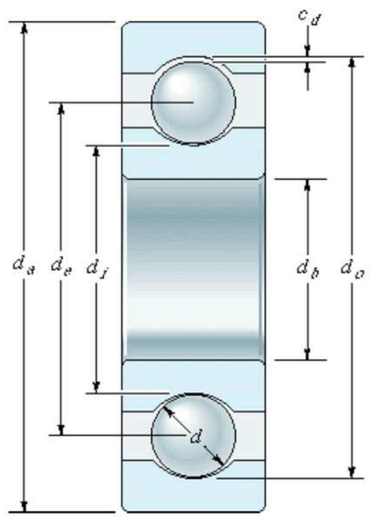
Rotational Frequencies
Ball-pass inner race (BPFI)

Balls or rollers passing outer race generate ball-pass inner-race frequency (BPFI):

Where n is the number of rollers

$$\mathrm{BPFI} = \frac{1}{2} n f_r \left[1 + \frac{d}{d_e} \cos\beta \right]$$





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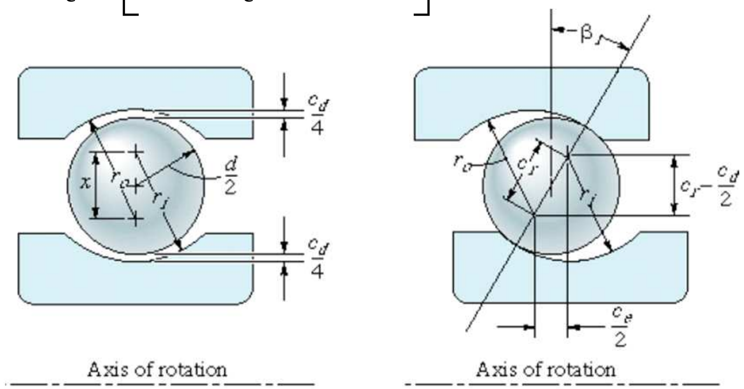
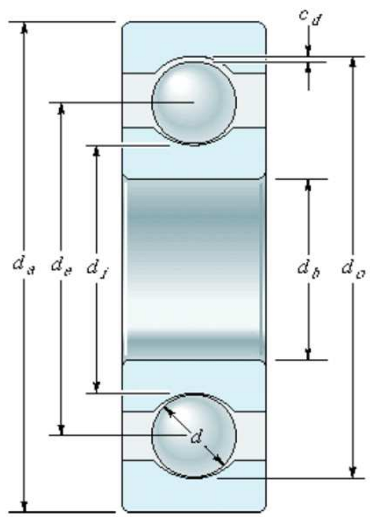
Rotational Frequencies
Ball-pass outer race (BPFO)

Ball-spin frequency - Each of balls or rollers rotates around own axis as it rolls around races. Speed of rotation (BSF) determined by geometry of bearing

Where β is the contact angle

$$\mathrm{BSF} = \frac{1}{2}\frac{d}{d_e} f_r \left[1 - \left(\frac{d}{d_e} \right)^2 \cos^2 \beta \right]$$





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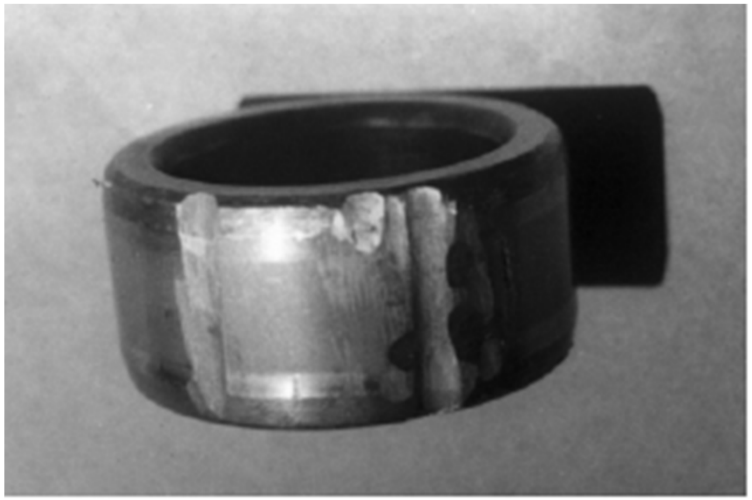
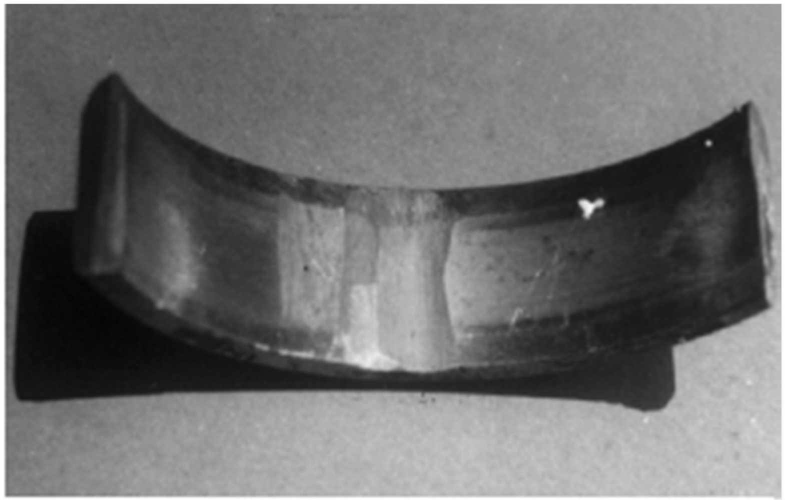
Defect Frequencies

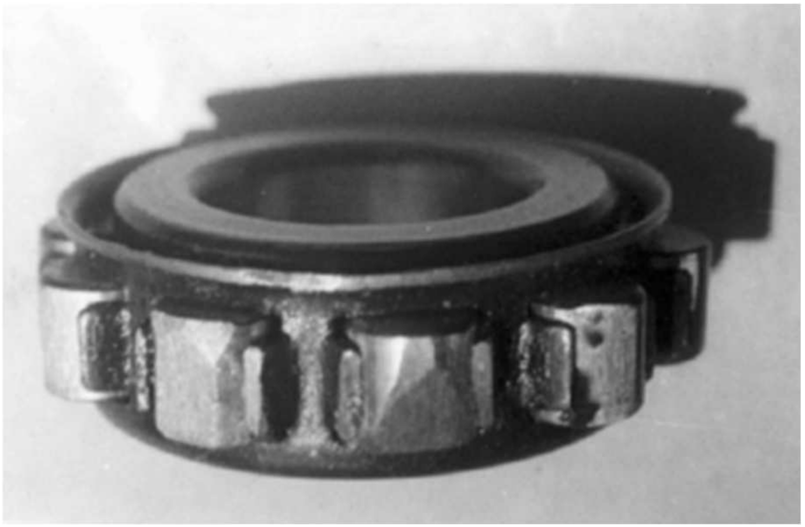
- Rolling element bearing defect frequencies are the same as the as their rotational frequencies, except BSF.
- For a defect on inner race: BPFI amplitude increases as balls/rollers contact defect.
- For a defect on outer race: BPFO amplitude increases as balls/rollers contact defect.





