Queen's University
Mechanical Engineering

Machine Condition Monitoring

and

Fault Diagnostics

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Course Overview

- Introduction to Machine Condition Monitoring and Condition Based Maintenance
- Basics of Mechanical Vibrations
- Vibration Transducers
- Vibration Signal Measurement and Display
- Machine Vibration Standards and Acceptance Limits (Condition Monitoring)
- Vibration Signal Frequency Analysis (FFT)

Course Overview

- Machinery Vibration Trouble Shooting
- Fault Diagnostics Based on Forcing Functions
- Fault Diagnostics Based on Specific Machine Components
- Fault Diagnostics Based on Specific Machine Type
- Automatic Diagnostic Techniques
- Non-Vibration Based Machine Condition Monitoring and Fault Diagnosis Methods

Current Topic

- Machinery Vibration Trouble Shooting
- Fault Diagnostics Based on Forcing Functions
- Fault Diagnostics Based on Specific Machine Components
- Fault Diagnostics Based on Specific Machine Type
- Automatic Diagnostic Techniques
- Non-Vibration Based Machine Condition Monitoring and Fault Diagnosis Methods

Fault Diagnostics Based on Forcing Functions

Unbalance

Mechanical Looseness

Rubs

Oil Whirl

Structural Vibrations

Hydraulic Forces

Misalignment

Soft Foot

Resonances

Oil Whip

Foundation Problems

Aerodynamic Forces

Fault Diagnostics Based on Forcing Functions

Forcing frequencies associated with machines

Source	Frequency (multiple of RPM)
Fault Induced	
mass unbalance	1x (frequency in once per revolution)
Misalignment	1x, 2x
bent shaft	1x
mechanical looseness	odd orders of x
casing and foundation distortion	1x
antifriction bearing	bearing frequencies, not integer ones
impact mechanisms	multi-frequency depending on waveform
Design Induced	
universal joints	2x
asymmetric shaft	2x
gear mesh (n teeth)	nx
coupling (m jaws)	mx
fluid-film bearings (oil whirl)	0.43x to 0.47x
blades and vanes (m)	mx
reciprocating machines	half &full multiples of speed, depending on design

UNBALANCE

- linear problem
- periodic time signal
- 360° cycle
- strong radial vibration at fundamental frequency (1 × rotational speed, 1X)
- if rotor is overhung - radial & axial vibration
- response amplitude proportional to rotational speed squared

Unbalance

Results in:

- excessive bearing wear (gears, bushings, etc.)
- fatigue in support structures
- decreased product quality
- power losses
- disturbed adjacent machinery

Unbalance

Causes of unbalance:

- excess of mass on one side of rotor
- centrifugal force pulls rotor toward heavy side
- low tolerances in fabrication (casting, machining, assembly)
- variation within materials - voids, porosity, inclusions, variable density, finishes, etc.
- non symmetry of design - motor windings, part shapes, locations

Causes of Unbalance

- non symmetry in use - distortion, size changes, shifting parts due to stress, aerodynamic forces, temperature changes, etc
- manufacturing processes are a major cause of unbalance
- cost plays a role - perfect balance is always possible but at a cost
- unbalance can be corrected by adding or removing weight from rotor at the appropriate location
- pros and cons to both practices
- consider unbalance problems last

Causes of Unbalance

Note: some machinery is designed to operate "out-of-balance" – shakers, sieves, materials transport….

Unbalance

Unbalance Correction Methods

Addition of mass:

- up to 20:1 vibration amplitude reduction on first try (if done carefully)
- if space limitations exist more than one addition of mass may be required

Unbalance Correction Methods

Addition of mass

a) addition of solder or epoxy

- centre of gravity difficult to control
- takes time

b) addition of standard washers

- bolted or riveted
- incremental sizes
- quick

Unbalance Correction Methods

Addition of mass

c) addition of pre-manufactured weights

- incremental sizes
- quick

d) addition of cut to size masses

- welded in place

Unbalance Correction Methods

Removal of mass - 10:1 vibration reduction first try

a) - drilling

- very accurate, quick

b) milling

- used for large corrections, accurate

c) grinding

- trial & error method
- accurate removal of mass difficult

Unbalance Correction Methods

Mass centering

- rotor principal axis of inertia found
- journal & shaft machined to match this axis
- very expensive

Unbalance

Units of Unbalance

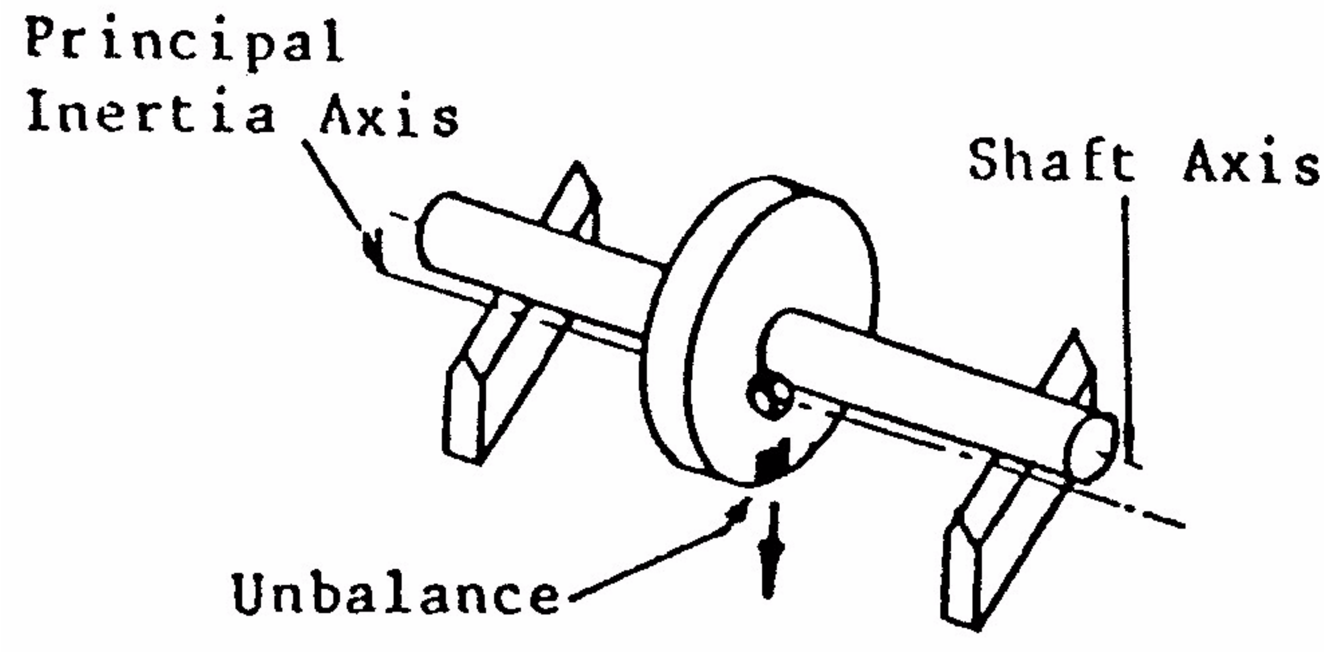
- gram - inches, gram - millimetres
- 100 g - in (10 g $\times$ 10 inches, 20g $\times$ 5 inches)
- rigid shafts may be balanced at any speed (theoretically)

Types of Unbalance

Static Unbalance (force unbalance)

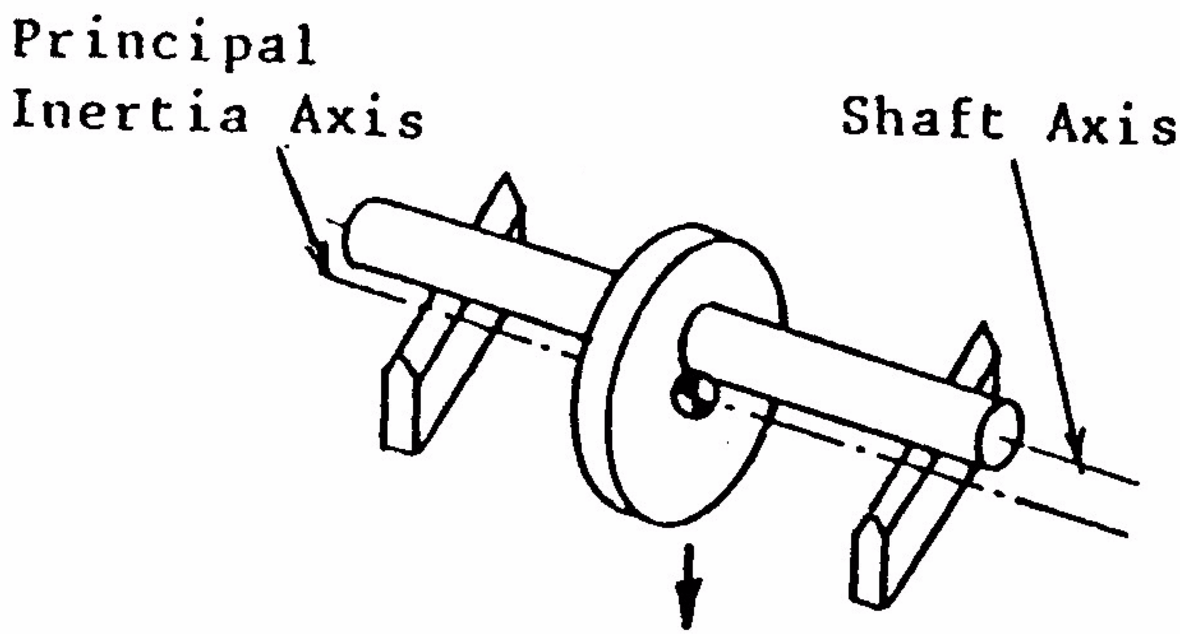
- principal axis of inertia is displaced parallel to the shaft axis
- found mostly in narrow, disc-shaped parts (fly wheels, turbine wheels)
- single mass correction placed opposite the centre-of-gravity in a plane $\perp$ to shaft axis and intersecting the centre-of-gravity
- knife edge balancing possible

Static Unbalance



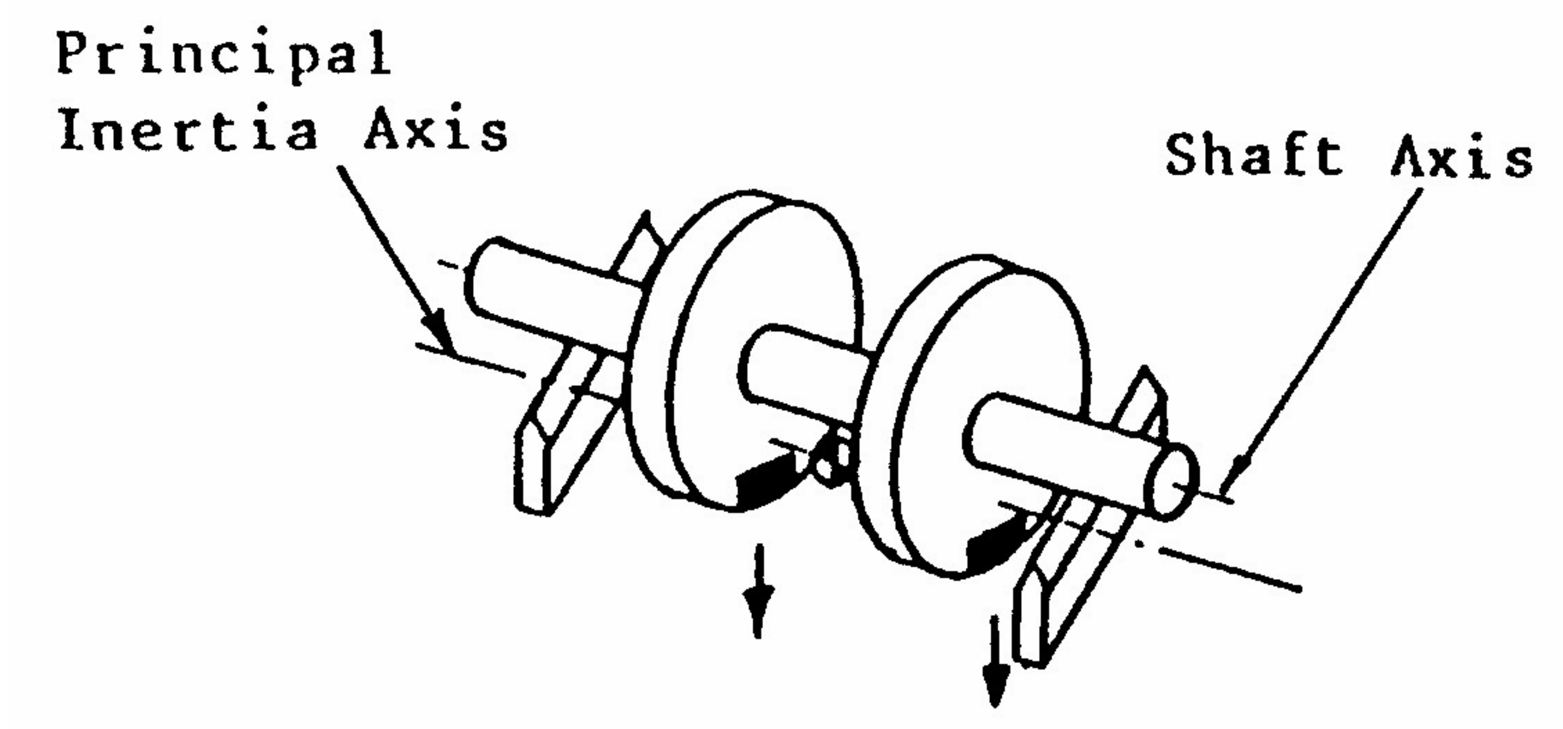
Concentric disc with Static Unbalance

Static Unbalance



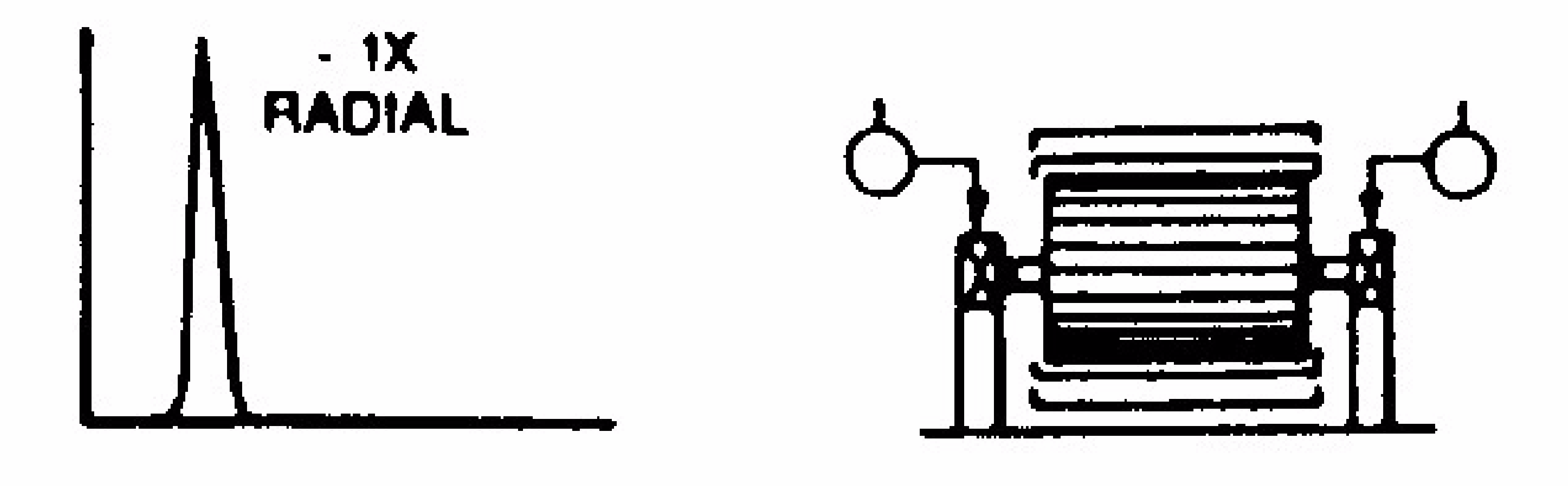
Eccentric disc - Static Unbalance

Static Unbalance



Two discs of Equal Mass and Identical Static Unbalance

Static Unbalance



Static Unbalance vibrations will be in-phase and steady.

Amplitude will increase as the square of the speed of rotation. (3X speed increase = 9X higher vibration)

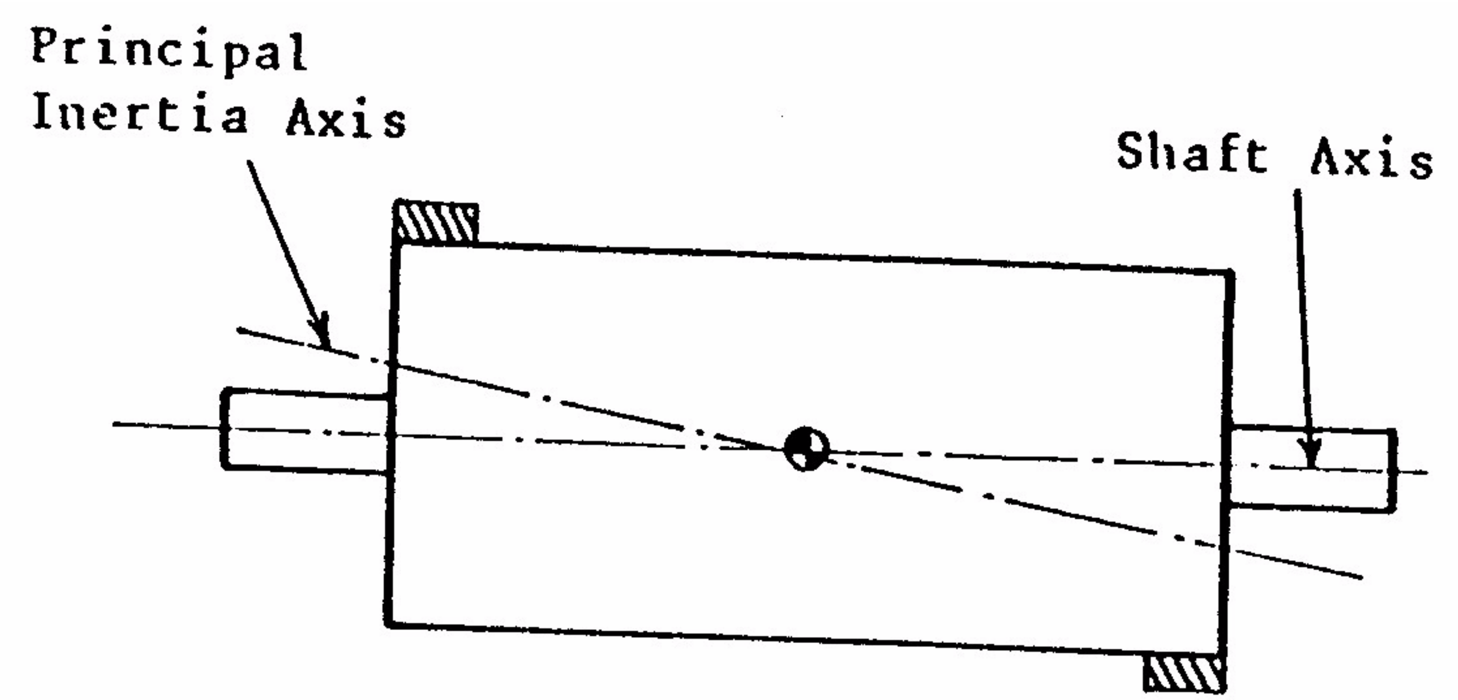
1X RPM vibration is always present and dominates spectrum.

