Machinery Fault Diagnosis

A basic guide to understanding vibration analysis for machinery diagnosis.
This is a basic guide to understand vibration analysis for machinery diagnosis. In practice, many variables must be taken into account. PRÜFTECHNIK Condition Monitoring and/or LUDECA are not responsible for any incorrect assumptions based on this information.
Unbalance is the condition when the geometric centerline of a rotation axis doesn’t coincide with the mass centerline.

\[ F_{\text{unbalance}} = m \cdot d \cdot \omega^2 \]

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

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

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

The static unbalance produces a vibration signal at 1X, radial predominant, and in phase signals at both ends of the rotor.
Pure couple unbalance is caused by two identical unbalance masses located at 180° in the transverse area of the shaft.

Pure couple unbalance may be statically balanced.

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

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

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

Frequency spectra before/after balancing and balancing diagram.

Before balancing

After balancing

Balancing diagram
Overhung Rotors

A special case of dynamic unbalance can be found in overhung rotors. The unbalance creates a bending moment on the shaft. Dynamic unbalance in overhung rotors causes high 1X levels in radial and axial direction due to bending of the shaft. The axial bearing signals in phase may confirm this unbalance.
Unbalance location

The relative levels of 1X vibration are dependant upon the location of the unbalance mass.
Misalignment is the condition when the geometric centerline of two coupled shafts are not co-linear along the rotation axis of both shafts at operating condition.

A 1X and 2X vibration signal predominant in the axial direction is generally the indicator of a misalignment between two coupled shafts.
Angular misalignment is seen when the shaft centerlines coincide at one point along the projected axis of both shafts.

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

These signals may be also visible in the radial direction at a lower amplitude and in phase.
Parallel misalignment is produced when the centerlines are parallel but offset.

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

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

Parallel Misalignment
Misalignment Diagnosis Tips

In practice, alignment measurements will show a combination of parallel and angular misalignment.

Diagnosis may show both a 2X and an increased 1X signal in the axial and radial readings.

The misalignment symptoms vary depending on the machine and the misalignment conditions.

The misalignment assumptions can be often distinguished from unbalance by:
- Different speeds testing
- Uncoupled motor testing

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

<table>
<thead>
<tr>
<th>Short &quot;flexible&quot; couplings</th>
<th>RPM</th>
<th>Alignment Tolerance in [mils]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>acceptable</td>
</tr>
<tr>
<td><strong>Offset</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>900</td>
<td>6.0</td>
<td>3.0</td>
</tr>
<tr>
<td>1200</td>
<td>4.0</td>
<td>2.5</td>
</tr>
<tr>
<td>1800</td>
<td>3.0</td>
<td>2.0</td>
</tr>
<tr>
<td>3600</td>
<td>1.5</td>
<td>1.0</td>
</tr>
<tr>
<td><strong>Angularity</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(gap difference at coupling edge per 10&quot; diameter)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>900</td>
<td>10.0</td>
<td>7.0</td>
</tr>
<tr>
<td>1200</td>
<td>8.0</td>
<td>5.0</td>
</tr>
<tr>
<td>1800</td>
<td>5.0</td>
<td>3.0</td>
</tr>
<tr>
<td>3600</td>
<td>3.0</td>
<td>2.0</td>
</tr>
<tr>
<td><strong>Spacer shafts and membrane (disk) couplings</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Offset</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>(per inch spacer length)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>900</td>
<td>2.0</td>
<td>1.2</td>
</tr>
<tr>
<td>1200</td>
<td>1.5</td>
<td>0.9</td>
</tr>
<tr>
<td>1800</td>
<td>1.0</td>
<td>0.6</td>
</tr>
<tr>
<td>3600</td>
<td>0.5</td>
<td>0.3</td>
</tr>
<tr>
<td><strong>Angularity</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[mrad]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>900</td>
<td>2.0</td>
<td>1.2</td>
</tr>
<tr>
<td>1200</td>
<td>1.5</td>
<td>0.9</td>
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<tr>
<td>1800</td>
<td>1.0</td>
<td>0.6</td>
</tr>
<tr>
<td>3600</td>
<td>0.5</td>
<td>0.3</td>
</tr>
<tr>
<td><strong>Soft foot</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>any</td>
<td></td>
<td>2.0 mils</td>
</tr>
</tbody>
</table>

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

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

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

2X and radial readings can also be visible.
Rotating looseness is caused by an excessive clearance between the rotor and the bearing.

