



# Engineering Encyclopedia

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## Saudi Aramco DeskTop Standards

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### COMMON MACHINE PROBLEMS

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## Common Machine Problems

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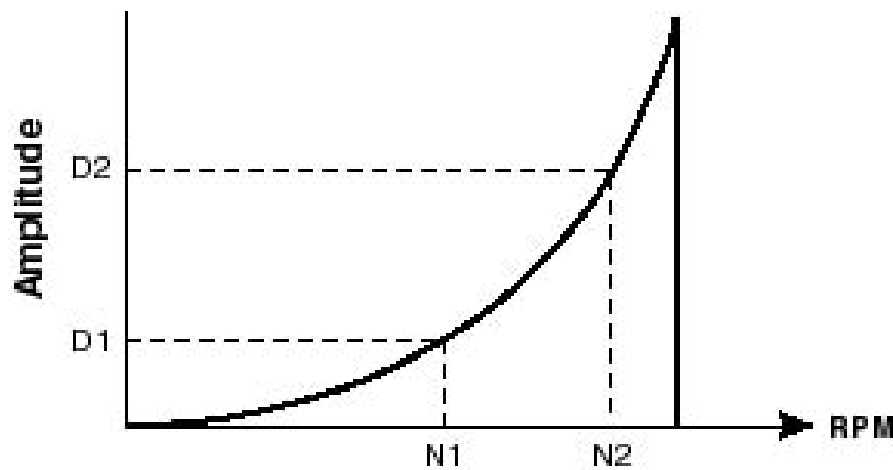
## UNBALANCE

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Unbalance in rotating machinery occurs simply when the center of mass differs from the center of rotation. Some of the causes of an unbalanced condition in rotating machinery include:

- Eccentric impeller
- Broken blades or missing parts
- Wear
- Eroded vanes
- Improper assembly
- Casting porosity or voids
- Differential thermal expansion (common in Gas Turbine)
- Process or dirt build-up
- Bent shaft

The vibration amplitude in an unbalanced condition varies proportionally with the machinery speed at approximately the square power. This condition is well identified from the coast down plot. The coast down plot is the vibration amplitude plot versus the machinery speed as the machine is shutdown. The coast down plot will be similar to exponential function  $Y \propto X^n$  where Y is the vibration amplitude, X is the machinery speed and n is the exponent factor. The exponent factor n is determined from the coast down data by selecting two points and the exponent factor can be calculated as shown in Figure 1.



**Figure 1. Exponent Determined From a Coastdown**

$$n = \frac{\text{Log}(D_2/D_1)}{\text{Log}(N_2/N_1)}$$

where: D1 is amplitude at speed N1 and D2 is amplitude at speed N2.

The unbalanced condition in machinery can be best characterized by:

- Sinusoidal vibration wave form at running speed (1X) frequency
- Periodic
- Simple non-impacting time wave form
- Unbalance vibration amplitude (1X) varies with speed (shut down or Bode' plot)
- Very low or non harmonic components (spectrum plot)
- Very little axial vibration amplitude
- Mainly radial vibration amplitude (orbit plot looks circular)

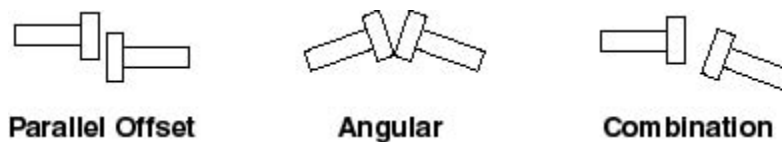
The exponent normally has a value in the region of 2 to 3 for a typical unbalance case. This is due to the fact that unbalance forces are centrifugal forces, and therefore, are a function of speed squared. If a value of 5 is derived, for example, it can mean that the exponent is influenced by the effects of a natural frequency. The exponent can therefore also be used to demonstrate the presence of a natural frequency or resonance.

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## MISALIGNMENT

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Misalignment can be thought of as the condition where the central shaft axis of two connected (coupled) machines does not fall on one continuous straight line. The two distinct types of misalignment are parallel offset and angular as shown in Figure 2, or a combination of the two. In most real cases a combination of the two exists.



**Figure 2. Misalignment of Central Shaft Axis of Two Connected Machines**

Misalignment is typically caused by the following conditions:

- Inaccurate assembly of components such as motors, pumps, etc.
- Relative position of components shifting after assembly
- Distortion due to forces exerted by piping, i.e. piping strain
- Distortion of flexible supports due to torque
- Temperature induced growth of machine structure, i.e. incorrect cold offset to compensate for thermal growth
- Coupling face not perpendicular to the shaft axis
- Soft foot or rusted shims, where the machine shifts when hold down bolts are torqued
- Foundation sag
- Causing case distortion or suppressed casing expansion, sometimes caused by rusted or seized expansion key ways
- Incorrect alignment procedure

Misalignment is best characterized by the following:

- High bearing cap vibration in radial and axial direction, with main components of 1X and 2X. This is usually seen on the bearings on either side of the coupling
- 180° phase difference in axial or radial direction across the coupling (out of phase)
- Repeatable periodic time wave form with one or two clear peaks per revolution
- High bearing temperature
- Unusual bearing wear pattern and sometimes metal flow
- Coupling wear or fatigue damage can also result

Misalignment causes a preload force on the bearing. When an orbit of relative vibration (shaft displacement) is viewed, the extent of preload will deform the circular orbit. Figure 3 shows the change in the orbit due to the change of the preload due to misalignment.

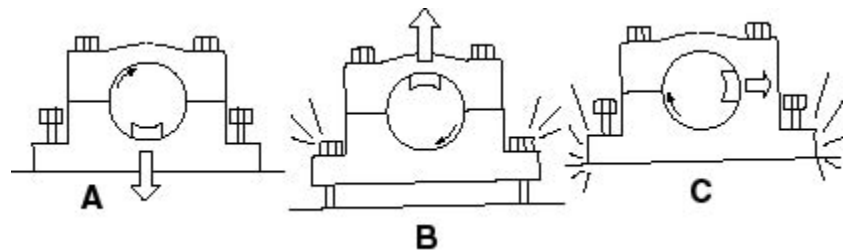


**Figure 3. Effect Of Preload Caused by Misalignment on the Displacement Orbit**

## MECHANICAL LOOSENESS

Mechanical looseness is generally a lesser problem than misalignment or unbalance. However, there are some cases where they can be put mainly into two categories which are internal and external.

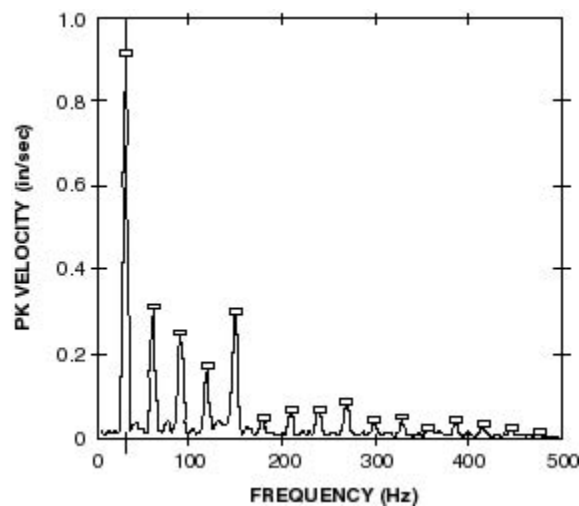
- *Internal* looseness of rotating element such as impellers, fans, bearings, coupling etc., is generally either due to excessive shaft to bearing, or bearing to housing clearance. It can also be caused by a lack of interference between the impellers and the shaft.
- *External* looseness of structure such as base mount, split casing, bearing caps, bearing support usually refers to lack of tightness of the pedestal, or hold-down bolts (see Figure 4). In both cases, the looseness can create a rocking motion.