Couple Unbalance

Couple Unbalance (moment unbalance)

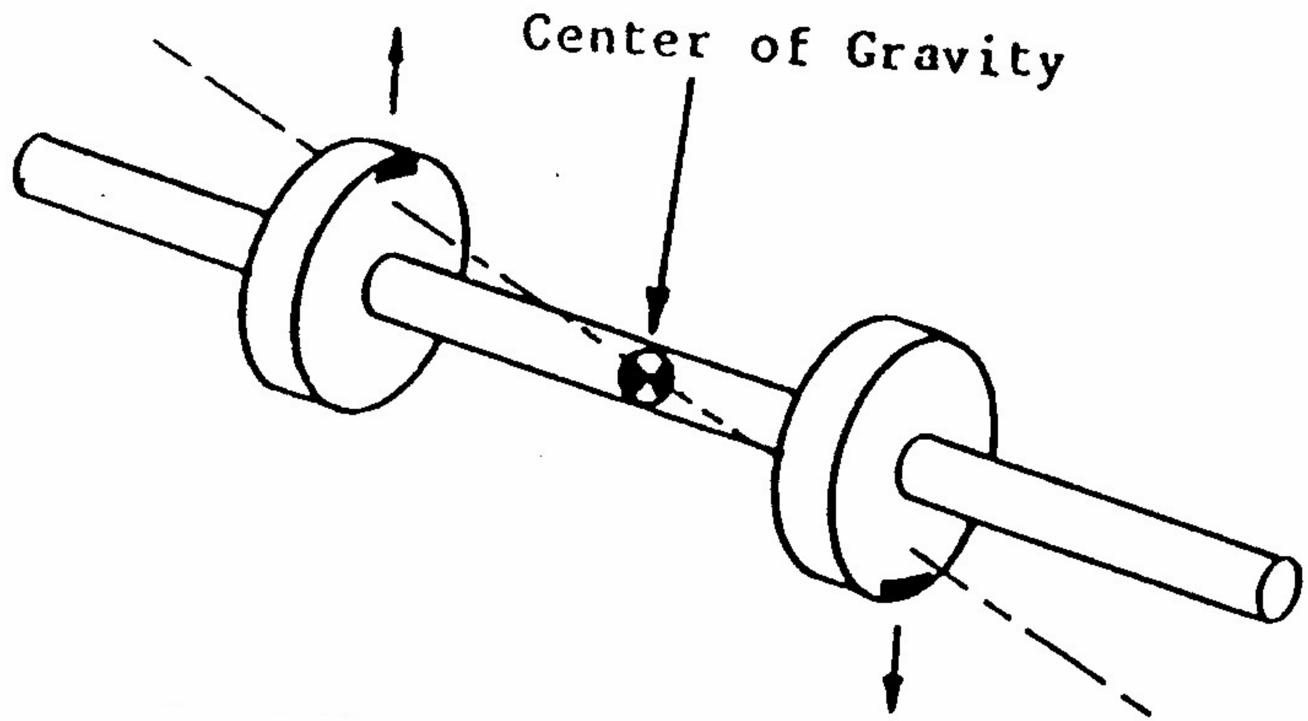
- principal axis of inertia intersects the shaft axis at the centre of gravity
- two equal unbalances at opposite ends of shaft and 180° apart
- dynamic balancing methods needed

Couple Unbalance



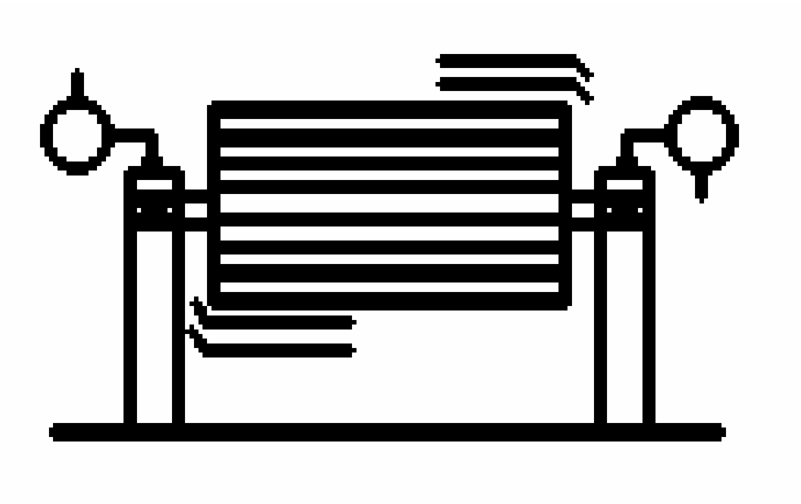
Couple Unbalance in a Solid Rotor

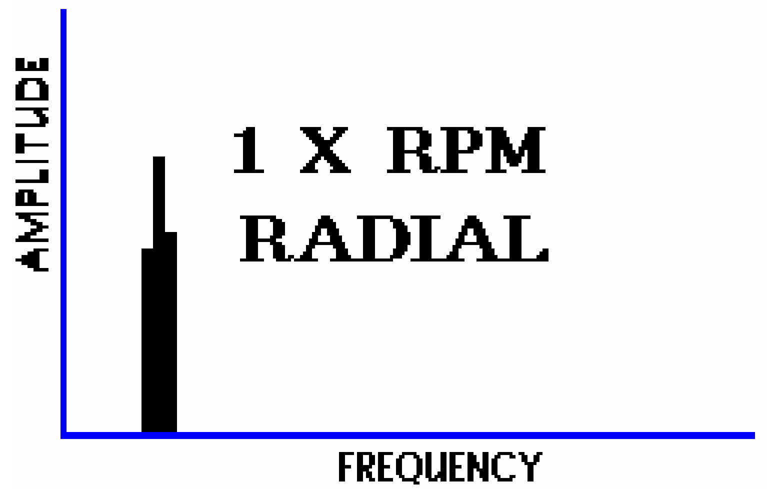
Couple Unbalance



Couple Unbalance in two Discs of Equal Mass

Couple Unbalance





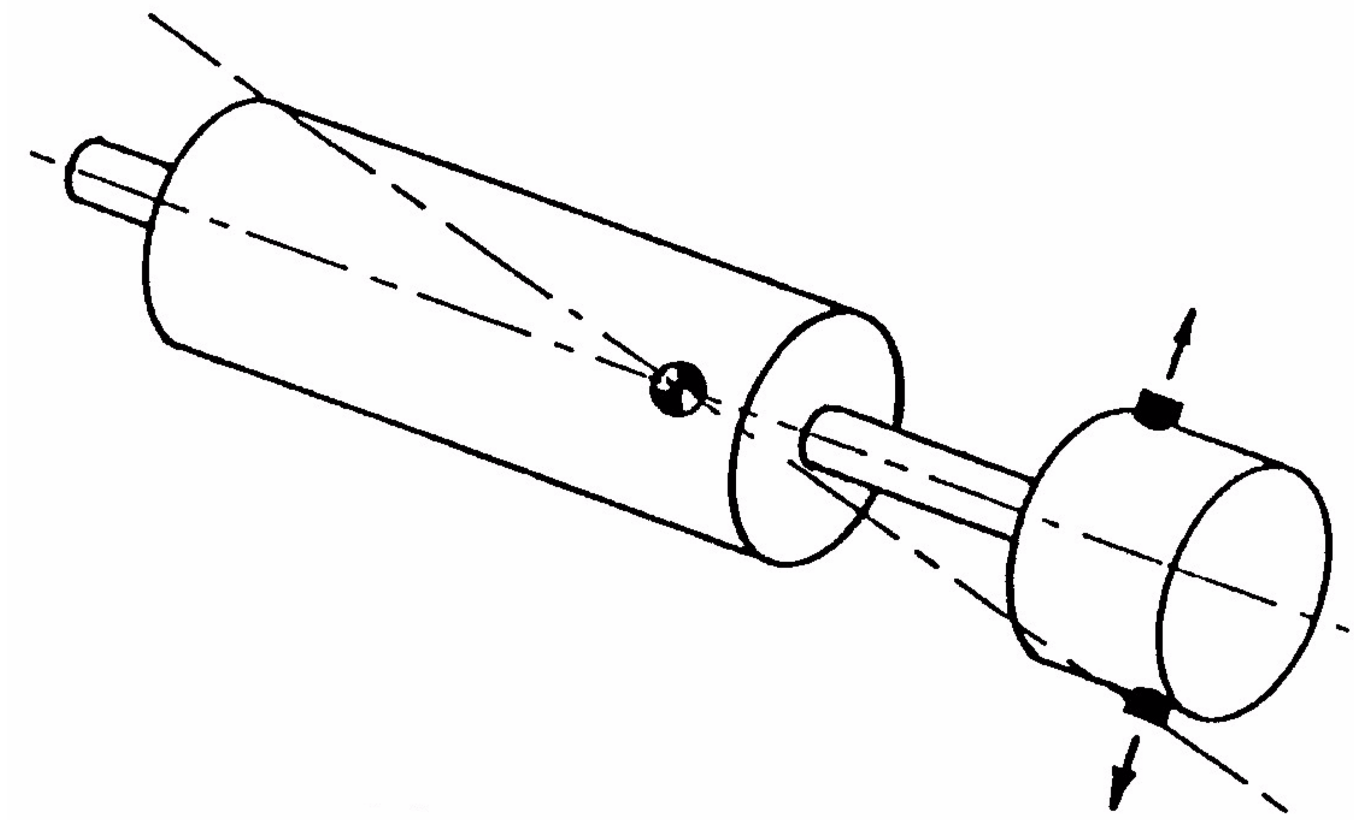
Couple Unbalance vibrations will be 180^{o} out-of-phase.

1X RPM vibration is always present and dominates spectrum.

Amplitude will increase as the square of the speed of rotation.

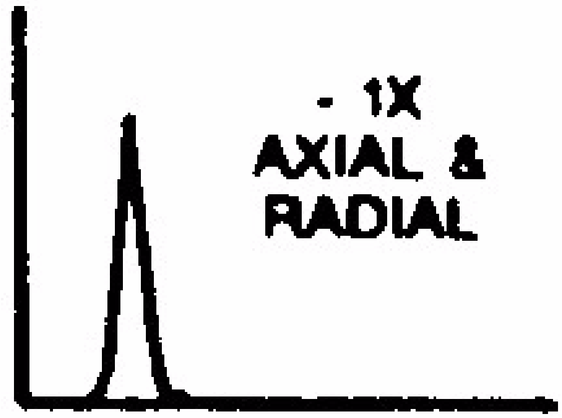
May be high axial vibrations as well as radial.

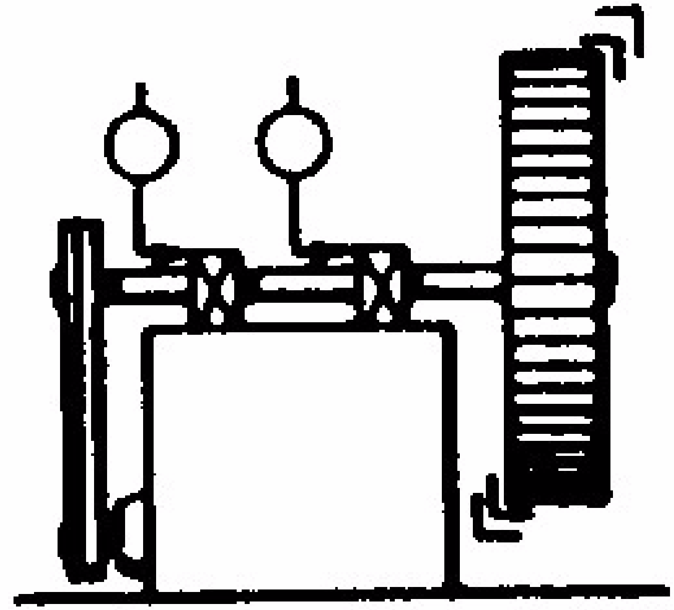
Couple Unbalance



Couple Unbalance in an Outboard Rotor Component

Couple Unbalance





Overhung Rotor Unbalance vibrations are at 1X RPM and in axial and radial directions.

Axial vibrations tend to be in-phase while radial vibrations may have unsteady phase readings.

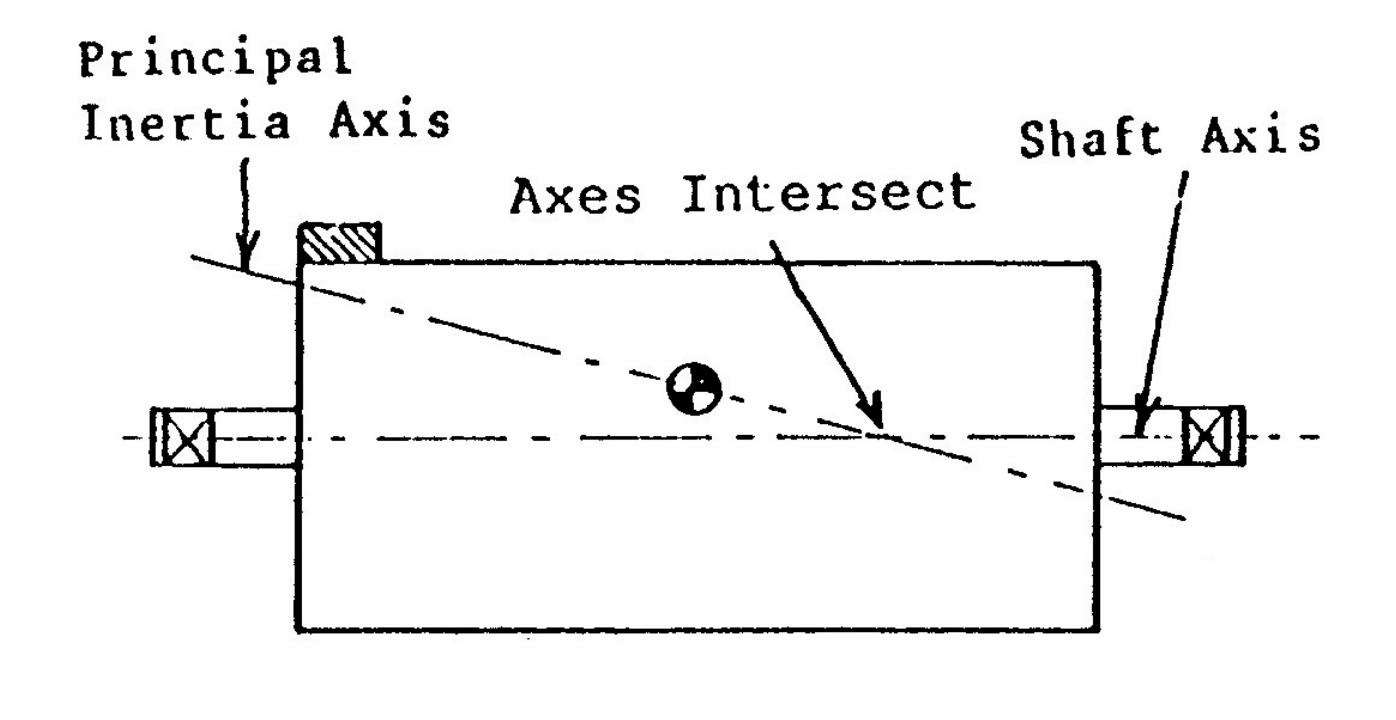
Overhung rotors usually have a combination of static and couple unbalance.

Types of Unbalance

Quasi-Static Unbalance

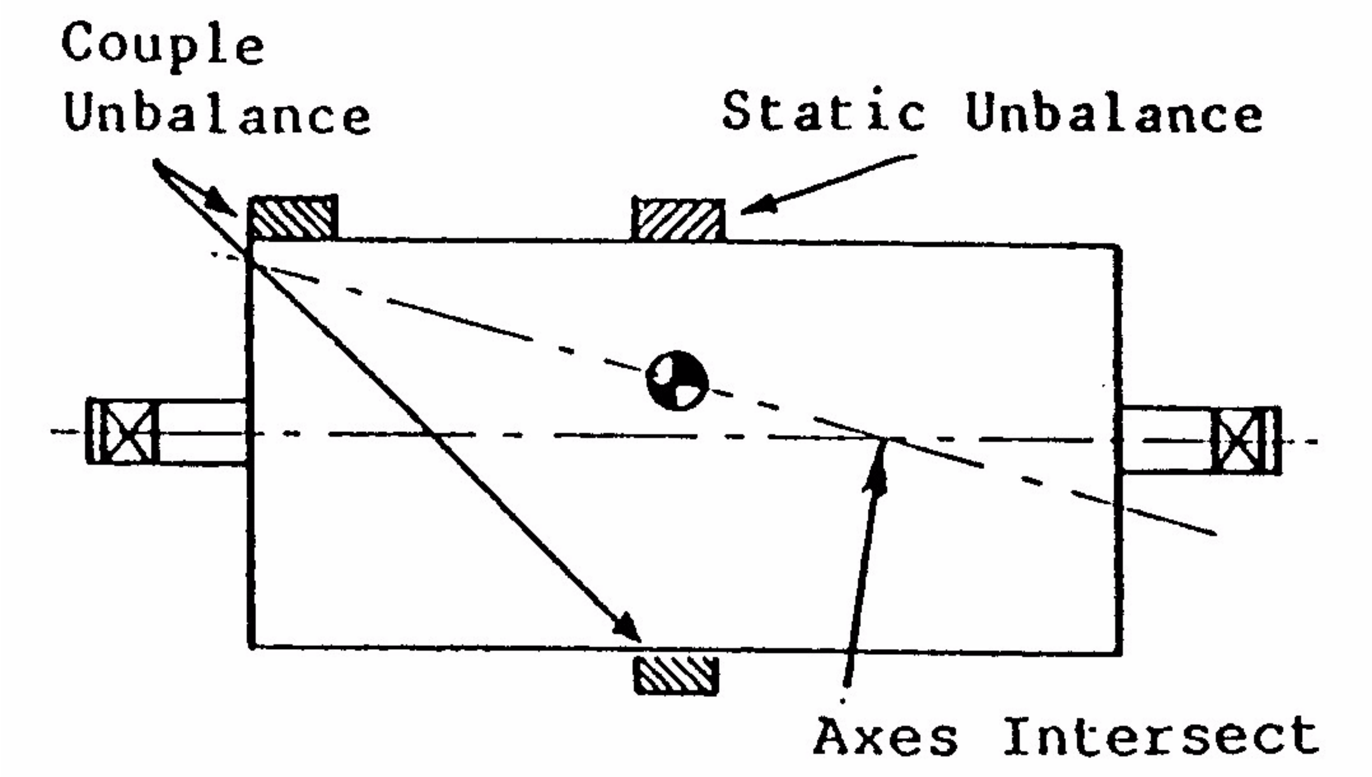
- principal axis of inertia intersects the shaft axis at a point other than the centre of gravity
- combination of static & couple unbalance

Quasi-Static Unbalance



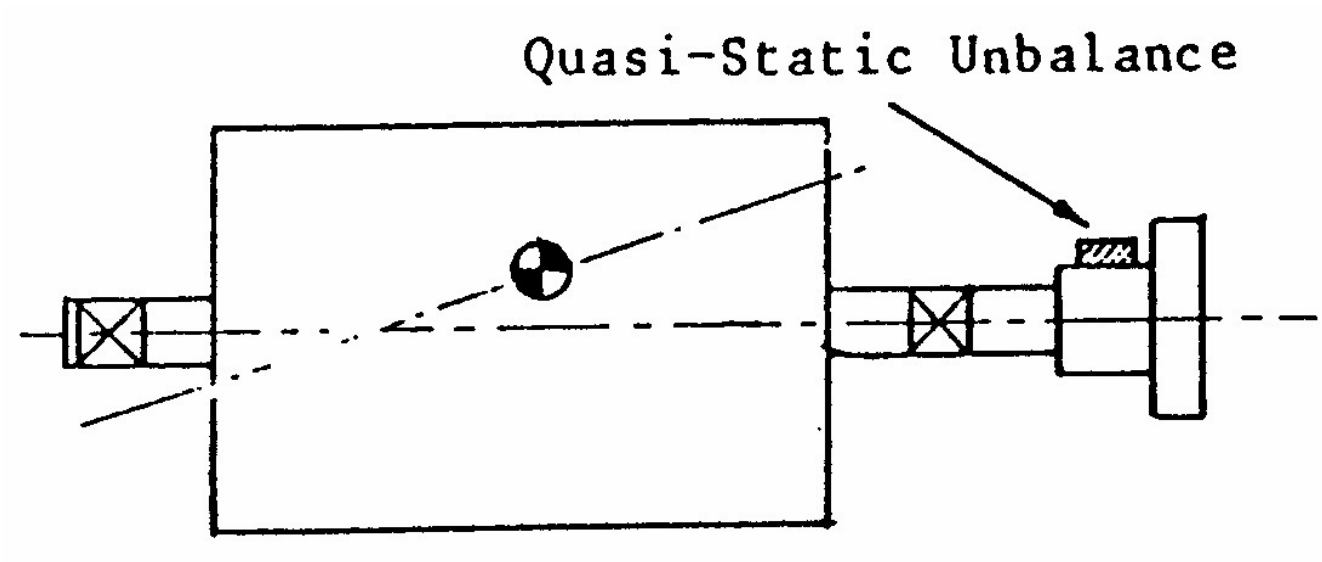
Quasi-Static Unbalance

Quasi-Static Unbalance



Couple Plus Static Unbalance – Quasi-Static Unbalance

Quasi-Static Unbalance



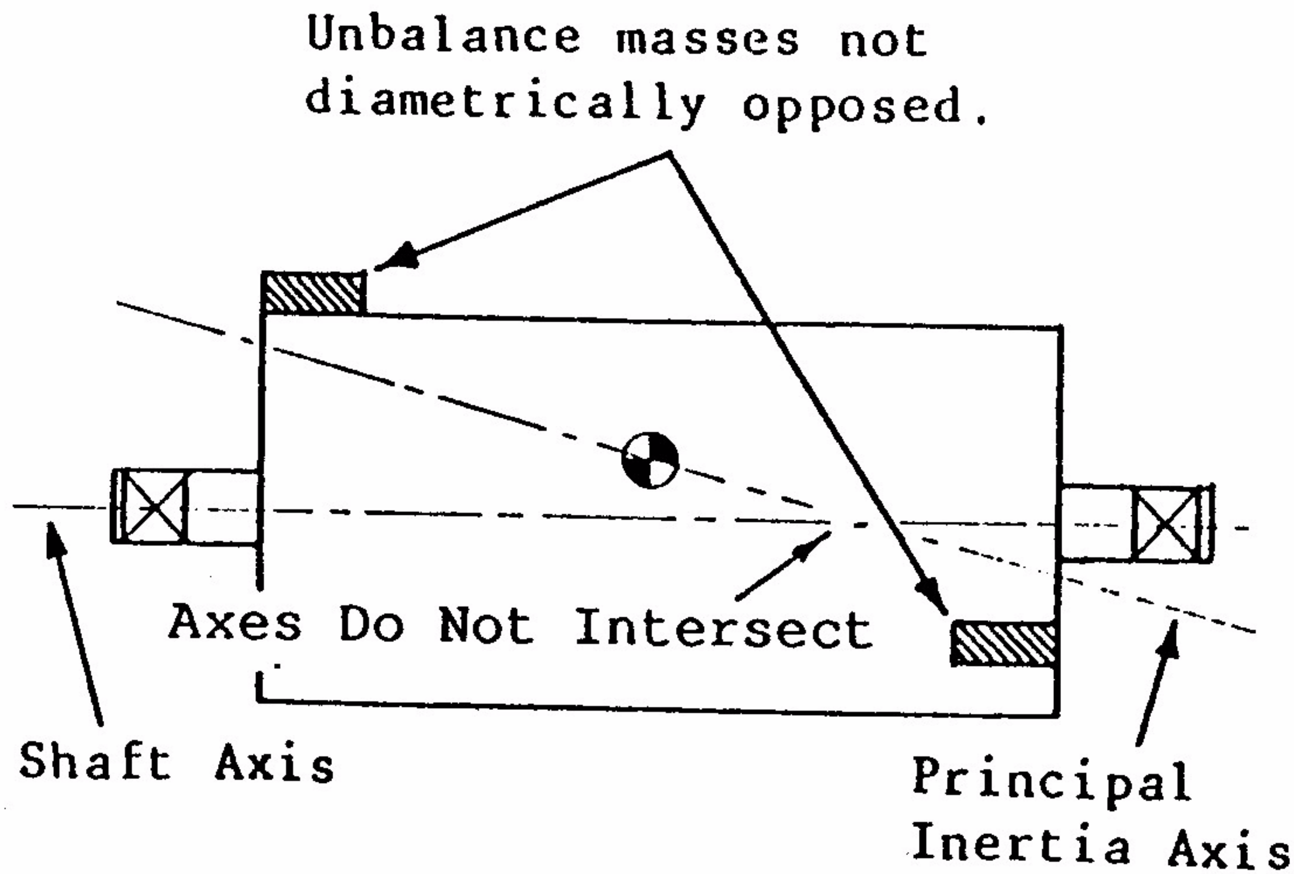
Rotor Assembly with Unbalance in Coupling
– Quasi-Static Unbalance

