Application Note

Journal bearing vibration

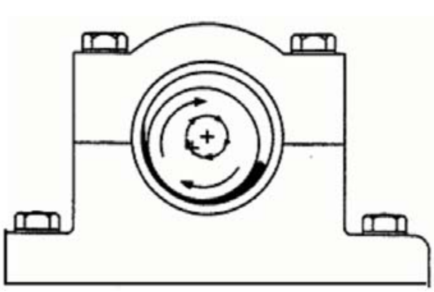
By Stuart Courtney and Allan Todd • SKF

Introduction to journal bearing monitoring

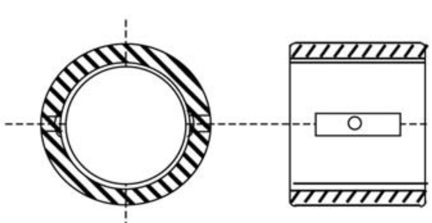
This section deals with plain bearings in high-speed rotating plants such as turbines and rotary compressors. It does not apply to reciprocating engines. The bearings of rotating plants include both journal and thrust bearings. In practice, only the journal bearings play a significant part in rotor vibration, and thus, they are the only bearings discussed here.

There are several types of journal bearings including:

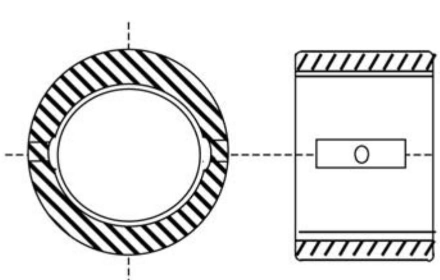
Simple hydrodynamic bearings



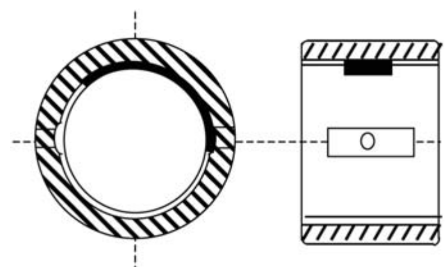
Pressure-fed sleeve bearings



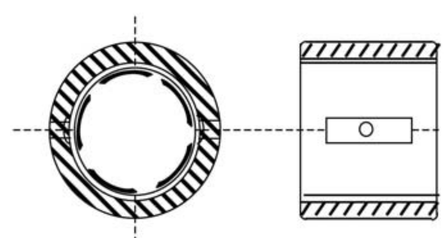
Lemon or elliptical bearings



Dam wall bearings



Tilting pad bearings



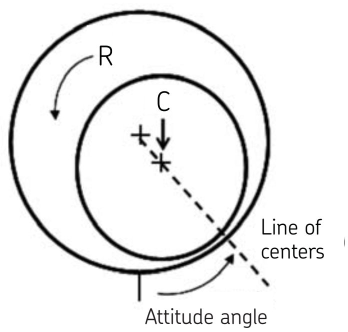
Each of these can show typical modes of vibration, which can in turn be diagnosed by the use of an orbit measurement using a pair of proximity probes.

Eccentricity and attitude angle

The operating condition of a journal bearing is described by its eccentricity and attitude angle.

The centre line of the shaft and the centre line of the bearing are generally different. The ratio between the lines is referred to as the eccentricity: a higher eccentricity means that the centre of the shaft is further from the centre of the bearing. Eccentricity will decrease as load decreases and/or as speed increases.

Attitude angle is a measure of where the shaft sits within the bearing while operating: it is calculated using the deviation from vertical of the line which connects the centre of the bearing from the centre of the shaft. This happens because as the shaft rotates, it climbs up the wall of the bearing on an oil-wedge. As this attitude angle increases, the bearing stability will decrease. Rotor instability will occur when bearing pre-load is not sufficient to keep the rotating shaft in a stable position.



Shaft instability (Oil whirl)

Oil whirl is probably the most common cause of sub-synchronous instability in hydrodynamic journal bearings. Normally, the oil film itself will flow around the journal to lubricate and cool the bearing. The shaft rides on the wedge of oil, rising slightly up the side of the bearing at a stable attitude angle and eccentricity. As the shaft rotates eccentrically relative to the bearing center, the oil wedge produces a pressurized load- carrying film.

If the shaft receives a disturbing force such as a sudden surge or external shock, it can momentarily increase the eccentricity from its equilibrium position. When this happens, additional oil is immediately pumped into the space vacated by the shaft. This results in an increased pressure of the load-carrying film, which creates additional force between the film and shaft. In this case, the oil film can actually drive the shaft ahead of it in a forward circular motion and into a whirling path around the bearing within the bearing clearance. If there is sufficient damping within the system, the shaft can be returned to its normal position and stability. Otherwise, the shaft will continue its whirling motion, and the amplitude of movement will progress to a point where the bearing clearances are exceeded, which in turn can lead to a catastrophic failure of the bearing.