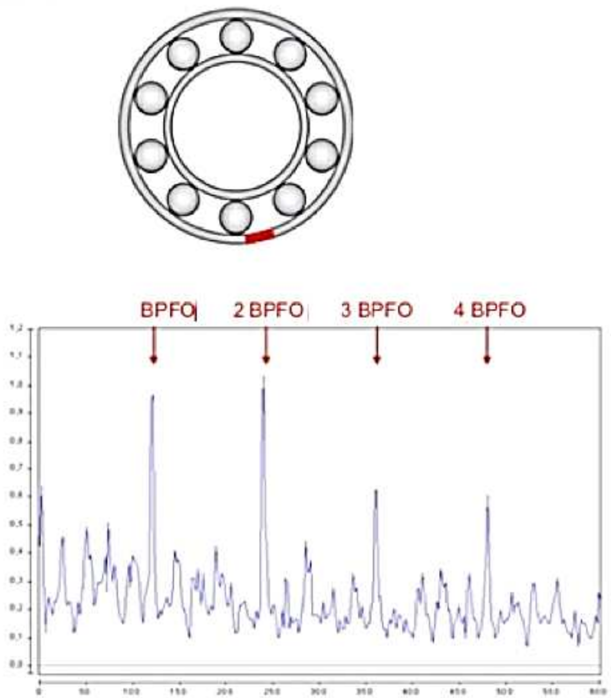
Defect Frequencies

When one or more of balls or rollers have defects, the defect impacts both the inner and outer race each time one revolution of the rolling-element is made. Therefore detection frequency is 2xBSF rather than 1 x BSF



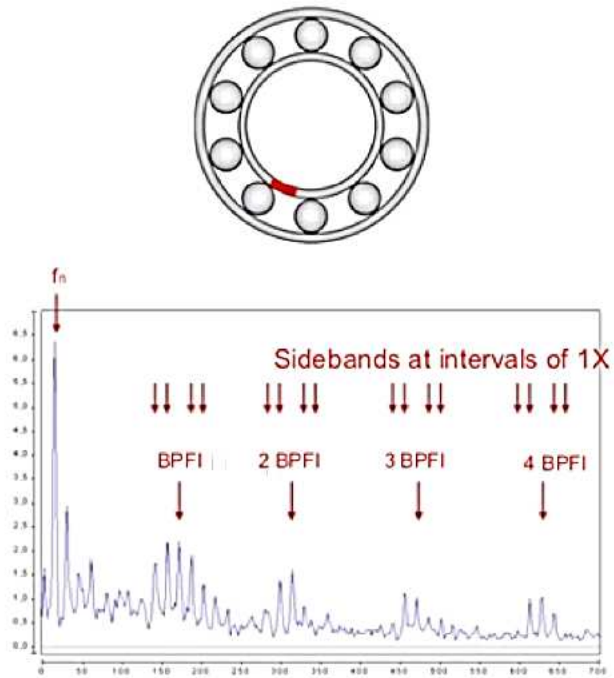
Example 1

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



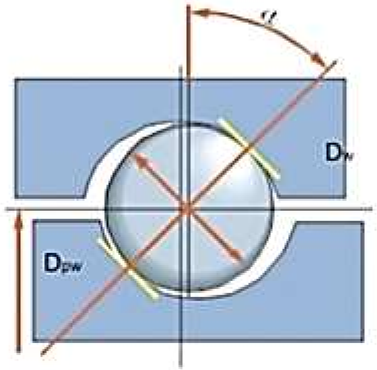
Outer race damage frequency BPFO as well as harmonics clearly visible

Inner race damage:
(Ball passing frequency, inner range BPFI)



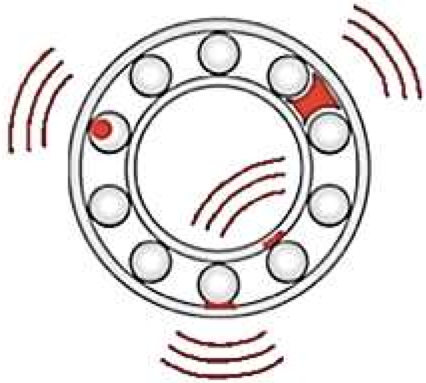
Inner race damage frequency BPFI as well as numerous sidebands at intervals of 1X.

Example 2



Angle of contact
D Arc diameter
d Rolling element diameter
Z Number of rolling elements
n Shaft RPM

Ball bearing SKF 6211
RPM, n = 2998 rev/min



1 - Outer race damage
2 - Inner race damage
3 - Rolling element damage
4 - Cage damage

Dimensions	Rollover frequencies
d =77.50 mm	BPFO = 203.77
D =14.29 mm	BPFI = 295.90 Hz
Z = 10	$2.f_w$ = 261,77 Hz
= 0	f_K = 20.38 Hz

Basic Frequency Spectrum Patterns

Vibration measurements display either of four basic spectrum (FFT) patterns:

Harmonics

Sidebands - Due to Amplitude Modulation or Frequency Modulation

Mounds/Haystacks - Random vibration occurring in a frequency range

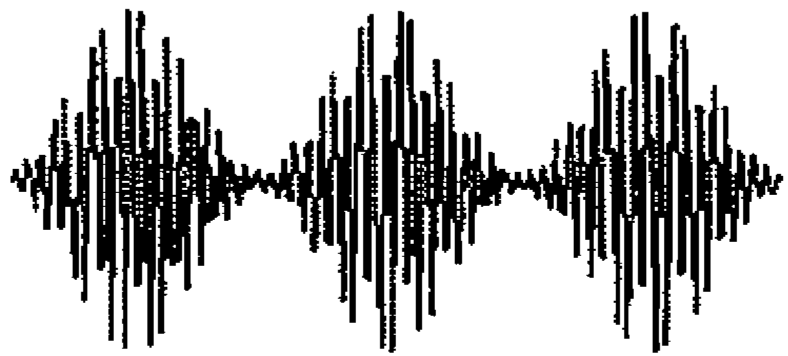
Raised Noise Floor - White noise or large random events

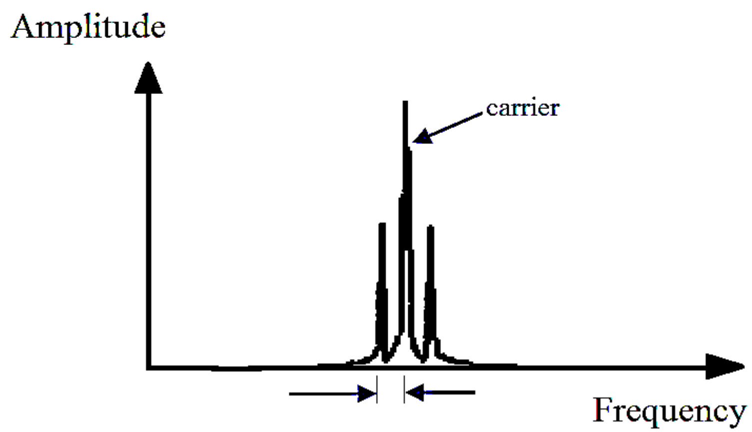
Demodulation

- Demodulating (Envelope) the Signal or Determination of the Peaks of the Repetitive Fault Frequency.
- Spanning the band for the station frequency (540-1600 kHz) and picking off the broadcasted signal.
- Incorporating a high-pass or band-pass filtering
- Eliminating any high amplitude signals associated with 1 x and multiples up to about 10 x
- Inclusion of only the fault frequencies exciting inherent resonance
- Intensifying and drawing out repetitive components of the fault
- Converting to frequency for display of the pattern
- Amplitudes will show up as a distinctive “saw-tooth” or “comb” harmonic pattern of the actual bearing fault

Amplitude Modulation

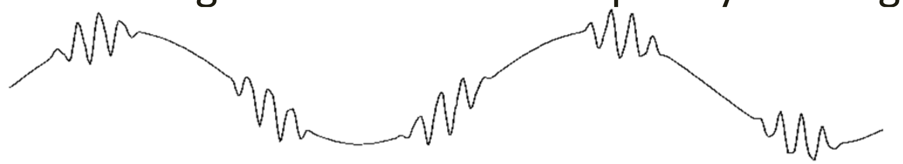
- Amplitude Modulation (AM)
- One frequency (carrier) is getting louder and softer at another frequency (the modulating frequency)
- AM is mono. Mono is ‘one’, which implies **one** sideband on each side of the carrier.