Types of Unbalance

Dynamic Unbalance

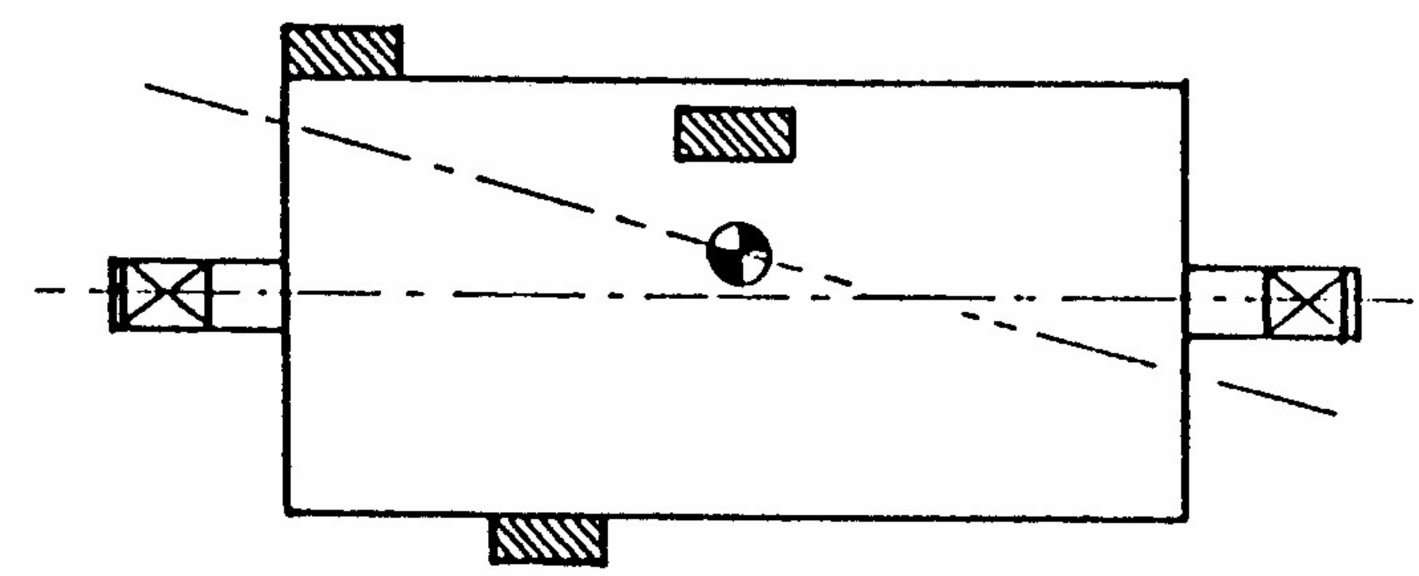
- principal axis of inertia is neither parallel to, nor intersects the shaft axis.
- most common type of unbalance
- corrected in at least two planes $\perp$ to the shaft axis

Dynamic Unbalance



Dynamic Unbalance

Dynamic Unbalance



Couple Plus Static Unbalance – Dynamic Unbalance

Unbalance

Rotor Motions

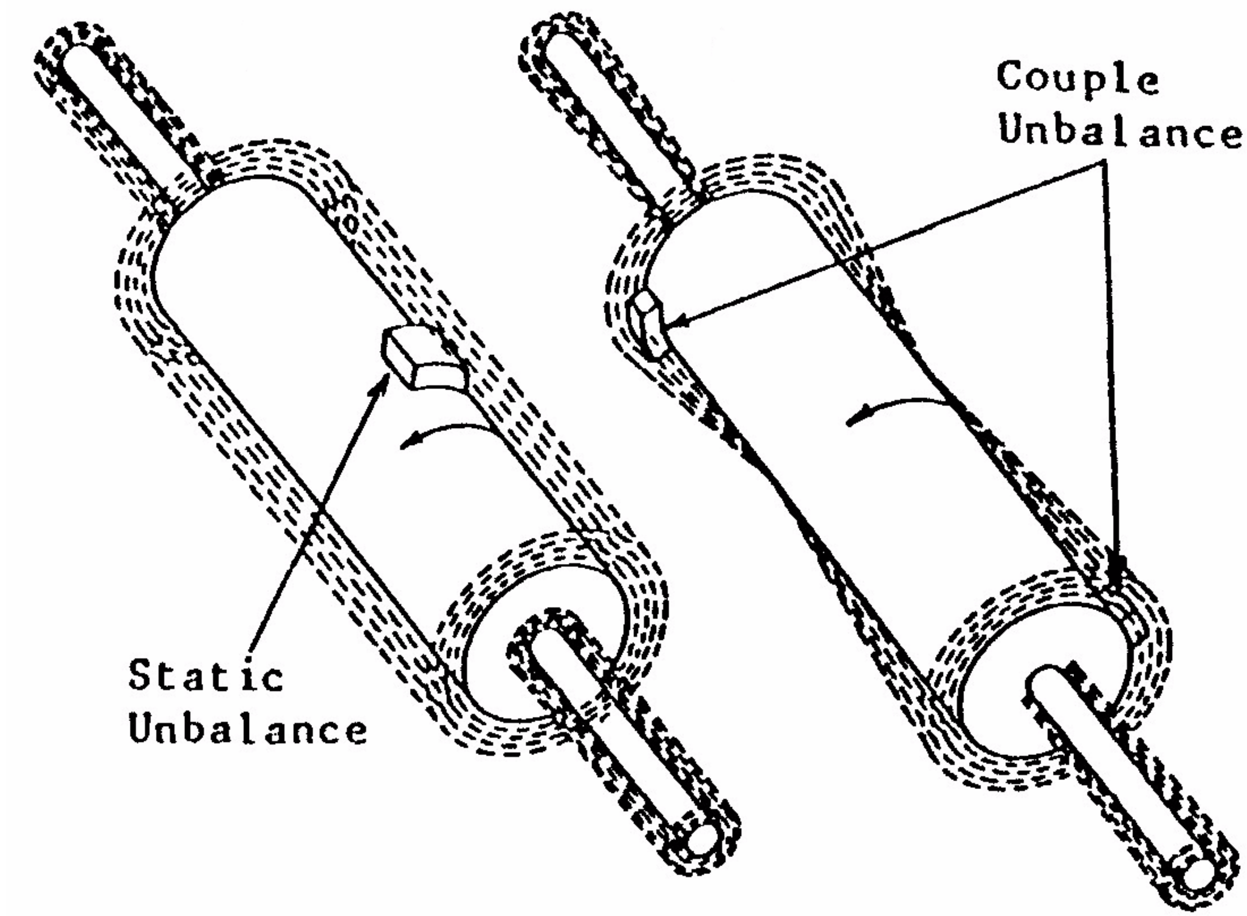
a) in phase - static unbalance

- all points vibrate in the same direction at the same time

b) out of phase - couple unbalance

- points at opposite ends vibrate in opposite directions

Unbalance



Effect of Unbalance on Free Rotor Motion

Unbalance

Rotor Motions

c) quasi - static unbalance (static & couple)

- apex of vibration is moved away from centre of gravity

d) dynamic unbalance - complex

Unbalance

Effects of Rotational Speed

- at low speeds

  high spot (maximum displacement of shaft) at same location as unbalance

- at increased speeds

  the high spot will lag behind the unbalance location

Effects of Rotational Speed on Unbalance

- at first critical speed (first resonance)

    lag reaches 90°

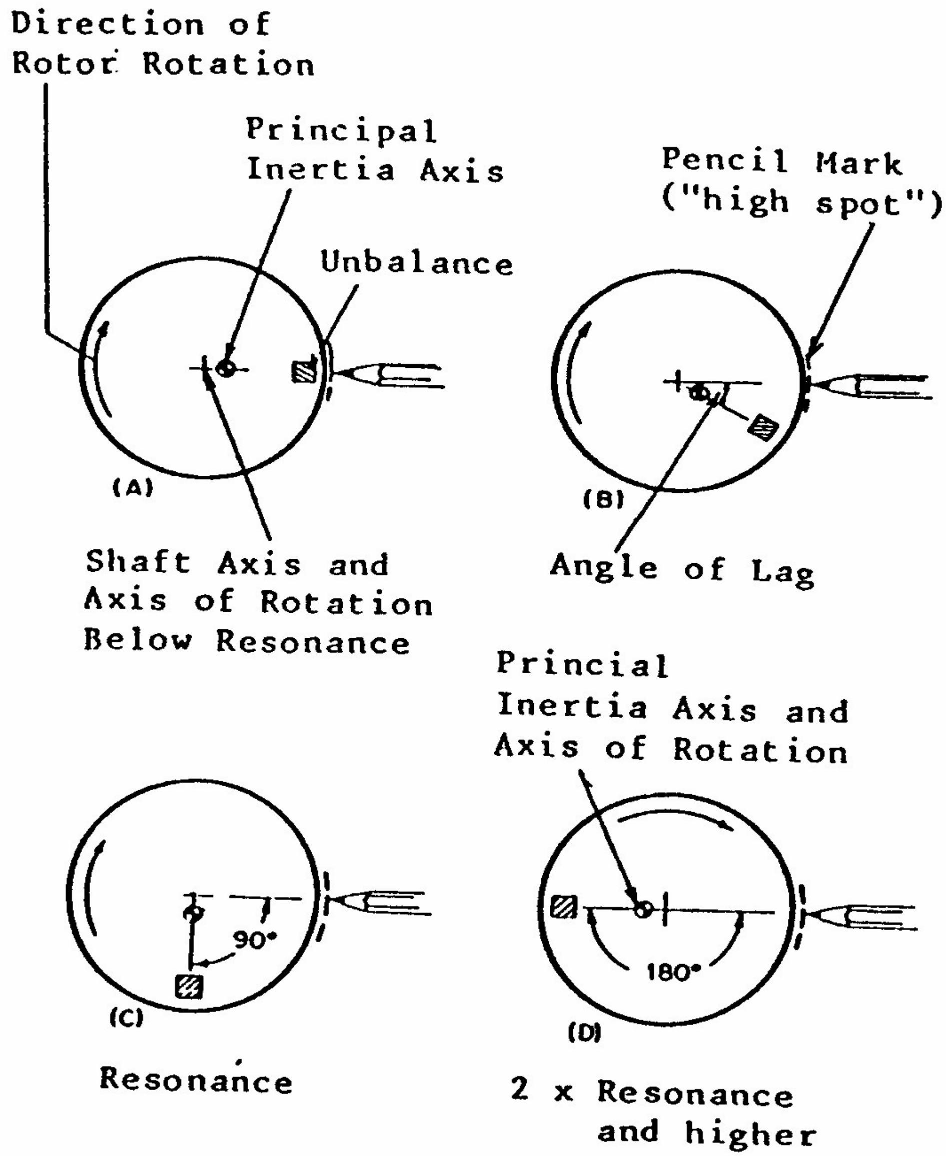
- at 2nd critical and above

    lag reaches 180°

- constant amplitude vibration about principal axis of inertia

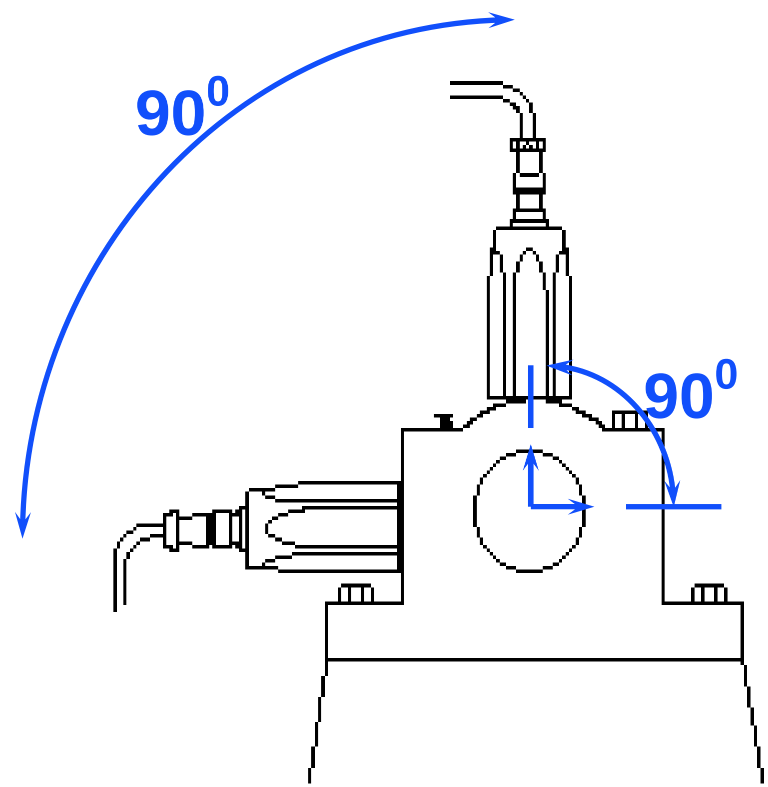
Effects of Rotational Speed on Unbalance

Angle of Lag and migration of Axis of Rotation



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

Correlating Center of Gravity Displacement with Unbalance

- important relationship when correcting for unbalance, setting balancing procedures, and tolerance selection.
- disc shaped rotor - simple
- long rotors - need to make assumptions

Unbalance

Disc Shaped Rotor Example:

- Weight of disc = 999 oz.
- Unbalance mass added = 1 oz.
- Total weight, W = 1000 oz.
- Weight added 10 inches from center of rotation
- Unbalance force, U = 10 in $\times$ 1 oz = 10 oz in.
- C of G displaced by, e = eccentricity

Unbalance

Consider the C of G rotating at a distance *e* about the shaft rotational axis.

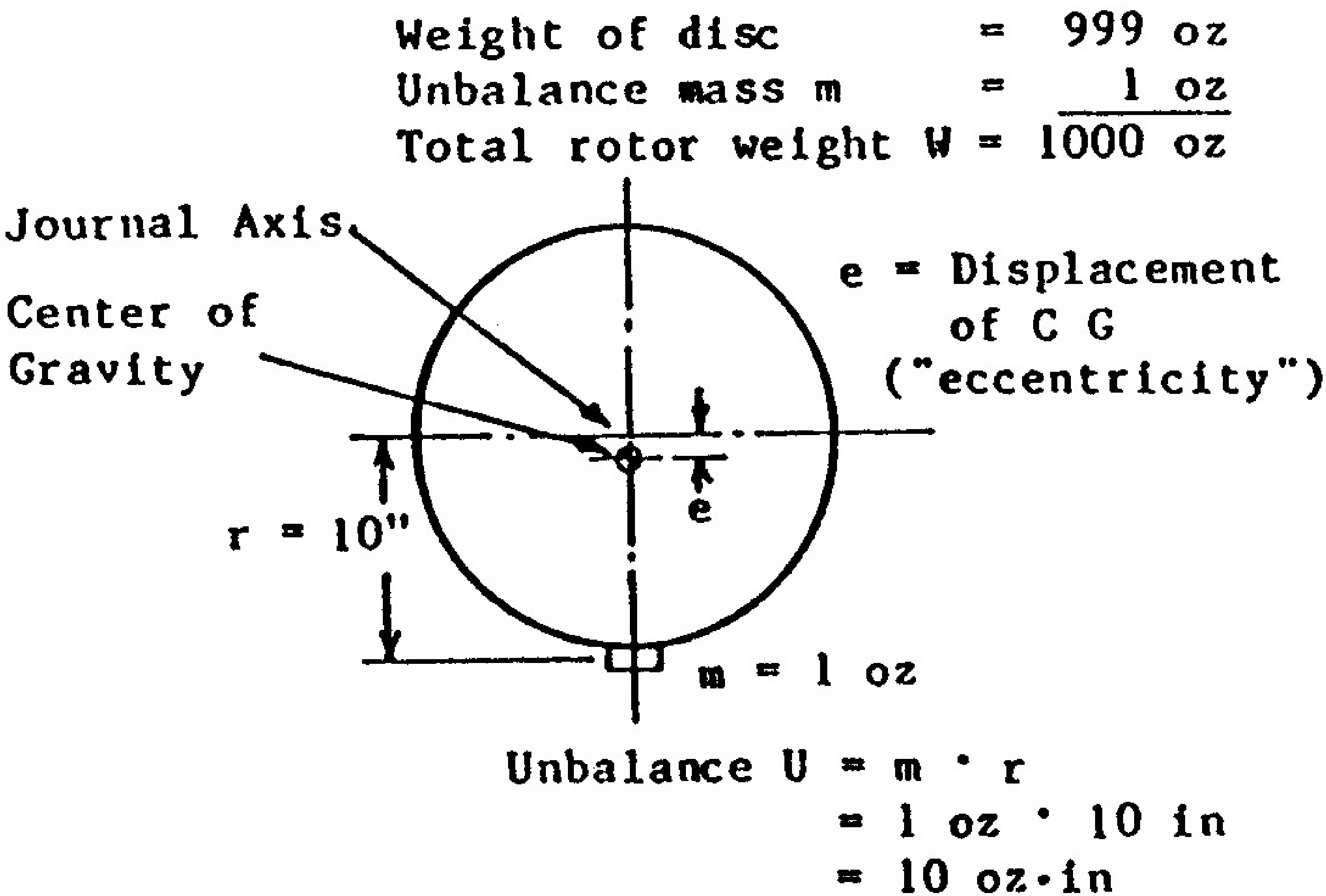
$$U = W \times e$$

$$10 \text{ oz. in} = 1000 \text{ oz.} \times e$$

$$e = \frac{10\ oz.in}{1000\ oz} = 0.01 \text{ in}$$

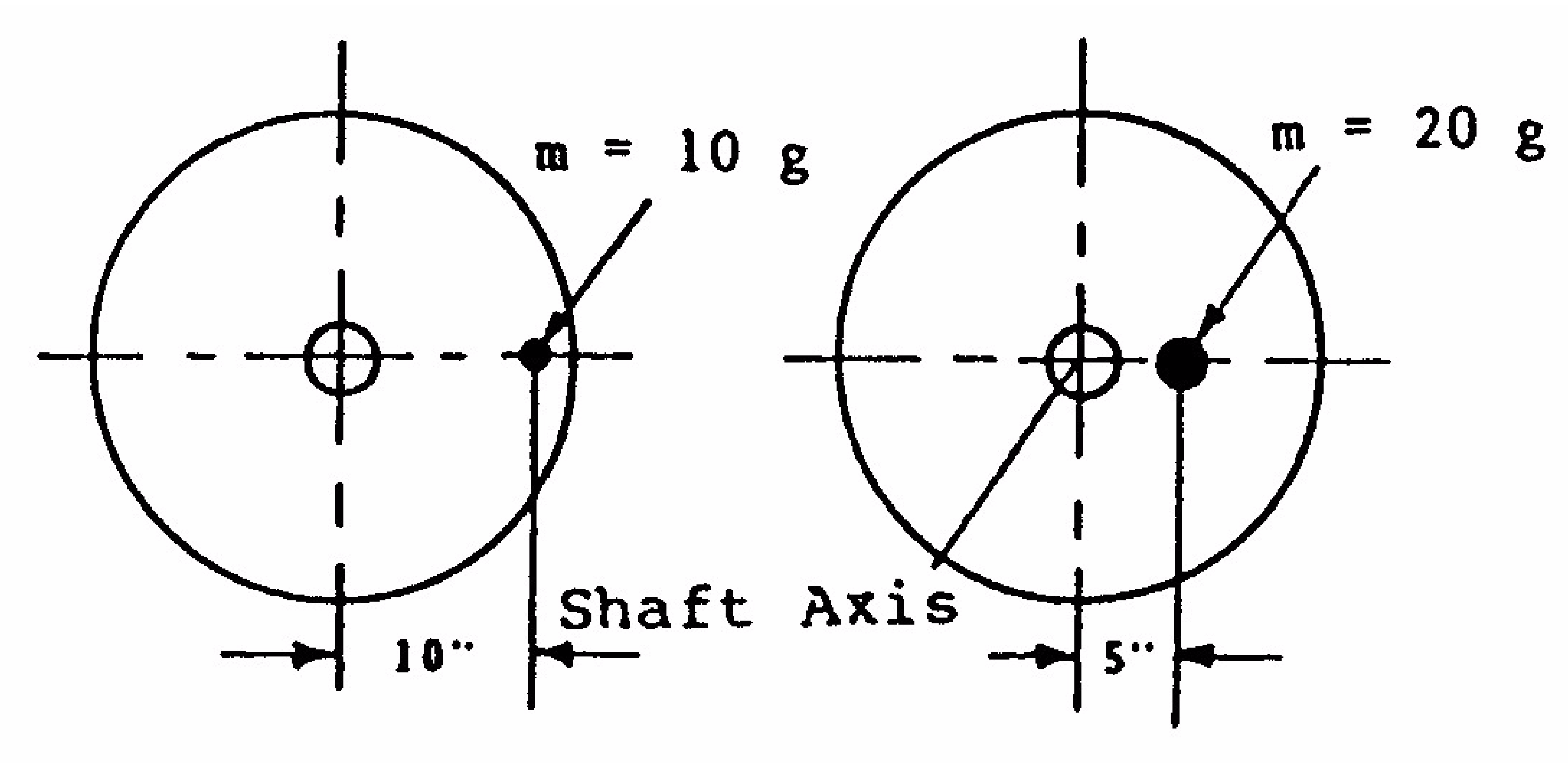
$$e = \frac{U\ \ oz.in}{W\ \ oz.}$$

Unbalance



Weight of disc = 999 oz
Unbalance mass m = 1 oz
Total rotor weight W = 1000 oz
Journal Axis
Center of Gravity
e = Displacement of C G ("eccentricity")
e
r = 10"
m = 1 oz
Unbalance U = m • r
= 1 oz • 10 in
= 10 oz•in

Unbalance



Unbalance

Note:

- displacement, e, is always only 1/2 of the measured relative vibration amplitude.

- for rotors longer than a disc shape,

  $e = \frac{U}{W}$ is true for static unbalance only.

Unbalance

For the case where the unbalance weight is near one end of the rotor:

$$d = \frac{m\,r}{W + m} + \frac{m\,r\,j\,h}{I_x - I_z}$$

d = displacement of principal axis of inertia from shaft axis at the bearing

W = rotor weight

m = Unbalance mass

r = radius of unbalance

h = distance from center of gravity to plane of unbalance

Unbalance

For the case where the unbalance weight is near one end of the rotor:

$$d = \frac{mr}{W + m} + \frac{mrjh}{I_x - I_z}$$

j = distance from center of gravity to bearing

I_x = moment of inertia around transverse axis

I_z = polar moment of inertia around shaft axis

Since I_x & I_z are not usually known, it is acceptable in most cases to assume that the unbalance causes parallel displacement of the principal axis of inertia.

