



Field Balancing of Rotating Machinery

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Topics

- Introduction
- Definition and Concepts
- Sensors & Measurement
- One Plane Balancing Methods
 - Influence Coefficient Method
 - Four run Method without phase
- Two Plane Balancing
 - Influence Coefficient Method
 - Four run Method without phase
- Why Balancing May Not Work
- Balancing Standard

HOW ROTORDYNAMICS IS DIFFERENT FROM STRUCTURAL VIBRATIONS

- Gyroscopic and Coriolis effects
- Speed-dependent bearing stiffness & damping
- Damping raises the critical speeds
- X and Y motions are coupled
- Rotor whirling can be self-excited at frequencies different from the forcing frequency
- Rotor damping can produce whirl instability

Introduction—some facts

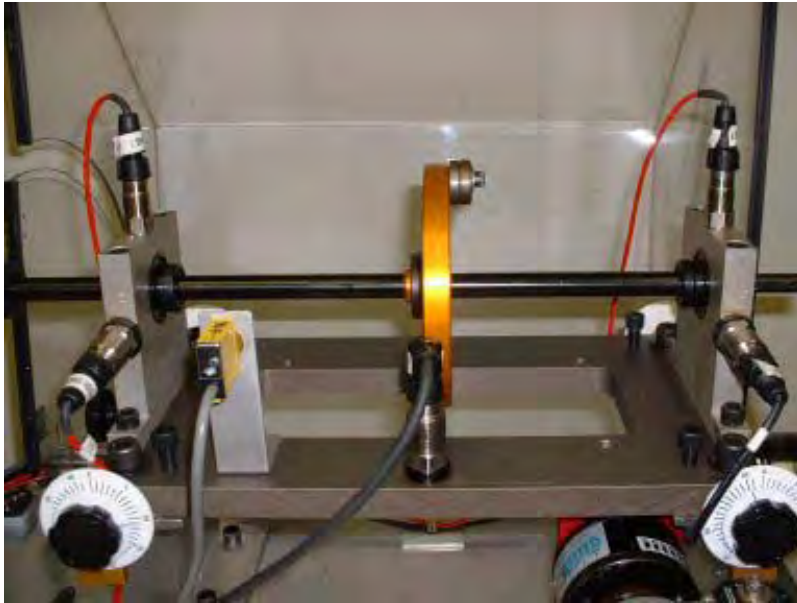
- Mass Unbalance is the most common malfunction of rotating machinery (50% or even more).
- It produces excessive synchronous forces on bearings and foundation, and also vibrations on the machine.
- Unbalance occurs due an unequal distribution of mass about the rotating centerline or shaft axis.
- Balancing is the process of attempting to improve the mass distribution of a body so that it rotates in its bearings within specified vibration level.
- Most balancing jobs are performed below the first critical.
- Balancing above the first critical may require several techniques and be even supported by a rotor model

Why Balancing?

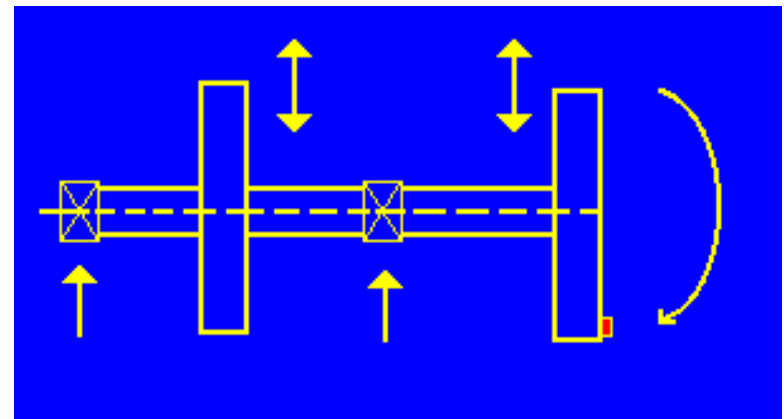
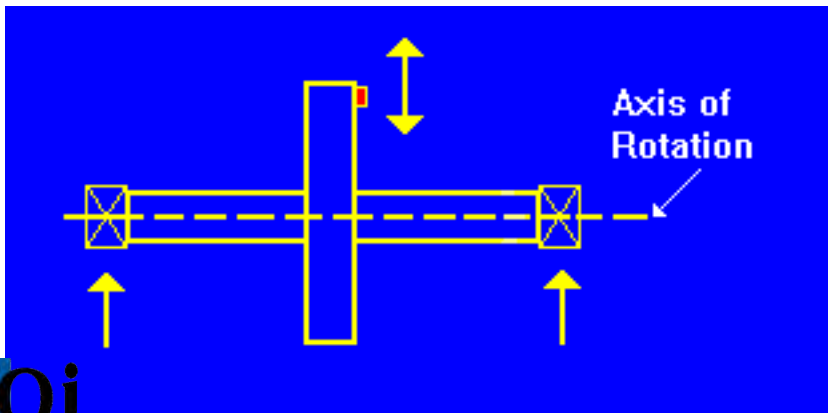
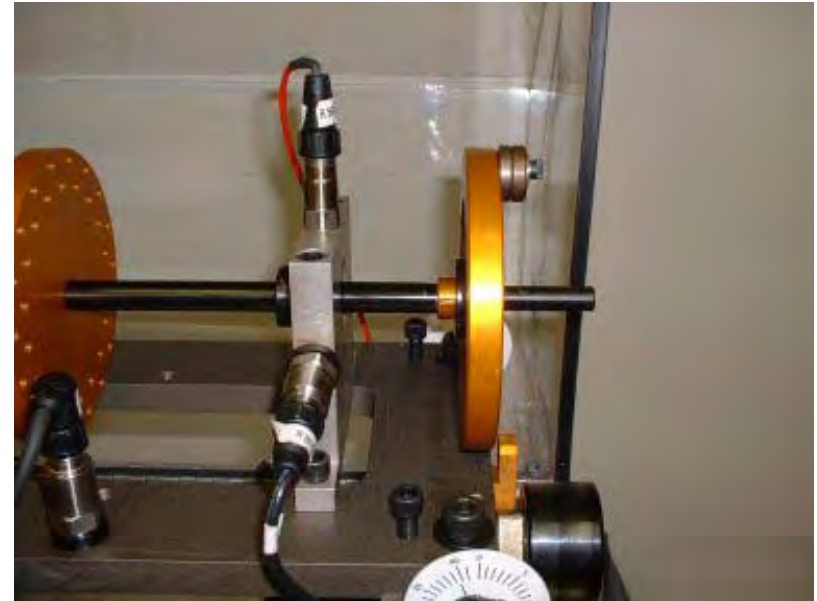
- Reliability-Increased bearing life
- Reliability-Decreased operating stresses that could lead to cracking and early failure
- Save energy (3 to 5% reduction is typical)
- Maintain optimum product quality
- Minimize operator fatigue
- Noise reduction to meet OSHA requirements
- Enhance Customer satisfaction

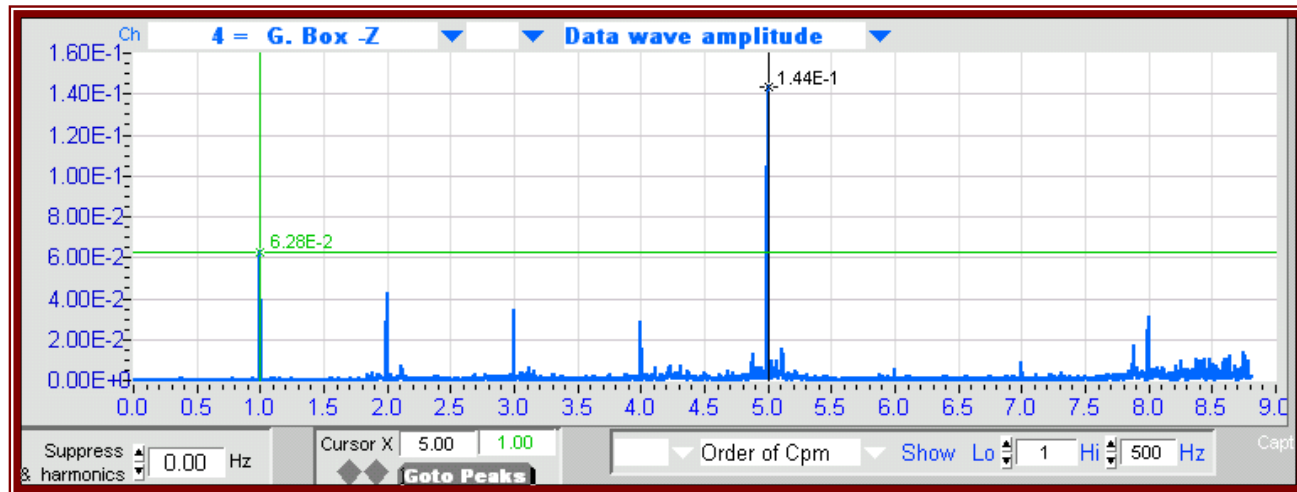
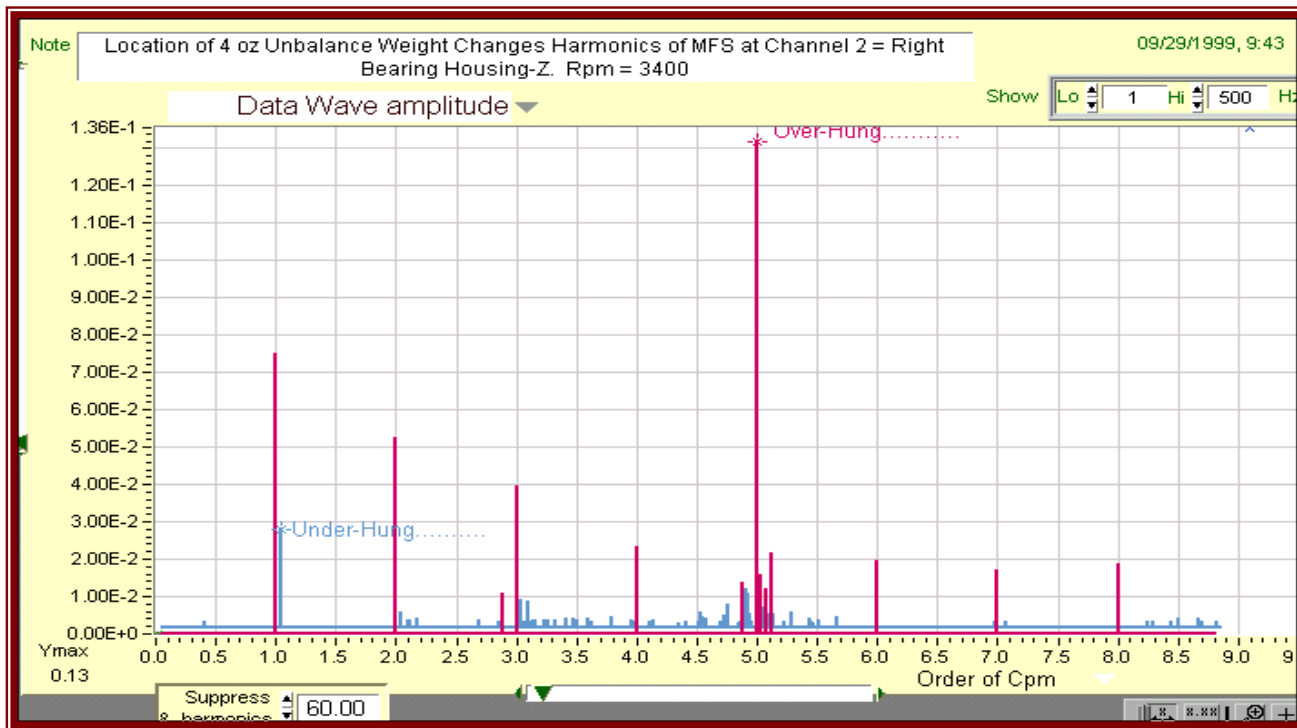
Unbalance