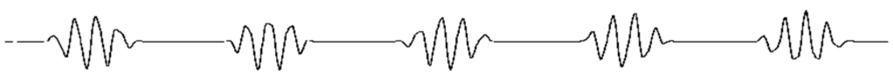


Signal Processing

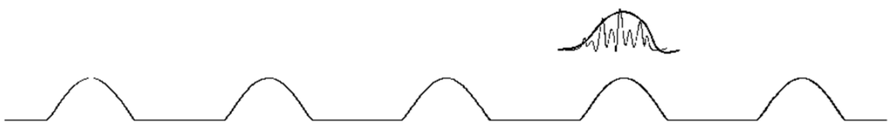
- The raw signal includes low frequency running speed harmonics



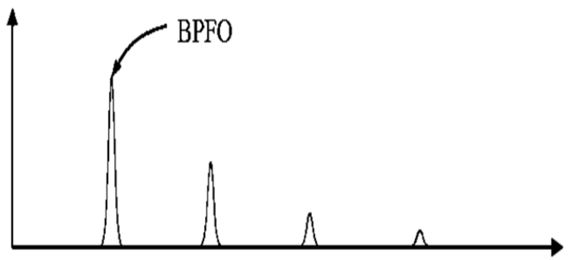
- These are removed by band-pass filtering



- Then envelope detection is applied



- Finally the result is displayed in the frequency domain:



Mechanical Condition Monitoring
Fault Diagnostics of Gears
Lecture 4

Outline

- Gear Basics
- Gear types
- Gear Meshing
- Vibration Profiles

Gears Types

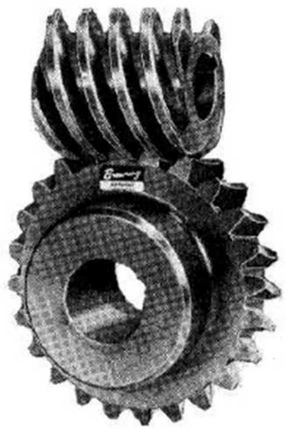

SPUR GEARS

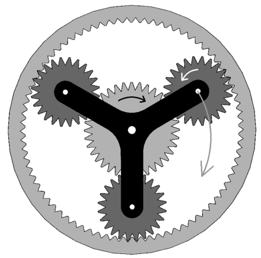
BEVEL GEARS



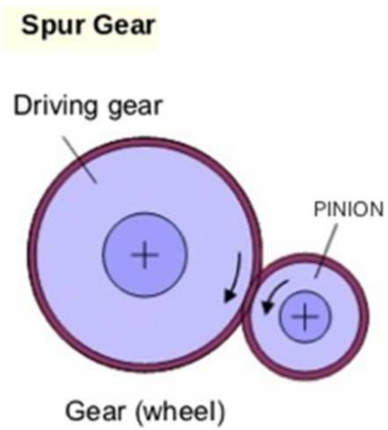
HELICAL GEARS

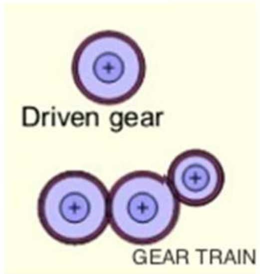

MITER GEARS

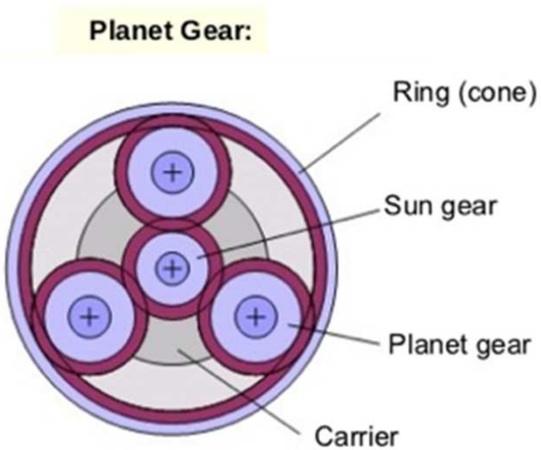

WORM GEARS

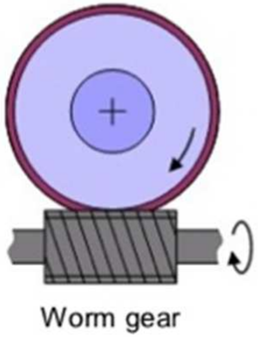

PLANETARY GEARS

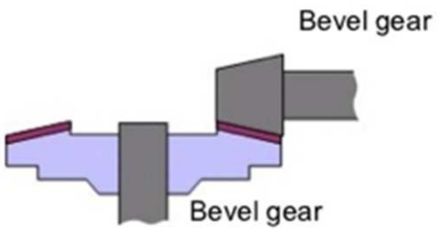
Gears Types


Spur Gear
Driving gear
PINION
Gear (wheel)


Driven gear
GEAR TRAIN


Planet Gear:
Ring (cone)
Sun gear
Planet gear
Carrier


Worm gear

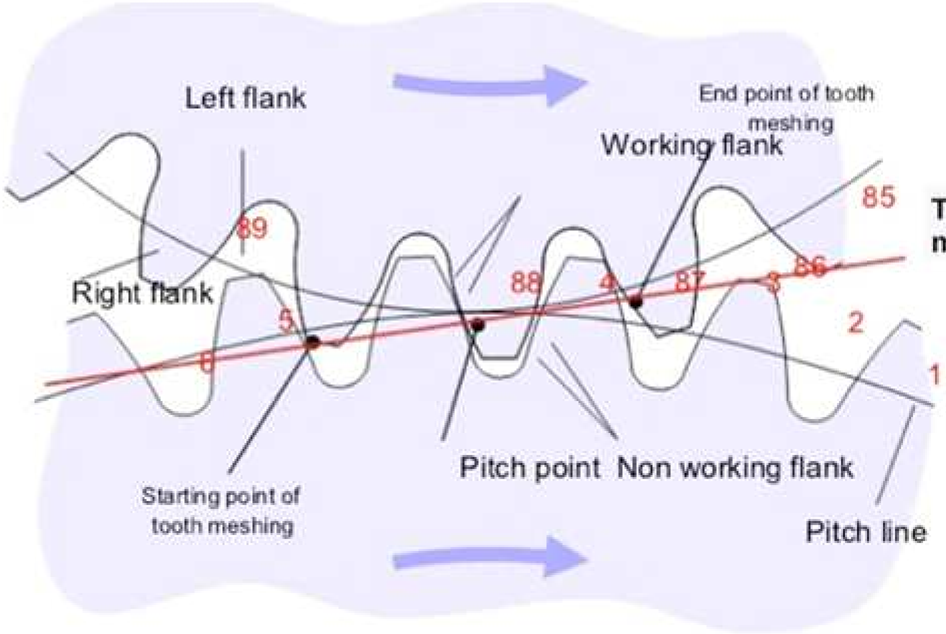

Bevel gear
Bevel gear

Gears

- Gearboxes contribute significantly to damage incidents and maintenance costs.
- Gearboxes comprise about 60% of mechanical faults in rotating machinery and about 30% of maintenance costs in aircraft industry due to gearboxes.
- Fault detection in helicopter gearboxes is one of THE most difficult problems in rotating machinery diagnostics

Gear Meshing

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

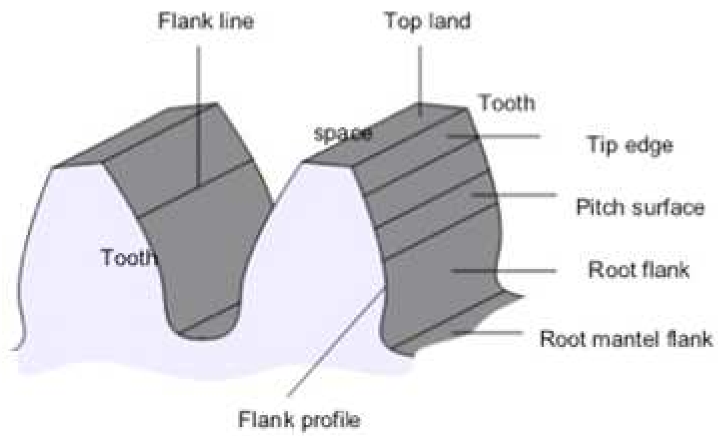


The red dotted line is the contact path where the meshing teeth will be in contact during the rotation.