**Figure 4. Vibration Due To External Mechanical Looseness**

## Symptoms

With internal looseness the major component is usually at 1X, with an unsteady amplitude and a string of harmonics (see Figure 5). Often directional in nature, horizontal and vertical amplitude may differ greatly when viewing the time display. The waveform is random, non-periodic. The frequency spectrum may show 1/2x. Loose impellers or fans can sometimes generate a strong 3rd or 4th harmonic due to a wobbling type motion.



**Figure 5. Mechanical Looseness Spectrum**

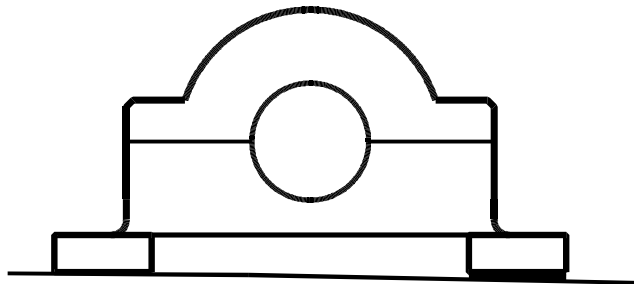
With external looseness similar characteristics exist but additionally a rocking motion can be generated and this can be detected by taking phase measurements.

With gear type couplings if excessive wear or looseness exists between the teeth it can cause an unbalance due to the spool-piece becoming eccentric, and the amplitude and phase can vary between successive runs. A similar effect can occur when tapered coupling hubs have an insufficient interference fit. This is usually evident by fretting on the taper and the key.

## SOFT FEET

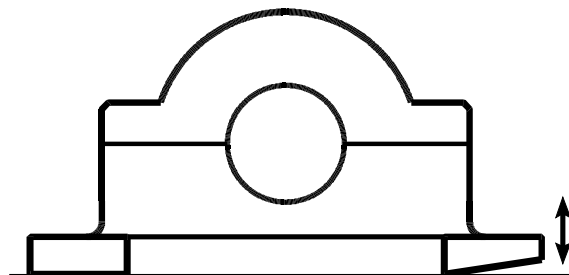
The condition of “soft-feet” is normally encountered with horizontal rigidly mounted machines. It is not normally a problem with flexible and split line mounted machines such as some centrifugal gas compressors. There are two common conditions that result in the effects referred to as “soft-feet.”

The first condition causing “soft feet” is excessive flexibility under a mounting foot due to a thick pack of shims, and this is often aggravated by corrosion (see Figure 6).



**Figure 6. Soft Foot Due To Thick Shims**

The second condition causing “soft feet” is machine feet and mounting base plate not being parallel (see Figure 7). This can be caused by distortion of the machine casing and feet, and also by sagging or cracking of the foundation.

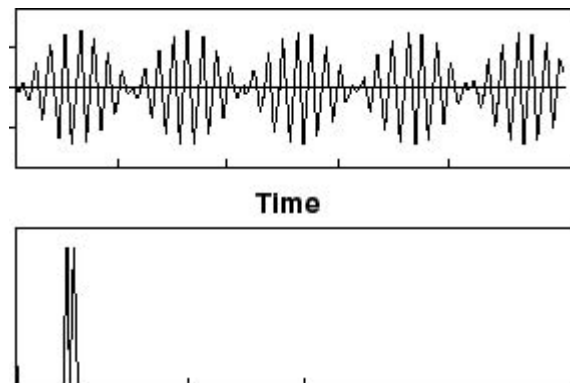


**Figure 7. Soft Foot Due To Skewed Feet**

## Characteristics

Characteristics of soft feet include:

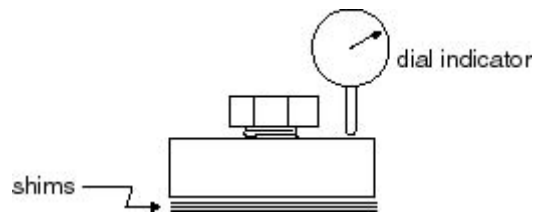
- High vibration in the vertical direction
- Much higher vibration on the machine foot than on the base plate
- A shift in casing natural frequency
- With induction electric motors, which are particularly sensitive to mounting distortion, the frequency components are generated at line frequency (60 Hz) and twice line frequency (120 Hz). These are not synchronous with, or harmonics of the running speed (RPM). Zooming the frequency spectrum or using resolution (i.e. more lines), can show both component closely to each other, i.e., (60 Hz and 1x for 3,600 rpm motor) (See Figure 8).



**Figure 8. Beat Frequencies Time Waveform and Spectrum**

## Testing

To test for a soft foot condition a dial indicator is mounted on the foot and the clock set on the machine foot (see Figure 9). The hold down bolt on that foot is then released, and also the adjacent bolts, and the total deflection recorded.



**Figure 9. Checking Soft Foot Using Dial Indicator**

In general, readings in the order of 1 to 2 mils are acceptable and readings in the order of more than 5 mils are not. If the machine foot is distorted it may be necessary to set more than one indicator on the foot.

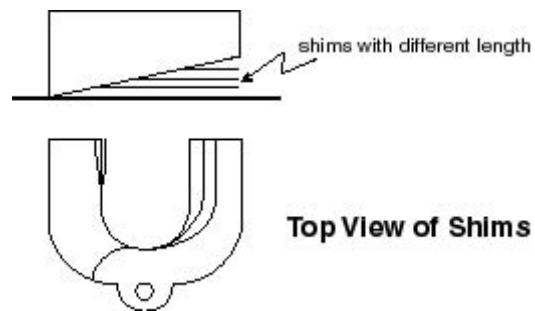
## Correction

Where a thick shim pack is the problem it should be removed and a thick machined block installed with one or two shims for adjustment (see Figure 10). Block and shims should be of stainless steel to avoid corrosion problems.



**Figure 10. Replacing Thick Shims Pack By Thick Machined Block**

Where the feet and the base are not parallel, it may require machining of the feet, but this could be a major task. An alternative solution may be to install tapered blocks or shims (see Figure 11).