Center-hung Rotors

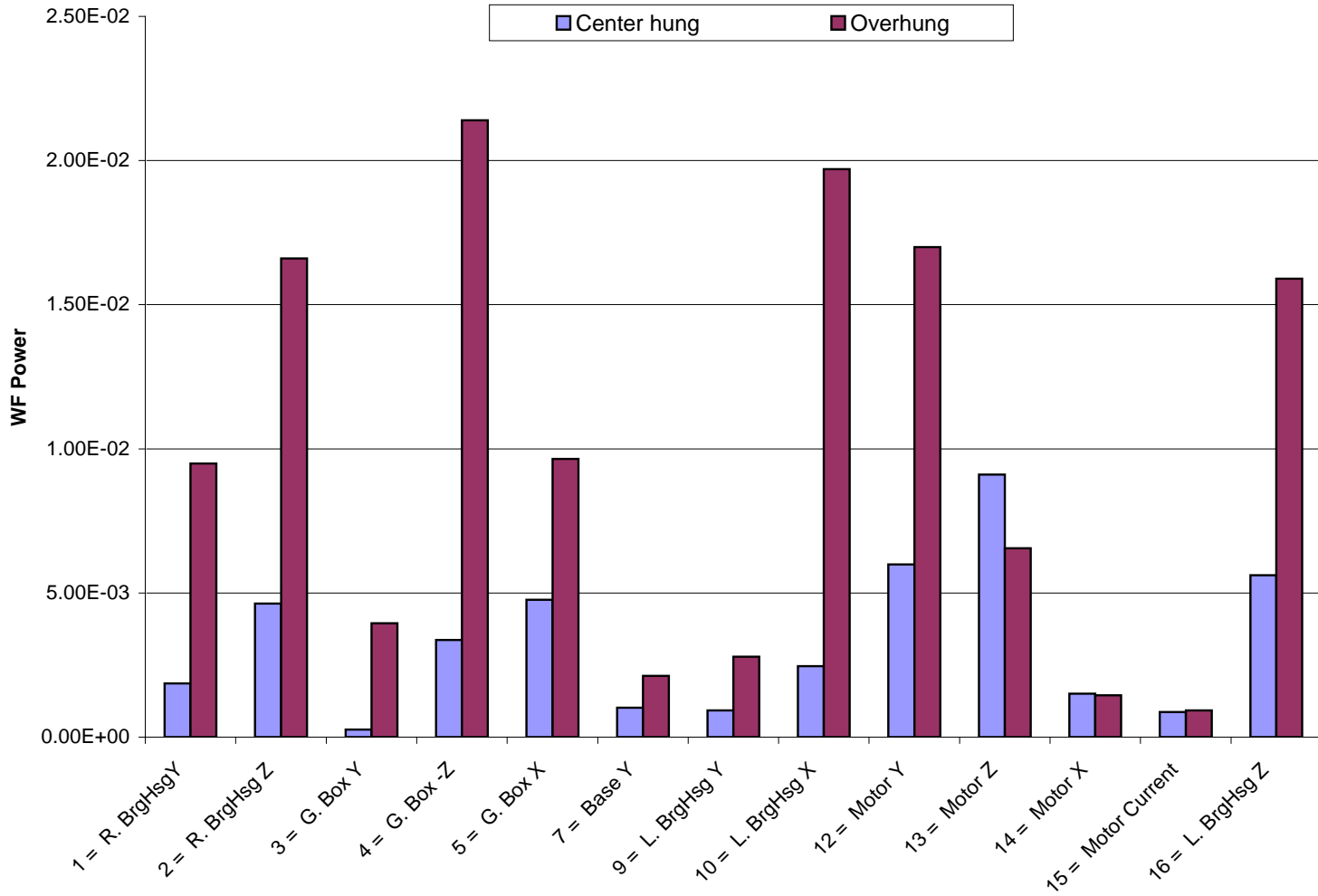


Over-hung Rotor





Vibration level comparison



Before Balancing

- ❑ Must establish that IX vibration is caused by mass unbalance
- ❑ A variety of other mechanisms can produce synchronous rotational speed vibration. Some examples:
 - Misalignment or Other Preload
 - Excessive Bearing Clearance
 - Bent Shaft or Rotor
 - Cracked Shaft
 - Soft Foot
 - Gyroscopic Effects
 - Load or Electrical Influence
 - Gear Pitch Line Runout
 - Locked Coupling
 - Foundation

Although the process of balancing is relatively straight forward, there are six basic issues that must be checked to assure a satisfactory outcome:

1. Check machine maintenance history
2. Check the rotor for mechanical integrity and remove dirt/debris so that it resembles as closely as possible the original condition
3. Verify that all fasteners are tight
4. Verify that the bearings are in good condition so that rotor is properly constrained
5. Verify that the rotor is securely attached to the shaft so that it cannot slip or move from the original position
6. Verify that the source of unbalance is constant so that unexpected movement cannot occur
7. Verify that the balancing operation will not be performed close to a natural frequency

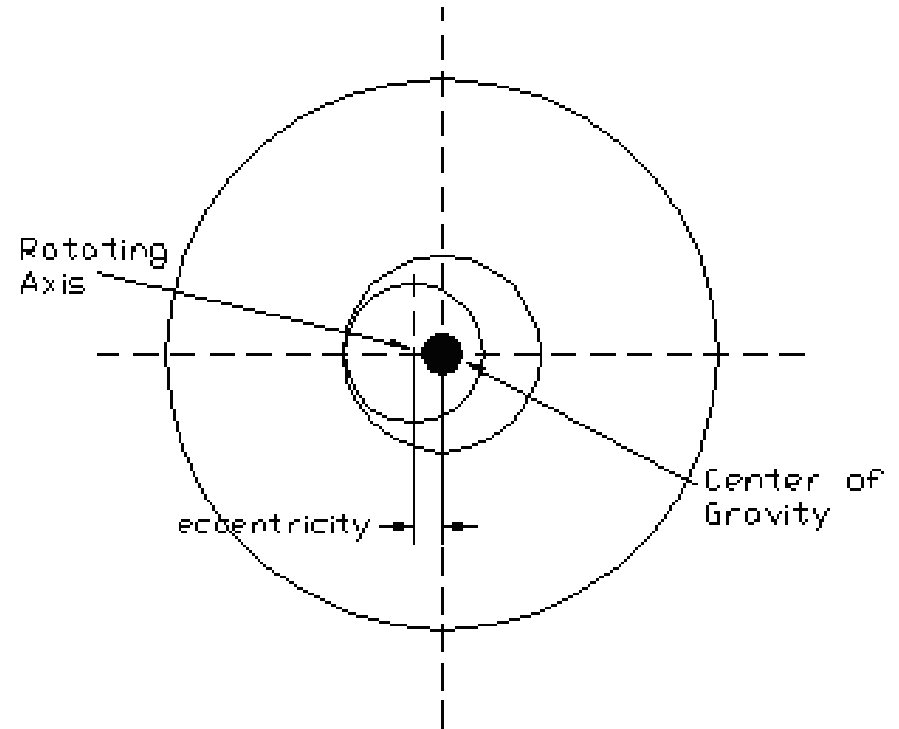
Definitions and Concepts

1. Unbalance (vibratory force or motion on bearings due to centrifugal forces.”)
2. Heavy Spot (unbalance spot on the rotor)
3. High Spot (location of vibration peak)
4. Synchronous (rotor speed or 1X) Vibration
5. Runout (Total radial displacement of the rotor)
6. Eccentricity (shaft radial displacement from the geometric axis, $U = eM$)
7. Units of Unbalance (mass times distance from shaft axis)
8. Centrifugal Force (operative force causing vibration)
9. Slow Roll and Slow Roll Compensation (slow shaft rotation to determine runout and subtracting it to balance the rotor)

Graphical Illustration

■ Eccentricity

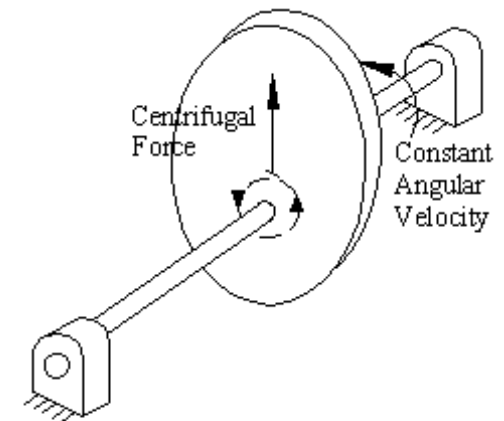
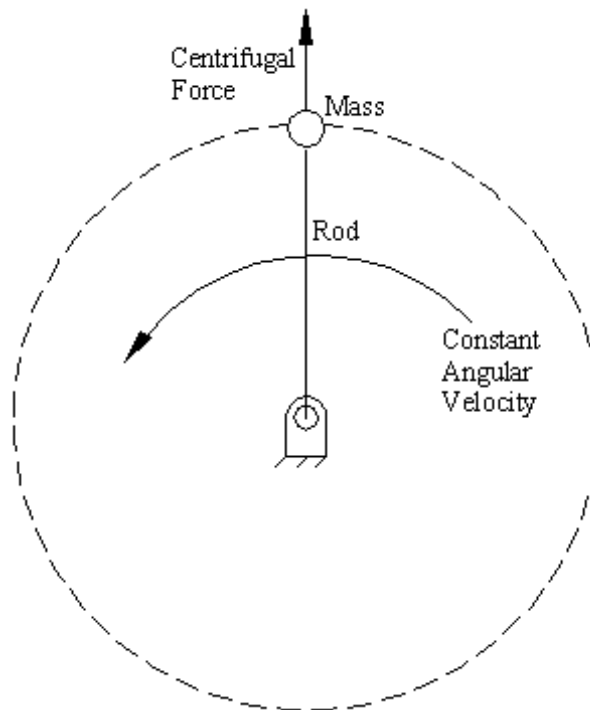
The eccentricity is the amount of displacement between the center of gravity and the rotating axis