**Rolling element bearing:**

![Diagram of a rolling element bearing]

**Journal bearing:**

![Diagram of a journal bearing]
Structural looseness occurs when the machine is not correctly supported by, or well fastened to its base.

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

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

Measurements should be made on the bolts, feet and bases in order to see a change in the amplitude and phase. A change in amplitude and 180° phase difference will confirm this problem.
Resonance is a condition caused when a forcing frequency coincides (or is close) to the natural frequency of the machine’s structure. The result will be a high vibration.

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

⇒ no harmonic relationship
Resonance

- Resonance can be confused with other common problems in machinery.
- Resonance requires some additional testing to be diagnosed.

Amplitude at rotation frequency $f_n$ by residual unbalance on rigid rotor.

Strong increase in amplitude of the rotation frequency $f_n$ at the point of resonance, step-up dependent on the excitation (unbalanced condition) and damping.

\[ \varphi_1 = 240^\circ \]

\[ \varphi_2 = 60...80^\circ \]
Resonance Diagnosing Tests

Run Up or Coast Down Test:

- Performed when the machine is turned on or turned off.
- Series of spectra at different RPM.
- Vibration signals tracking may reveal a resonance.

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

Bump Test:

**Excitation – force pulse**

- F/a
- Double beat 5 ms

**Response – component vibration**

- s
- Decaying function

**Shock component, natural vibration, vertical**

- Frequency response, vertical
- Natural frequency, vertical

- Frequency response, horizontal
- 1st mod. 2nd mod.
Journal bearings provides a very low friction surface to support and guide a rotor through a cylinder that surrounds the shaft and is filled with a lubricant preventing metal to metal contact.

High vibration damping due to the oil film:
- High frequencies signals may not be transmitted.
- Displacement sensor and continuous monitoring recommended

Clearance problems (rotating mechanical looseness).

Oil whirl
- Oil-film stability problems.
- May cause 0.3-0.5X component in the spectrum.
1. Wear:

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

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

Increased level of shock pulses.
Rolling Element Bearings

2. Race Damage:

Roller bearing geometry and damage frequencies:

- \( \alpha \) Angle of contact
- \( D \) Arc diameter
- \( d \) Rolling element diameter
- \( Z \) Number of rolling elements
- \( n \) Shaft RPM

Ball pass frequency, outer race:

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

Ball pass frequency, inner race:

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

Ball spin frequency:

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

Fundamental train frequency:

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

Example of rollover frequencies:

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

<table>
<thead>
<tr>
<th>Dimensions</th>
<th>Rollover frequencies</th>
</tr>
</thead>
<tbody>
<tr>
<td>( d = 77.50 ) mm</td>
<td>( BPFO = \frac{n}{60} \cdot 4.0781 = 203.77 ) Hz</td>
</tr>
<tr>
<td>( D = 14.29 ) mm</td>
<td>( BPFI = \frac{n}{60} \cdot 5.9220 = 295.90 ) Hz</td>
</tr>
<tr>
<td>( Z = 10 )</td>
<td>( 2f_w = \frac{n}{60} \cdot 5.2390 = 261.77 ) Hz</td>
</tr>
<tr>
<td>( \alpha = 0 )</td>
<td>( f_K = \frac{n}{60} \cdot 0.4079 = 20.38 ) Hz</td>
</tr>
</tbody>
</table>

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**Rolling Element Bearings**

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

Outer race damage frequency BPFO as well as harmonics clearly visible

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

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

Rolling element damage:
(Ball spin frequency BSF)

Rolling elements rollover frequency BSF with harmonics as well as sidebands in intervals of FTF

Cage damage:
(Fundamental train frequency FTF)

Cage rotation frequency FTF and harmonics visible

Sidebands in intervals of FTF
↓↓↓↓ 2.BSF  ↓↓↓↓ 4.BSF  ↓↓↓↓ 6.BSF  ↓↓↓↓ 8.BSF

FTF and 2<sup>nd</sup>, 3<sup>rd</sup>, 4<sup>th</sup> harmonics
Rolling Element Bearings

Lubrication Problems:

Lubricant contamination
- Race damage
- Defective sealing
- Contaminated lubricant used

Insufficient lubrication
- Insufficient lubricant
- Underrating

Over-greasing
- Maintenance error
- Defective grease regulator
- Grease nipple blocked

Major fluctuation in level of shock pulses and damage frequencies

→ Subsequent small temperature increase

Large temperature increase after lubrication
Rolling Element Bearings

Incorrect mounting.