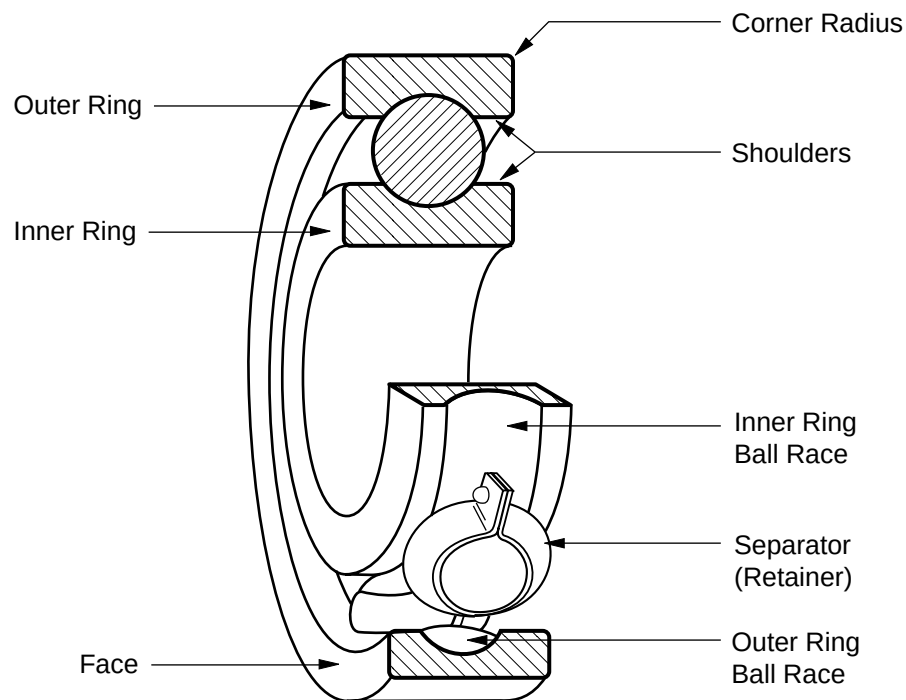
**Figure 11. Correction of Skewed Foot**

## ANTI-FRICTION BEARING

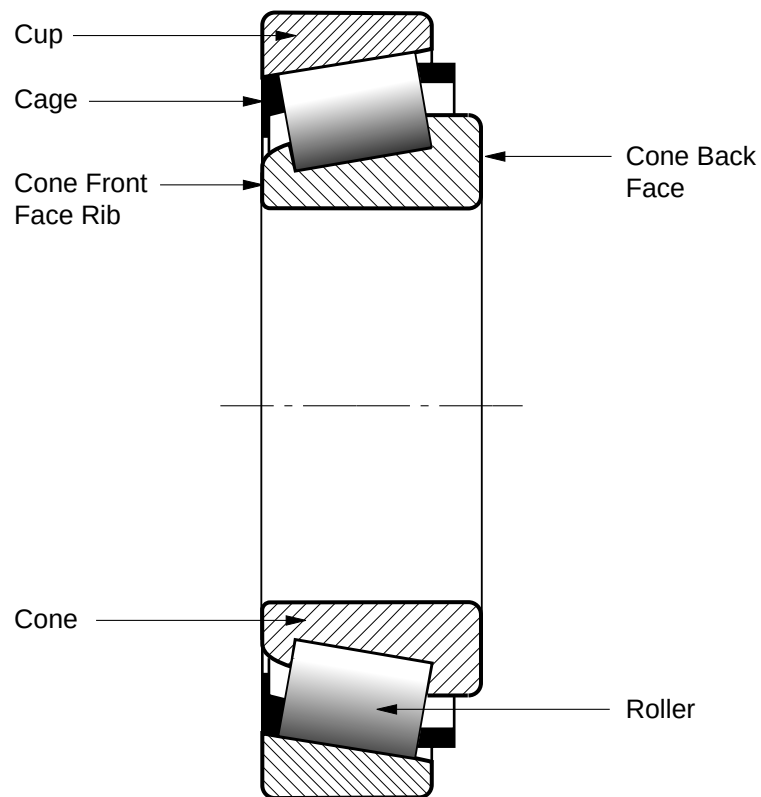
*Anti-friction bearing*, rolling contact bearing or rolling element bearing are terms used to describe the group of bearings in which the main load is transferred through elements in rolling contact rather in sliding contact.

### Configuration

Bearings are designed and built to take pure radial load, pure thrust load, or a combination of the two. There are four essential parts in a ball bearing or tapered roller bearing. These are the outer ring, the inner ring, the balls and the separator. The separator sometimes is omitted depending on the application, but it is important to eliminate rubbing contact between the balls (see Figure 12 and Figure 13).



**Figure 12. Anti-Friction Bearing Configuration – Roller Bearing**



**Figure 13. Anti-Friction Bearing Configuration – Tapered Roller Bearing**

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## Anti-Friction Bearing Vibration

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Five frequencies are associated with defective bearings. They are:

- Ball Pass Frequency of Outer race, BPFO
- Ball Pass Frequency of the Inner race, BPFI
- Fundamental Train Frequency, FTF
- Ball Spin Frequency, BSF
- Rotating Unit frequency

### Ball Passing Frequencies

Ball passing frequencies are generated as the balls or rollers pass over a defect on the raceway. If the defect is in the outer race, the frequency generated at the outer race is roughly equal to 40 % of the number of balls times the revolutions per seconds (RPS). If the defect is in the inner race, the frequency is approximately 60 % of the product of the number of balls and RPS. This means 40 % of the balls passing the defect on the outer race during each revolution and 60 % passing over the defect on the inner race during each revolution.

$$\text{BPFO (Hz)} = 0.40 \times \text{number of balls} \times \text{rps}$$

$$\text{BPFI (Hz)} = 0.60 \times \text{number of balls} \times \text{rps}$$

### Ball Spin Frequency

Ball spin frequency is generated when there is a defect on the ball or the roller strikes the raceway. The frequency generated can be as much as two times the BSF because the defect strikes both races during each revolution. Ball spin frequency can also be produced when the balls are thrusting against the cage.

## Fundamental Train Frequency

The fundamental train frequency is the rotating speed of the balls or rollers and the cage assembly. This frequency is not encountered very often but it can occur when some defects affect the rotation of the train. This defect generates a frequency at 40% of running speed.

$$\text{FTF (Hz)} = 0.4 \times \text{rps}$$

## Rotating Unit Frequency

The rotating unit frequency generated by the speed of the rotating unit is caused by some type of rotating unbalance.

## Mathematical Relations

The following are mathematical relationships used in the study of mechanical bearing rotation.

$$\text{RPS} = \text{RPM} / 60$$

$$\text{FTF (Hz)} = \text{RPS} / 2 \times (1 - B_b / P_d \times \cos \theta)$$

$$\text{BPFI (Hz)} = N_b / 2 \times \text{RPS} (1 + B_d / P_d \times \cos \theta)$$

$$\text{BPFO (Hz)} = N_b / 2 \times \text{RPS} (1 - B_d / P_d \times \cos \theta)$$

$$\text{BSF (Hz)} = P_d / 2 B_d \times \text{RPS} (1 - (B_d / P_d)^2 \times \cos^2 \theta)$$

$$\text{RPS} = \frac{\text{RPM}}{60}$$

$$\text{FTF} = \frac{\text{RPS}}{2} \left( 1 - \frac{B_d}{P_d} \times \cos \theta \right)$$

- RPM, revolution per minute
- RPS, revolution per second
- FTF, fundamental train frequency
- BPFI, ball pass frequency of the inner race
- BPFO, ball pass frequency of the outer race
- BSF, ball spin frequency
- Bd, ball or roller diameter
- Nb, Number of balls or rollers
- Pd, pitch diameter.
- $\theta$ , contact angle, deg.