Unbalance

Field Balancing (in situ)

Advantages:

- rotor balanced on own bearings
- balanced at normal operating speed
- balanced at normal load
- rotor driver same as normal operation
- no tear down, re-assembly & re-alignment
- in place trim balance not required

Field Balancing

Advantages:

- down time greatly reduced
- generally simple procedures which require only
  - starting and stopping machine (may be time consuming)
  - adding or removing correction weights

Pre-Balancing Checks

Before starting balancing

- Determine if mass unbalance is the problem by performing a complete vibration analysis
- If mass unbalance is not the problem, look into and correct other problems: excessive bearing clearance, looseness, resonance, and misalignment, etc
- If mass unbalance is the cause, continue with pre-balancing checks

Pre-Balancing Checks

- Nature of unbalance problem (do a vibration analysis)
- Determine whether or not the rotor is clean
- Assess rotor stability (structural, thermal)
- Determine critical speeds (star-up/coast-down tests)
- Locate balance weights already in place
- Know details of balance planes or rings

Measurements

Sensors for vibration

- Proximity probes – the most direct measure
- Velocity transducers – indirect measure
- Accelerometers – indirect measure

Sensors for phase

- Strobe light – less accurate phase reading
- Photoelectric sensor –accurate phase reading
- Proximity probe – accurate phase reading

Field Balancing

Single Plane Balancing

- dynamic balance conducted in only one plane
- the centrifugal force developed by an unbalance is

$$F_c = \omega^2 \times \frac{U_b}{g}$$

$$U_b = M \times r$$

U_b = Unbalance (oz. in)

M = weight of unbalance (oz.)

r = radius from rotor center to M (in.)

Field Balancing

Single Plane Balancing

$\omega = 2 \pi f$

f = frequency (Hz)

g = acceleration due to gravity (ft/sec^2)

F_c = centrifugal force

Note:

The shaft motion induced by an unbalance will be orbital, that is, the shaft center will move in a circular path. The vibration transducer only sees the motion that is parallel with its principal axis of operation.

Field Balancing

Single Plane Balancing – Vibration Measurements

The instantaneous vibration amplitude measured by a displacement transducer is,

$$d = \frac{D}{2} \times \sin(\omega t)$$

d = instantaneous displacement

$\omega = 2 \pi f$,   f = frequency (Hz) , t = time (sec)

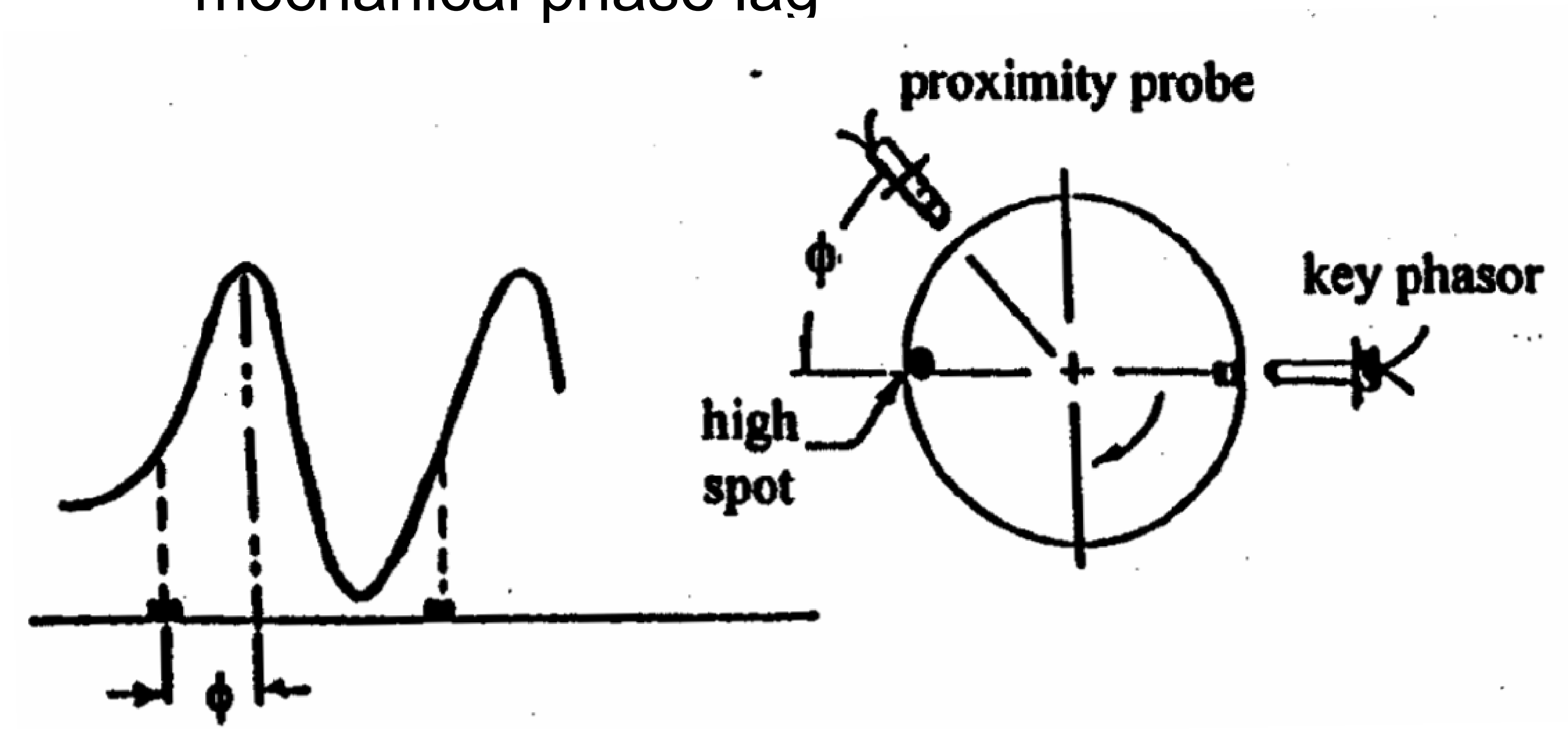
D = Peak-to-Peak displacement

Note: using displacement for balancing simplifies phase measurements and calculation of correction weight placement

Field Balancing

Proximity probe measurement

- No electronic phase lag between signals with mechanical phase lag



Field Balancing

Single Plane Balancing – Vibration Measurements

The instantaneous vibration amplitude measured by a velocity transducer is,

$$v = \frac{d}{dt}d = \frac{\omega D}{2} \times \cos(\omega t)$$

v = instantaneous velocity

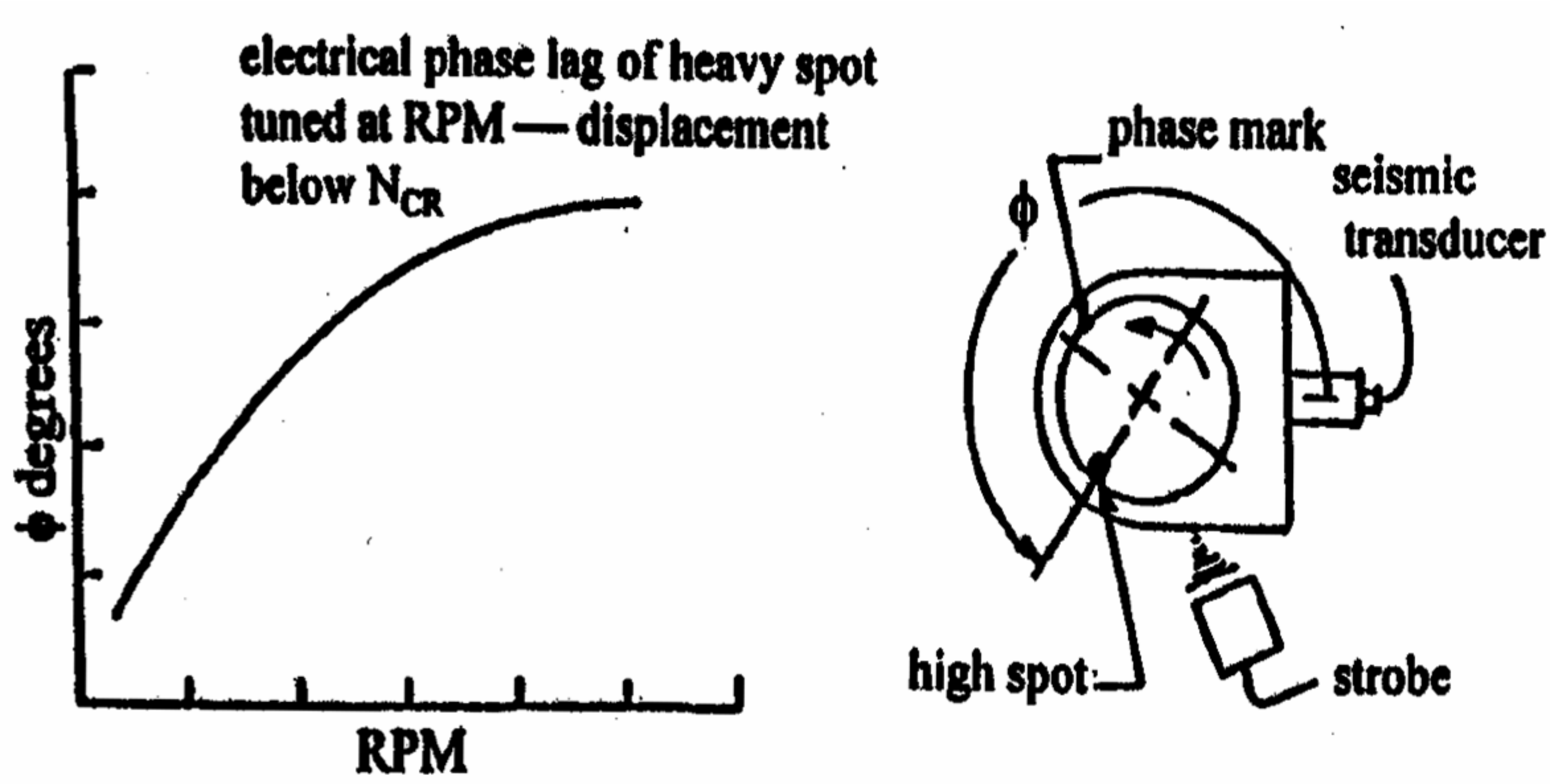
$\omega = 2\,\pi\,f$, f = frequency (Hz) , t = time (sec)

D = Peak-to-Peak displacement

Field Balancing

Strobe/velocity measurement

- With both electronic and mechanical phase lags



Field Balancing

Single Plane Balancing – Vibration Measurements

The instantaneous vibration amplitude measured by an accelerometer transducer is,

$$a = \frac{d}{dt} v = \frac{-\omega^2 D}{2} \times \sin(\omega t)$$

a = instantaneous acceleration

$\omega = 2 \pi f$, f = frequency (Hz) , t = time (sec)

D = Peak-to-Peak displacement

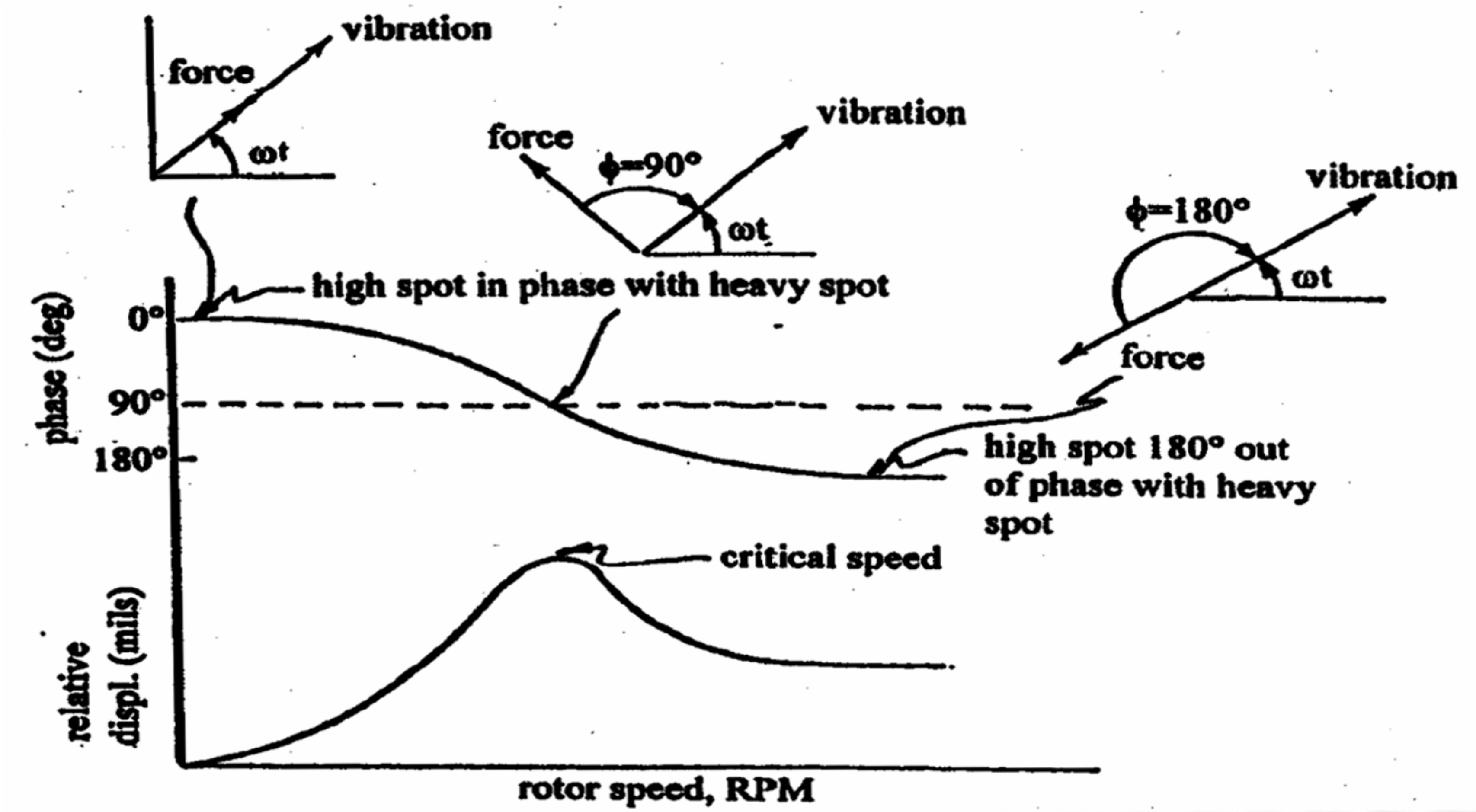
Mass Unbalance and Phase Relationships

The force by mass unbalance (heavy spot) leads the vibration peak (high spot) by 0º to 180º depending the operating speed

- 0º - 90º : operating speed is less than the first critical speed
- 90º: operating speed is close to the first critical speed
- 90º - 180º: operating speed is beyond to the first critical speed

Mass Unbalance and Phase Relationships

Heavy/high spot relationship – mechanical phase lag



Unbalance

Single Plane Balancing – Procedure

1 - “as is” run

2 - trial weight run

3 - calculate and make permanent correction

4 - the final run

Single Plane Balancing

Single Plane Balancing – the “as is” run

- provides baseline data
- ensure repeatability
- record vibration amplitude and phase

Example: Vibration Amplitude = 10 at 30° (10 mils peak-to-peak at 30° phase relative to some reference)

We wish to know - location and weight of unbalance causing this vibration

Single Plane Balancing

Single Plane Balancing – the trial weight run

- this tells us how a known unbalance weight in a known location affects rotor dynamics
- how closely the machine is operating to a resonance is important (just below, just above, well above)
- watch phase change immediately after shut down & before coast down
- dramatic changes in phase mean operational speed is close to a resonance

Single Plane Balancing

- when calculating the unbalance weight include an extra 10% onto the weight of the rotor to account for vibrations absorbed into the bearings & supports
- the trial weight is equal to the unbalance weight divided by an amplification factor (determined from the location of the operating frequency on the system response curve)
- add trial weight opposite existing unbalance location
- collect trial weight run vibration data

Single Plane Balancing

Some general guidelines

- A trial weight should not yield a force of more than 10 % the static weight of the rotor
- Trial weight (W_T) calculation formula

$$W_T = 56{,}375.5\ (W/N^2e)\ \text{(ounces)}$$

  W - rotor weight (lb), N – rotor speed (RPM), e – trial weight eccentricity (in)

- If no vibration response is obtained, either the trial weight is too small or the problem is not mass unbalance

Single Plane Balancing

Single Plane Balancing – final calculation

Correction = $\dfrac{\textit{initial unbalance vector}}{\textit{trial weight vector}} \times$ Magnitude of trial weight

Where: the trial weight resultant vector

= (trial weight vector) + (initial unbalance vector)

Single Plane Balancing

Single Plane Balancing – final run

- restart machine to check vibration levels
- if unacceptable treat as a new “as is” run
- do not tinker with weights already added.

Single Plane Balancing

Single-plane vector balancing procedure with trial weight (review)

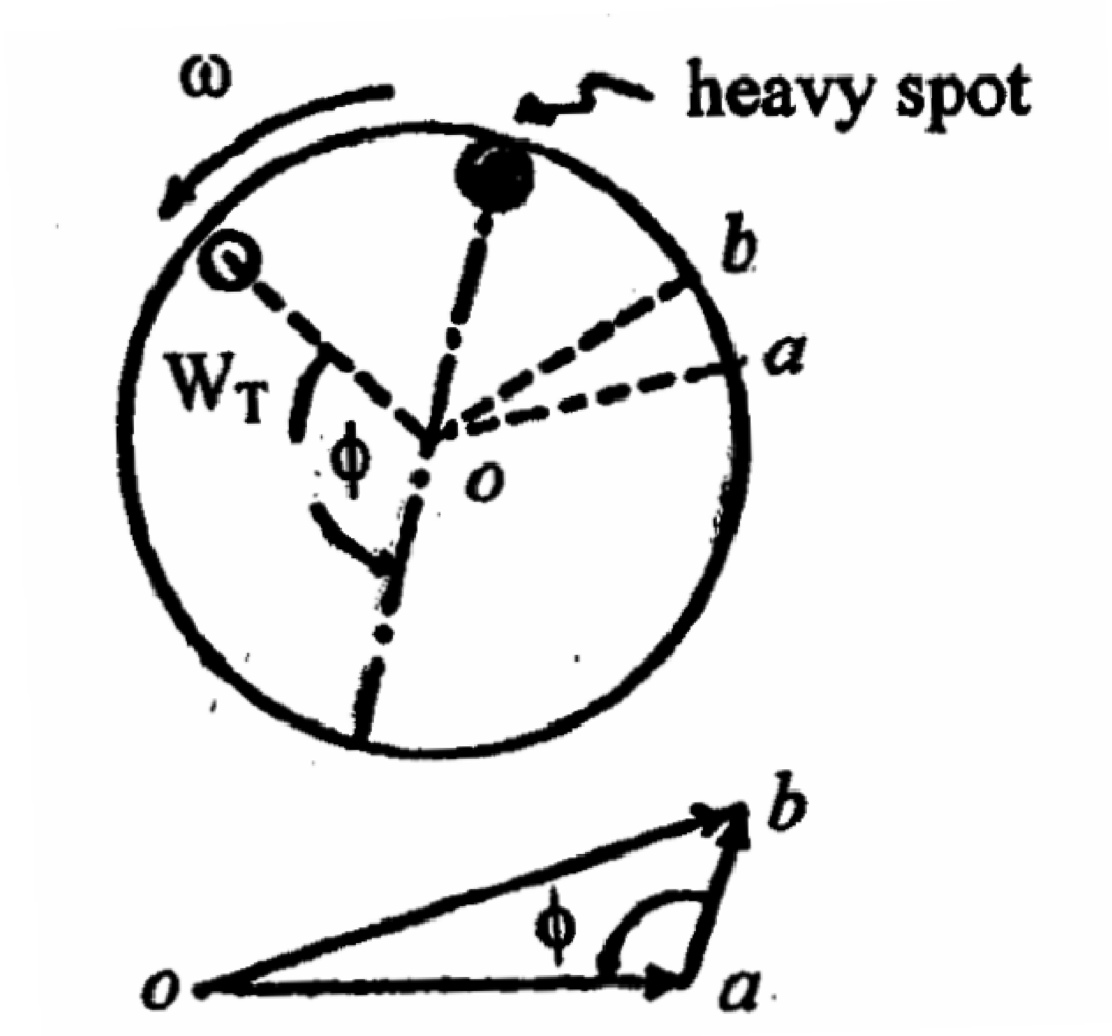
- Measure and record signal
- Install trial weight
- Measure and record trial run
- Calculate vectors
- Correct trial weight
- Measure and record trial run

Vector Method with Trial Weight

Single-plane balancing – a simple example

- Mark the high spot (***a***) and its amplitude (***oa***)
- Place trial weight $\mathbf{W_T}$ at selected direction and mark the new high spot (***b***) and its amplitude (***ob***)
- The vector difference ***ab = ob – oa*** is the effect of $\mathbf{W_T}$ alone
- Move the W_T in the same direction and angle ϕ to make (***ab)*** parallel to and opposite (***oa)***
- The trial weight is increased or decreased in the ratio (***oa/ab)*** equal to the original unbalance

Vector Method with Trial Weight



Vector Method with Trial Weight

Detailed procedures for constructing a vector diagram for single-plane balancing

1. Mark the direction of rotor rotation on the graph
2. Mark the direction of positive phase angle
3. Establish a scale of numbers of mils per division so the vectors are large but do not exceed the graph
4. The original vibration **O** (5 mils at 190º in Figure) is plotted on the graph

Vector Method with Trial Weight

5. The location of the trial weight (**W_T**) is plotted (30º) and its size (75 grams) are noted on the graph
6. Plot the vibration (**O+T**) obtained after the trial weight has been added to the rotor. The rotor must be operated at the same speed as when the original data (**O**) were acquired
7. The difference between **O** and (**O+T**) is the effect of the trial weight

Vector Method with Trial Weight

8. The effect of the trial weight is obtained by drawing a line between **O** and (**O+T**)
9. **O + T** must be equal to (**O+T**). The arrow on T must point to (**O+T**). Vectors add heads to tails and subtract heads to heads
10. T is now repositioned with its tail at the origin by moving it parallel and maintaining the same length
11. Draw a line opposite **O** from the origin