Graphical Illustration

■ Centrifugal Force

Rotating Shaft with an Eccentric Mass Moving in a circular path experiences centrifugal force

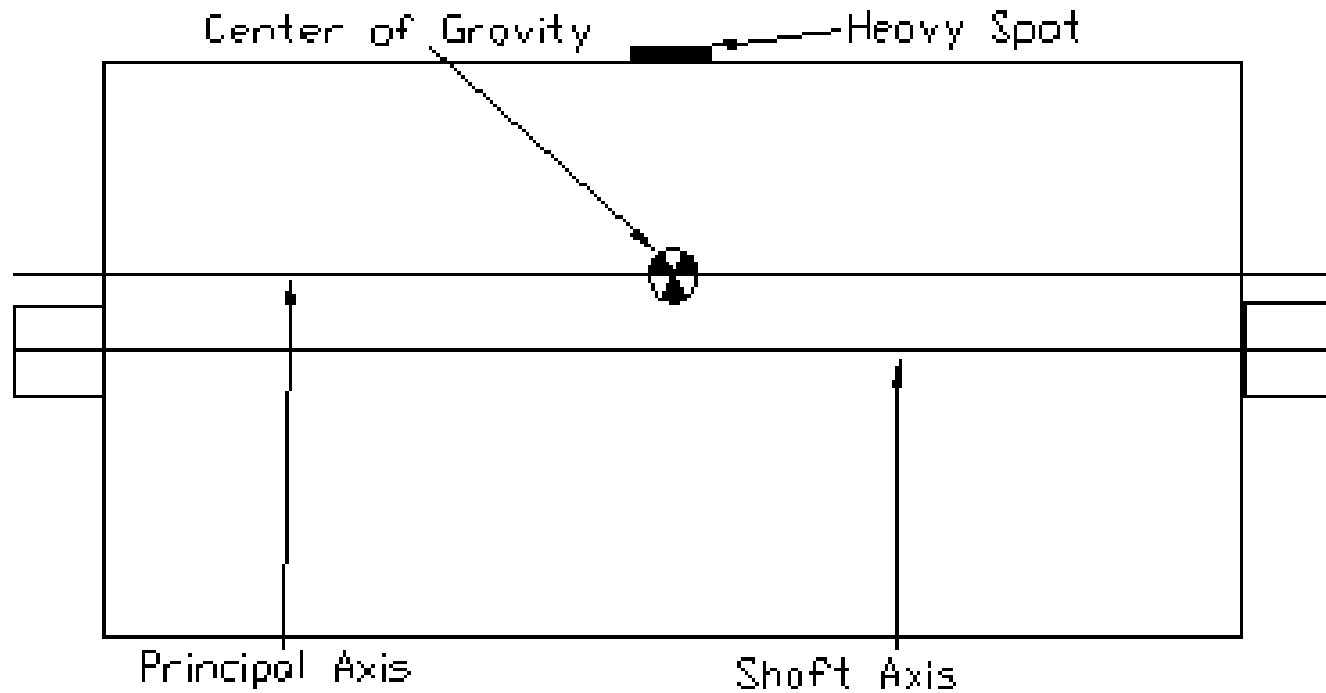


Types of Unbalance

1. Static Unbalance
2. Quasi-Static Unbalance
3. Couple Unbalance
4. Dynamic Unbalance

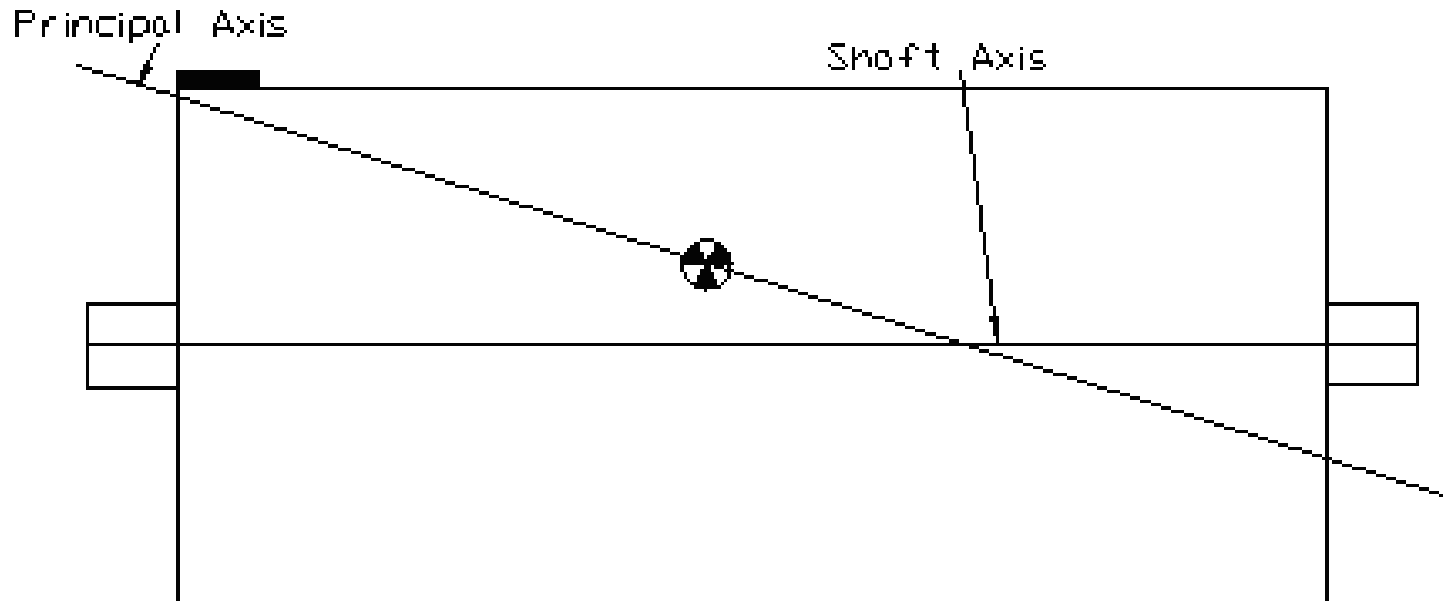
Graphical Illustration

- Static Unbalance



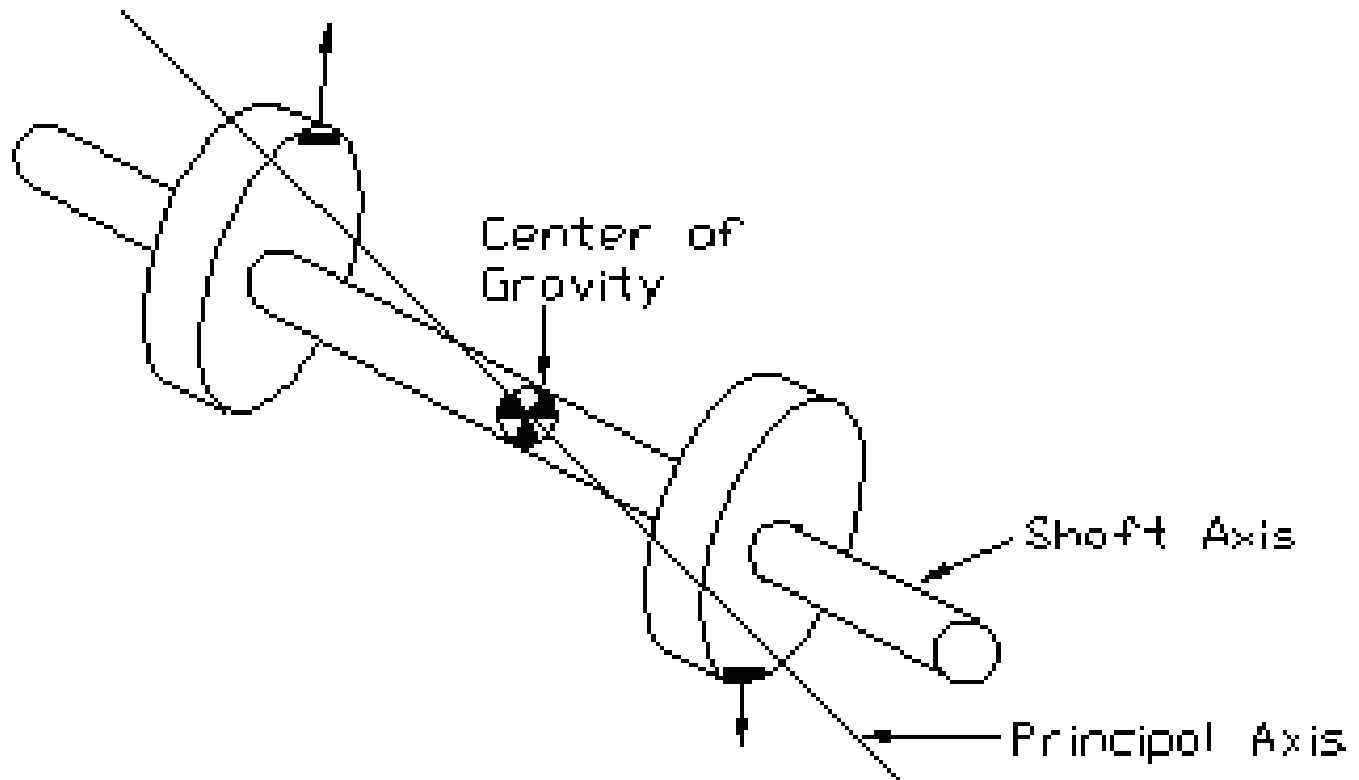
Graphical Illustration

- Quasi-static unbalance



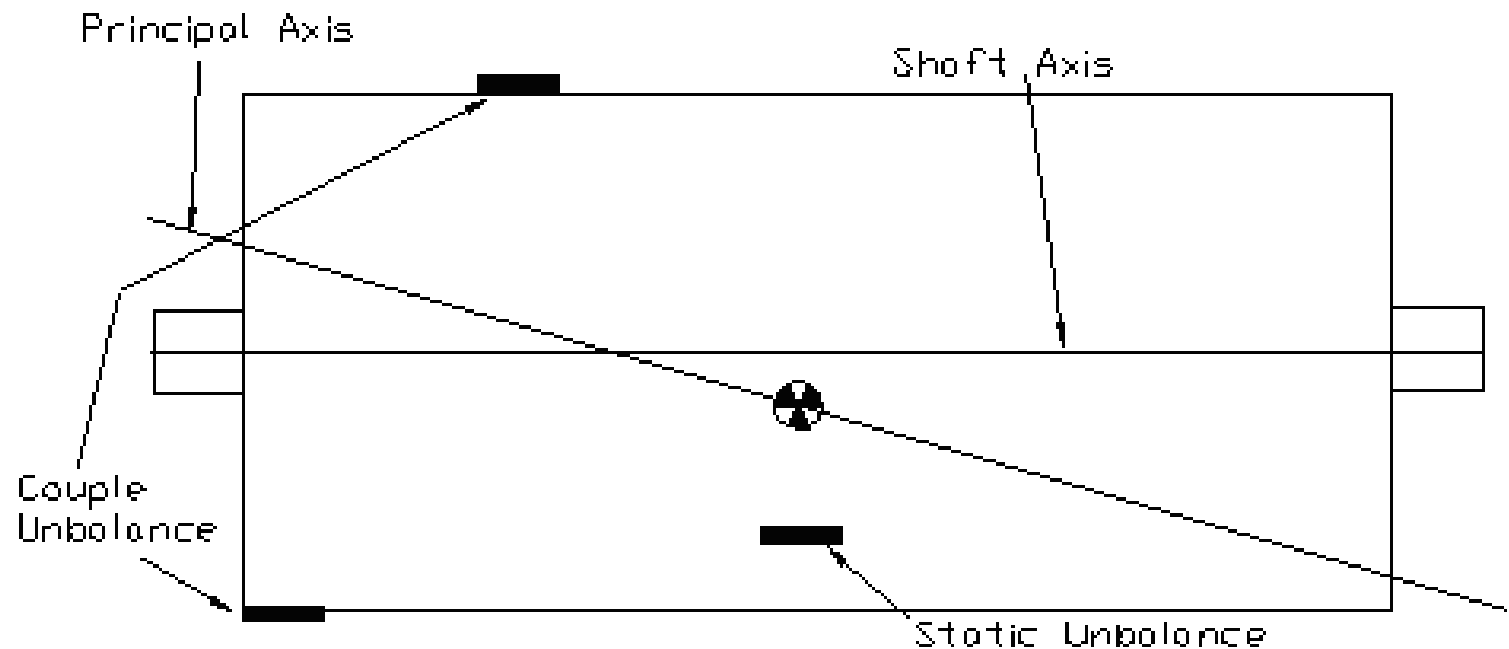
Graphical Illustration

- Couple Unbalance



Graphical Illustration

- Dynamic Unbalance



Balancing Instrumentation/issues

- Phase Measurement
 1. Trigger Sensor Method
 2. Strobe Method
- Sensors for Measurement
 1. Displacement Probes and Mountings
 2. Velocity Probe
 3. Accelerometers and Mountings
- Angle Convention
- Labeling Convention

Balancing Instrumentation

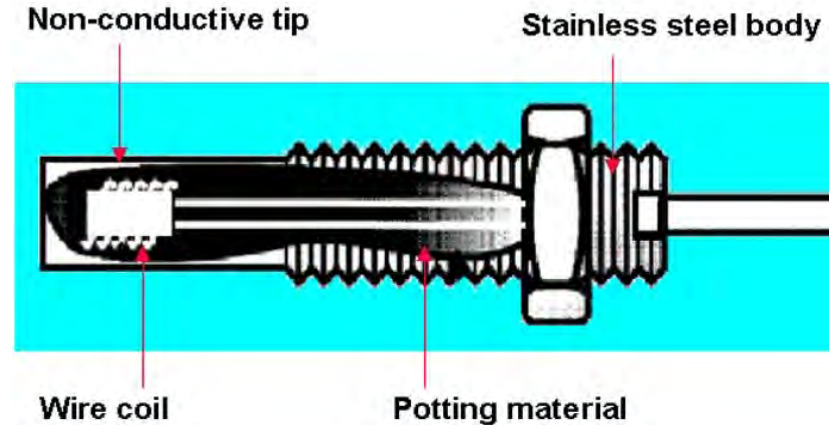
- Data Acquisition and software
 - Vibration meter
 - Single channel analyzer with phase trigger
 - Dual channel analyzer with phase trigger
 - Accurate phase measurement is essential for a success full balancing

Sensors for Vibration Measurement

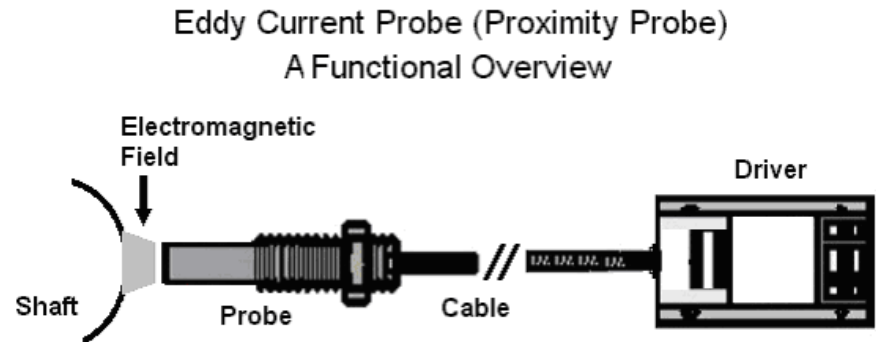