Example:

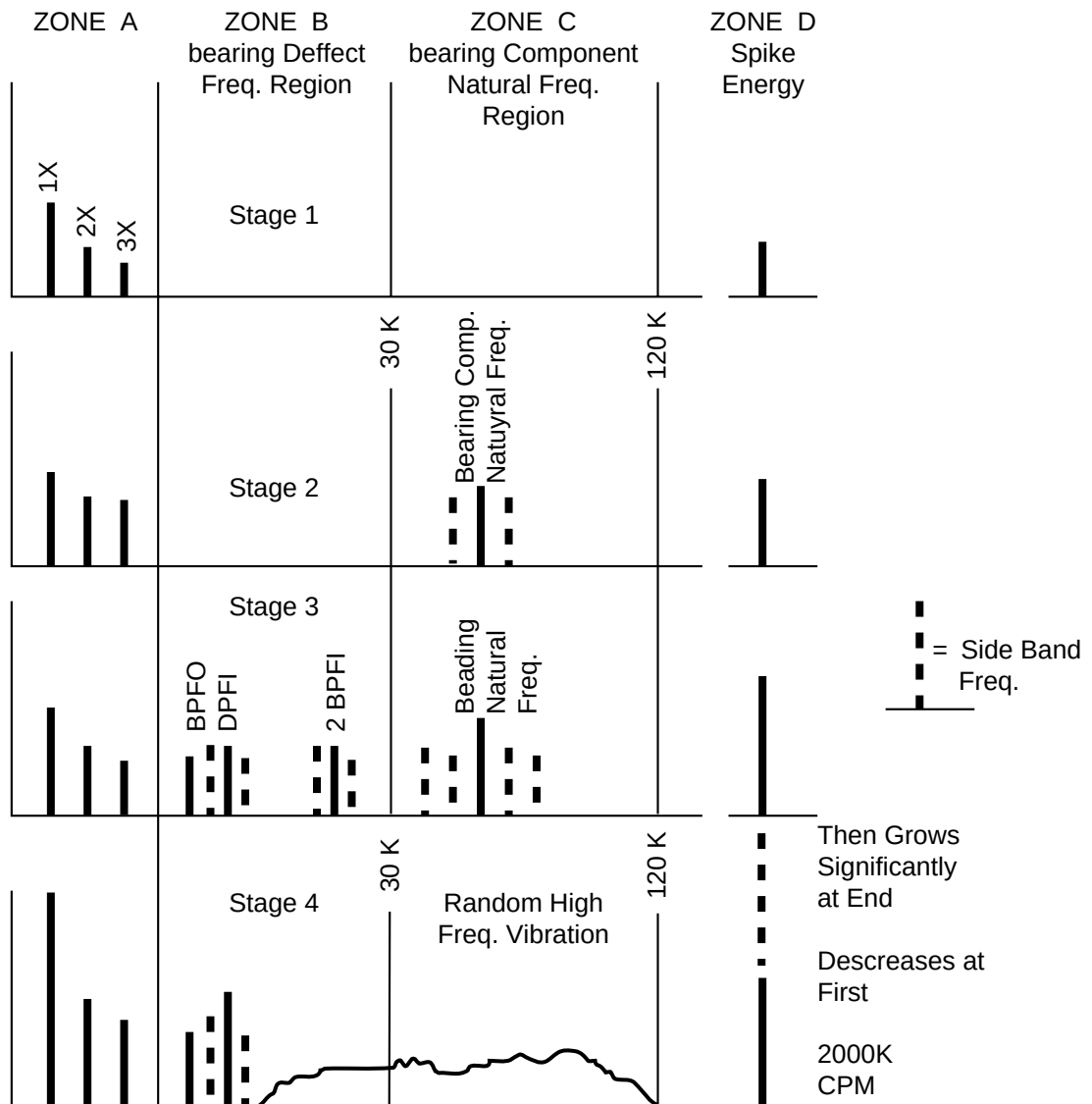
Single row ball bearing SKF 206 Nb = 13, Pd = 1.811 in.,  
Bd = 0.375 in.,  $\theta = 0$  deg., Speed = 1800 RPM.

Therefore:

BFO = 154.62Hz, BSF = 69.318 Hz., BPFI = 235.44 Hz.,  
FTF = 11.89 Hz.

## Failure Modes

There are four stages for rolling element bearing failure (see Figure 14) as follows:



**Figure 14. Anti-Friction Bearing Failure Modes**

1. At an early stage of bearing failure ultrasonic frequencies in the range of 20 KHz to 60 KHz starts to appear. These are frequencies evaluated by spike energy (see Figure 8). Spike energy is defined as “ vibration energy generated by short-duration, metal-to-metal impacts and random vibration that propagate through the structure”.
2. The next stage, slight bearing defects begin to excite bearing component natural frequencies which predominantly occur in 500 Hz to 2000 Hz range. Side-band frequencies appear above and below natural frequency.
3. At the third stage bearing defect frequencies and harmonics will start to appear. When wear progresses, more defect frequency harmonics appear and number of side-bands grow; spike energy continue to increase.
4. At the last stage, even the amplitude of 1X RPM is affected. It grows, and normally causes the growth of many running speeds harmonics. Discrete bearing defect and component natural frequencies actually begin to disappear and are replaced by random, broad band high frequency noise.

## **Bearing Failures and Some of Their Causes**

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Most bearing failures can be attributed to some or more of the following causes.

### **Defective Bearing Seats on Shafts and in Housing**

There are factors that produce shaft seats and housing bores that are oversized or undersized, tapered or oval. The same condition can be produced by seating the bearing in a housing with a correctly made bore but where the housing is distorted when it is secured to machine frame.

When the contact between a bearing and its seat is not intimate, relative movement results. Small movements between the bearing and its seat will produce condition called fretting.

### **Misalignment**

Misalignment occurs when an inner ring is seated against a shaft shoulder that is not square with the journal, or where a housing shoulder is out of square with the housing bore. Misalignment arises when two housings are not on the same centerline.

### **Inadequate Lubrication**

There are three factors comprising the adequacy of lubrication. One must consider first its properties, secondly the quantity applied to the bearing, and thirdly the operating conditions. If any of these factors do not meet the requirements, the bearing is said to have failed from inadequate lubrication.

### **Vibration**

Rolling element bearings exposed to vibration while the shafts are not rotating are subject to a damage called false brinelling. False brinelling occurs most frequently during transportation of assembled machines.

**Electric Current  
Through the  
Bearing**

In certain applications of bearings in electrical machinery there is a possibility that electrical current will pass through a bearing. When the current passes at the contact surfaces between the rolling elements and raceways, arcing results and this produces very localized high temperature and consequent damage called fluting. For this reason bearings are often insulated.

## GEAR DEFECTS

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Gears are usually used in industry to change speed of the driven equipment from that of the driver, to change direction of rotation or the direction of power flow, and to transfer motion between input and output machinery.

Vibration in gears could be caused by the following gear defects

- Unbalance
- loose foundation bolt
- coupling misalignment
- inadequate foundation
- wear in bearing and gears
- lateral and torsional critical speed response
- coupling lockup and wear

Vibration data is necessary for accurate identification and analysis of gear problems. The time domain signal is required. The ability to process and view this signal with various long time periods is also essential. The frequency domain spectra are also required. The ability to view these spectra on various ranges and zoom in on certain frequencies is equally essential.