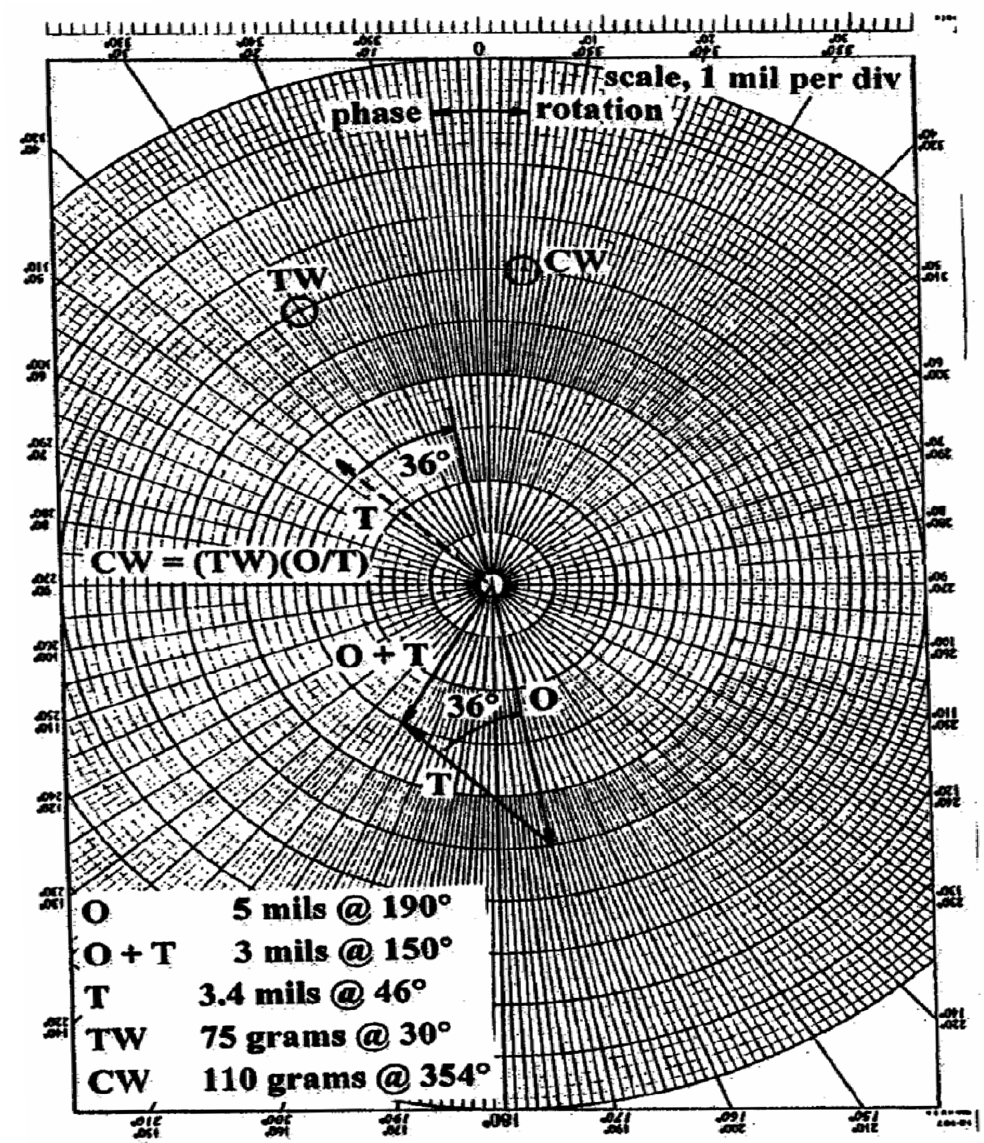
Vector Method with Trial Weight

12. The goal in balancing is to add a trial weight that will create a **T** vector directly opposite and equal to **O**
13. The angle between T and the line opposite **O** 36º determines how far and in what direction the trial weight must be moved
14. The trial weight is multiplied by the ratio of the original vibration to the effect of the trial weight (5/3.4)angle between **T** and the line to determine the balance weight

$$75 \text{ g } (5/3.4) = 110 \text{ g}$$

Vector Method with Trial Weight

- Procedure of single-plane balancing

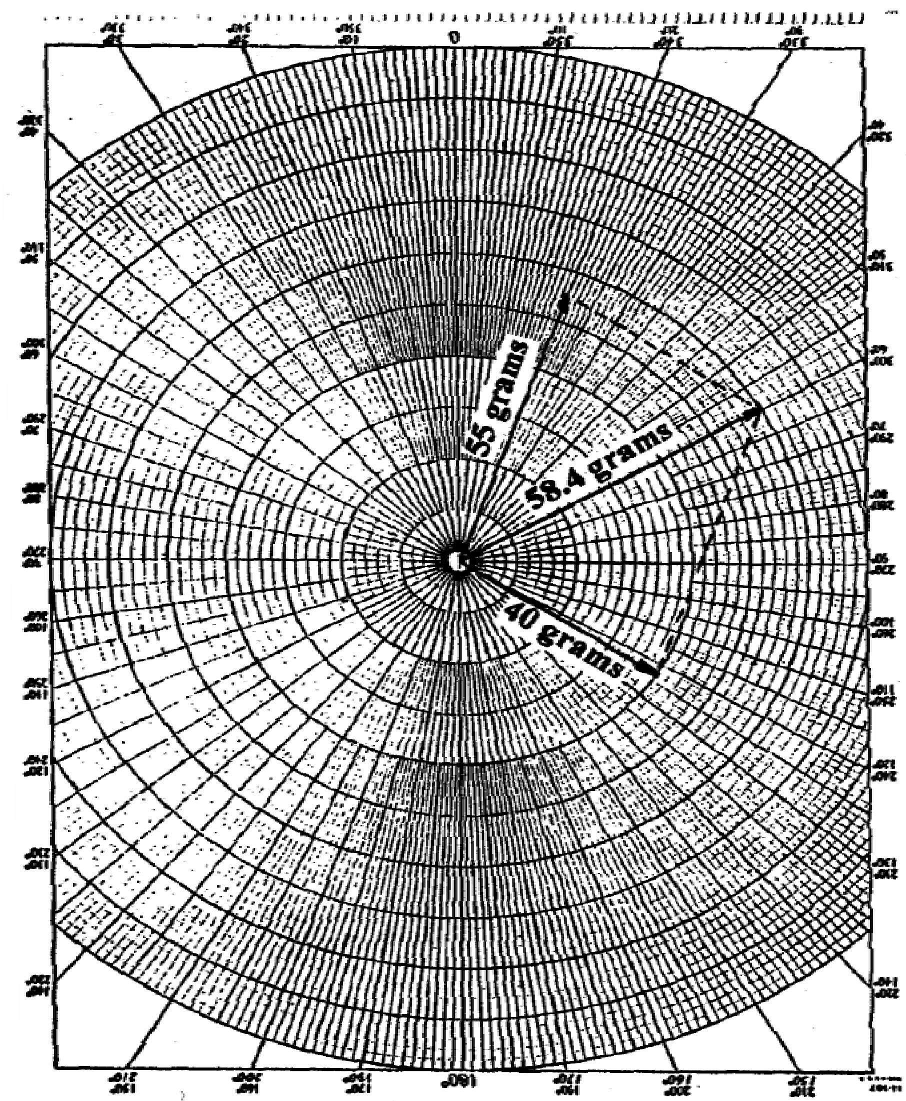


Weight Splitting and Consolidation

- Weight splitting is to place the weights at the desired locations (***a*** and ***b***)
- A parallel rule can be used to determine graphically the magnitudes of the weights at desired locations (***a*** and ***b***) by the lengths of the vectors
- Weight combination is the inverse process used to determine the location and magnitude of the combined weight

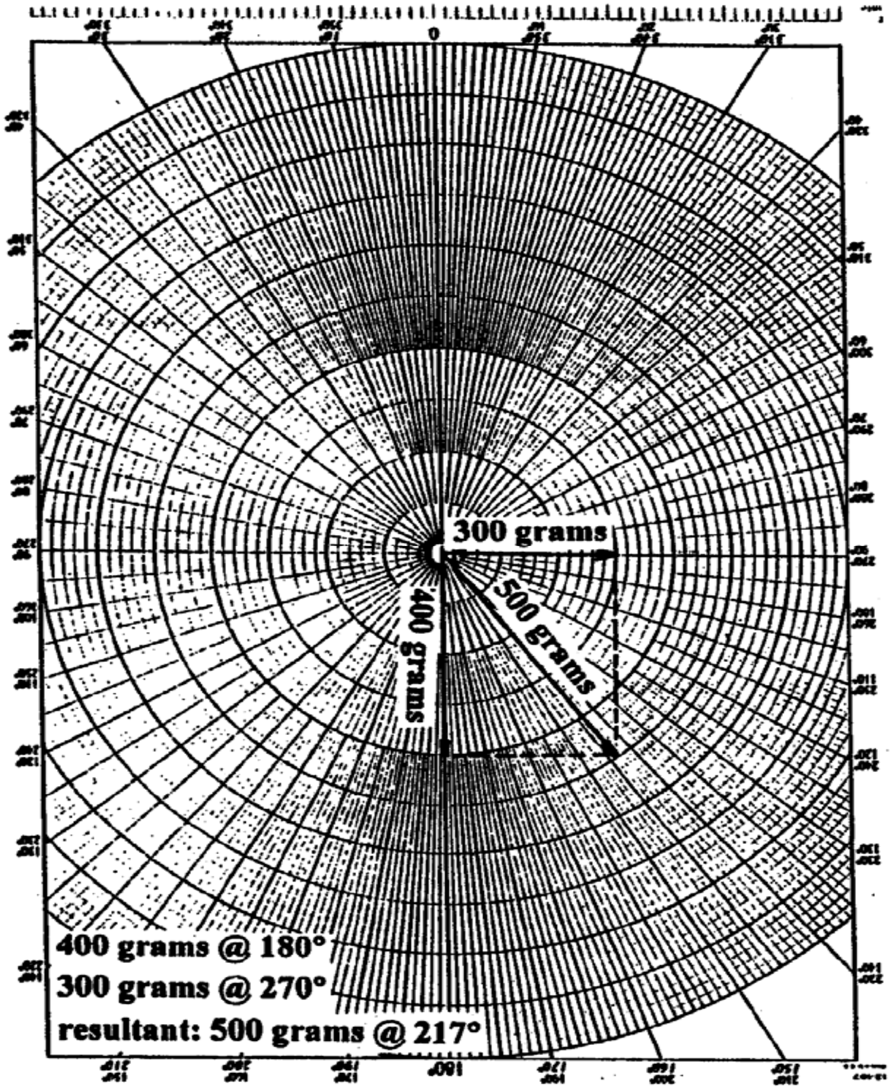
Vector Method with Trial Weight

- Weight splitting



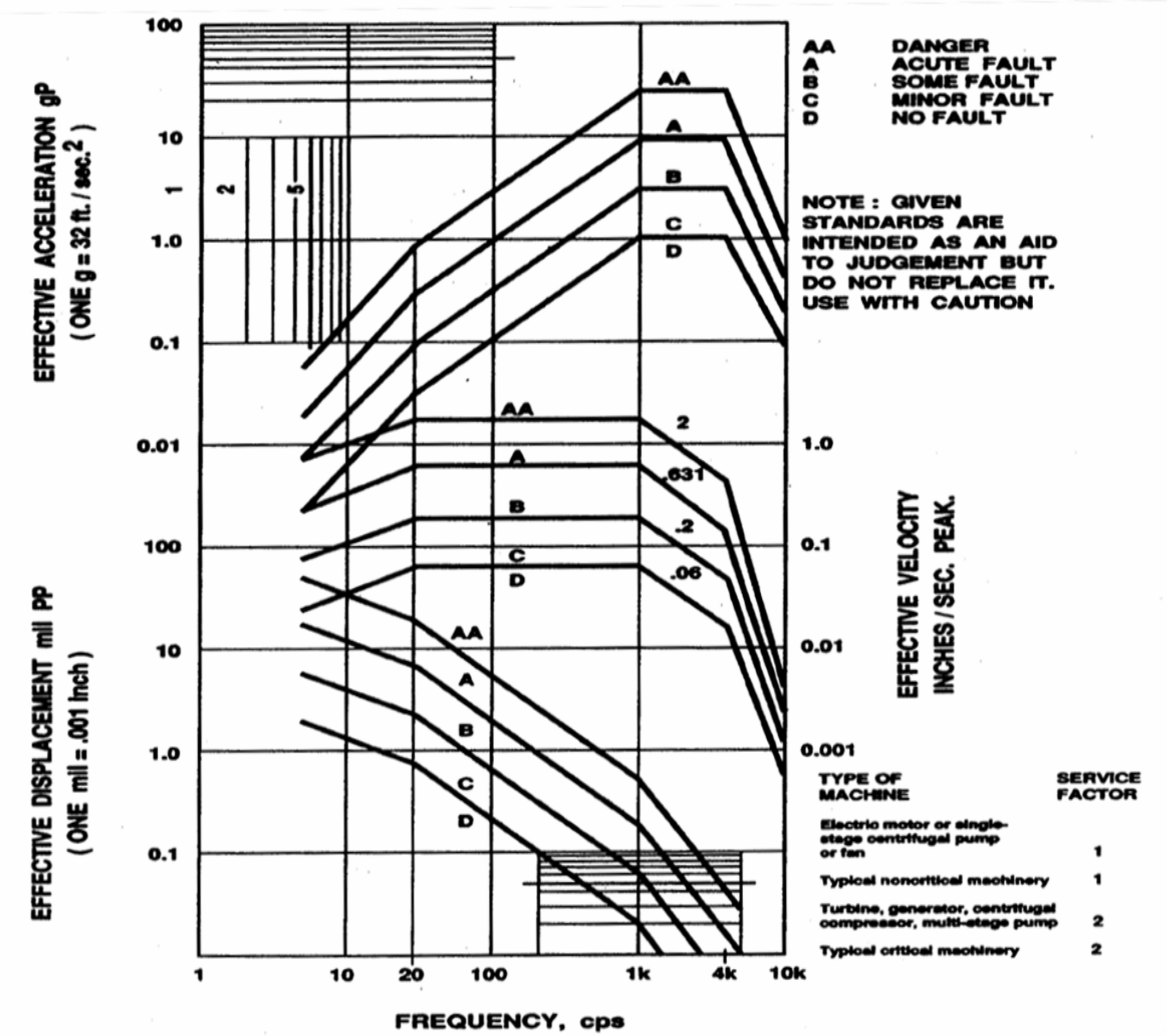
Vector Method with Trial Weight

- Weight combination



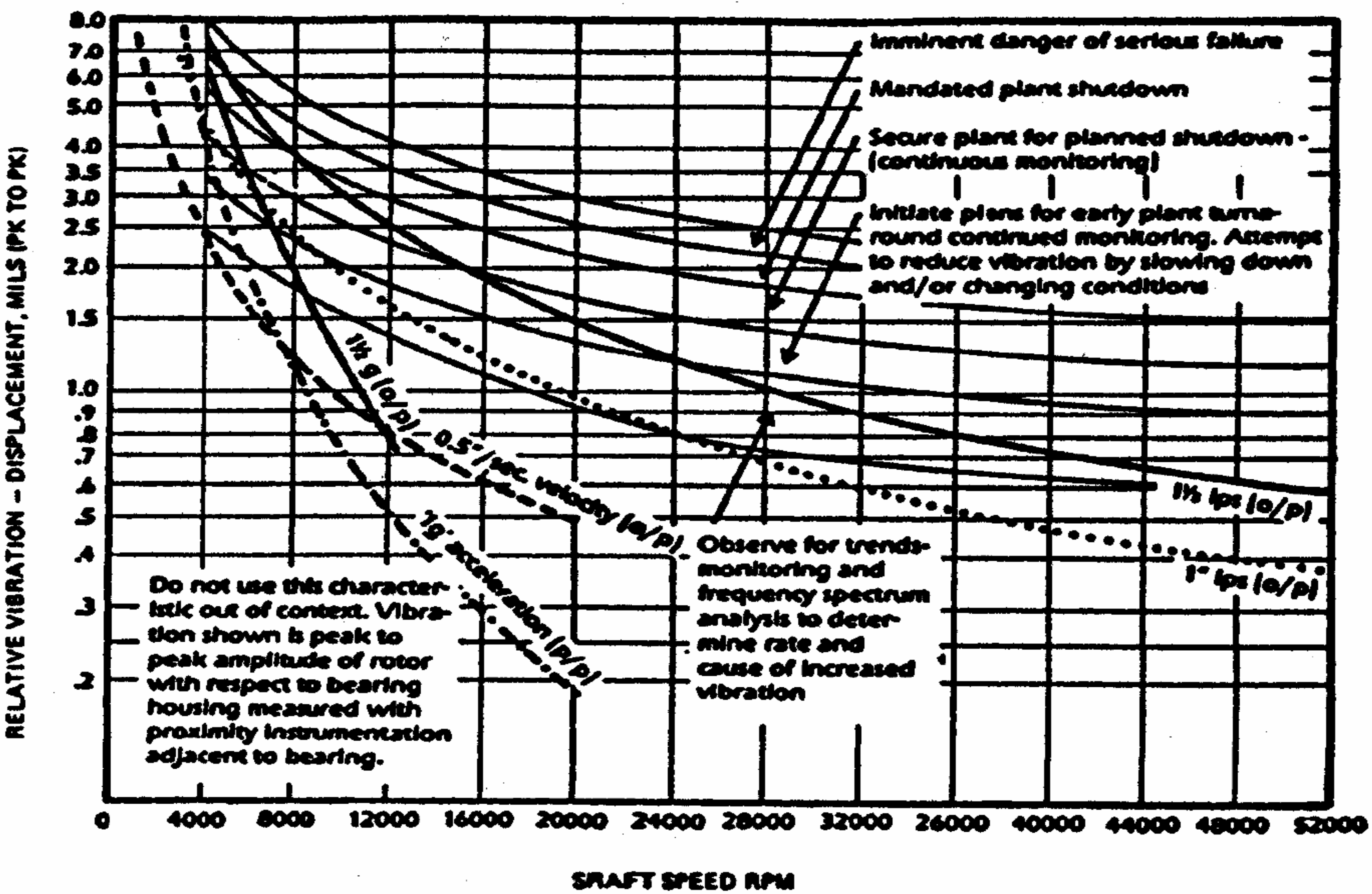
Acceptable Vibration Levels

Modified Blake chart for field balancing



Acceptable Vibration Levels

Proximity probe measurements – Dresser-Clark chart



Unbalance

Multiple Plane Balancing

- correction planes = number of bearings plus one
- depends on flexibility of rotor.
- several trial runs.
- cross plane effects
- matrix solution

Summary

- Mass unbalance of a rotor results when the mass center is not at the same location as the geometric center
- Mass unbalance causes a rotating force at the frequency of shaft speed
- The amount of mass unbalance force depends on the location of the mass center from the geometric center, the weight of the object, and the square of the speed

Summary

- Balancing is a procedure in which a balance weight that creates a force equal to the mass unbalance is placed opposite the effective location of the mass unbalance
- The heavy spot is the angular location of the mass unbalance on the rotor
- The high spot is the angular location of the peak of vibration (displacement)

Summary

- The high spot is measured during the balancing process; however, the balance weight must be positioned opposite the heavy spot.
- Either displacement, velocity, or acceleration can be measured; however, displacement is preferred
- The high spot lags the heavy spot as a result of electronic (instrument) and mechanical lag

Summary

- Balancing should not be performed until it is evident that misalignment, excessive bearing clearance, looseness, and distortion are not the cause of the vibration at operating speed
- The rotor should be clean and structurally sound prior to balancing
- Trial or calibration weights are used to obtain the mechanical lag

Summary

- The rule of thumb for selecting a trial weight is that it should create a force of not more than 10% of the rotor weight
- The vector method is used to determine the size and location of the correction weight
- Vibration is measured on the machine with and without the trial weight.
- The vectorial difference is determined to access the effect of the trial weight.

Summary

- The trial weight is moved relative to the effect vector so that it is opposite the original unbalance vector
- The size of the trial weight is adjusted so that the effect vector is the same length as the original unbalance vector
- Allowable field unbalance values are obtained from vibration severity levels in ISO 2372 (rms) and the modified Blake chart

Machinery Vibration Forcing Functions

BENT SHAFT (Bowed Rotor)

- a special form of unbalance (same vibrations)
- bent shaft – outside the machine housing
- bowed rotor – inside the housing
- common on large machines with heavy shafts
- when idle - gravity causes sag (slow rotate when not in use), difficult to correct
- may also be caused by local (uneven) heating of the shaft (see rubs)

Machinery Vibration Forcing Functions

SHAFT MISALIGNMENT

- a major cause of excessive machinery vibration
- due to improper machine installation
- flexible coupling can tolerate some shaft misalignment
- slight misalignment necessary for proper gear teeth lubrication in a gear coupling
- otherwise - shafts of coupled machines should be as closely aligned as possible

Machinery Vibration Forcing Functions

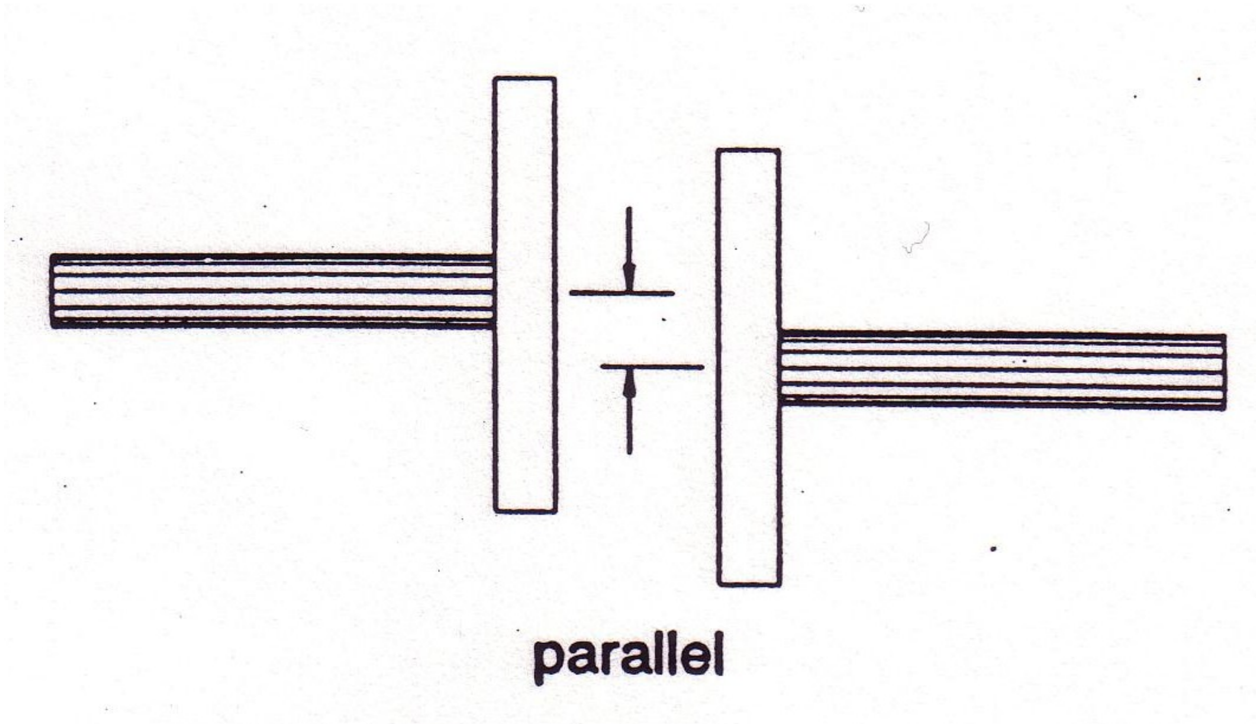
Types of Misalignment

Parallel Misalignment (offset)