Gear mesh frequency f_z can be calculated:

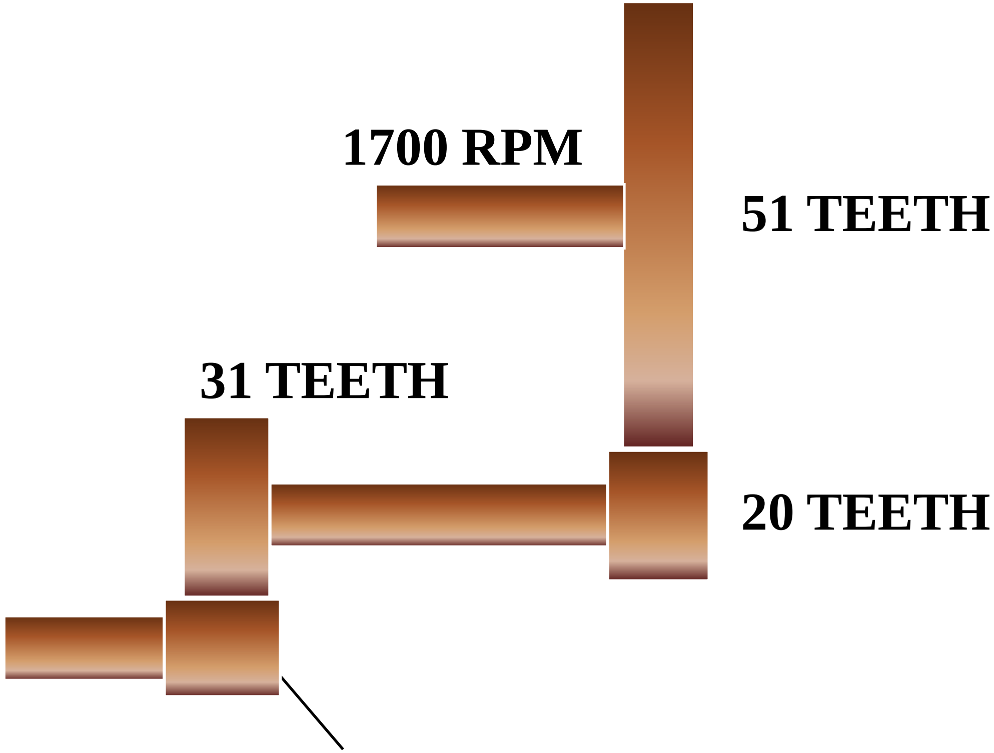
$F_z = z f_n$

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



Basic Gear Relations

RPM1 x T1 = RPM2 x T2

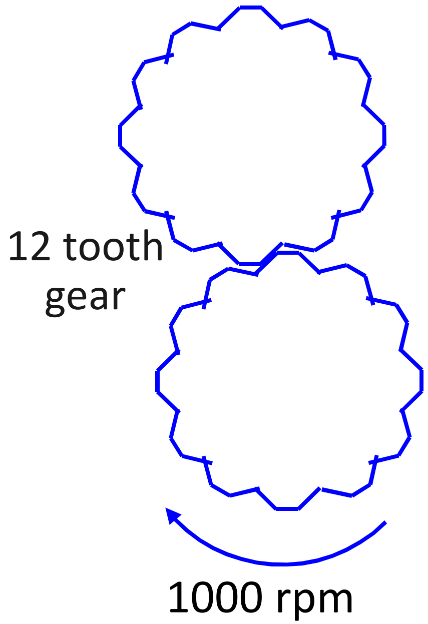


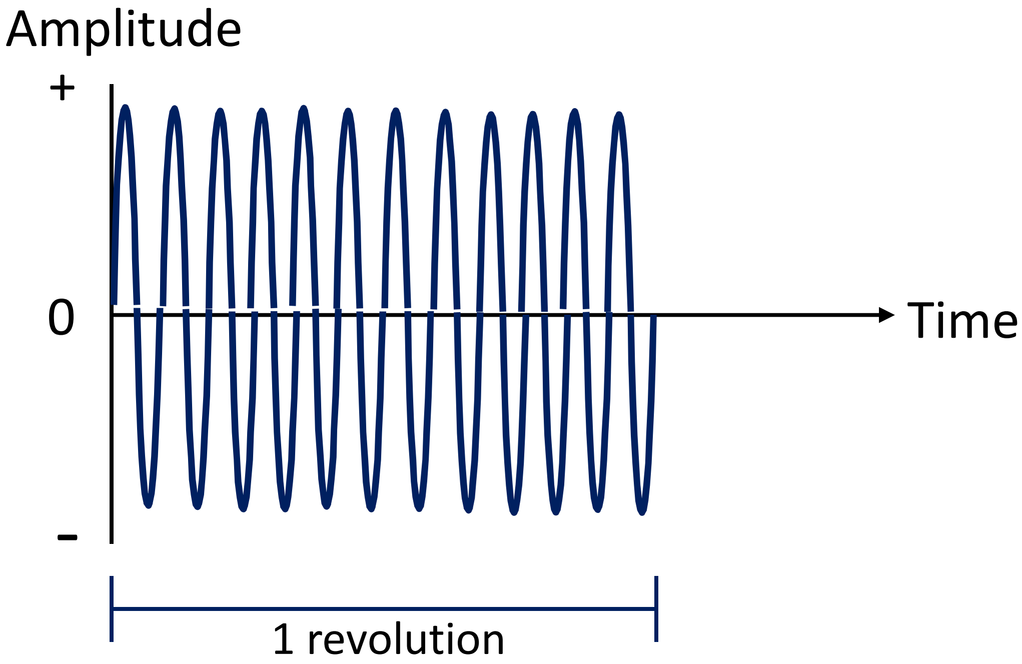
If 8959 RPM - HOW MANY TEETH ARE ON THIS GEAR?

Gears

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

Gear Meshing (GMF)