Bearing rings out of round, deformed.
- Incorrect installation
- Wrong bearing storage
- Shaft manufacturing error
- Bearing housing overtorqued.

Bearing forces on floating bearing.
- Incorrect installation
- Wrong housing calculation
- Manufacturing error in bearing housing

Cocked bearing.
- Incorrect installation

Shock pulse ↑

Damage frequencies envelope

Air gap

Dirt

Severe vibration
Bearing temperature increases

Fixed bearing

Floating bearing

Axial 1X, 2X and 3X.
Blade and Vanes

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

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

$B_n$ = # of blades or vanes

$N$ = rotor speed in rpm

Identify and trend $f_{BP}$.

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

Example characteristic frequency:

3 struts in the intake; $x=3$.

9 blades; $B_n=9$.

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

Characteristic frequency = $N \cdot 27$
There are two basic moving fluid problems diagnosed with vibration analysis:

- Turbulence
- Cavitation
Belt Drive Faults

Belt transmission a common drive system in industry consisting of:

- Driver Pulley
- Driven Pulley
- Belt

The dynamic relation is: \( \dot{\varnothing}_1 \omega_1 = \dot{\varnothing}_2 \omega_2 \)

Belt frequency:

\[
 f_B = \frac{3,1416 \dot{\varnothing} A}{l}
\]

\( l \): belt length
Belt Drive Faults

Belt Worn:

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

2 $f_B$ generally dominates the spectrum

Pulley Misalignment:

1X of diver or driven pulley visible and predominant in the axial reading.

Offset

Angular

Twisted
Belt Drive Faults

**Eccentric Pulleys:**

- The geometric center doesn’t coincide with the rotating center of the pulley.
- High 1X of the eccentric pulley visible in the spectrum, predominant in the radial direction.
- Easy to confuse with unbalance, but:
  - Measurement phase in vertical and horizontal directions may be 0° or 180°.
  - The vibration may be higher in the direction of the belts.

**Belt Resonance:**

- If the belt natural frequency coincides with either the driver or driven 1X, this frequency may be visible in the spectrum.
Gear Faults

**Spur Gear:**
- Driving gear
- Driven gear
- Gear (wheel)
- Pinion

**Worm Gear:**
- Gear
- Worm gear

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

**Bevel Gear:**
- Bevel gear
Gear mesh frequency $f_z$ can be calculated:

$$F_z = z f_n$$

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

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

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

Incorrect tooth meshing

Wear

Detail of X:

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Gear Faults

Incorrect tooth shape

Tooth break-out
Gear Faults

Eccentricity, bent shafts

Detail of X:

“Ghost frequencies” or machine frequencies

Gearwheel being manufactured
Cutting tool
Worm drive part of the gear cutting machine

f\(_{z}\) and harmonic sidebands

f\(_{z}\)  
f\(_{M}\) “Ghost frequency”
Electromagnetic forces vibrations:

Twice line frequency vibration: \(2 \cdot f_L\)

Bar meshing frequency: \(f_{\text{bar}} = f_n \cdot n_{\text{bar}}\)

Synchronous frequency: \(f_{\text{syn}} = 2 \cdot \frac{f_L}{p}\)

Slip Frequency: \(f_{\text{slip}} = f_{\text{syn}} - f_n\)

Pole pass frequency: \(f_p = p \cdot f_{\text{slip}}\)

\(f_L:\) line frequency
\(n_{\text{bar}}:\) number of rotor bars
\(p:\) number of poles

- Stator eccentricity
- Eccentric rotor
- Rotor problems
- Loose connections
Stator Eccentricity:

- Loose iron
- Shorted stator laminations
- Soft foot

1X and 2X signals
f_L without sidebands
Radial predominant
High resolution should be used when analyzing two poles machines.
Electrical Motors

Eccentric Rotor:

- Rotor offset
- Misalignment
- Poor base

$f_p$, 1X, 2X and 2$f_L$ signals.
1X and 2$f_L$ with sidebands at $f_p$.
Radial predominant.
High resolution needed.

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

1. Rotor thermal bow:
   - 1X Radial
   - Unbalanced rotor bar current
   - Unbalance rotor conditions
   - Observable after some operation time

2. Broken or cracked rotor bars:
   - 1X, 2X, 3X, 4X Radial
   - 1X and harmonics with sidebands at $f_p$
   - High resolution spectrum needed
   - Possible beating signal
3. Loose rotor bar:

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

Loose connections:

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