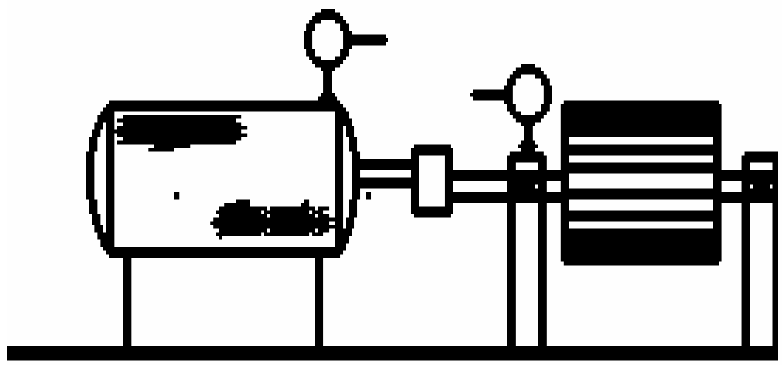
- shaft centre lines are parallel but offset from one another
- horizontal, vertical or combination

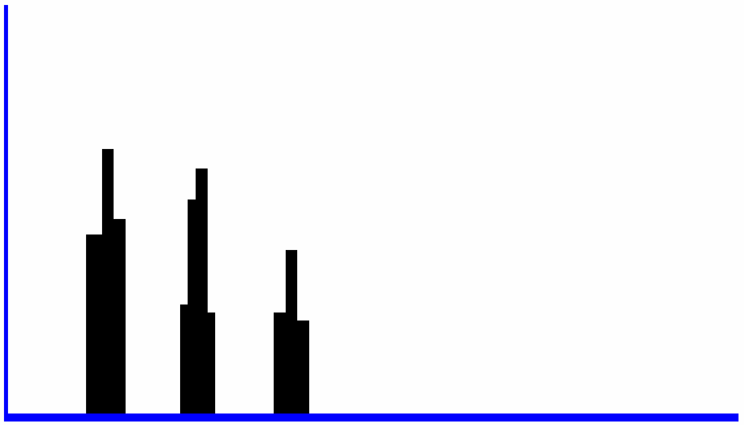
Machinery Vibration Forcing Functions

Parallel Misalignment (offset)



Machinery Vibration Forcing Functions





- High radial vibration 180^0 out of phase
- Severe conditions give higher harmonics
- 2X RPM often larger than 1X RPM
- Similar symptoms to angular misalignment
- Coupling design can influence spectrum shape and amplitude

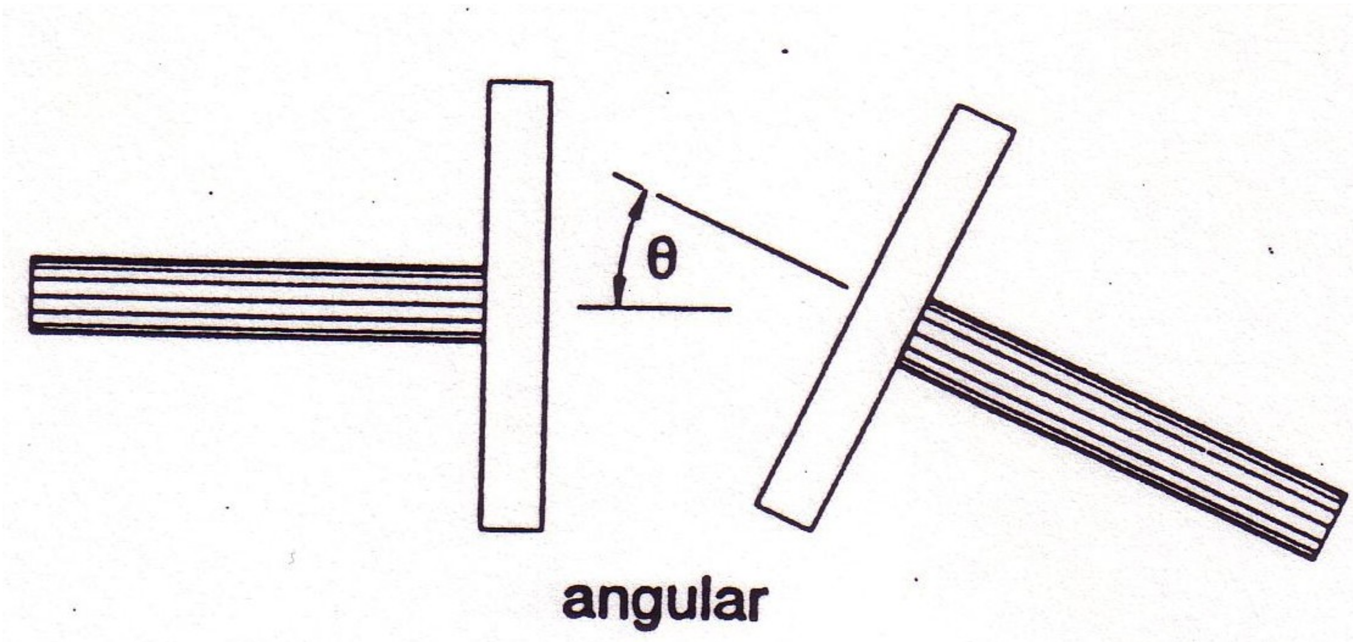
Machinery Vibration Forcing Functions

Angular Misalignment

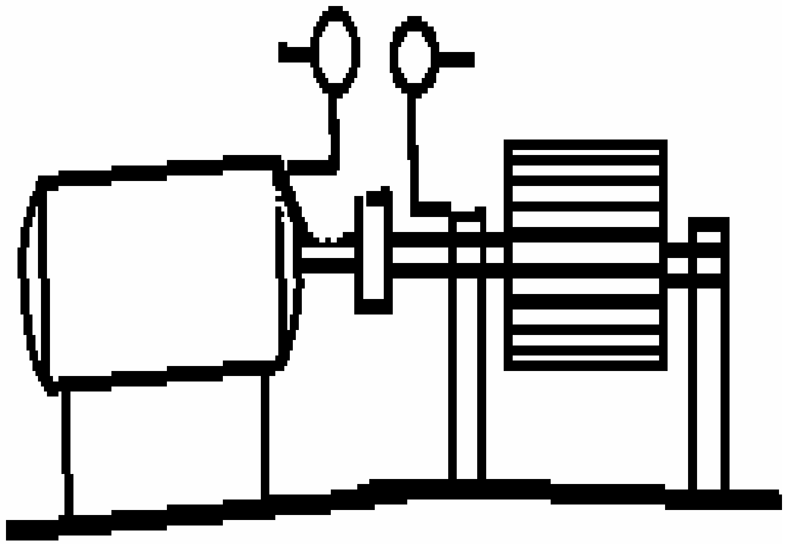
- shaft center lines meet at an angle
- intersection may be at driver or driven end, between units or behind units
- most misalignment is a combination of Parallel and Angular misalignment

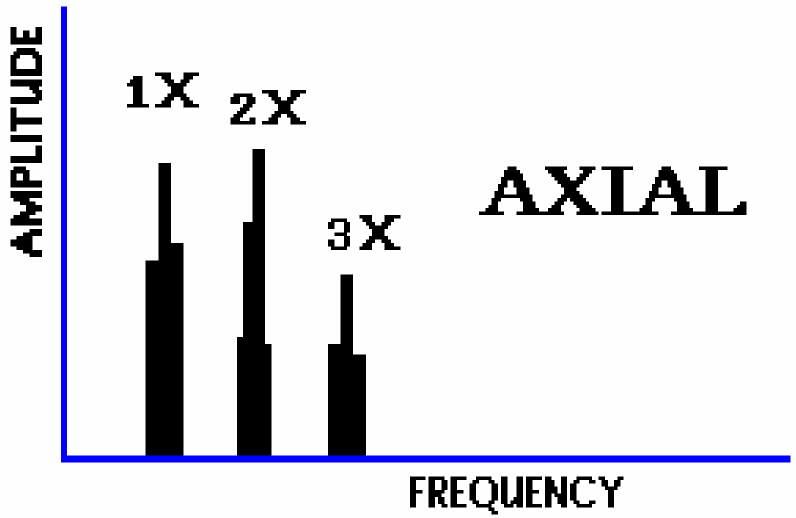
Machinery Vibration Forcing Functions

Angular Misalignment



Machinery Vibration Forcing Functions





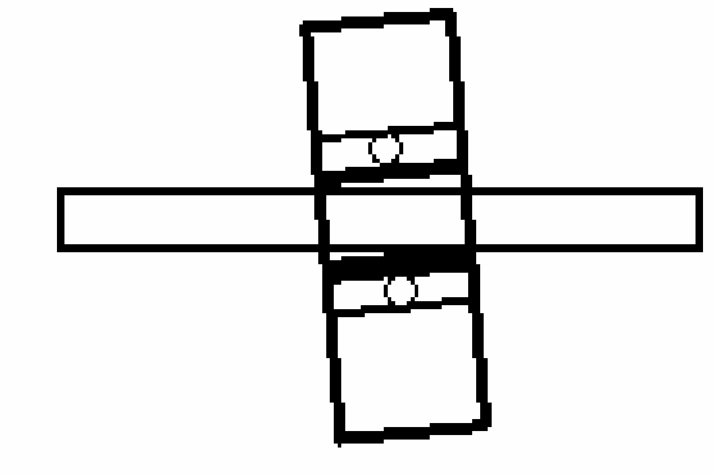
- Characterized by high axial vibration
- 180^0 phase change across the coupling
- Typically high 1 and 2 times axial vibration
- Not unusual for 1, 2 or 3X RPM to dominate
- Symptoms could indicate coupling problems

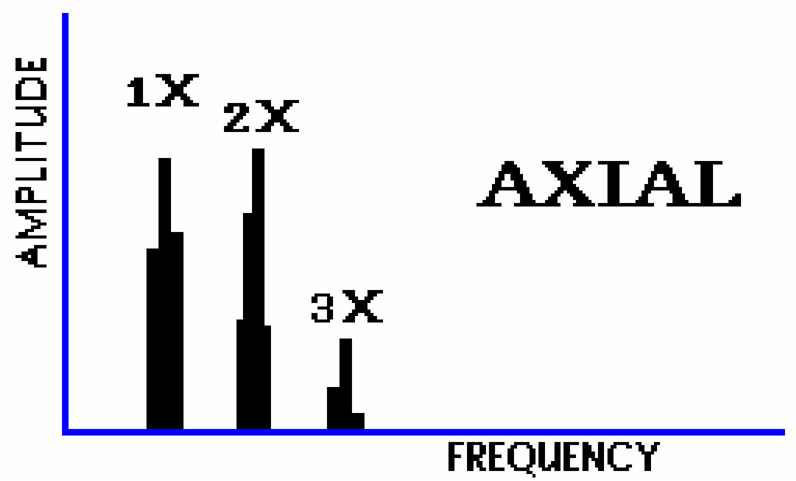
Machinery Vibration Forcing Functions

Bearing Misalignment

- shaft center lines are properly aligned
- bearings on one side of coupling are misaligned
- not mounted in the same plane
- not normal to shaft
- machine distorts in use (soft foot, uneven base, thermal growth)

Machinery Vibration Forcing Functions





- 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

Alignment Methods

- reverse dial method
- face and rim method
- dial indicators or laser sensors

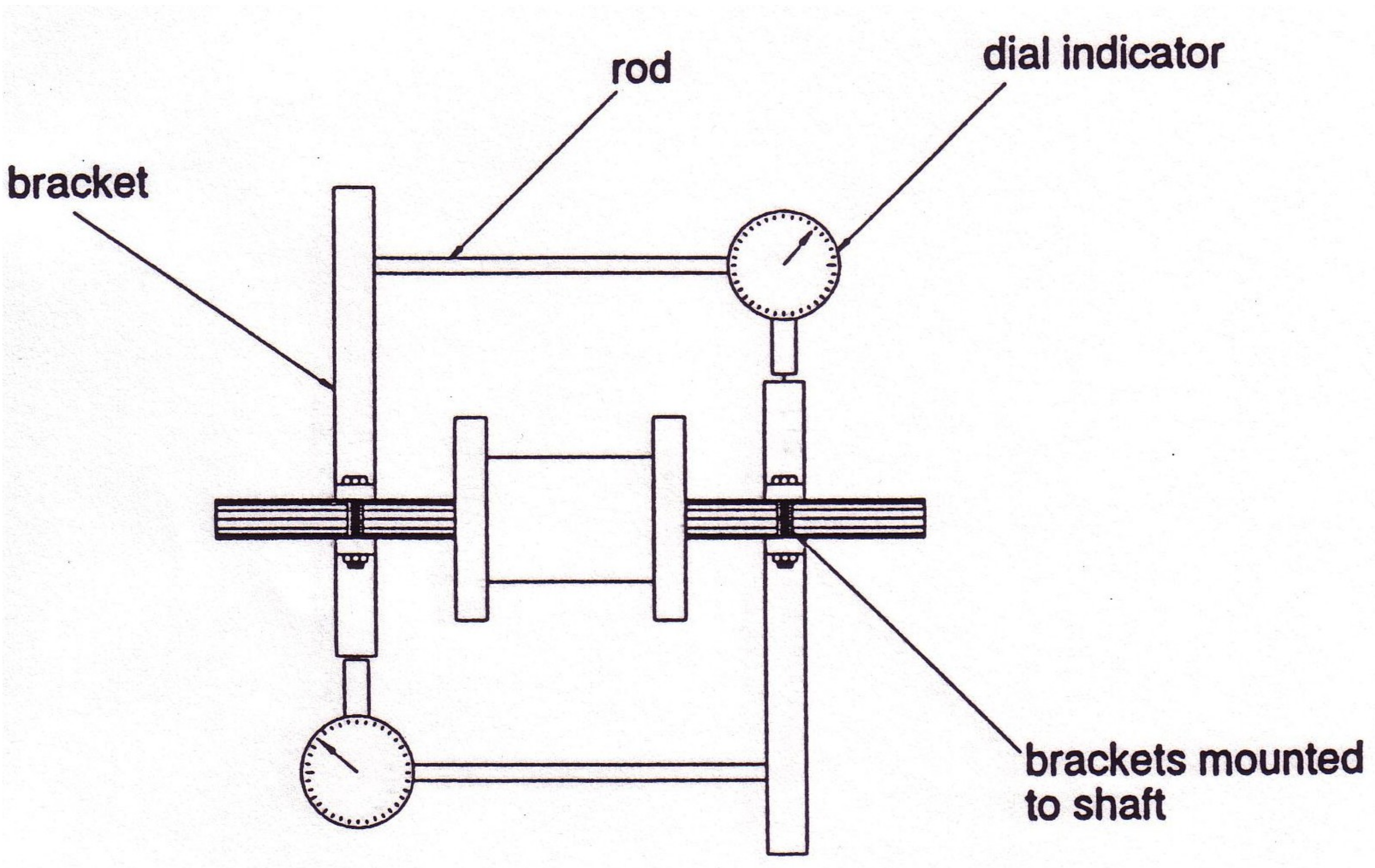
Machinery Vibration Forcing Functions

Reverse Dial Alignment Method

- simple & accurate
- brackets on both shafts, opposite sides of coupling
- each bracket holds a rod which spans the coupling
- both rods rest on dial indicators attached to the opposite bracket
- long rods between brackets improve accuracy
- for long spool pieces - position dial indicator stems against spool piece surface

Machinery Vibration Forcing Functions

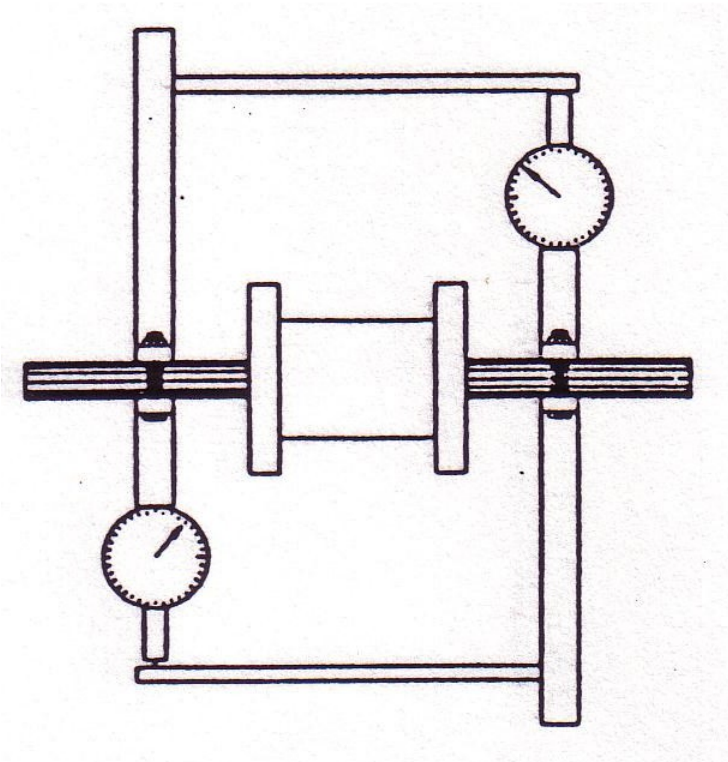
Reverse Dial Alignment Method



Machinery Vibration Forcing Functions

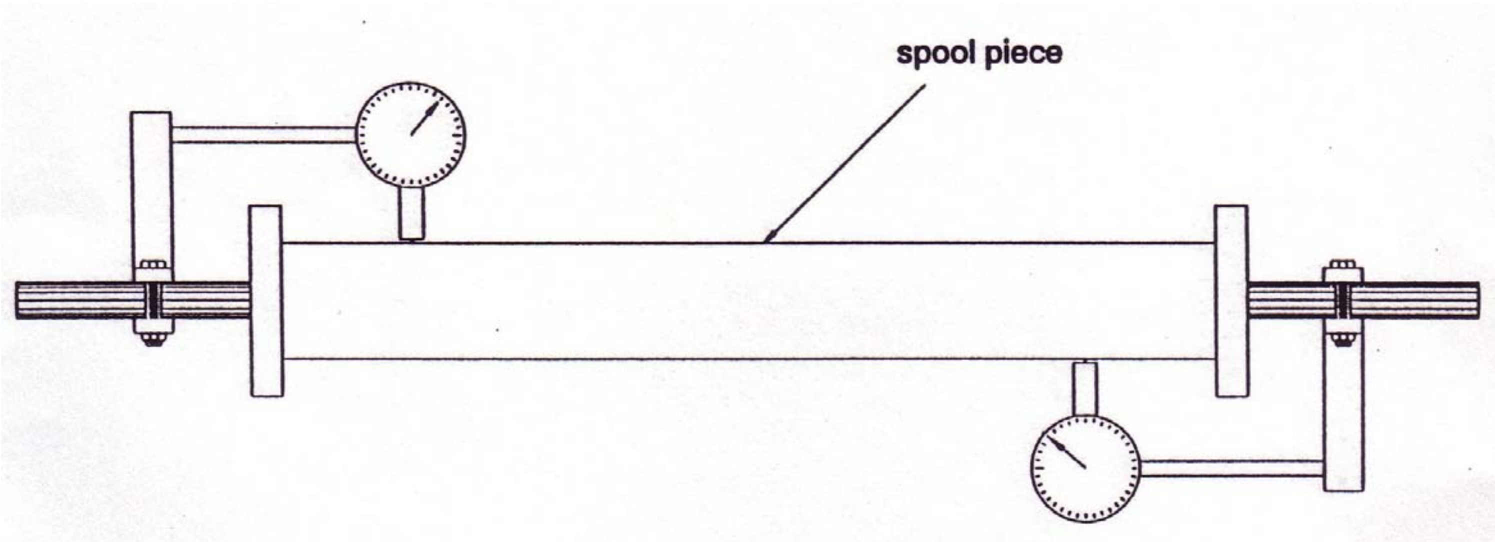
Reverse Dial Alignment Method

- mounting dial on bracket reduces rod sag



Machinery Vibration Forcing Functions

Reverse Dial Alignment Method



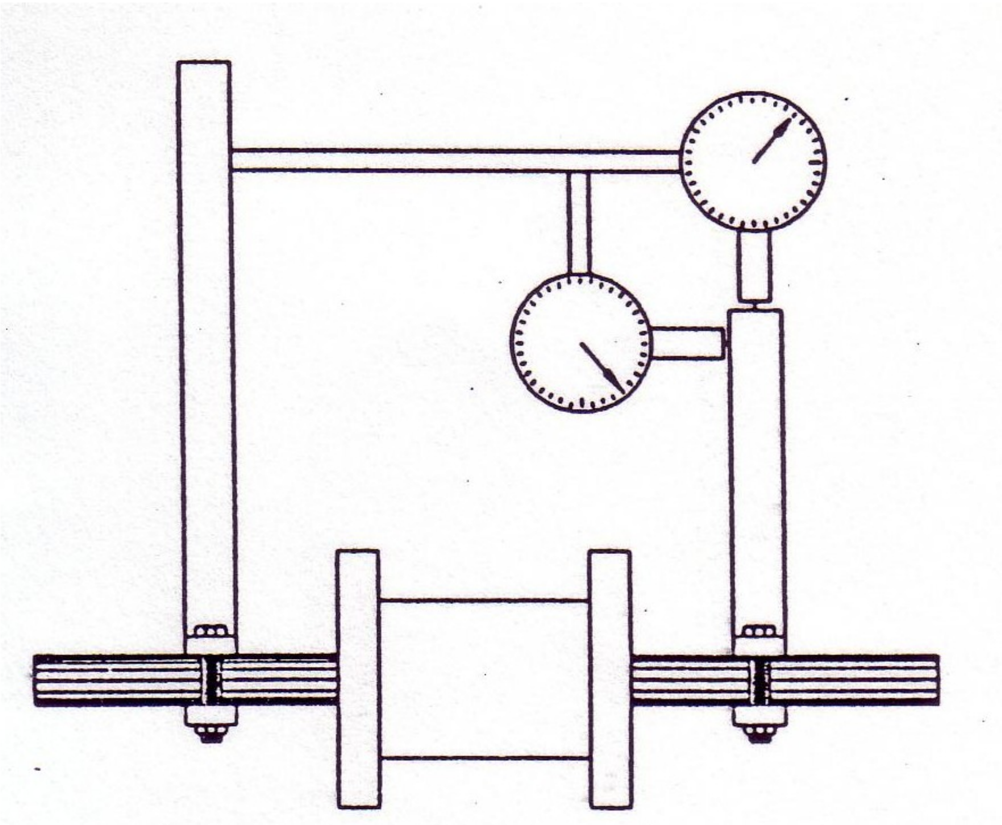
Machinery Vibration Forcing Functions

Face and Rim Alignment Method

- one bracket holds both dials
- brackets as far apart as possible
- rod as high above shaft center line as possible (this amplifies even small angles of misalignment)
- long rods sag due to their own weight & the dial indicator weight