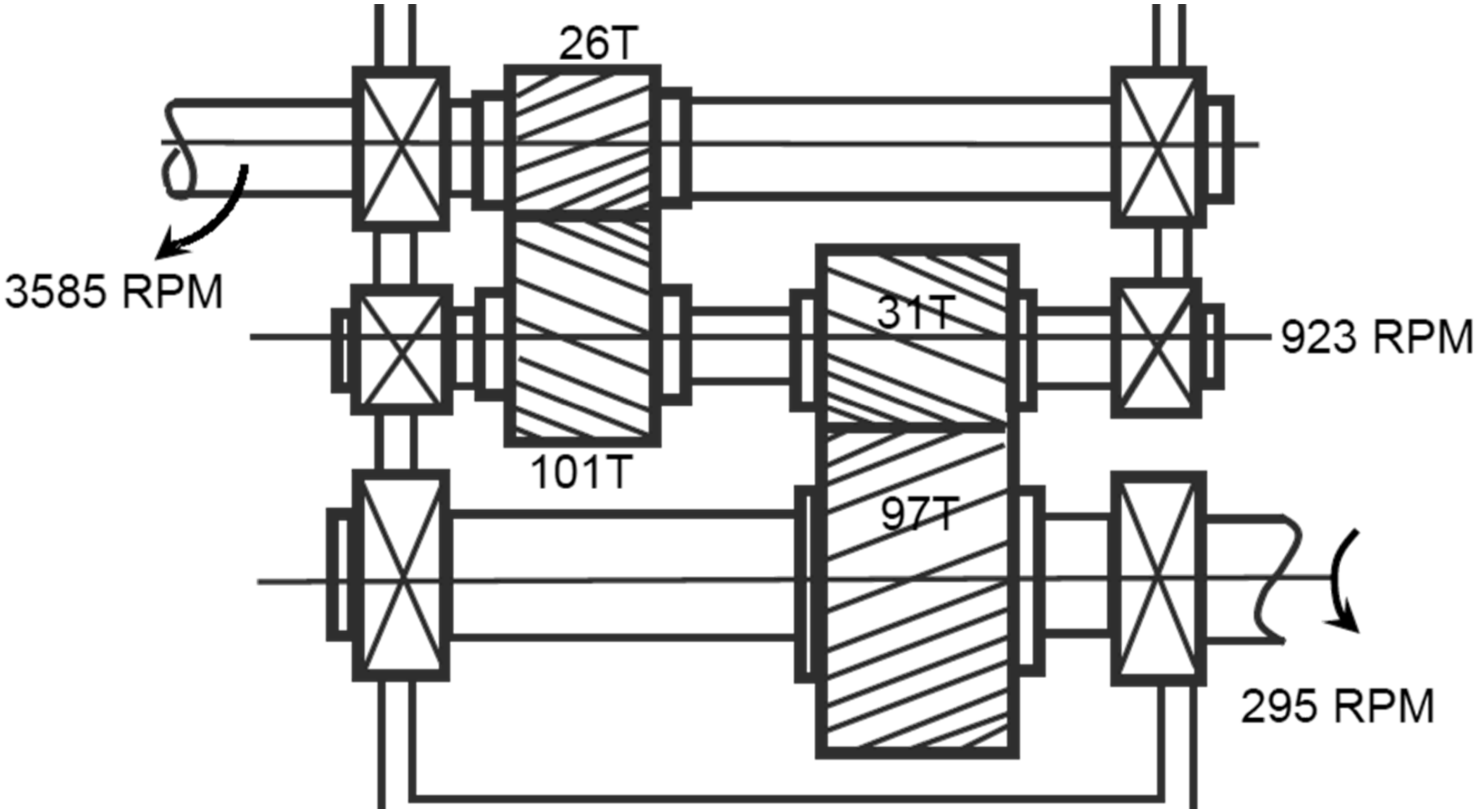


12 teeth are meshing every revolution of the gear

12 x 1000 rpm = vibration occurs at 12,000 cycles per minute

GMF = 12,000 RPM = 200 Hz

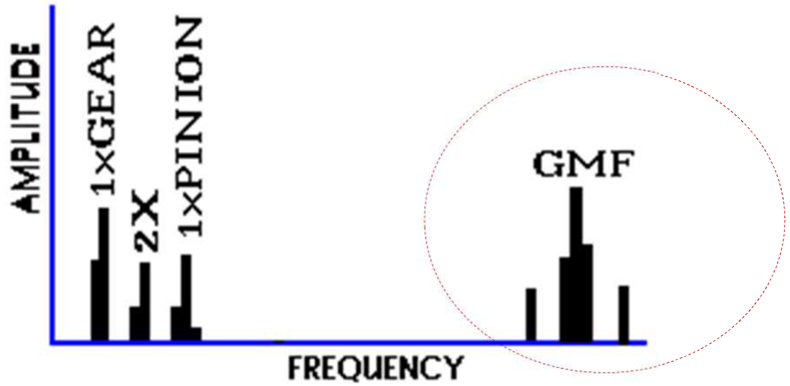
Normal Vibration Profile - E.g. GMF calculation



Normal Vibration Profile - Example GMF calculation

- Input shaft speed = 3,585 RPM
- Intermediate shaft speed = (3,585 RPM) [(26 T)/101 T] = 923 RPM
- Output shaft speed= 923 x 31T/97T=295 RPM
- High-speed gear mesh:
- 3,585 RPM x 26T = 93,210 CPM (1,553.5 Hz)
- Low-speed gear mesh:
- 922.87 RPM x 31 T = 28,609 CPM (476.8 Hz)

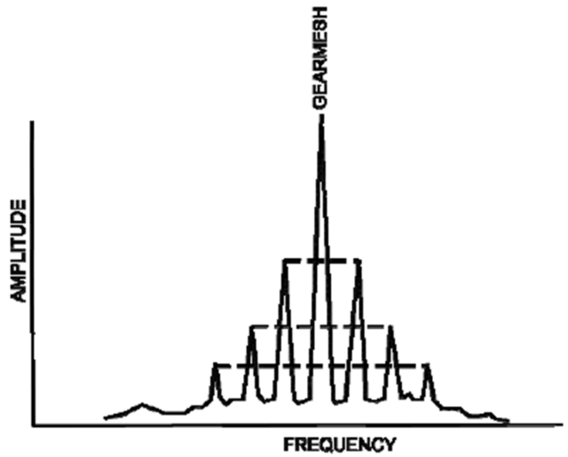
Normal Vibration Profile

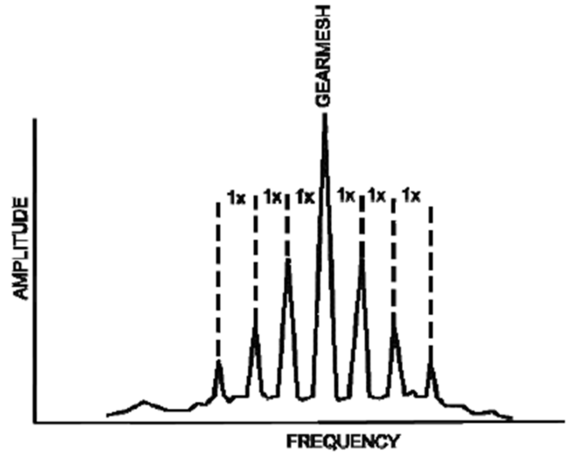


- **Normal profile** of **undamaged gear set** shows **1X** and **2X** and **gear mesh frequency GMF**
- GMF commonly will have sidebands of running speed
- All peaks are of low amplitude and **no natural frequencies** are present

Normal Vibration Profile – GMF Frequency

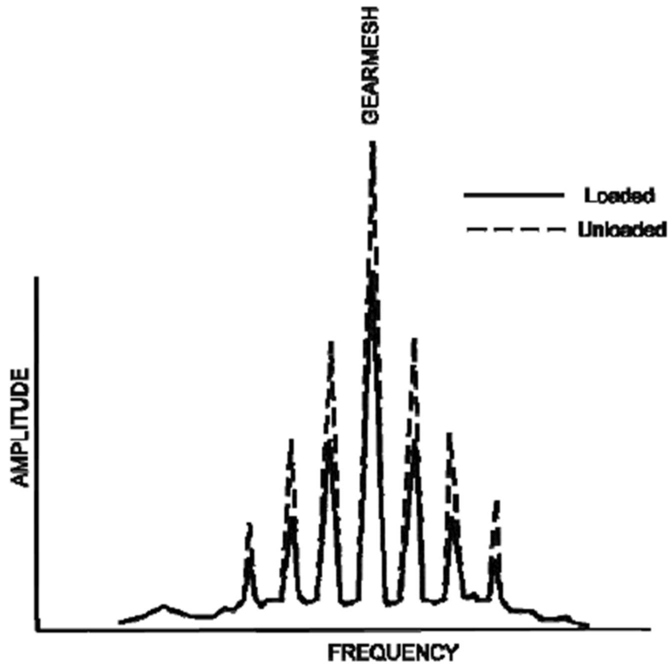
- GMF Sidebands occur **in pairs**, one below and one above the gear-mesh frequency, and amplitudes are the same.
- Any deviation from this profile indicates a gear problem.