- Proximity Probe
- Velocity Pick up
- Accelerometer

Proximity Probe

Diagram of tip of proximity probe



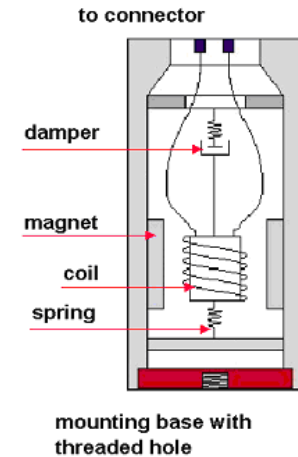
General layout for proximity probe



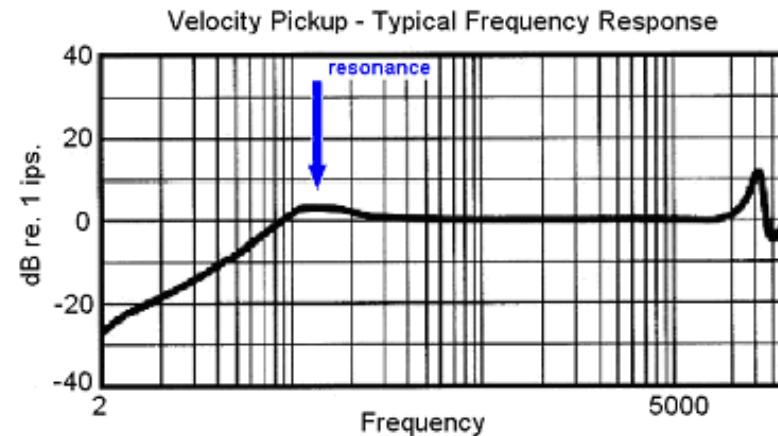
Velocity Probe

Diagram of velocity probe

- Velocity transducer consists of a permanent magnet surrounding a floating coil.
- The coil is coupled to the housing through a spring-damper.
- The magnet is rigidly attached to the sensor housing and follows the vibratory motion of the structure on which it is mounted.
- In a different configuration, the coil is attached while the magnet floats.
- The important thing here is the relative motion between the coil and magnet which produces the signal

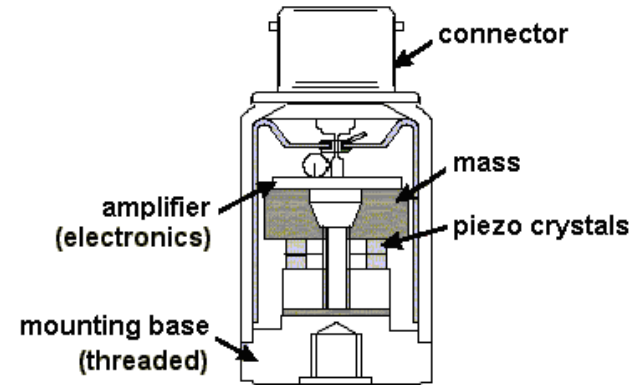


Typical frequency response for a velocity probe



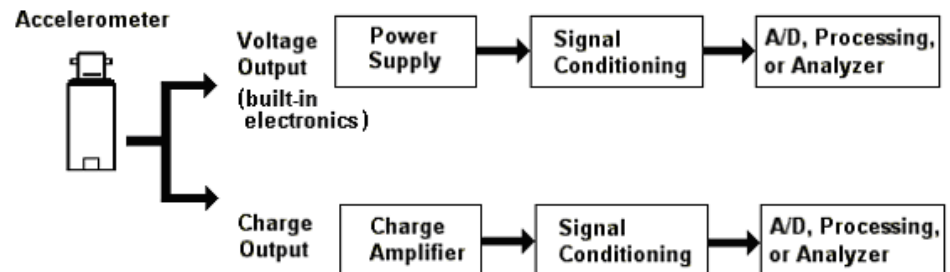
Accelerometer

Cutaway drawing of accelerometer



Accelerometer signal proces

Processing Accelerometer Signal



Types of Balancing

■ Shop Balancing

- Often use balancing small rigid rotor/components of at low speed
- Do not mimic real machine support and constraints
- High speed turbomachinery balancing is done in vacuum tight pits