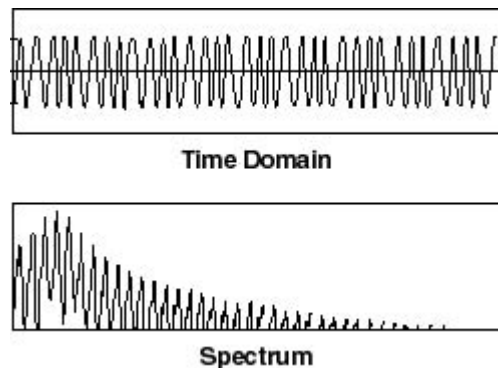
## Gear Mesh Frequency

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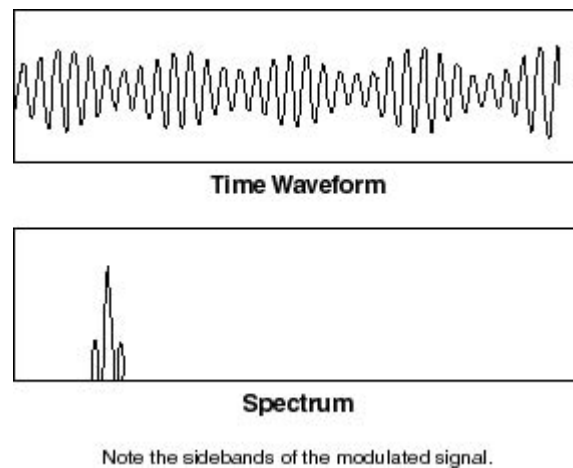
The gear mesh frequency can be computed by multiplying the number of teeth on a gear by the speed of the gear. For example if a pinion gear has 43 teeth and rotates at 1776 RPM (29.6 Hz) the gear mesh would be  $29.6 \times 43 = 1272.8$  Hz.

## Gear Modulation

A set of properly meshing gears may generate a gear mesh frequency of very low amplitude. Increases in the amplitude can be caused by loading, bottoming of the gears, improper backlash, and eccentricity in the shaft, teeth, or gear. A constant load will cause the gear mesh to remain at a fairly constant amplitude. However, a changing load will cause variations in the amplitude of the gear mesh frequency. This phenomenon in effect modulates the amplitude of the gear mesh frequency. Analysis of the modulation frequency in either the time or frequency domain can identify the rate at which the load is changing (see Figure 15 and Figure 16). Note the side bands of the modulated signal.



**Figure 15. Frequency Modulation**



**Figure 16. Amplitude Modulation**

The varying load amplitude modulates the amplitude of the gear mesh frequency and this modulation is presented as side band to the gear mesh frequency. The difference frequency between the spectral lines is the repetition rate of the load change.

When a set of gears are too closely engaged they will bottom out. When this occurs a very high amplitude gear mesh frequency may be generated. A second, third and fourth gear mesh harmonic may also be generated. If one of the shafts or gears is eccentric, the gear may bottom out for only a few teeth each revolution. When this occurs, the gear mesh frequency will vary in amplitude and become modulated with the speed of the gear/shaft that is eccentric.

## The Natural Frequencies of Gears

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If a gear is small relative to the shaft size, the natural frequency may be so high that you need not be concerned with it. If a gear is large relative to the shaft size, it can have several natural frequencies in the audio range. These natural frequencies can be identified by performing an 'impact' test on the gear. Empirical data indicate that after the gear is installed, some of the natural frequencies may no longer be present or they are damped out by the system. When a problem occurs with a gear the natural frequency of the problem gear or its meshing gear may be excited. The excited natural frequency is modulated with the frequency of the event that is exciting the natural frequency.

We often refer to the natural frequency of a gear as though it were a single frequency. The natural frequency is actually the center frequency of a band of frequencies. The bandwidth of these frequencies is often in excess of 50 Hz.

In rare cases, the gear mesh frequency is near the natural frequency of the gear. When this occurs very high amplitudes of the gear mesh frequency may be present and can cause serious analysis errors.

## Irregularities and Physical Defects

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A gear that has small spalls on the face of each tooth may not generate a unique signal. Therefore, these spalls or irregularities are not considered defects.

A large spall on one tooth, a cracked tooth, a chipped tooth, or a tooth completely broken-off will generate a unique signal in the form of a pulse. These occurrences are considered defects. The pulse generated by a defective gear has four measurable characteristics: they are pulse frequency, pulse width, repetition rate, and amplitude. On some slow speed machinery the defect can be heard as a dull knock or clank and may occur once each revolution of the defective gear.

If the configuration of the gears is such that natural frequencies are not excited the frequency of the pulse may be the gear mesh frequency or any available frequency. In most cases, however, a natural frequency is excited and the frequency of the pulse generated is the same as the natural frequency.

The pulse width is very short in duration and often lasts only a few cycles. Empirical data indicate the signal is damped very quickly because of the gear mesh or rotation of the gear or a combination of the two.

In rotating machines, vibration frequencies generated by the rotation are equal to the number of events times the speed. This is true of unbalance, vane pass frequency, blade pass frequency, and ball pass frequency. This is also true of gears. For example, if a gear has a defective tooth, the pulse generated will occur once each revolution, or the repetition rate is the speed of the gear. If two adjacent teeth are broken the repetition rate may be twice the speed of the gear. The limited data available indicate that if some distance separates the two broken teeth, the pulses will appear as two separate events. The data indicate that two separate pulses or events are occurring each revolution.

These data can be observed in the time domain and in the frequency domain if you can recognize a Fourier analysis of a pulse. Such analysis presents a series of spectral lines whose difference frequency is equal to the repetition rate of the pulse. Observing a once per revolution marker and the pulse generated by a defective tooth on a dual channel oscilloscope can reveal the location of the defective tooth with respect to the marker.

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## Identification of Defective Gears

Identification of a defective gear in a single reduction gear box is a relatively simple matter. The repetition rate of the pulses equals the speed of the defective gear. This identification process is not nearly so simple for multiple reduction units, and multiple defects.

When a tooth breaks, the impact created each time the broken tooth hits the other gear can excite the natural frequencies of the gear involved. If the pinion gear is small relative to the shaft size, its response at the natural frequency, if excited, will be so high a velocity-transducer may not pick up the signal. However, the broken tooth on the pinion can excite the natural frequency of the driven gear. If the driven gear has a broken tooth and the gear is relatively large compared to shaft, the natural frequency of the driven gear can be excited once each revolution when the broken tooth hits. To identify which gear is defective on a shaft that has more than one gear, the natural frequency of each gear should be excited, measured, and recorded.

## ELECTRICAL FAULT

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Electromagnetic problems are those that make the vibration level disappear instantly as soon as power is cut-off. These problems are normally associated with the magnetic field within the motor. Actually, they are mechanically based but the effect of the magnetic field affects the rotor. Such problems are broken rotor bar(s), shorted endrings, exciter or pole winding damage, open or short circuited windings, eccentricity, etc.

### Broken Rotor Bars and Shorted Rings

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The symptoms of the above are as follows:

- Large running speed component that “beats” with slip frequency
- Large running speed with 2X slip frequency sidebands
- Large portion of the running speed will disappear on shutdown

Since the running speed component is beating with the slip frequency sidebands, it is essential to zoom in at the running speed to view the individual components. If the component was found to be at rotor speed, then the problem is in the rotor itself.

If the zoomed component was found to be at the AC (Alternating Current) line frequency, then the problem is definitely in the stator. It could be windings, short circuit, etc.

## Eccentric Rotor

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When the rotor is not round or concentric with its center line, it cannot be made concentric with the stator and hence produces uneven air gap between the rotor and the stator.