Normal Vibration Profile – Effect of Load

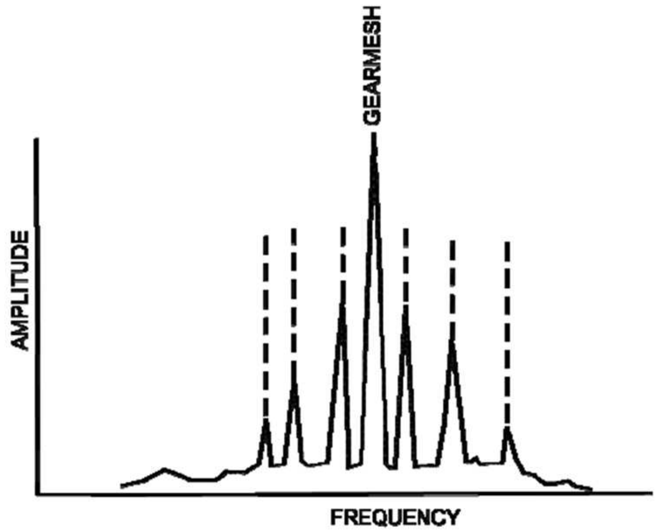
- Energy and vibration profiles of gear sets change with load.
- Gear Mesh Frequencies are often sensitive to load
- High GMF amplitudes do not necessarily indicate a problem
- When the gear is fully loaded, profiles exhibit amplitudes as shown before.
- When gear is unloaded, same profiles are present, but amplitude increases dramatically.
- Difference is due to gear-tooth roughness. More looseness is present on the non-power side of the gear.



Around the GMF Frequency

Fault Diagnostics

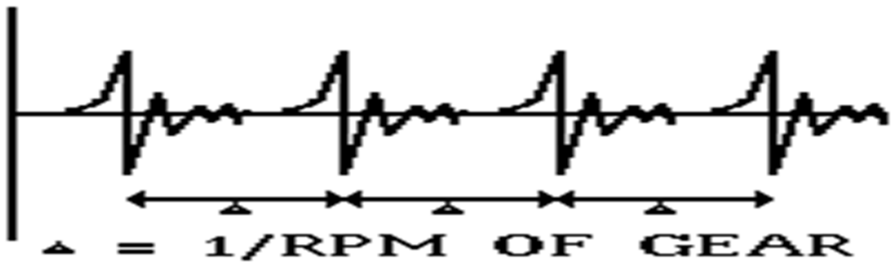
- If the gear set develops problems, the **amplitude** of the gear-mesh frequency and **symmetry** of the sidebands changes
- When a gear set becomes worn the spacing between the sidebands becomes erratic and symmetry of amplitudes alters.



Around the GMF Frequency

Fault Diagnostics - Chipped/Broken tooth

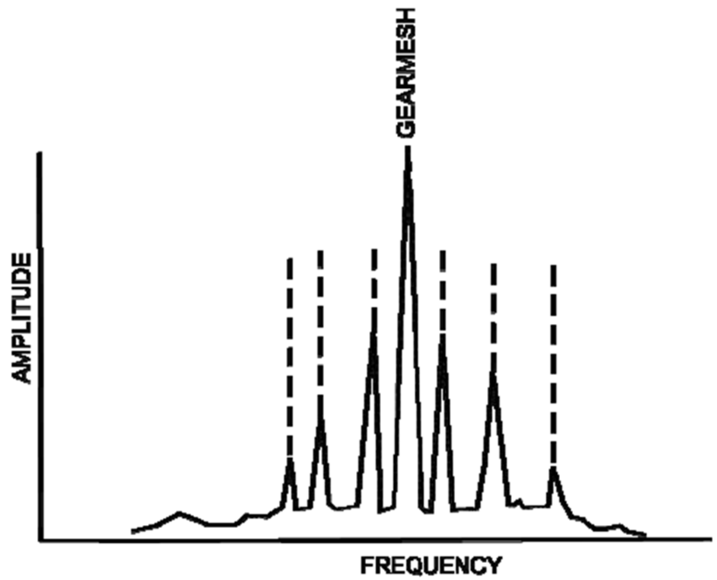
TIME WAVEFORM



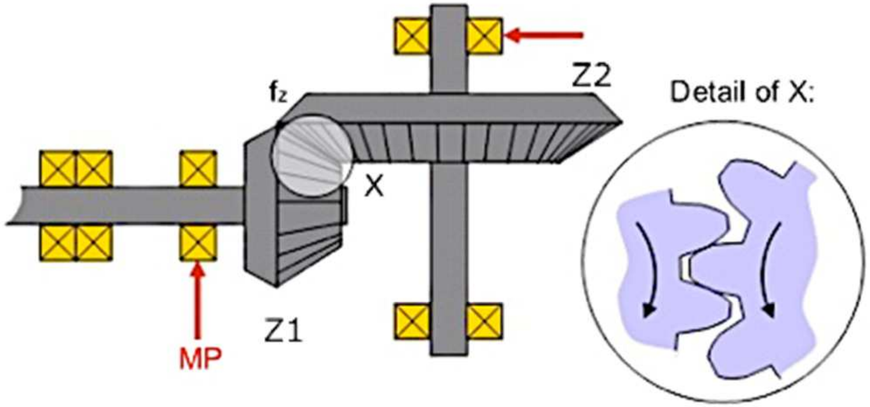
- A cracked or broken tooth will generate a high amplitude **at 1X RPM of the gear**
- It will excite the gear natural frequency which will be sidebanded by the running speed fundamental
- Best detected using the time waveform
- Time interval between impacts will be the reciprocal of the 1X RPM

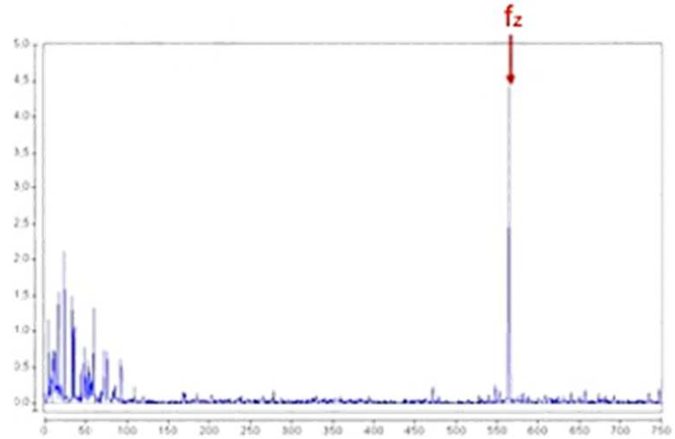
Fault Diagnostics - Chipped/Broken tooth Effect on GMF

- The profile shown signifies the effect of a chipped or broken tooth in the gear set.
- As the gear rotates, the space left by the chipped or broken tooth increases the **mechanical clearance** between pinion and bullgear
- Result is low-amplitude sidebands to left of gear-mesh frequency (lower freq), and
- Sidebands to the right of the mesh frequency have higher amplitude