■ Field Balancing

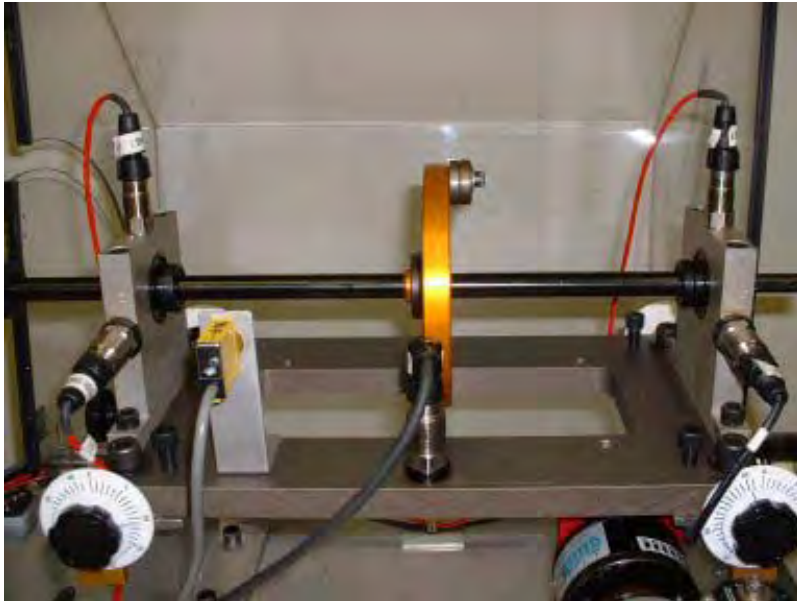
- Single Plane
- Two or more plane

Balancing-Rotor Configurations

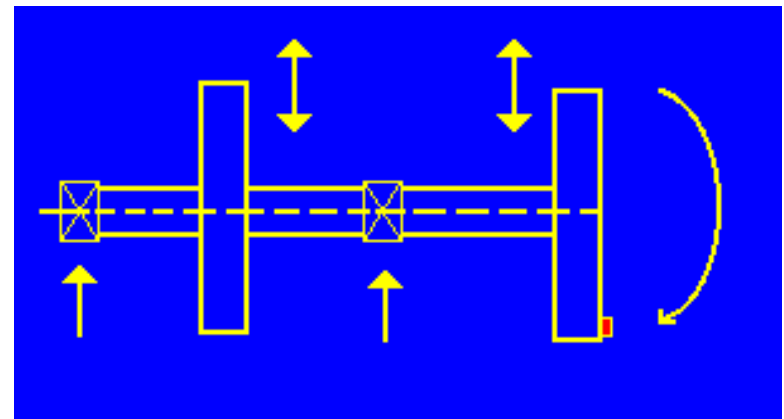
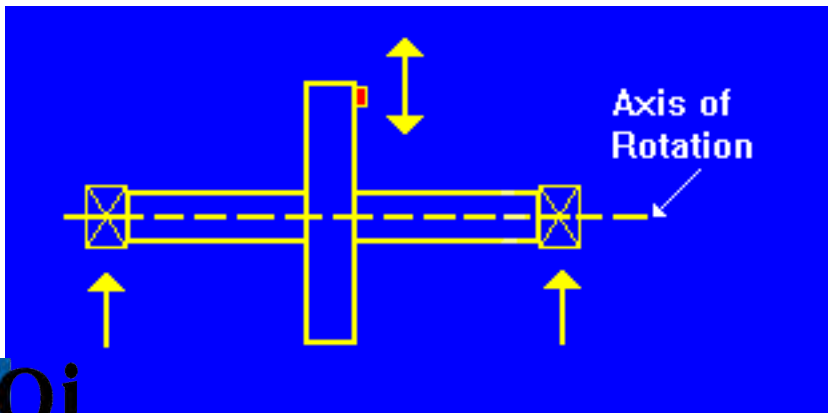
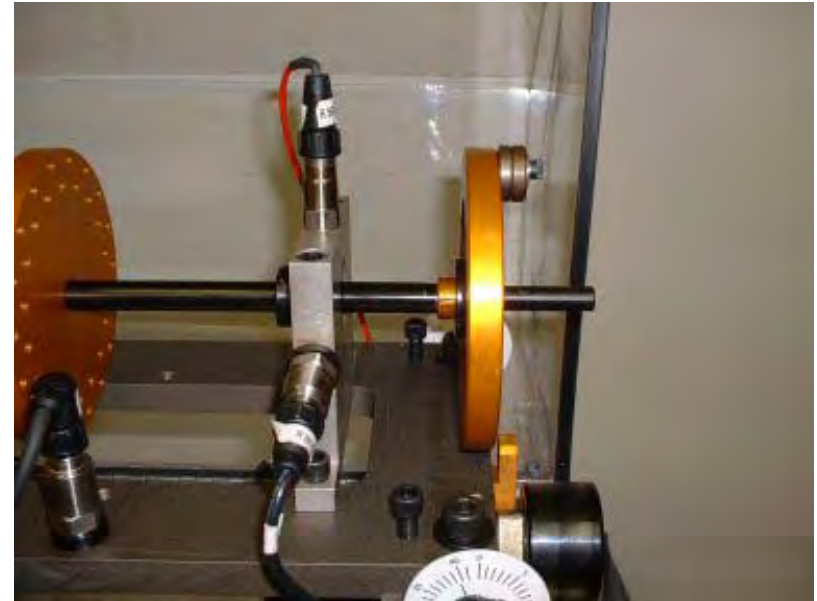
- Center Hung Configuration (thin rigid rotors)
- Over Hung Configuration (Many fan),
Vibration levels are much more severe and much harder to balance, usually due resonance and gyroscopic effects.

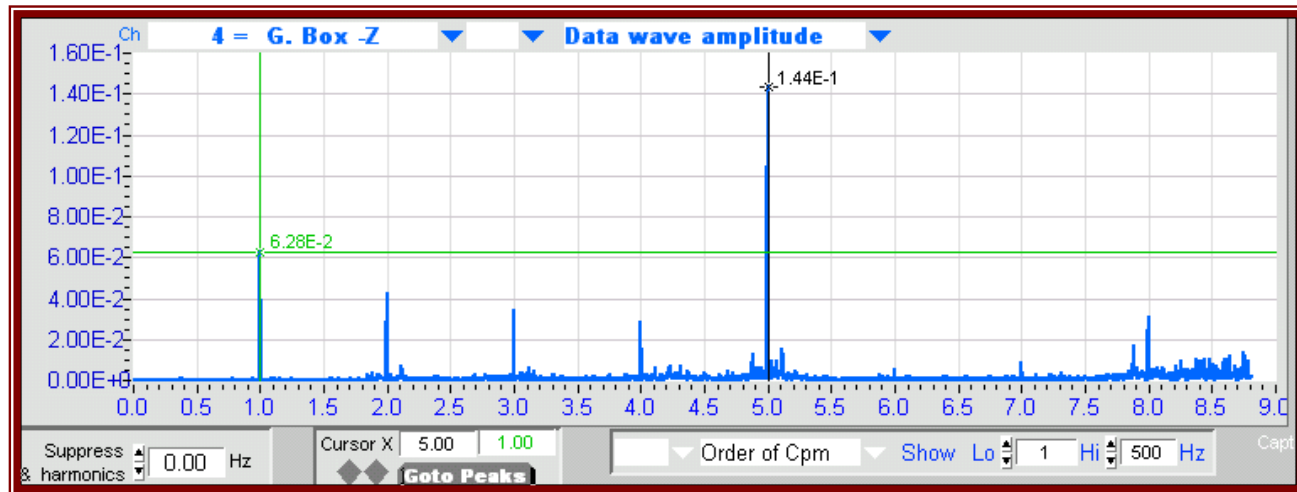
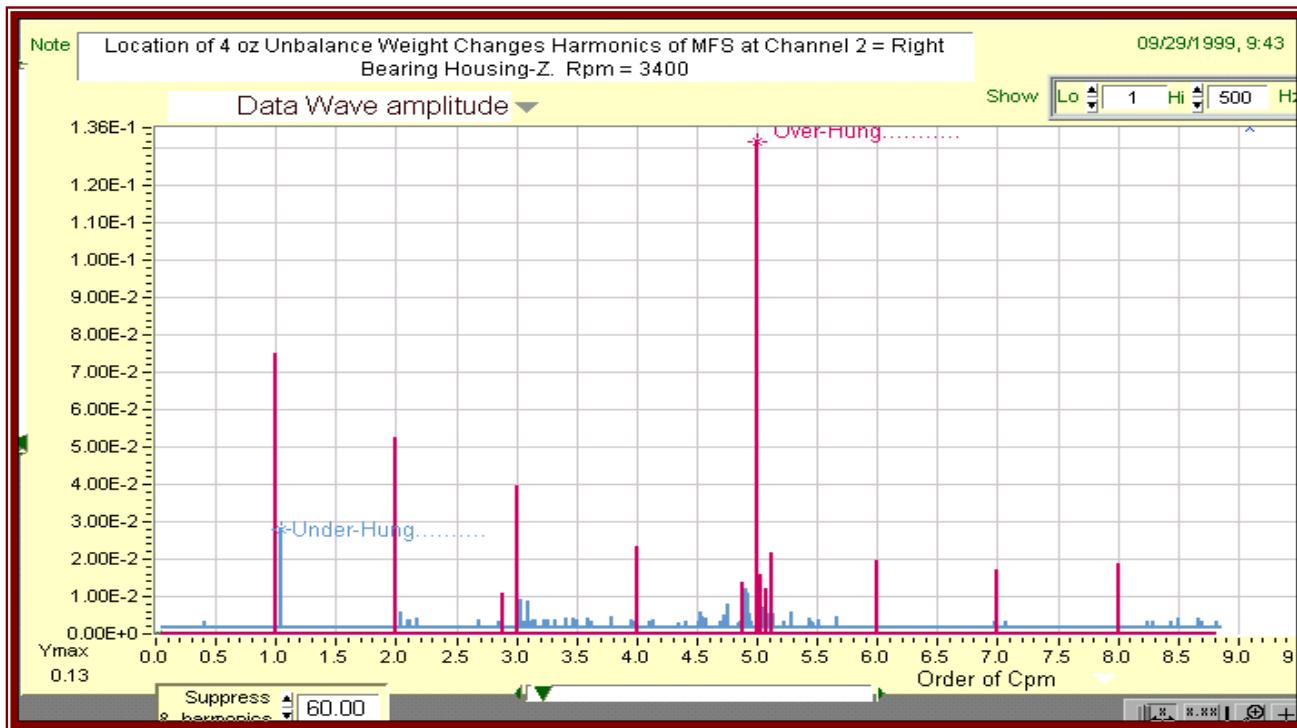
Unbalance