### Causes of Rotor Eccentricity

Rotor eccentricity to stator can be caused by:

- Rotor not centralized in a stator
- A bowed rotor
- A rotor bow increase due to local heating (hot-spot) in the rotor caused by shorted rotor lamination or broken rotor bars
- A rotor with high mechanical runout at its journals causing the rotor to run eccentric to the stator producing a varying air gap
- In rare cases, the rotor cage not concentric with rotor shaft

Eccentric rotors normally give a 2X line frequency component, and a 2X slip frequency sidebands of running speed and twice running speed.

**Problem**  
**Correction of**  
**Rotor Eccentricity**

To correct such a problem, perform the following:

1. Centralize the rotor by moving bearing housings to give correct air gap between rotor and stator.
2. Machine the rotor journals round.
3. Machine the rotor with respect to the journals
4. Straighten the rotor shaft where possible. Small bows can be machined. If the bow is excessive, the rotor should be replaced.
5. Locate possible hot spots by using infrared techniques. For hot spots on the surface, the problem area can be machined out by removing material 360 degrees in the radial direction from the rotor so as not to cause an unbalance. Internal hot spot can not be repaired and total rotor replacement is recommended.

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## Eccentric Stator

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An eccentric stator is one that is not perfectly round or whose centerline does not coincide with the rotor axis of rotation which produces a stationary differential air gap between the rotor and the stator. This will cause a 2X line frequency vibration component, which will be directional depending on the largest air gap differential. The differential in air gap between the rotor and the stator should not exceed 5 % for induction motors and 10 % for synchronous motors. Manufacturers, however, normally allow slightly higher percentage of differential air gaps.

### Causes of Stator Eccentricity

Stator eccentricity can be caused by:

- Poor build of stator lamination stampings.
- Loose laminations.
- Soft mounting feet causing distortion in the stator.
- Warped base, again causing distortion in the stator.
- A hot spot in the stator which may be caused by:
  - Shorted lamination
  - Uneven heat distribution caused by bad stacking
  - Dirty stator

For poor build of stator lamination stamping, it may be necessary to line bore the stator to ensure the unit is concentric. Line boring may also be necessary to relieve shorted laminations causing uneven heating in the stator core and ultimately distortion.

**Problem**  
**Correction of**  
**Stator Eccentricity**

Loose lamination in the stator can be overcome by:

- Dipping in varnish and baking
- Re-stacking lamination
- Replacing the stator

Soft mounting feet and warped bases should be machined and shimmed so as not to cause distortion in the base and thus the stator. It can be detected by taking vibration measurements on the feet and by soft foot testing with dial indicator.

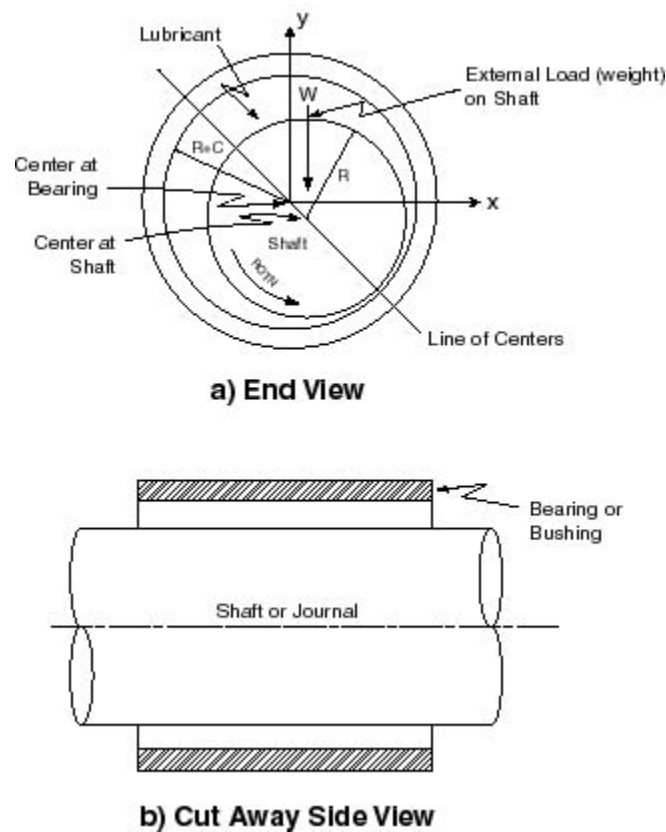
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## JOURNAL BEARING (FLUID FILM BEARING)

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Many machines require two pieces of metal to move relative to one another at fairly high speeds. A fluid lubricant is often used between the two pieces to prevent wear and failure of the metal surfaces. The resulting situation is called a fluid film bearing, or journal bearing. The primary function of a journal bearing is to support the rotor on a film of oil as it rotates. The primary function of the oil is to separate the rotating shaft from the stationary bearing, and the secondary function of the oil is to remove heat generated by bearing friction or elsewhere in the rotor.

The most widely used of bearing types is the plain journal bearing, shown in Figure 17. It has a circular shaft rotating in a slightly larger non-rotating bushing while the space between them is filled with the lubricating fluid. Fluid is pulled into the region under the shaft due to the shear forces generated by shaft rotation. Forcing the fluid into the converging film thickness at the bottom of the shaft produces high pressure. This high pressure supports the weight of the rotating machine component and prevents the shaft from touching the bushing surface. Normally each of the two journal bearings on the end of a shaft supports half of the rotor weight.

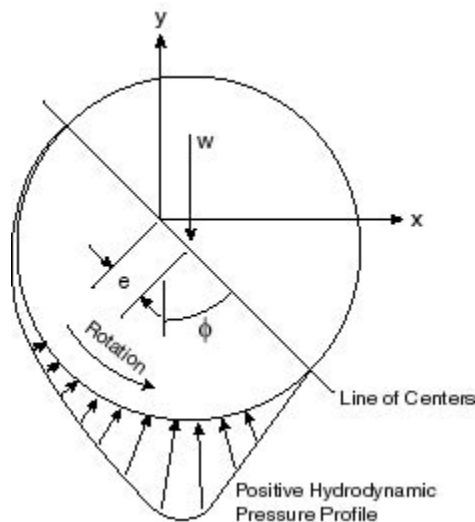


**Figure 17. Journal Bearing**

Journal bearings can be incredibly long lived. If sufficient lubricant for cooling the bearing is supplied through a pump and contaminants kept out of the oil, there is not really any reason why they should wear out. Many industrial bearings have operated for 20 or 30 years with only minor periodic maintenance every several years. In order for this system to operate properly there has to be a suitable clearance between the shaft and the bearing. This clearance is typically in the order of 1.5 thousandth, plus 1 thousandth for each inch of diameter. It also depends on factors such as load, RPM, and bearing type.