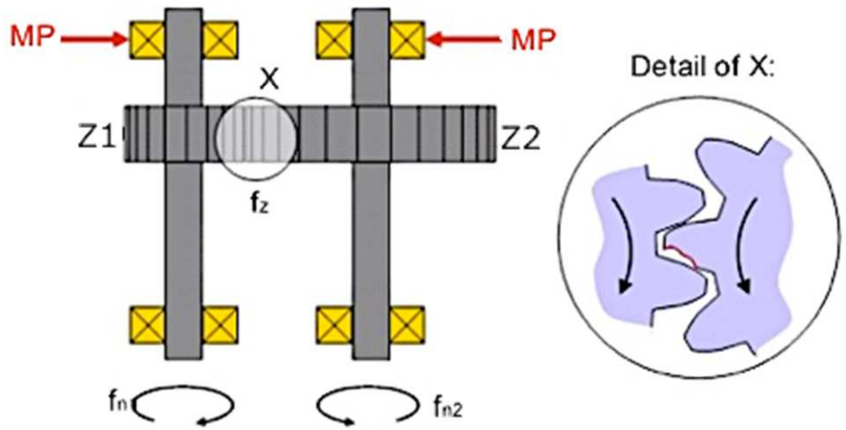


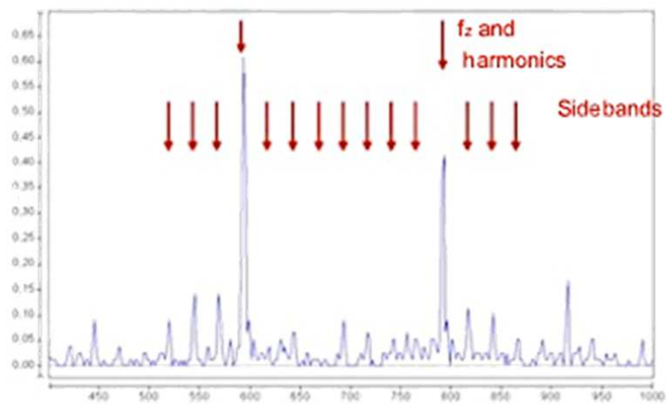
INCORRECT TOOTH SHAPE



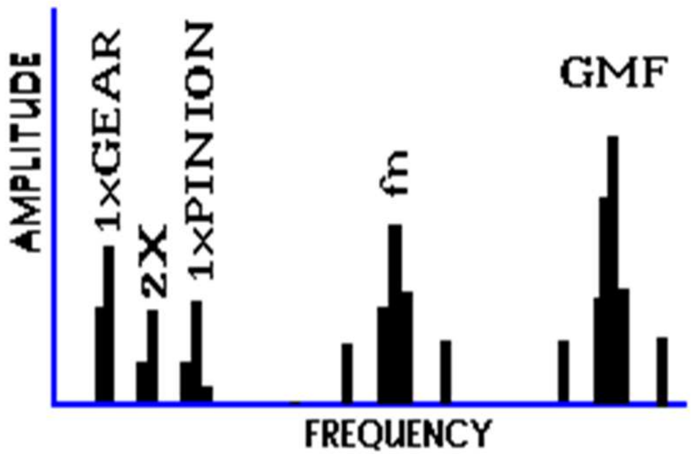


TOOTH BREAK-OUT



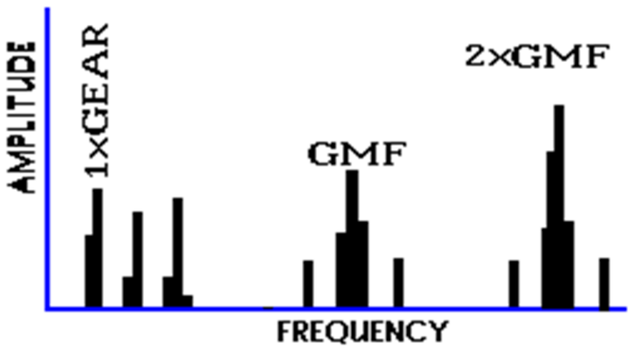


Fault Diagnostics – Tooth Wear



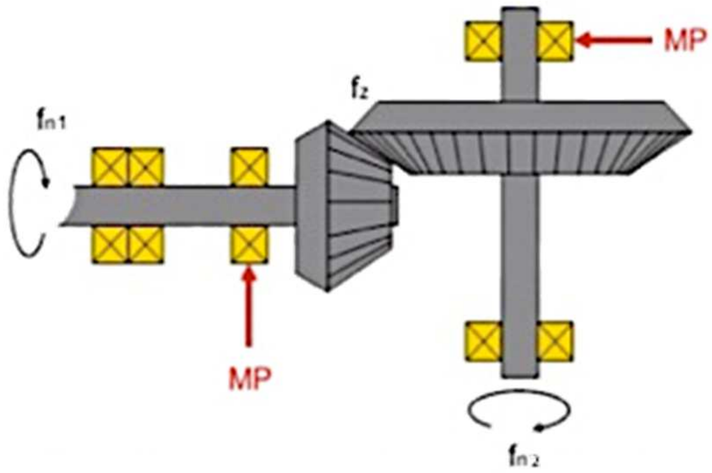
- Wear is indicated by excitation of natural frequencies along with sidebands of 1X RPM of the bad gear
- Sidebands are a better wear indicator than the GMF
- GMF may not change in amplitude when wear occurs

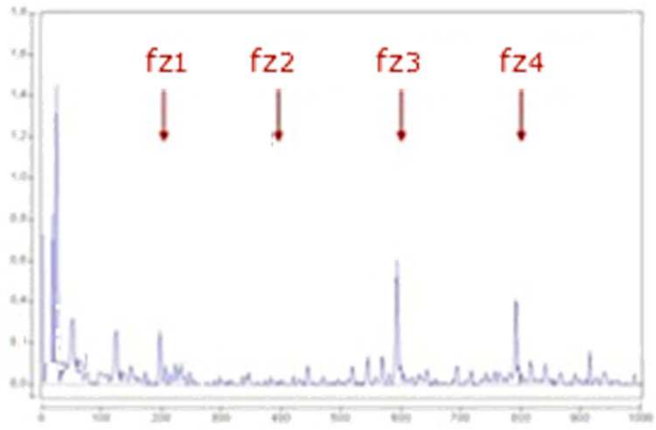
Fault Diagnostics – Misalignment of Gears



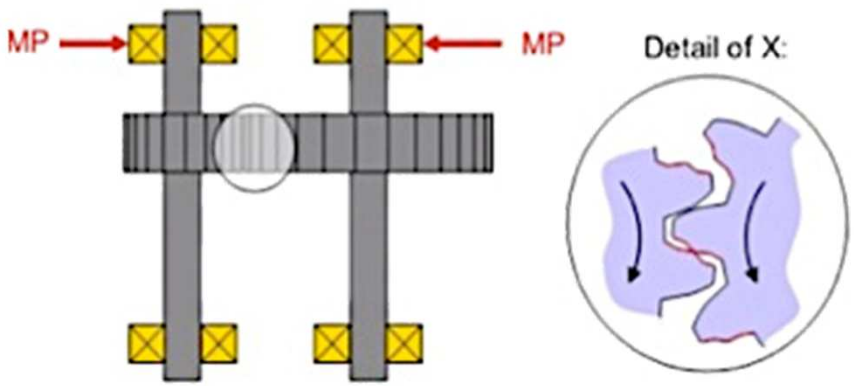
- Gear misalignment almost always excites **second order** or higher harmonics with sidebands of running speed
- Small amplitude at 1X GMF but higher levels at 2X and 3X GMF