Machinery Vibration Forcing Functions

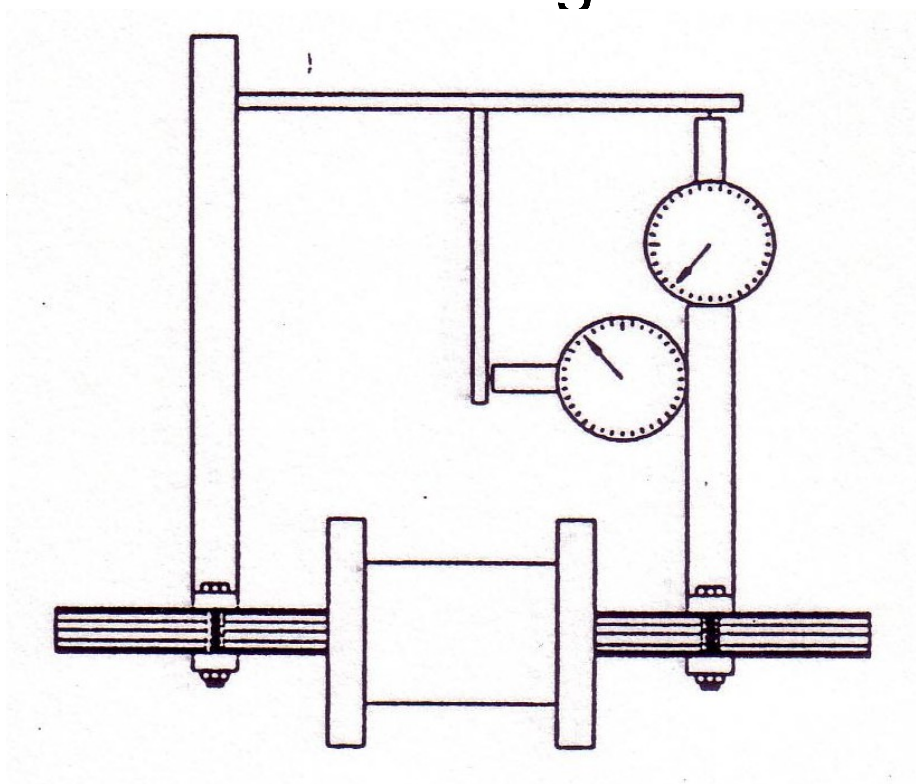
Face and Rim Alignment Method



Machinery Vibration Forcing Functions

Face and Rim Alignment Method

- can move dial indicator to bracket and counter weight rod to reduce sag



Machinery Vibration Forcing Functions

Alignment Procedures

- set dial indicators to zero after mounting
- take four equally spaced readings (rotate brackets 90° each time)
- if a full rotation is not possible, three spaced at 90° will do
- the sum of opposite readings should always be equal
- shafts must rotate together

Machinery Vibration Forcing Functions

Alignment Procedures (Cont'd)

- do not uncouple shafts during alignment (axial shaft movement causes inaccuracies)
- always rotate shaft in the same direction (coupling backlash - inaccuracies)
- if coupling allows some axial movement this must be restricted

Laser Based Alignment Methods

- laser beam used in place of rod
- detector measures beam deflection as shaft is rotated.
- emitter and detector on opposite shafts
- long coupling spans
- no sag (high accuracy)
- quick set up
- costly to purchase, but quick to use

Machinery Vibration Forcing Functions

Alignment Preparation (General)

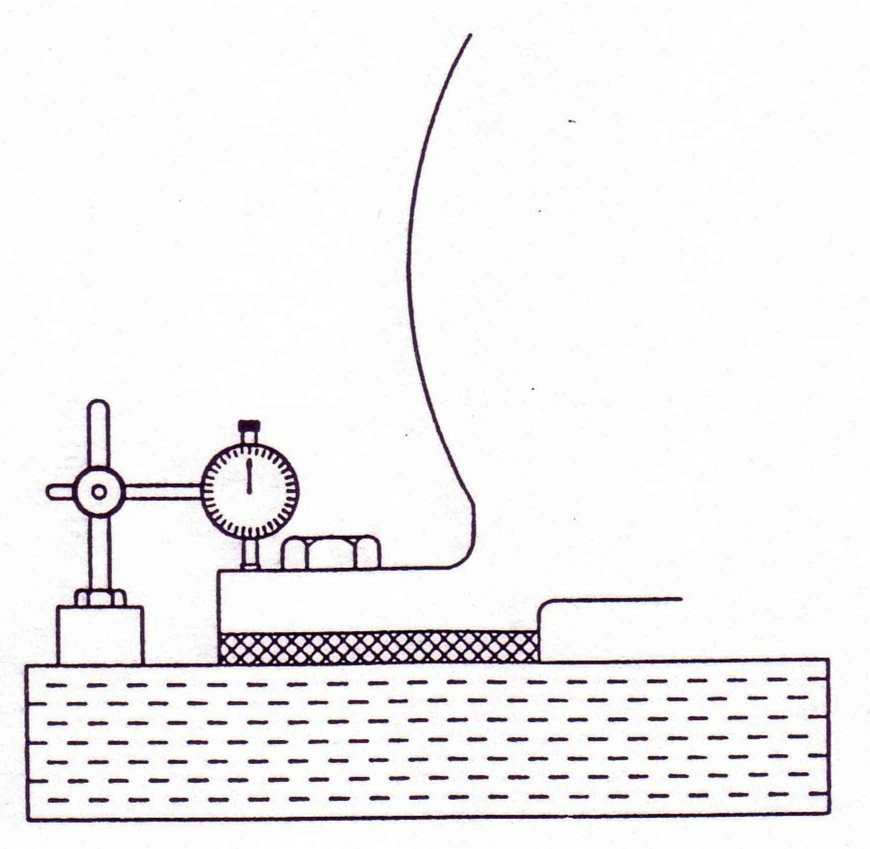
Base preparation

- base clean and free of defects

Soft feet and frame twists

- set dial indicator on foot
- loosen hold down nut
- foot movement of more than 1 mil. (1/40 mm) should be shimmed
- check each foot, tighten and loosen in a set sequence

Detecting Soft Feet



Machinery Vibration Forcing Functions

Lifting and moving machine

- use jack screws wherever possible for vertical & horizontal movements
- never move a machine with a sledge hammer

Piping connections

- disconnect during alignment
- keep dial indicators in place during reconnection as a check
- piping should have its own supports

Machinery Vibration Forcing Functions

Dial indicator rod sag

- can be calculated
- should not be ignored
- can be measured using a straight piece of pipe

Machinery Vibration Forcing Functions

Sag in round rods

$$S_g = 2.829 \times 10^{-5} \times L^3 \times \left(\frac{K_r}{D^4 - d^4}\right)$$

S_g - dial indicator sag (mils)

L - length of rod (inches)

D - Outside diameter (inches)

d - Inside diameter (inches)

K_r - spring constant of circular rod (mild steel)

$$K_r = 1.334 \times (D^2 - d^2) \times (L + W_d)$$

W_d - weight of dial indicator (ounces)

Machinery Vibration Forcing Functions

Sag in square rods

$$S_g = 2.833 \times 10^{-5} \times L^3 \times \left(\frac{K_s}{D^4 - d^4} \right)$$

S_g - dial indicator sag (mils)

L - length of rod (inches)

D - Outside width or height (inches)

d - Inside width or height (inches)

K_s - spring constant of square rod (mild steel)

$$K_s = 1.699 \times (D^2 - d^2) \times (L + W_d)$$

W_d - weight of dial indicator (ounces)

Machinery Vibration Forcing Functions

Thermal growth

- heat of operation causes metal to expand
- vertical growth most important
- may be minimal but should always be measured
- laser equipment, micrometers, alignment bars are all subject
- take readings when cold and again after normal operating temperature has been reached

Machinery Vibration Forcing Functions

Calculating machine moves

- some computer based alignment systems do this automatically
- plot shaft center lines from movable and fixed machine
- difference between center lines on graph paper is the distance the movable machine must be moved
- check in vertical and horizontal directions

Machinery Vibration Forcing Functions

Vibration from misalignment

- causes excessive radial loads on bearings
- premature bearing failure
- high 1X vibration with high harmonics up to 6th
- also seen as one without the other
- this could be mistaken as unbalance, looseness or excessive clearance
- high horizontal to vertical vibration amplitude ratios (greater than 3 : 1) may also indicate misalignment

Machinery Vibration Forcing Functions

Distinguishing Unbalance from Misalignment

Unbalance	Misalignment
High 1X response in frequency spectra.	High harmonics of 1X relative to 1X.
Low axial vibration levels.	High axial vibration levels.
Measurements at different locations are in phase.	Measurements at different locations are 180° out of phase.
Vibration levels are independent of temperature.	Vibration levels are dependent on temperature (change during warm-up).
Vibration level at 1X increases with rotational speed.	Vibration level does not change with rotational speed.
Centrifugal force increases as the square of the shaft rotational speed.	Forces due to misalignment remain relatively constant with changes in shaft rotational speed.

Machinery Vibration Forcing Functions

LOOSENESS

Types of Looseness

Bearing loose on shaft

- modulated time waveform
- harmonics (many)
- varying time period of modulation
- truncated time signal (clipped)

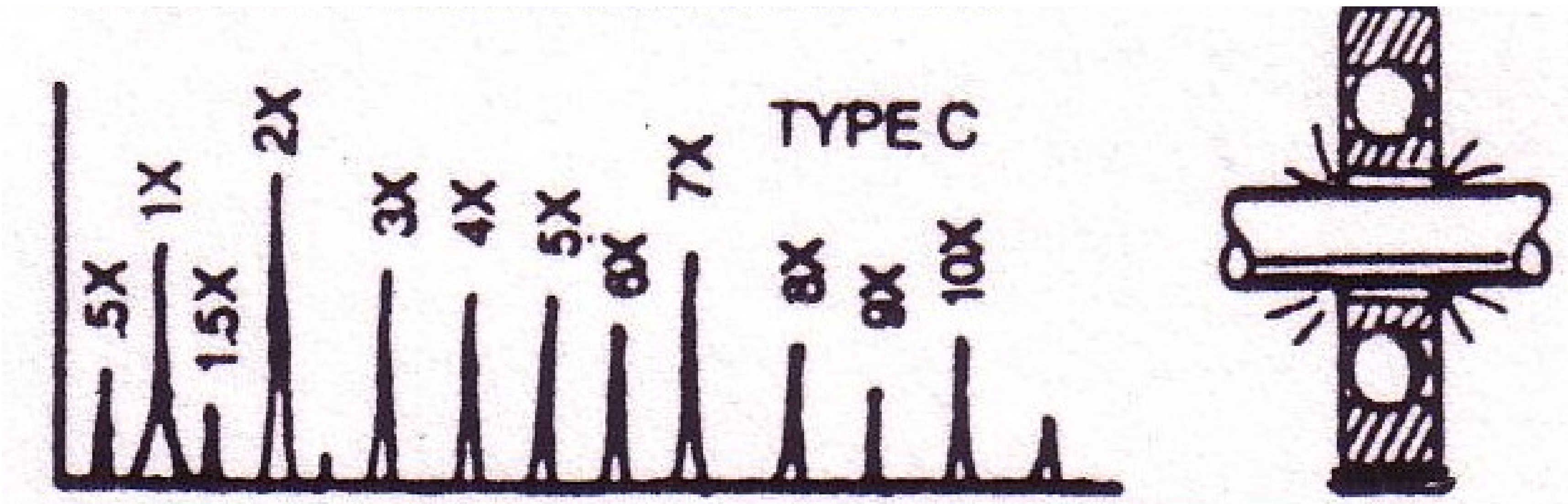
Machinery Vibration Forcing Functions

Bearing loose in housing

- fourth harmonic distinctive
- beware of 4 blade fans (blade pass frequency will mask looseness signal)
- may also look like rolling element bearing characteristic frequencies
- wideband noise
- further deterioration results in fractional harmonics (½, ⅓, 1½, 2½) increasing in amplitude

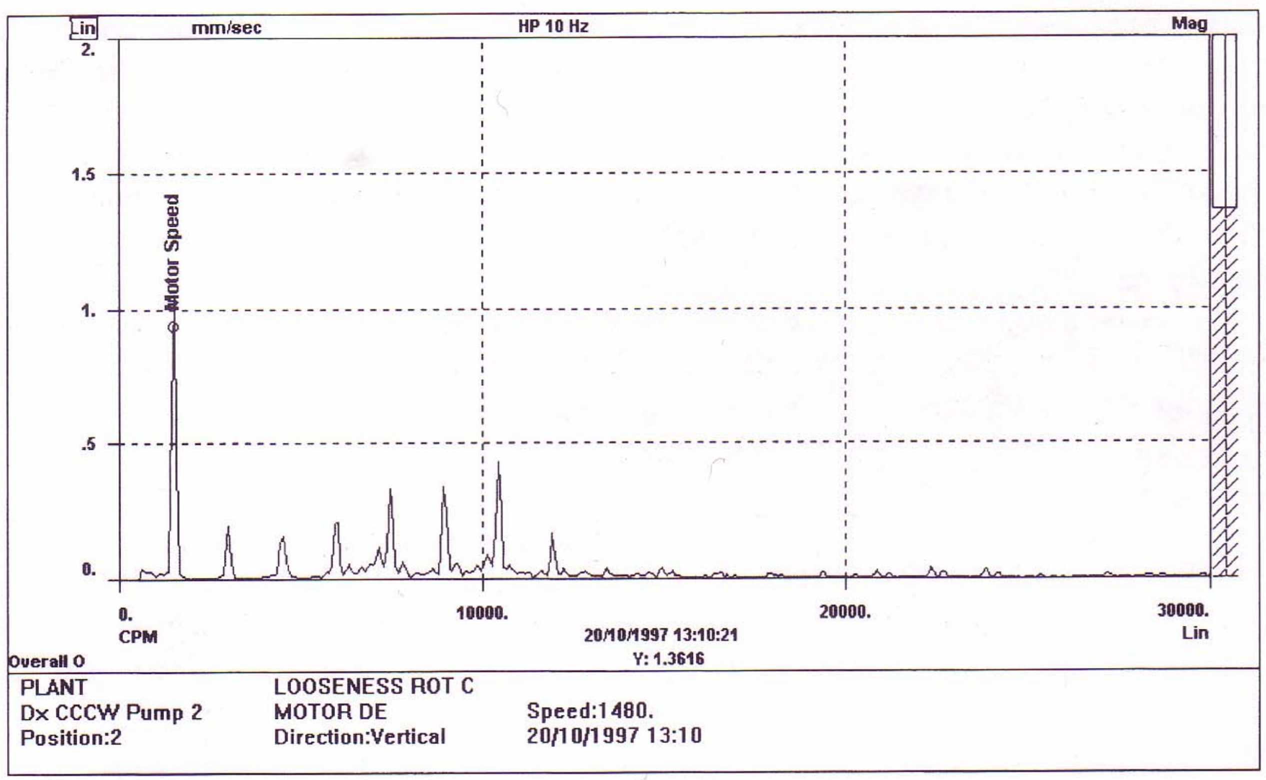
Machinery Vibration Forcing Functions

Bearing loose in housing



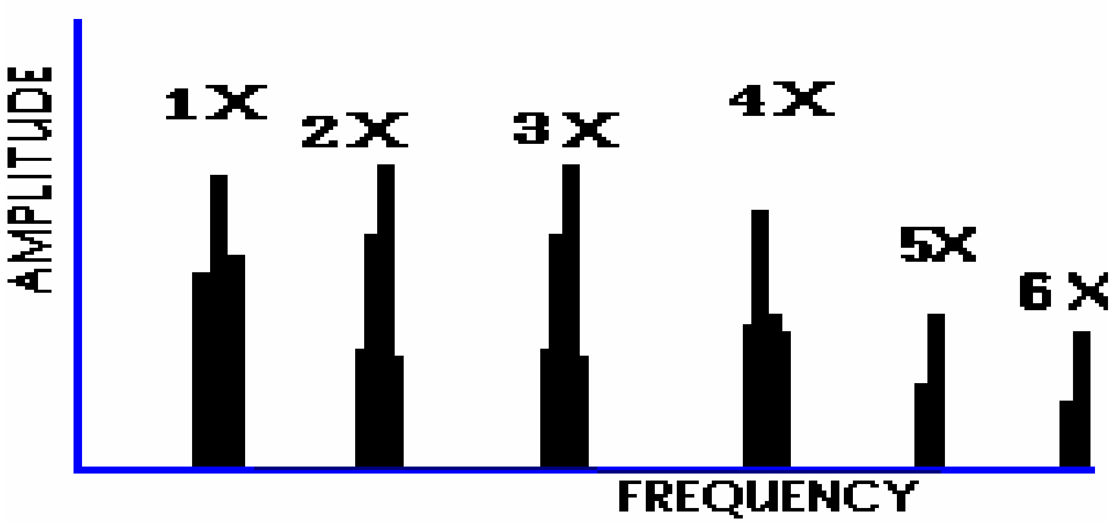
Machinery Vibration Forcing Functions

Coupling looseness between motor and pump



Machinery Vibration Forcing Functions

Loose Sleeve Bearing



- 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

Machinery Vibration Forcing Functions

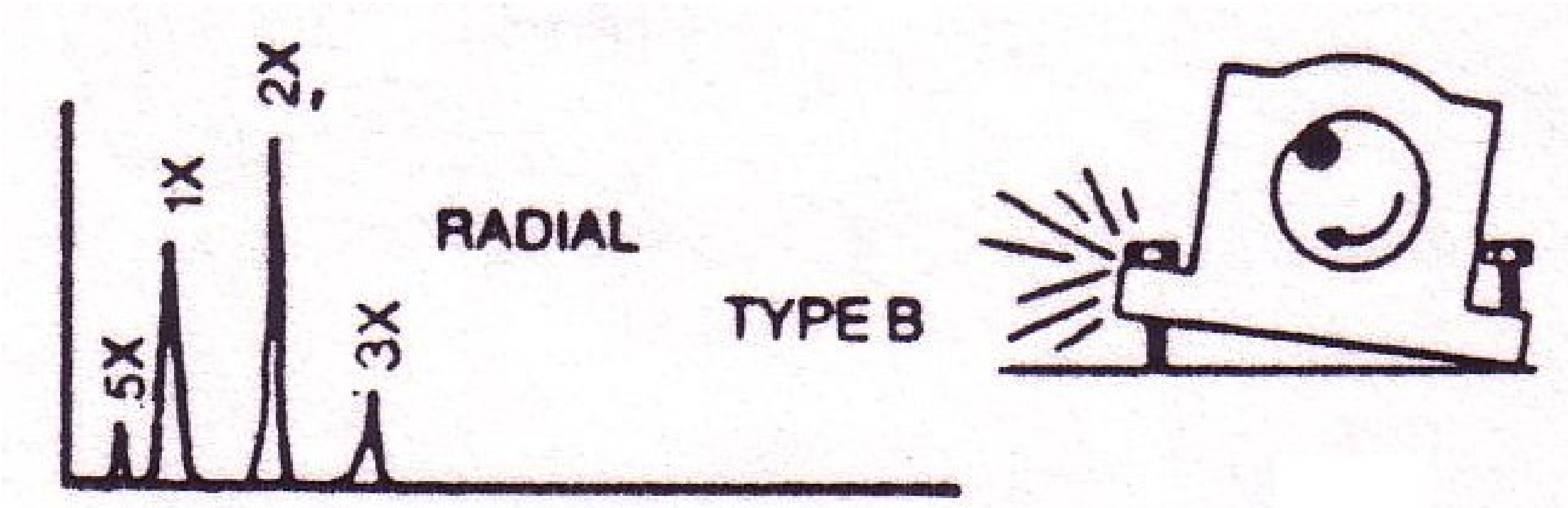
SOFT FOOT

(another type of mechanical looseness)

- loose hold-down bolts cannot resist dynamic forces of machine
- harmonics due to opening and closing of gap (impacts cause non-linear vibration signal)

Machinery Vibration Forcing Functions

SOFT FOOT



Machinery Vibration Forcing Functions

SOFT FOOT

(another type of mechanical looseness)

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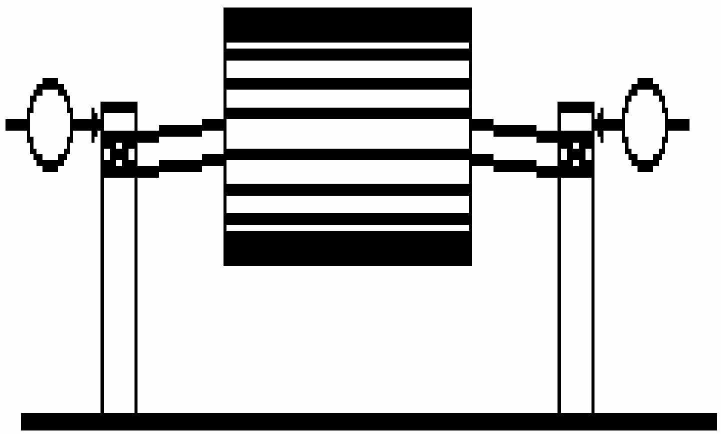
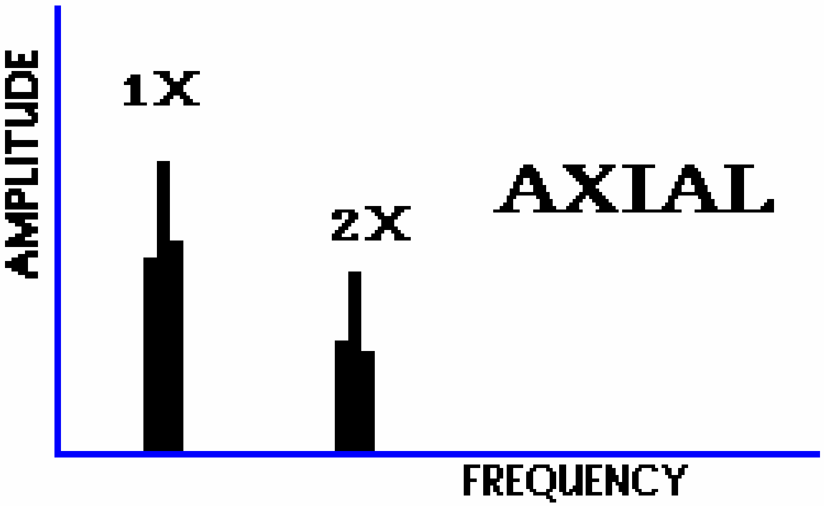
Machinery Vibration Forcing Functions