## Pressure In Bearing

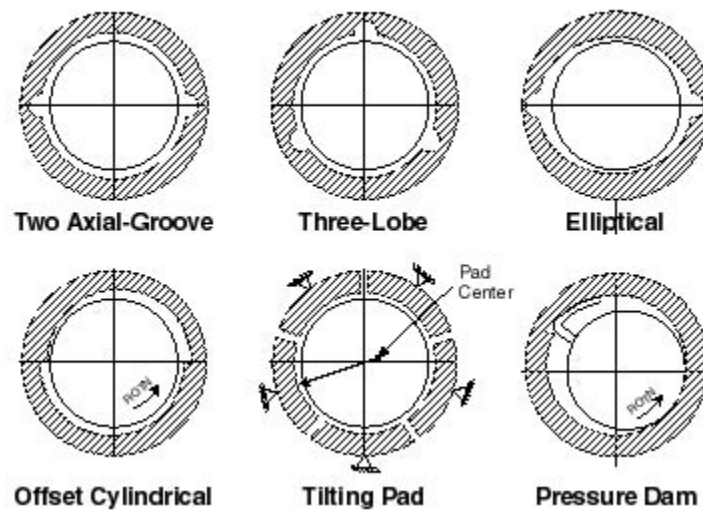
The rotating shaft does not touch the non-rotating bearing surface in spite of the applied downward load. The only way that shaft is supported is by a pressure generated in the fluid film. As the force is applied, the shaft moves downward in the bearing as shown in Figure 17(a). This creates a thinner fluid film on the bottom of the bearing than on top. As the shaft rotates, it drags the lubricant into this region. A high pressure is generated in the fluid below the shaft as the incompressible oil is forced through the minimum clearance area. Figure 18 illustrates the pressure profile.



**Figure 18. Hydrodynamic Pressure Profile Acting On Shaft**

## Bearing Types

Various bearing types are shown in Figure 19 and their characteristics are shown in Table 1.



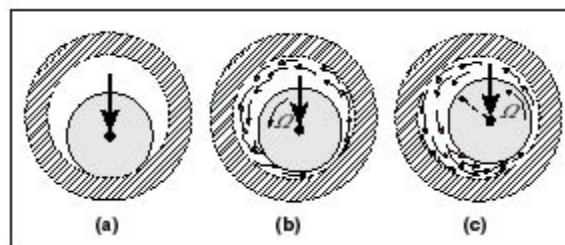
**Figure 19. Various Bearing Types**

**Table 1. Bearing Characteristics**

| <b>Bearing Type</b>                 | <b>Advantages</b>                                                                            | <b>Disadvantages</b>                                                                                                                         | <b>Comments</b>                                                                               |
|-------------------------------------|----------------------------------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------|-----------------------------------------------------------------------------------------------|
| Plain Journal                       | Easy to make.<br>Low cost.                                                                   | Very subject to oil-whirl.                                                                                                                   | Round bearings are nearly always “crushed” to make elliptical bearings.                       |
| Axial Groove                        | Easy to make.<br>Low cost.                                                                   | Subject to oil whirl.                                                                                                                        | Round bearings are nearly always “crushed” to make elliptical or multi-lobe.                  |
| Elliptical                          | Easy to make.<br>Low cost.<br>Good damping at critical speeds.                               | Subject to oil whirl at high speeds.<br>Load direction must be known.                                                                        | Probably most widely used bearing at low or moderate speeds.                                  |
| Three & Four Lobe                   | Good suppression of whirl.<br>Overall good performance<br>Moderate cost.                     | Some types are expensive to make properly.<br>Subject to whirl at high speeds.                                                               | Currently used by some manufacturers as standard bearing design.                              |
| Pressure Dam (Single Dam)           | Good suppression of whirl.<br>Low cost.<br>Good damping at critical speeds.<br>Easy to make. | Goes unstable with little warning.<br>Dam may be subject to wear or build up over time.<br>Load direction must be known.                     | Very popular with Petrochemical industry.<br>Easy to convert elliptical over to pressure dam. |
| Offset Half (With Horizontal Split) | Excellent suppression of whirl at high speeds.<br>Low cost.<br>Easy to make.                 | Fair suppression of whirl at moderate speeds.<br>Load direction must be known.                                                               | Has high horizontal stiffness and low vertical stiffness may become popular, used outside US. |
| Hydrostatic                         | Good suppression of oil whirl.<br>Wide range of design parameters.<br>Moderate cost.         | Poor damping at critical speeds.<br>Requires careful design.<br>Requires high pressure lubricant supply.                                     | Generally high stiffness properties used for high precision rotors.                           |
| Tilting Pad                         | Will not cause whirl (no cross coupling)<br>Wide range of design parameters.                 | High cost.<br>Requires careful design.<br>Poor damping at critical speeds.<br>Hard to determine actual clearances.<br>High horse power loss. | Widely used bearing to stabilize machines with subsynchronous non-bearing excitation.         |

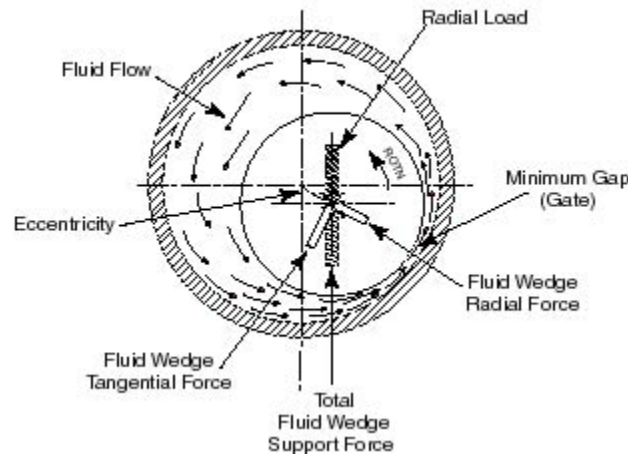
## Shaft Position

Figure 20 shows a cross sectional view of the shaft position within a journal bearing. At rest (Figure 20a), the shaft is at rest in the bottom of the bearing. The only force acting on the shaft center is the gravitational force. For a rotor in between bearing, the bearing supports half the rotor weight (assuming the center of gravity coincides with geometrical center). For counter clockwise shaft rotation, and as the fluid accumulates at the left side, pressure unbalance at left and at right result in rotor motion to the right (Figure 20b). When the rotor starts rotating, the fluid tends to pass and the rotor is moved up (Figure 20c).



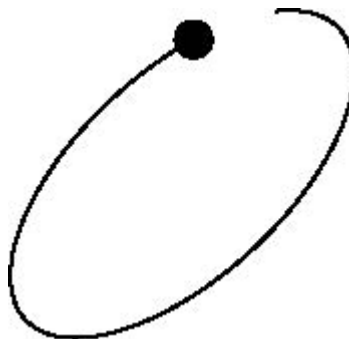
**Figure 20. Typical Shaft Position Within Journal Bearing**

The forces acting on the shaft are shown in Figure 21 below. The radial load is acting downward due to gravitational direction and the fluid wedge force which has radial and tangential components. The total fluid wedge should counteract the shaft radial load and this determines the shaft center position within the journal bearing.



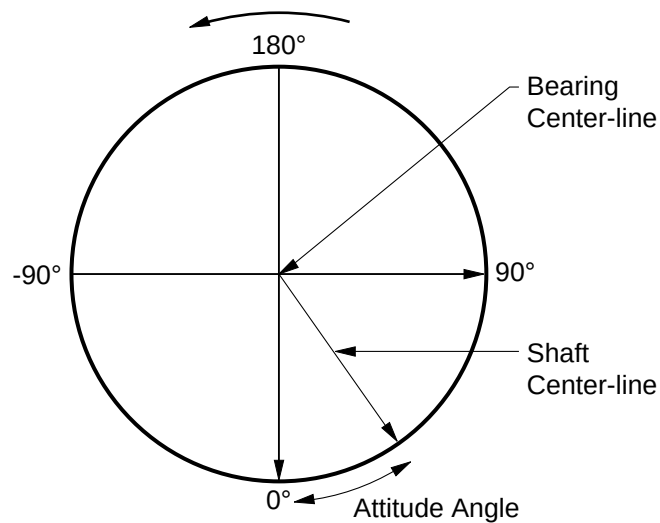
**Figure 21. Shaft/Journal Bearing Dynamic**

The shaft position (moved right and upward for counter-clock rotation) is considered the normal shaft centerline *normal* position. The displacement orbit associated with this position is shown in Figure 22.