INCORRECT TOOTH MESHING

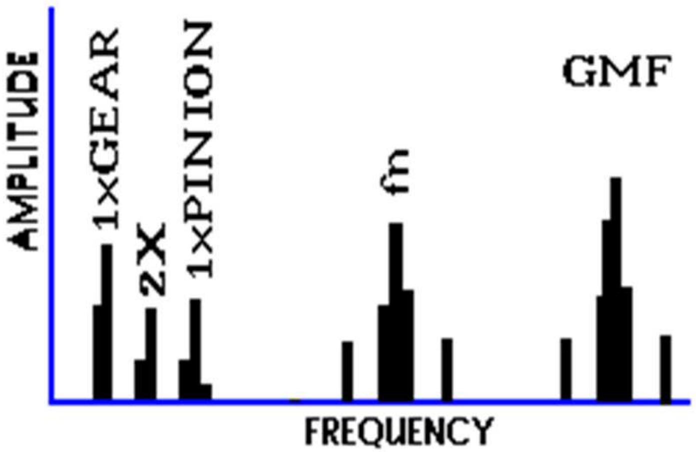




WEAR

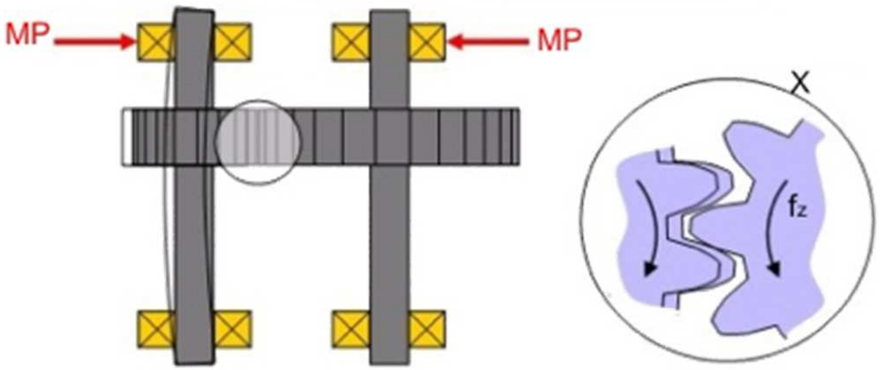


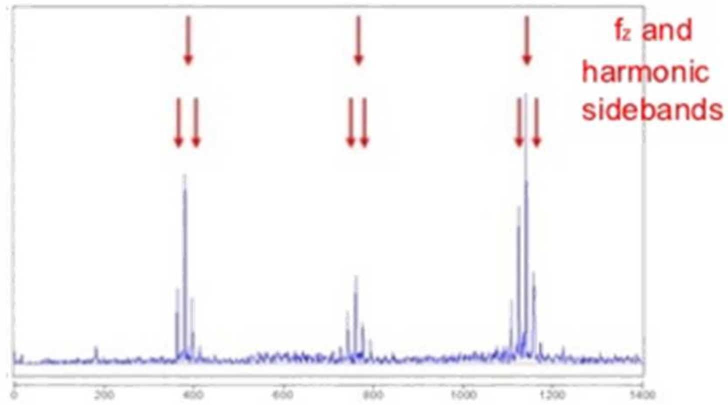
Fault Diagnostics – Eccentricity and Backlash



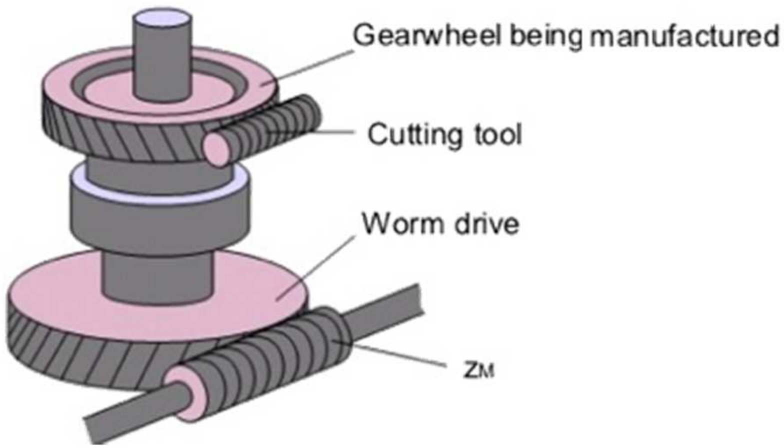
- Fairly **high amplitude sidebands** around GMF suggest eccentricity, backlash or non parallel shafts
- The problem gear will modulate the sidebands
- Incorrect backlash normally excites gear natural frequency

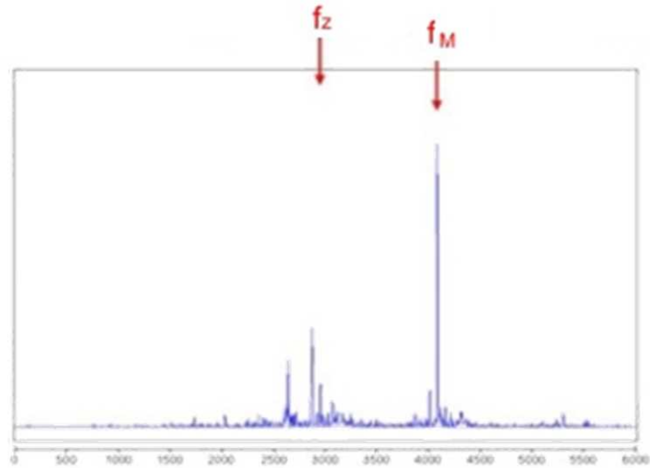
ECCENTRICITY, BENT SHAFTS





GHOST FREQUENCIES





Gear Hunting Problem

- Is the phenomenon in which two teeth - one on each gear - that are damaged contact one another at a particular frequency.
- During the normal rotation of these gears, those two teeth will enter the mesh area simultaneously and contact one another. This is a relatively low frequency - lower than the RPM of either gears.
- It is determined by the common factors of the number of teeth on each gear.
- Hunting Tooth Frequency – FHT:

$$\boxed{\mathrm{FHT} = \frac{\mathrm{GMF} \times \mathrm{CF}}{\mathrm{T_{Gear}} \times \mathrm{T_{Pinion}}}}$$

Gear Hunting frequency - FHT

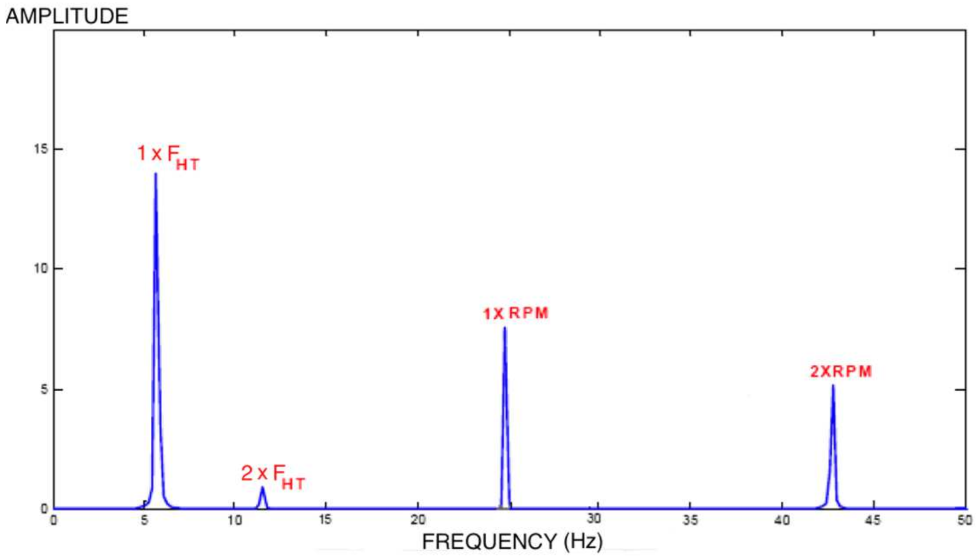
To calculate the common factors of each gear (CF), list all the multiplication possibilities for each tooth number and compare.

Example

24 Tooth Gear		**84 Tooth Gear**	
	1 x 24		1 x 84
	2 x 12		2 x 42
	3 x 8		3 x 28
	4 x 6		4 x 21
			6 x 14
			7 x 12

The numbers that appear in each column are: 1, 2, 3, 4, 6 and 12. The highest common factor is 12 in this example

Hunting Tooth Symptoms



- Amplitude peaks at 1 x FHT and possibly 2 x FHT.
- Sidebands of FHT around 1x rpm (of each shaft).
- Sidebands of FHT around 1x GMF and harmonics.