Center-hung Rotors

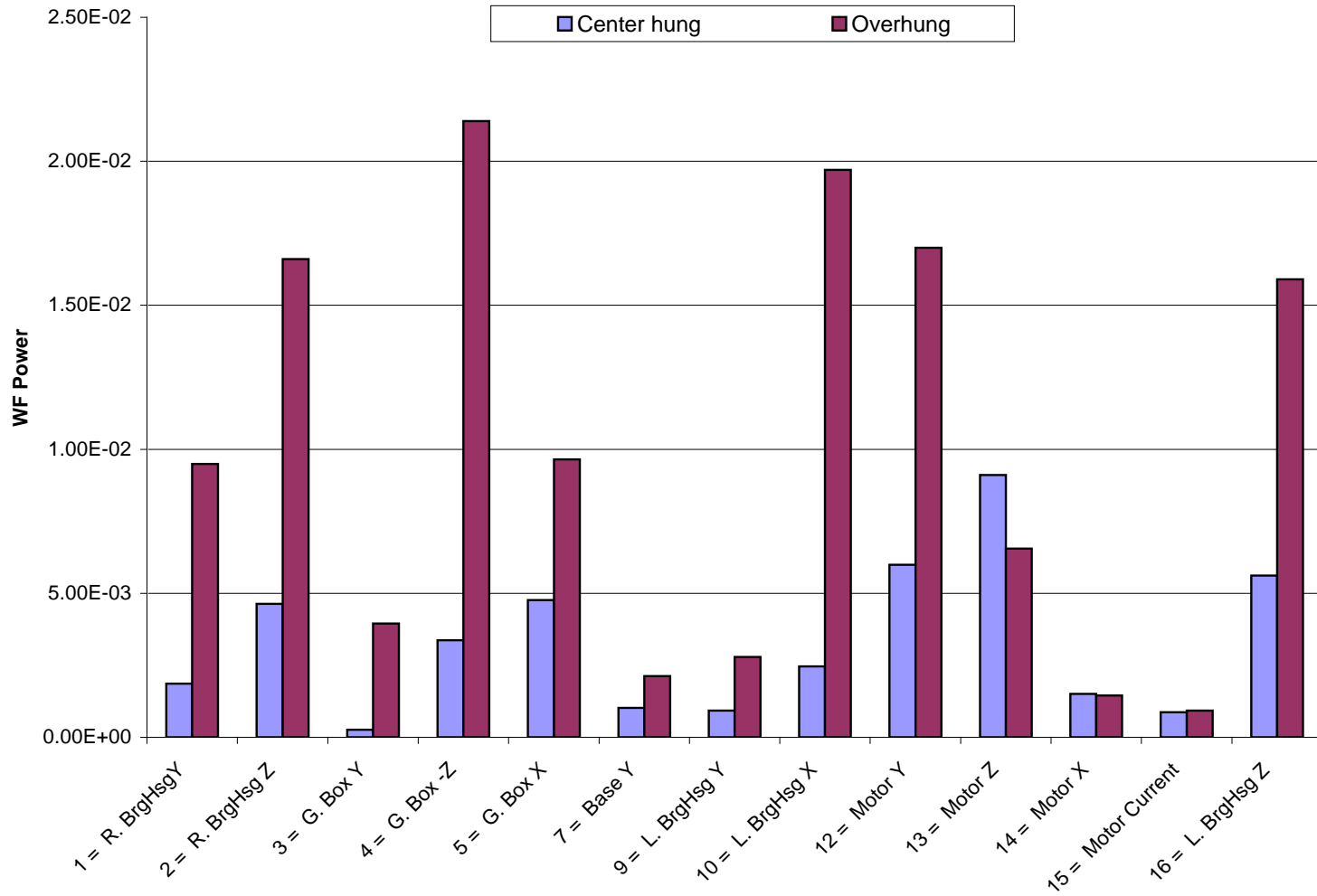


Over-hung Rotor



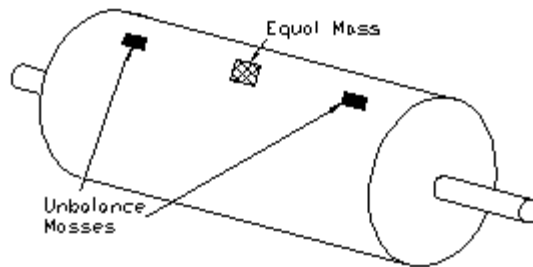


Vibration level comparison

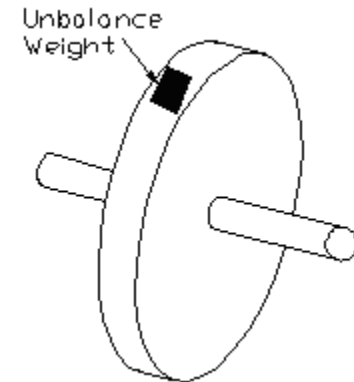


Single Plane Balancing

1. Single plane balancing is used to correct unbalance in one plane.
2. Single plane balancing corrects for static unbalance only and cannot help a machine that is experiencing a couple or dynamic unbalance
3. Any rotor that is classified as a thin rotor can be single plane balanced. Also, single plane balancing may be used for long rotors when a majority of the unbalance is in one plane.



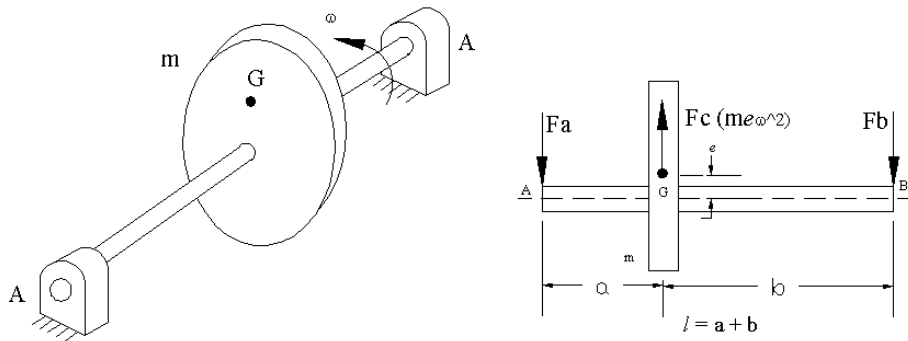
Long rotor with
single plane
unbalance



Thin rotor with single
plane unbalance

Single Plane Balancing Methods

- 1-Plane Balancing with Influence Coefficient Method.
- 1-Plane Balancing with 4 Run Without Phase Method.

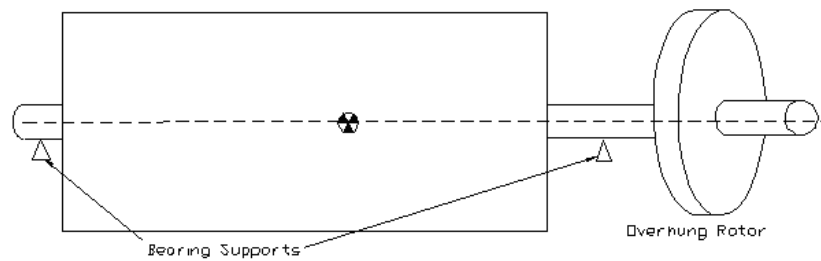


$$F_A = \frac{b}{l} F_c = \frac{b}{l} me\omega^2$$

$$F_B = \frac{a}{l} F_c = \frac{a}{l} me\omega^2$$

$$F_s = \frac{a}{l} me\omega^2 + \frac{b}{l} me\omega^2 = me\omega^2$$

Over hung Rotor



Influence Coefficient

1. An influence coefficient or influence vector, is a complex number that describes how the 1X vibration of a machine will change when a balance weight is added to the machine. Or, how a balance weight will influence the amplitude and phase of the vibration of the rotating machine.
2. With a valid influence coefficient, the required correction weight can be calculated in one step.
3. When balancing, it is important to remember that a change in the machine since the last balancing operation may have changed the influence coefficient.

Procedure for Single - Plane Balancing Using Influence Coefficient Method.

1. Determine the state of the machine and establish that unbalance is indeed the problem.
2. With the machine at rest, turn the rotor until tachometer exciter trigger the tacho; then make a mark 'P' on the rotor under the vibration transducer
3. Run the machine at the selected speed and measure the synchronous vibration amplitude and phase lag of the tacho pulse to the next peak of the synchronous vibration signal. The phase lag is the angle from the mark P on the rotor to the vibration vector. Positive angle is measured opposite to the direction of the rotation.
4. Stop the rotor and attach a trial mass m' at an angle from the mark P, using the same convention as step 3.

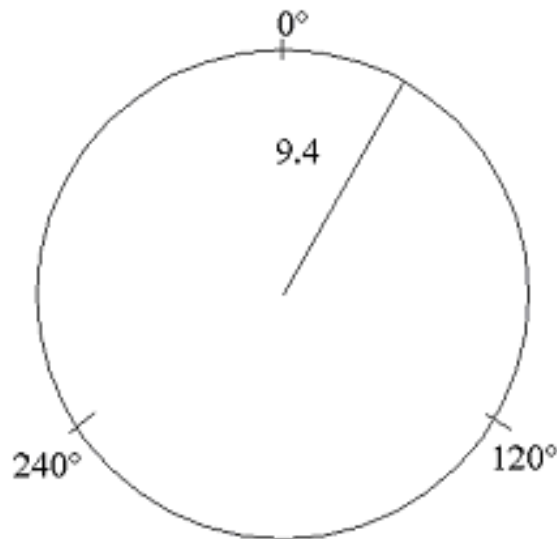
Procedure for Single - Plane Balancing Using Influence Coefficient Method Cont..

5. Start the rotor and record its vibration amplitude and phase measured at the same speed as step #3.
6. Determine the size of the correction mass and its position on the rotor.
7. Remove the trial weight and add the calculated correction mass.
8. Start the rotor and record its vibration amplitude and phase at the same speed
9. If the new vibration data is within an acceptable range, then the balancing job is complete; otherwise, either a trim balance is required (back to Step #4) or unbalance may not be the problem.

General Procedure of 4 Run Without Phase Balancing Method (Example)

First Run

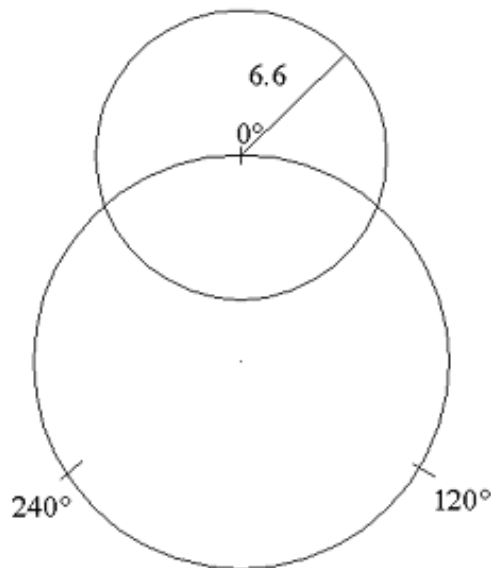
1. Measure the original vibration amplitude (9.4 mils).
2. Draw a circle with a radius equal to the original vibration amplitude (9.4 mils or units).



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

2nd Run

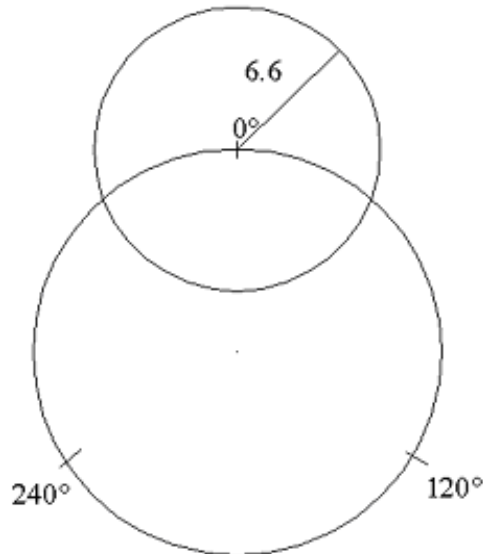
3. Choose a 0° location on the rotor and mark it (after rotor has stopped). Rotate the rotor and mark subsequent 120° and 240° locations.
4. Determine the size of the trial weight (100 g in this example) and place it at the 0° location on the rotor.