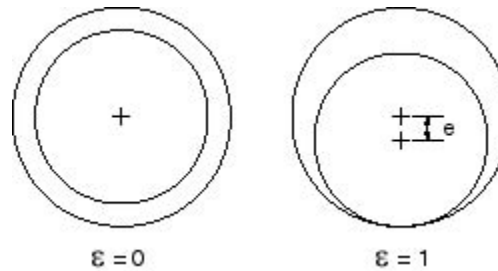
**Figure 22. Shaft Orbit**

This position is often characterized by the term “Attitude Angle” (refer to Figure 23). For normal bearing loading the attitude angle is usually in the region of  $20^{\circ}$ - $60^{\circ}$  degrees when measured from the vertical.



**Figure 23. Attitude Angle Shown Opposite**

Another useful characterization of shaft position is the eccentricity ratio  $\epsilon = e/c$ . This is calculated by dividing the eccentricity  $e$  (the distance between the shaft centerline and the bearing centerline) by the radial clearance  $c$ . This results in a dimensionless number from zero to one. Figure 24 shows the two extreme values of  $\epsilon$ .



**Figure 24. Eccentricity Ratio  $0 \leq e/c \leq 1$**

## Wear Pattern

In the normal shaft position the bottom half of the bearing will have a wear pattern. For the horizontally cocked bearing the wear pattern will run diagonally across the bearing, and for the vertically cocked case the wear pattern will be localized to one end on the bottom and possibly the opposite end on the top. When investigating these problems it is necessary to personally inspect the bearings for abnormalities.

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## WHIRL/WHIP VIBRATIONS

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Rotor lateral vibrations caused by fluid interaction are known as whirl and whip. They occur as self-excited, limit cycle vibrations after the threshold of stability is exceeded. Whirl/whip vibrations occur when the system mechanical resonance coincides with the fluid-induced resonance.

Whirl frequency is proportional to the rotative speed with the proportionality coefficient close to the fluid circumferential average velocity ratio. Whip frequency asymptotically approaches the high eccentricity natural frequency of the rotor lateral mode, modified by non-linearity of the fluid film.

Self-excited vibrations are induced by a constant force, and sustained by a constant energy supply. The system has an internal energy transfer mechanism which delivers the energy in a periodic manner. The frequency at which the energy is provided usually corresponds to a natural frequency of the system.

The threshold of stability depends not only on fluid characteristics, but also on the rotor dynamic characteristics; thus it is a rotor/bearing/seal system property. In some machines the threshold of stability may be high enough that only whip vibrations are observed. In other machines, it may be low enough to observe whirl, while the operating speed is not high enough to allow whip.

It is well known in vibration theory that self-excited vibrations occur at frequencies that are close to the natural frequencies of the system. The whirl and whip frequencies are close to the mechanical and fluid-induced natural frequencies of the system. The whip frequency approaches the high eccentricity, natural frequency of the first lateral mode of the rotor (first balance resonant frequency, the "mechanical" or "direct" type of the system natural frequencies).

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## Characteristics of Fluid Induced Instabilities (Whirl and Whip Condition)

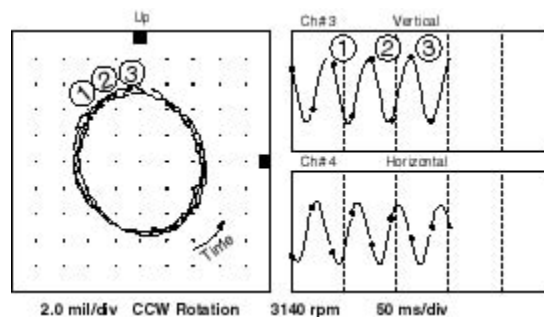
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The general signal characteristics of fluid induced instabilities include the following:

- The precession of the whirl/whip self-excited vibration is always forward. This means the vibration rotation is in the same direction as the shaft rotation.
- The orbit shape is circular, or nearly circular. When observed on an oscilloscope, the Key-phasor dots appear to revolve slowly in the direction opposite to rotation for whirl vibration frequency slightly less than  $\frac{1}{2} X$ .
- The self-excited vibration frequency of a whirl instability is approximately the fluid circumferential average velocity ratio as a percentage of the shaft rotative speed. The whirl frequency is usually slightly less than half of rotating speed and is normally in the range of 38 to 49 % of rotative speed.
- The self-excited vibration frequency of a whip instability is asymptotic to the high eccentricity (e/c ratio), lateral natural frequency of the rotor. In other words, the vibration frequency is almost the same as the “critical” frequency, and changes little or not at all with increasing rotative speed (a condition referred to as locked critical).

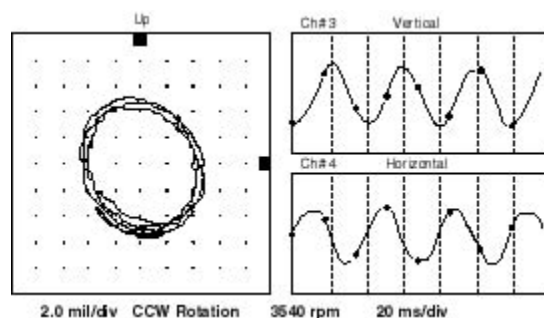
### Fluid Induced Stability Orbit/Time Waveform Plots:

The whirl orbit shown in Figure 25 is nearly circular with forward precession. The period of the Whirl vibration cycle is slightly more than a half of the period of the shaft rotation.



**Figure 25. Whirl Orbit and Time Waveform**

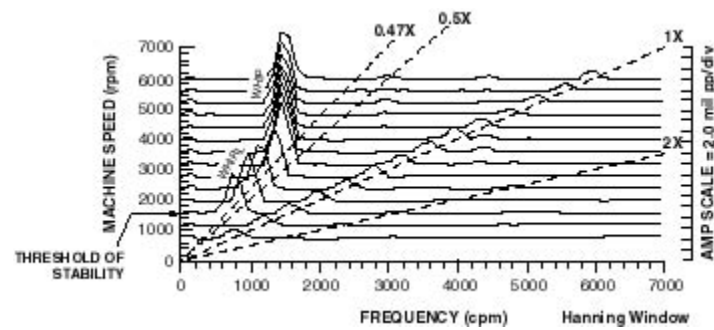
The whip orbit shown in Figure 26 is also nearly circular with forward precession. Depending on the system damping, the vibration amplitude could reach the bearing clearance value at which rubbing may occur.



**Figure 26. Whip Orbit and Time Waveform**

## Fluid Induced Cascade Plots

The cascade plot in Figure 27 shows the frequency relationship between the rotative speed and the frequencies of the self-excited whirl/whip vibration.



**Figure 27. Fluid Induced Instability (Whirl/Whip) Cascade Plot**