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

Machinery Vibration Forcing Functions

RUBS

- caused by excessive mechanical looseness or oil whirl
- moving parts come into contact with stationary ones
- vibration signal similar to looseness
- high levels of wide-band noise (caused by impacts)
- if impacts are repetitive, there may be strong spectral responses at the striking frequency

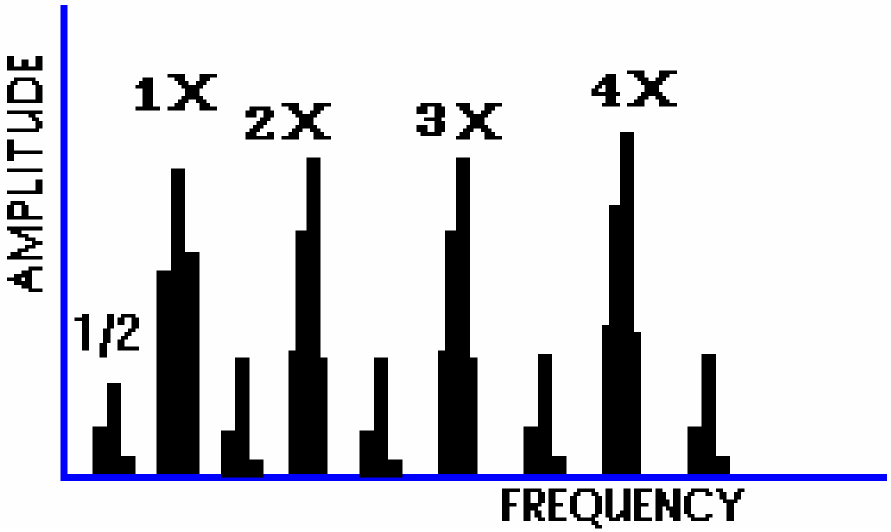
Machinery Vibration Forcing Functions

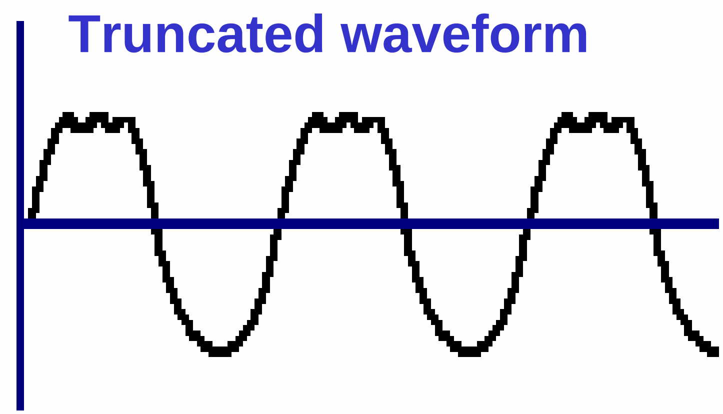
RUBS

- if rotor presses too hard against a seal - the rotor will heat up unsymmetrically and develop a bowed shape
- vibration signal shows unbalance
- to diagnose - note that the unbalance is absent until the machine comes up to normal operating temperature

Machinery Vibration Forcing Functions

RUBS

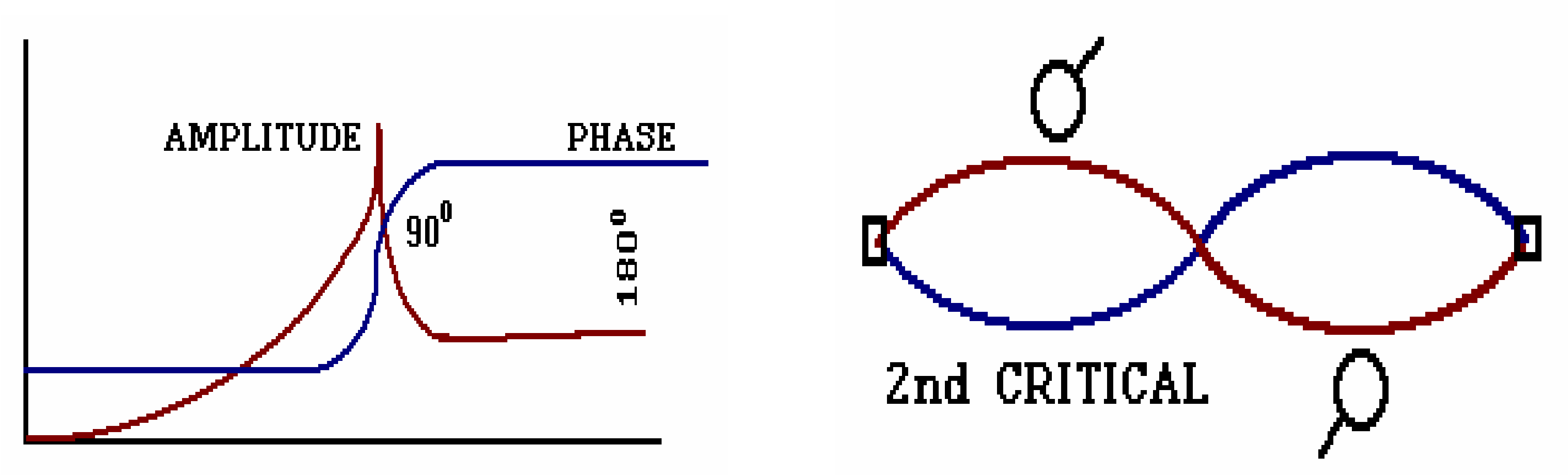




- Similar spectrum to mechanical looseness
- Usually generates a series of frequencies which may excite natural frequencies
- Subharmonic frequencies may be present
- Rub may be partial or through a complete revolution.

Machinery Vibration Forcing Functions

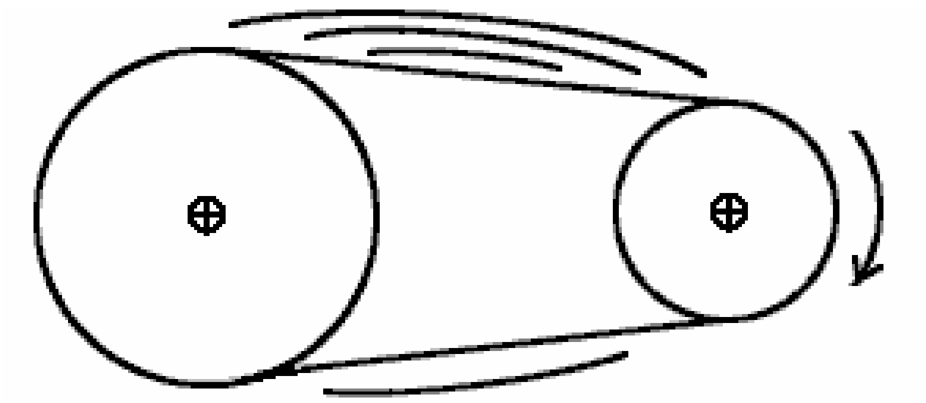
Resonance Excitation

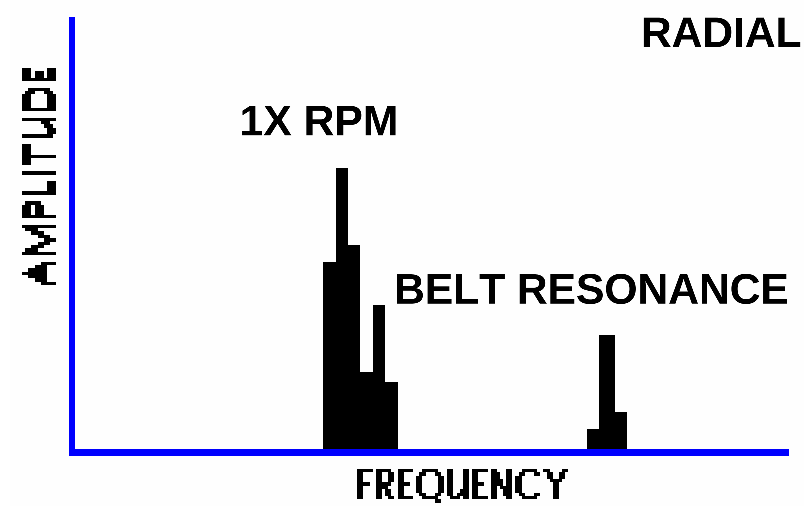


- Resonance occurs when the Forcing Frequency coincides with a Natural Frequency
- 180^0 phase change occurs when shaft speed passes through resonance
- High amplitudes of vibration will be present when a system is in resonance

Machinery Vibration Forcing Functions

Belt Resonance

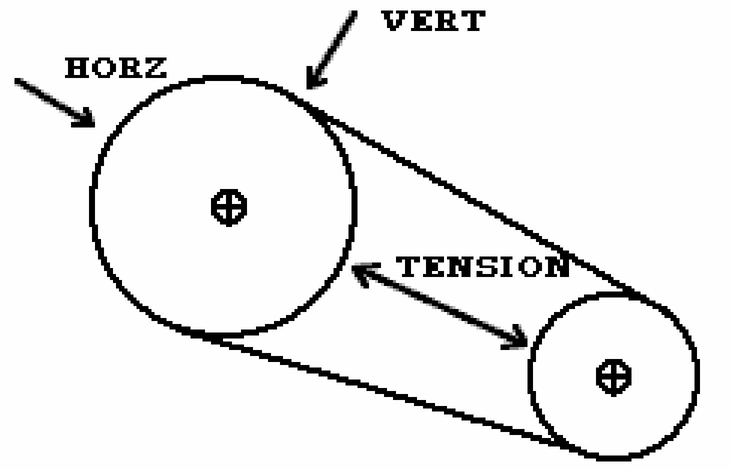


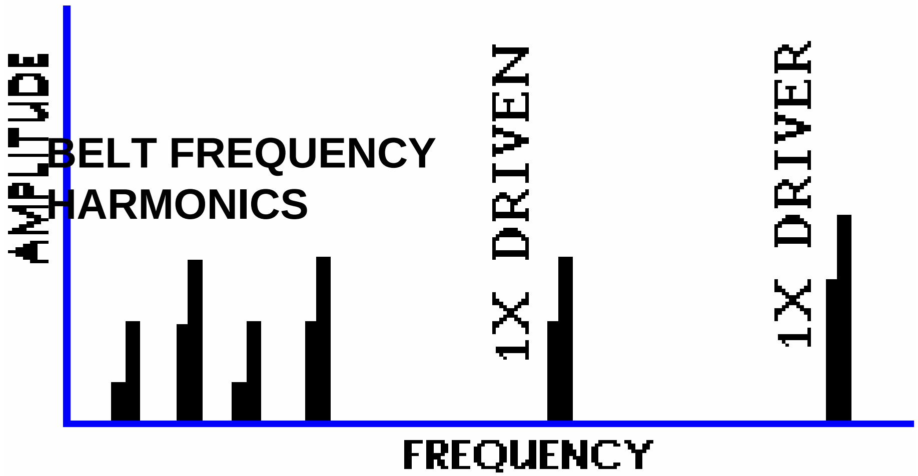


- High amplitudes can be present if the belt natural frequency coincides with driver or driven RPM
- Belt natural frequency can be changed by altering the belt tension

Machinery Vibration Forcing Functions

Worn, Loose or Mismatched Belts

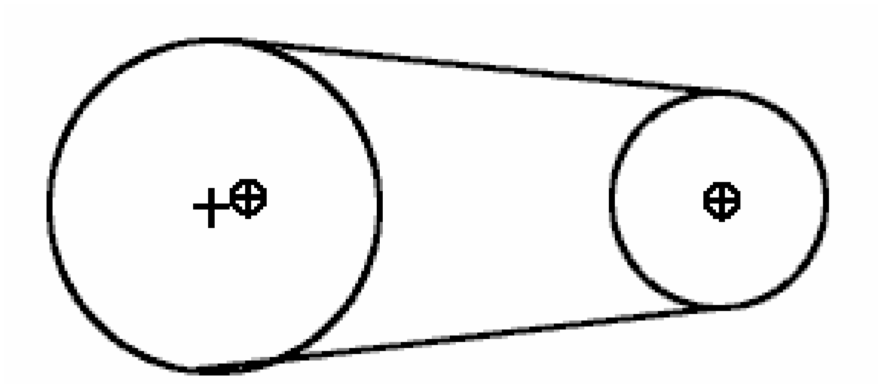


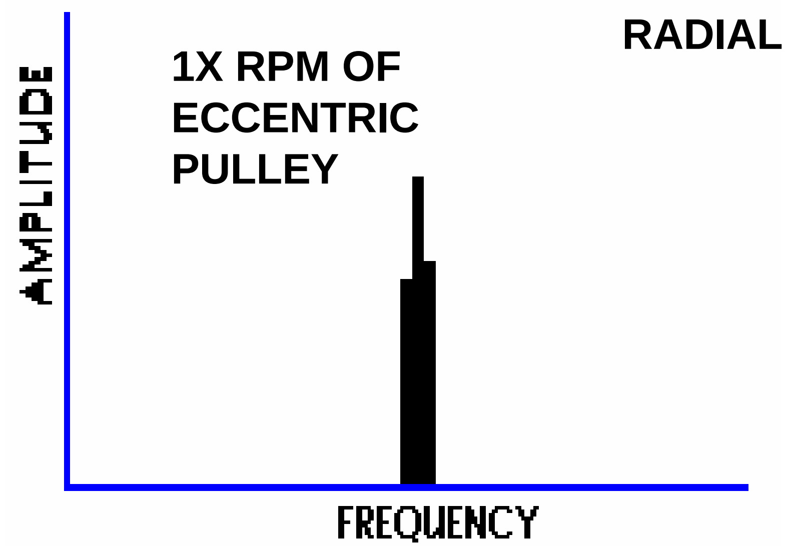


- Often 2X RPM is dominant
- Amplitudes are normally unsteady, sometimes pulsing with either driver or driven RPM
- Wear or misalignment in timing belt drives will give high amplitudes at the timing belt frequency
- Belt frequencies are below RPM of both driver and driven

Machinery Vibration Forcing Functions

Eccentric Pulleys

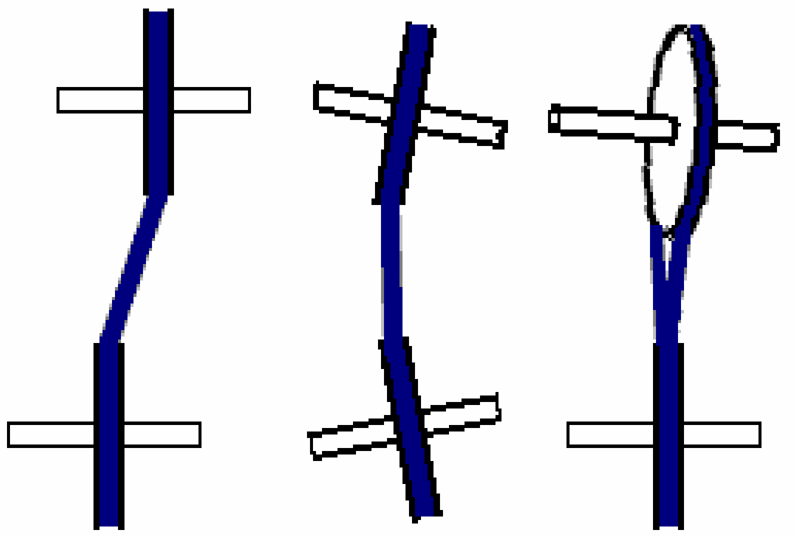


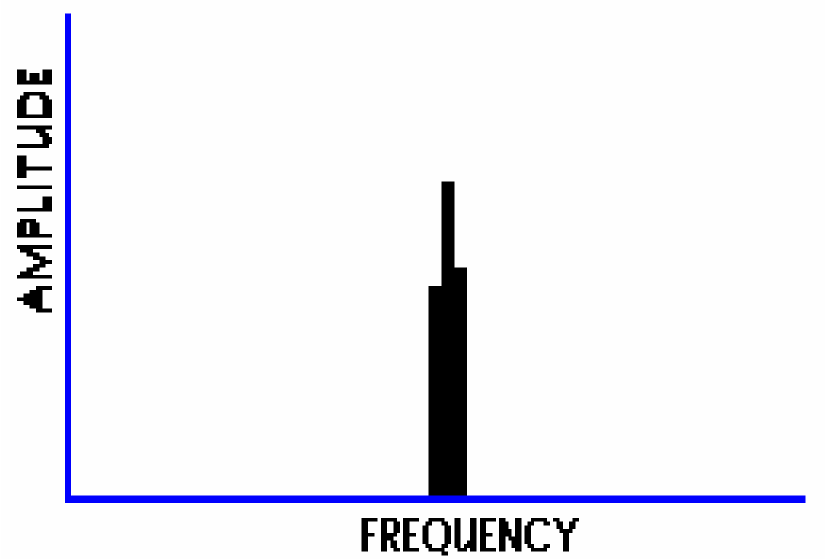


- Eccentric or unbalanced pulleys will give a high 1X RPM of the pulley
- The amplitude will be highest in line with the belts
- Beware of trying to balance eccentric pulleys

Machinery Vibration Forcing Functions

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

Machinery Vibration Forcing Functions

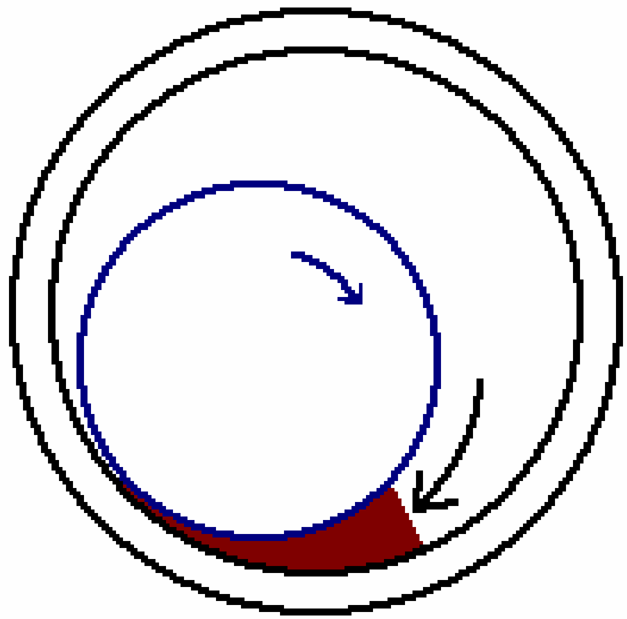
OIL WHIP & OIL WHIRL

Oil Whirl

- bearing can not exert sufficient force on shaft to maintain a stable operating position
- corrected by using pressure dams or tilt pad designs
- shaft rides on an oil pressure gradient
- rotates within bearing clearance at just less than one half shaft rotational speed (~0.42X)

Machinery Vibration Forcing Functions

OIL WHIRL



- Vibration amplitudes are sometimes severe
- Whirl is inherently unstable, since it increases centrifugal forces therefore increasing whirl forces

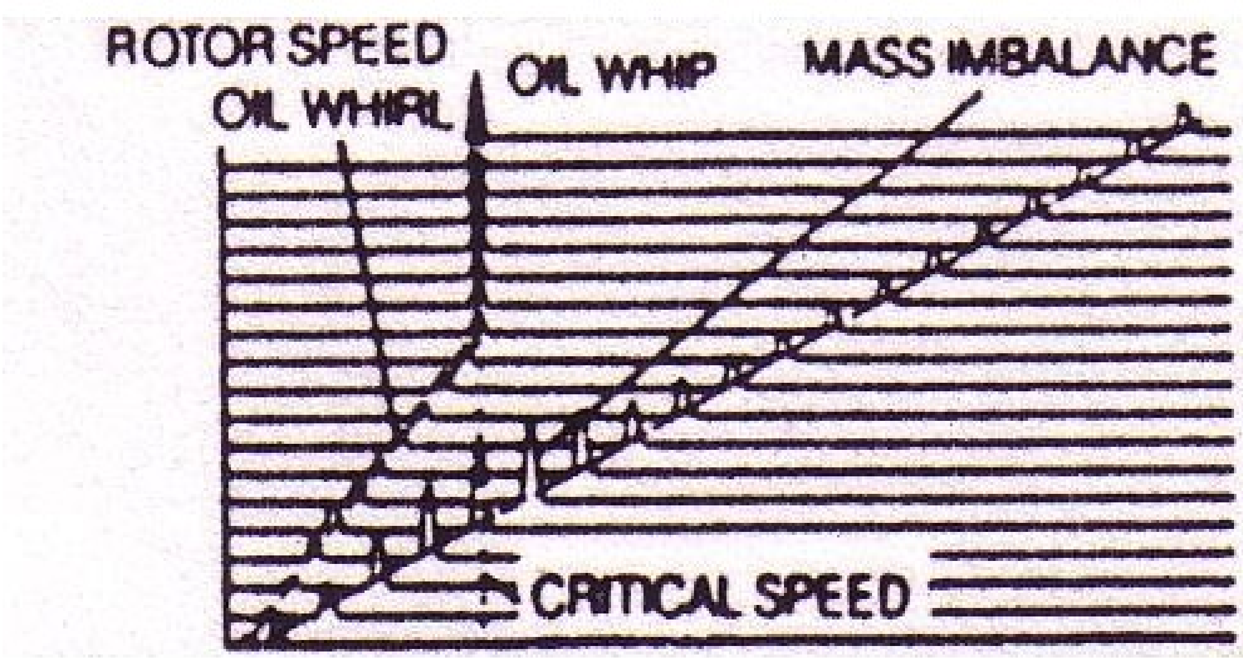
Machinery Vibration Forcing Functions

Oil Whip

- occurs when sub synchronous instability (oil whirl) excites a critical speed (resonance)
- excitation remains at a constant frequency regardless of speed changes
- 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 whipfrequency will not increase.

Machinery Vibration Forcing Functions

OIL WHIP & OIL WHIRL



A spectral map showing Oil Whirl becoming Oil Whip instability as shaft speed reaches twice critical.

Next Time

- Machinery Vibration Trouble Shooting
- Fault Diagnostics Based on Forcing Functions
- Fault Diagnostics Based on Specific Machine Components
- Fault Diagnostics Based on Specific Machine Types
- Automatic Diagnostics Techniques
- Non-Vibration Based Machine Condition Monitoring and Fault Diagnosis Methods