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

2nd Run

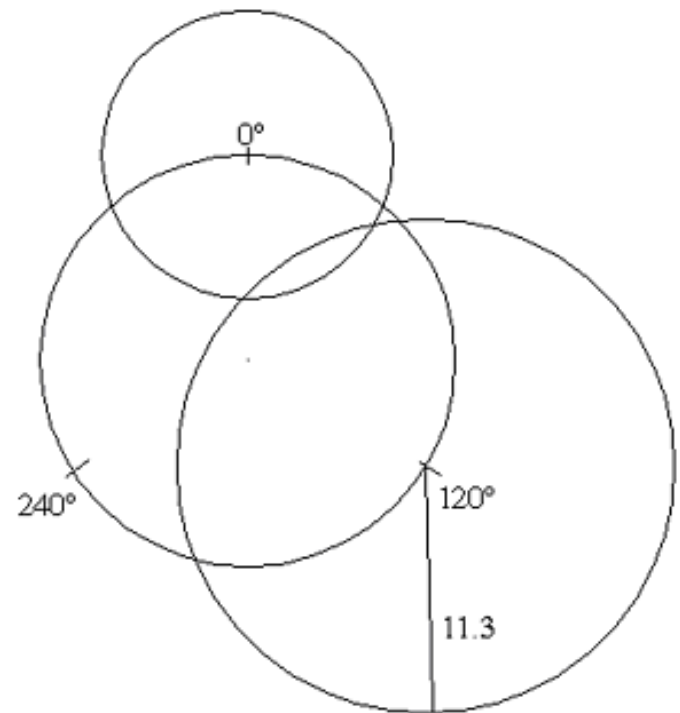
5. Start the rotor and let it reach typical operating speed.
6. Measure the new vibration amplitude. For the example, the vibration amplitude is 6.6 mils.
7. Draw a circle with radius equal to the new vibration amplitude (6.6 units) from the 0° location on the original circle as the center in Step #2.



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

3rd Run

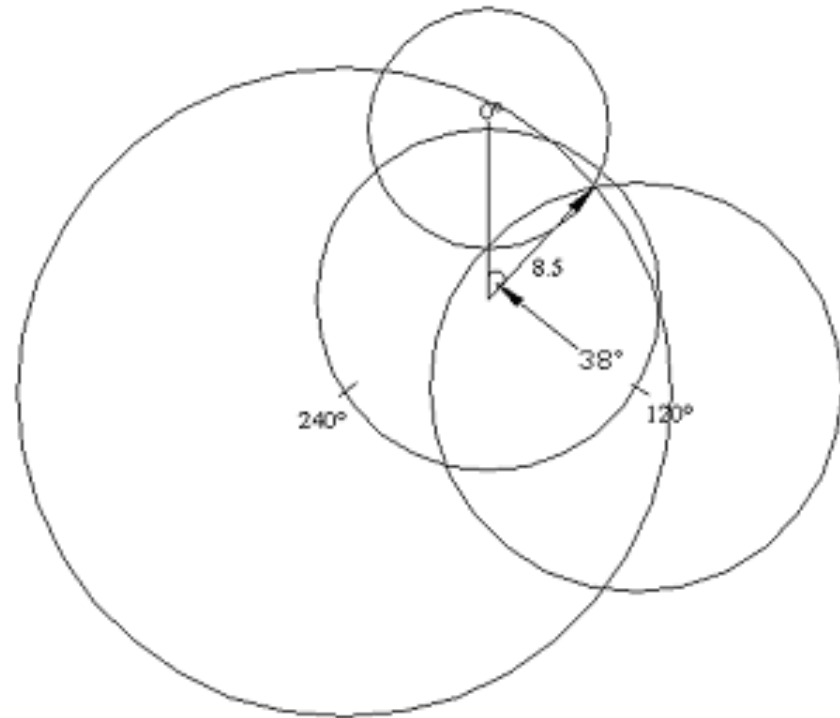
8. Stop the rotor and move the trial weight to the 120° location on the rotor.
9. Start the rotor and let it reach typical operating speed.
10. Measure the new vibration amplitude. For the example, the vibration amplitude is 11.3 mils.
11. Draw a circle with radius equal to the new vibration amplitude (11.3 units) from the 120° location on the original circle as the center in Step #2.



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

4th Run

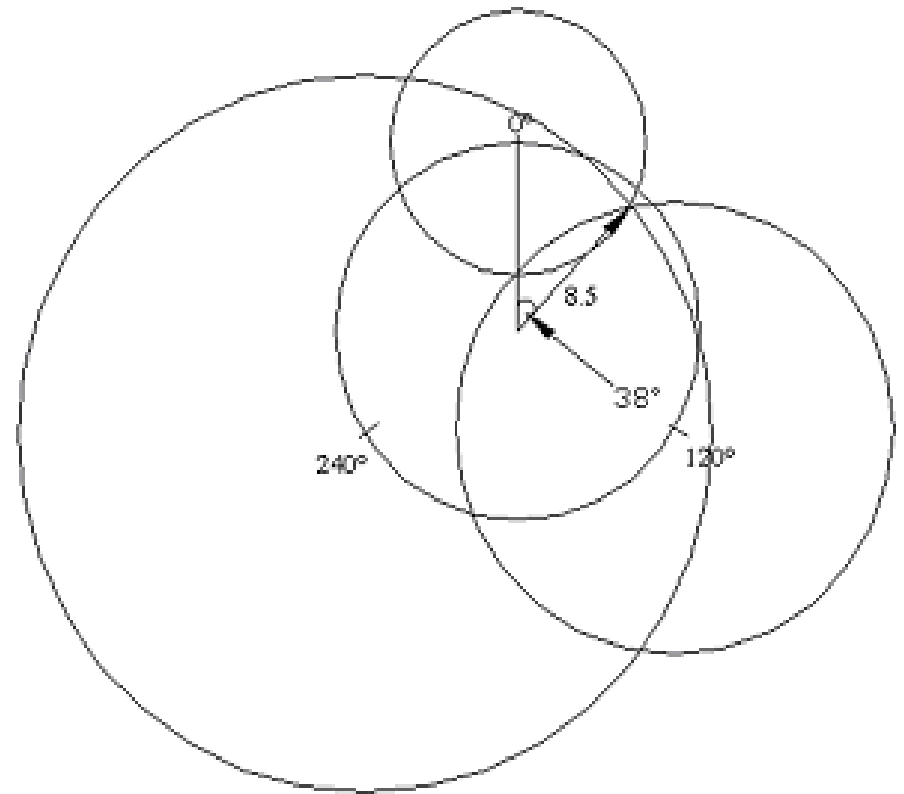
12. Stop the rotor and move the trial weight to the 240° location on the rotor.
13. Start the rotor and let it reach typical operating speed.
14. Measure the new vibration amplitude. For the example, the vibration amplitude is 17.9 mils.
15. Draw a circle with radius equal to the new vibration amplitude (17.9 units) from the 240° location on the original circle as the center in Step #2.



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

Solution

16. All three trial weight runs should intersect at one point. Draw a line from the center of the original circle in Step #2 to this point. Measure the length of this vector using the same scale as before. The vector should be labeled "T." For the example, "T" is 8.5 units.
17. The size of the correction weight can now be calculated using the same formula from single-plane balancing.



General Procedure of 4 Run Without Phase Balancing Method (Example Cont'd)

18. The correct angle to place the correction place weight must now be determined. Measure the angle between 0° and “T” using a protractor. For the example, the angle is 38°.
19. Remove the trial weight
20. Add the correction weight at the location in Step #18 (110.6 g at 38°) while the rotor is at rest.
21. Start the rotor and let it reach typical operating speed.
22. Measure the new vibration amplitude to confirm a decrease in the original measurements

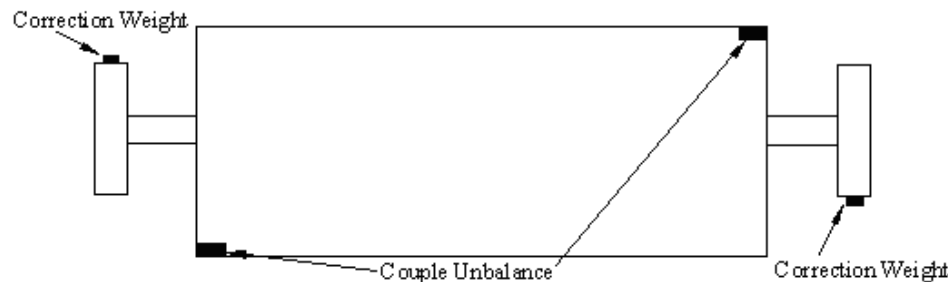
Two Plane Balancing Using Influence Coefficient Method

- Introduction
- Cross Effect
- Procedure of Influence Coefficient Method for 2-Plane Balancing

Two Plane Balancing Using Influence Coefficient Method

Introduction

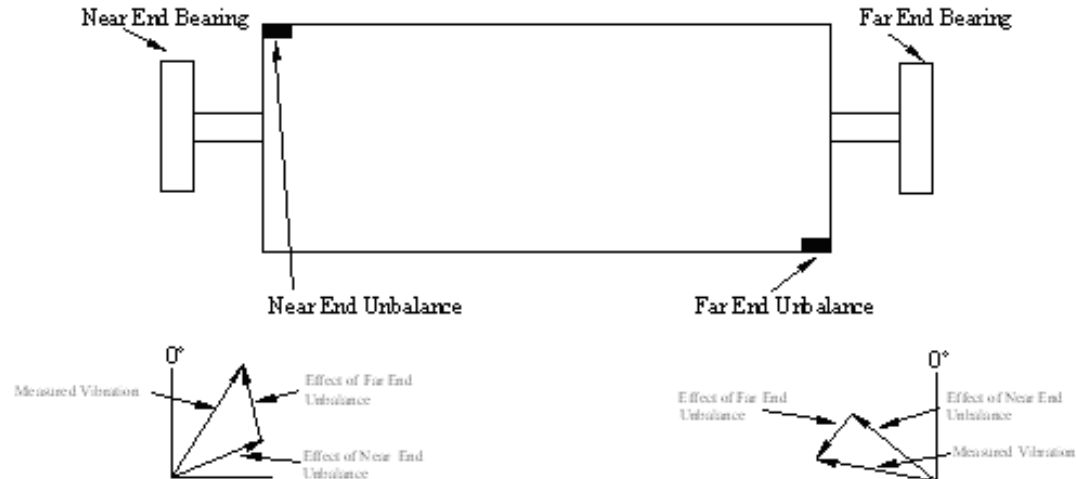
1. Two-plane balancing is performed only after unsuccessful single-plane balancing
2. Two assumptions are required for two-plane balancing: a rigid rotor and flexible (or semi-flexible), non-resonant supports.
3. Couple or dynamic unbalance is the main cause for performing two-plane balancing.
4. Two-plane balancing is almost required when a significant cross-effect are present



Two Plane Balancing Using Influence Coefficient

Cross Effect

1. The term cross-effect describes how unbalance forces are transmitted between supports
2. Cross-effect is a property of every type of rotor.
3. Two assumptions must be made when determining cross-effect.
 - a). The superposition principle applies, meaning that the two unbalance effects at each bearing can be combined in an additive manner or broken into components
 - b). Secondly, two correction weights can be used to correct any dynamic unbalance condition.



Two Plane Balancing Using Influence Coefficient

Procedure of Influence Coefficient Method

1. Verify that the cause of the vibration is unbalance.
2. With the machine at rest, turn the rotor until tachometer exciter trigger the tacho; then make a mark 'P1 & P2' on the rotor under the vibration transducers
3. Start the rotor and let it reach normal operating speed.
4. Record the vibration amplitude and phase measurements at both bearings.
5. Stop the rotor.
6. Install a trial weight at one side to establish the first correction plane.

Two Plane Balancing Using Influence Coefficient

Procedure of Influence Coefficient Method

6. Install a trial weight at one side to establish the first correction plane.
8. Start the rotor and let it reach the same speed as Step #1.
9. Record the vibration amplitude and phase measurements at both bearings.
8. Stop the rotor and remove the first trial weight.
9. Install a trial weight (could be the same weight at Step #4) at the other side to establish the second correction plane.
10. Start the rotor and let it reach the same speed as Step #1.
11. Record the vibration amplitude and phase measurements at both bearings.

Two Plane Balancing Using Influence Coefficient

Procedure (Cont'd)

12. Using the data recorded from the previous steps, calculate the correction masses and locations. The following measurements should have been recorded:

- a). Six vibration amplitudes and phase angles
- b). Two trial-weight amounts and locations

13. Stop the rotor and remove the trial weights.

14. Install the correction weights in the locations determined in Step #11.

15. Start the rotor and let it reach the same speed as Steps #1, 5, and 9.

Record the vibration amplitude and phase measurements at both bearings to confirm an improvement of the unbalance. If the results are not satisfactory, perform a trim balance.

Two-Plane Balancing Procedure without phase measurements

[L. J. Everett, Texas A&M, 1987]

1. Run unbalanced rotor, measure vibration magnitudes at 2 planes (V1).
2. Locate a trial mass on **plane 1**, run and measure vibration magnitudes in two planes.
3. Change position of trial mass in **plane 1**, run and measure vibrations.
4. Perform step 3 again for a new position in **plane 1**, measure vibrations.
5. Perform steps 2-3-4 for **plane 2**.

At this point the measured vibration magnitudes can be used to obtain unbalances graphically:

6. Draw three circles. Centers and radii of circles are based on measured vibration magnitudes in **plane 1** and angles.
7. Measure the distance (d) from the origin to the approximate intersection and angle (Θ) from the real axis to the line between the origin and intersection.

Two-Plane Balancing Procedure without phase measurements

[L. J. Everett, Texas A&M, 1987)

8. Divide d by trial mass magnitude (w) to obtain I_{11} . Subtract Θ from 180 deg to obtain the angle.
9. Draw three more circles as in step 6, using centers and radii vibration magnitudes in plane 2. Use the circles to compute I_{21} and the angle as was done in step 7-8
10. Again draw three circles as in step 6 using centers and radii measured in plane 1 from 3 more unbalanced mass in plane 1. Again compute I_{12} and the angle as was done in step 7-8
11. Repeat step 10 drawing circles with centers and radii from vibration magnitudes in plane 2 from 3 more unbalanced mass in plane 1.
12. Use obtained values for I 's and vibration magnitudes to solve the equation for unbalance.

Why Balancing May Not Work?

- Other Causes Instead of Unbalance
 1. Reciprocating forces
 2. Resonance coincident with running speed
 3. Looseness
 4. Distortion (i.e., bowed or bent shaft)
- Misalignment
- Compounding Effects
- Axial Offset
- Instability
- Nonlinear Response

Balancing Standard

Shop Balancing Standards

1. ISO Standard No. 1940 on “Balance Quality of Rotating Rigid Bodies” provides an easy guide for determining recommended balance tolerances.
2. American National Standards Institute (ANSI) adopted this standard in 1989 with ANSI S2.19-1989.
3. API has different set of standards

Shop Balancing Standards

- Balance quality grades for various groups of representative rigid rotors Source: ISO 1940-/1

Balance quality grade	Product of the relationship ($\epsilon_{per} \times \omega$) ^{1), 2)} mm/s	Rotor types – General examples
G4 000	4 000	Crankshaft/drives ³⁾ of rigidly mounted slow marine diesel engines with uneven number of cylinders ⁴⁾
G1 600	1 600	Crankshaft/drives of rigidly mounted large two-cycle engines
G630	630	Crankshaft/drives of rigidly mounted large four-cycle engines Crankshaft/drives of elastically mounted marine diesel engines
G250	250	Crankshaft/drives of rigidly mounted fast four-cylinder diesel engines ⁴⁾
G100	100	Crankshaft/drives of fast diesel engines with six or more cylinders ⁴⁾ Complete engines (gasoline or diesel) for cars, trucks and locomotives ⁵⁾
G40	40	Car wheels, wheel rims, wheel sets, drive shafts Crankshaft/wheels of elastically mounted fast four-cycle engines (gasoline or diesel) with six or more cylinders ⁴⁾ Crankshaft/drives of engines of cars, trucks and locomotives
G16	16	Drive shafts (propeller shafts, cardan shafts) with special requirements Parts of crushing machines Parts of agricultural machinery Individual components of engines (gasoline or diesel) for cars, trucks and locomotives Crankshaft/drives of engines with six or more cylinders under special requirements
G6,3	6,3	Parts of process plant machines Marine main turbine gears (merchant service) Centrifuge drums Paper machinery rolls; print rolls Fans Assembled aircraft gas turbine rotors Flywheels Pump impellers Machine-tool and general machinery parts Medium and large electric armatures (of electric motors having at least 80 mm shaft height) without special requirements Small electric armatures, often mass produced, in vibration insensitive applications and/or with vibration-isolating mountings Individual components of engines under special requirements
G2,5	2,5	Gas and steam turbines, including marine main turbines (merchant service) Rigid turbo-generator rotors Computer memory drums and discs Turbo-compressors Machine-tool drives Medium and large electric armatures with special requirements Small electric armatures not qualifying for one or both of the conditions specified for small electric armatures of balance quality grade G6,3 Turbine-driven pumps
G1	1	Tape recorder and phonograph (gramophone) drives Grinding-machine drives Small electric armatures with special requirements
G0,4	0,4	Spindles, discs, and armatures of precision grinders Gyroscopes

1) $\omega = 2\pi n/60 = \pi/10$, if n is measured in revolutions per minute and ω in radians per second.

2) For allocating the permissible residual unbalance to correction planes, see clause 7.

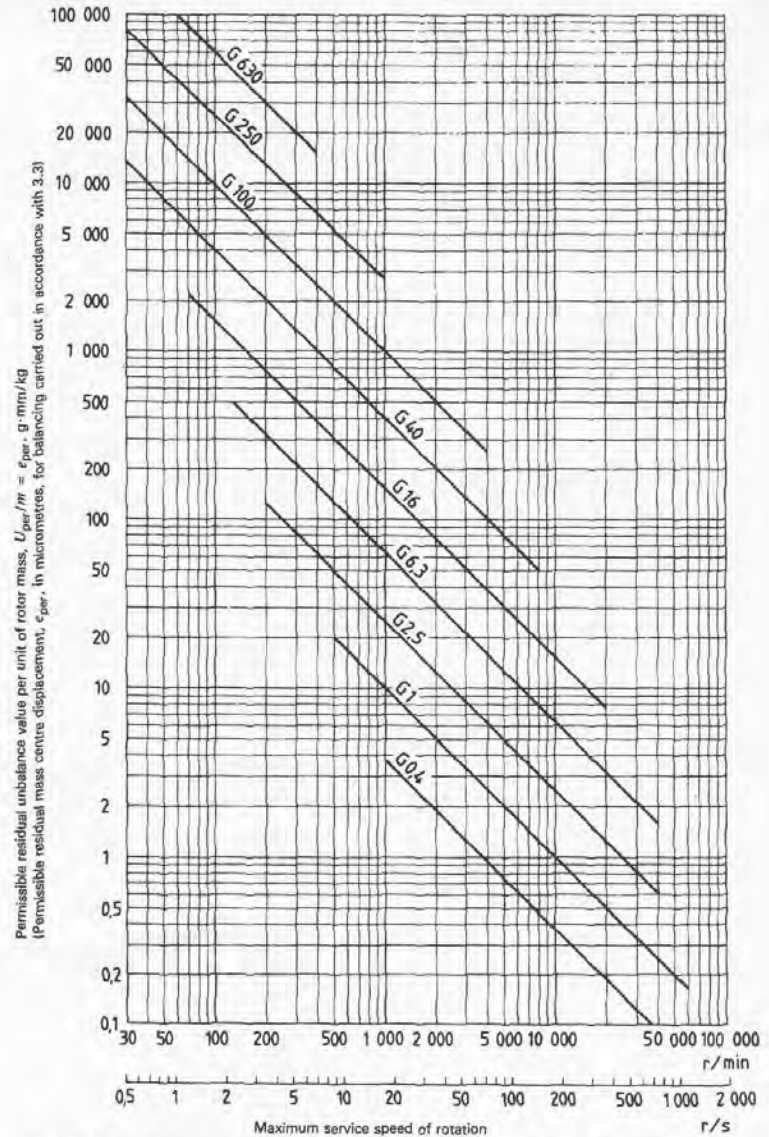
3) A crankshaft/drive is an assembly which includes a crankshaft, flywheel, clutch, pulley, vibration damper, rotating portion of connecting rod, etc. (see 3.5).

4) For the purposes of this part of ISO 1940, slow diesel engines are those with a piston velocity of less than 9 m/s; fast diesel engines are those with a piston velocity of greater than 9 m/s.

5) In complete engines, the rotor mass comprises the sum of all masses belonging to the crankshaft/drive described in note 3 above.

Shop Balancing Standards

- Maximum permissible residual specific unbalance value corresponding to various balance quality grades
Source: ISO 1940-1 (SI Units)



NOTE — The numerical value after the letter G is equal to the product of ($\epsilon_{per} \times \omega$), expressed in millimetres per second.

Balancing Standard

- Field Balancing Standards

1. There is no set of standards like the ISO 1940 for field balancing.
2. The best criterion is to use the manufacturer's set of tolerances for a given machine.