The oil whirl condition can be induced by several conditions including:

- light dynamic and pre-load forces
- excessive bearing wear or clearance
- changes in oil properties (such as temperature and viscosity)
- changes in oil pressure
- improper bearing design (the use of theoretical shaft loading instead of actual shaft loading)

Sometimes machines can exhibit oil whirl intermittently that have nothing to do with the condition of the sleeve bearing, but rather from external vibratory forces transmitting into the unit or from sources within the machinery itself. In these cases, these vibratory forces have the same frequency as the oil whirl. This vibration can transmit from other machinery through attached structures such as piping and braces, or even through the floor and foundation. If this does occur, it may be necessary to isolate this machine from surrounding machinery.

Oil whirl can be easily recognized by its unusual vibration frequency which is generally on the order of 0.40X to 0.48X shaft RPM. Oil whirl occurring at 0.43X shaft speed, instability may occur at the first critical speed.

Oil whirl is considered severe when vibration amplitudes reach 50% of the normal bearing clearance. At this point, corrective action must be taken.

Temporary corrective measures include:

- changing the temperature of the oil (and therefore, the oil viscosity)
- purposely introducing a slight unbalance or misalignment to temporarily increase the loading
- shifting the alignment by heating or cooling support legs
- scraping the sides or grooving the bearing surface to disrupt the lubricant "wedge"
- changing the oil pressure

More permanent corrective steps to solve oil whirl problem include:

- installing a new bearing shell with proper clearances
- preloading the bearing with an internal oil pressure darn
- completely changing the bearing type to oil film bearings which are less susceptible to oil whirl

Alternative bearing types include lemon bearings, dam wall bearings or tilting pad bearings. The tilting pad bearing is possibly one of the best choices since each segment or pad develops a pressurized oil wedge tending to center the shaft in the bearing, thereby increasing the system damping and overall stability.

Monitoring journal bearings

It is generally accepted that the best method of measuring vibration in sleeve bearings is by using a proximity probe. These are sometimes known as non-contact pickups, or eddy current probes. Displacement probes monitor shaft vibration and static shaft position. It is important to note that displacement probes measure only the relative motion of the shaft or rotor relative to the casing of the machine. If the machine and rotor are moving together, displacement is measured as zero (while in fact the machine could be vibrating heavily).

Displacement probes have excellent low frequency response. Probably the best sleeve bearing condition data at lower frequencies up to approximately 5X RPM is captured by non-contact probes reading relative shaft vibration. Probes are also often used to provide shaft tachometer (speed) signals by sensing a shaft surface mark, hole, or projection. This is known as a key phasor.

Some of the disadvantages of these probes are that they are somewhat difficult to install and replace, require external power sources, and must be calibrated to the surface material.

When taking spectral data from a proximity probe, it is important to point out that it is quite normal to see several running speed harmonics. This is unlike velocity spectra taken from bearing caps in which normally only the first 2 or 3 harmonics are seen, and each succeeding harmonic normally is only about 1/3 the height of the former (if no problems are present). Still, even with proximity probe shaft vibration data, the harmonics should also disappear into the spectral noise floor.

Using the SKF Microlog Analyzer GX series to monitor journal bearings

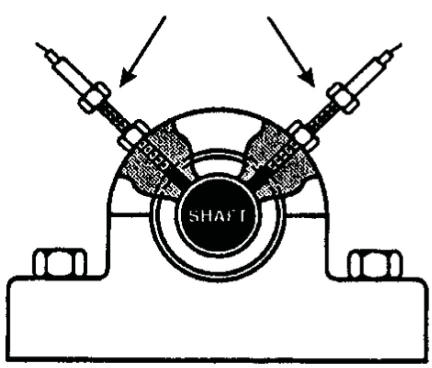
The SKF Microlog Analyzer GX series support two-channel measurements in both NonRoute (GX v1.02 and above) and Route (GX v2.00, SKF @ptitude Analyst v4.00 and above). The two-channel measurements supported are:

- orbit, including gap readings for a shaft-centerline plot
- cross-channel phase
- simultaneous 2-channel FFT and/or time

To diagnose journal bearing vibration problems, the orbit is a key tool.

Orbits

An orbit measurement is taken with a pair of proximity probes (non-contact probes), mounted radially, 90° apart (or as near as is practical). Typically the probes are mounted at ±45° as shown below. When setting up an orbit point in SKF @ptitude Analyst, the actual probe angles and tacho (key phasor) angle can be specified: the orbit trace is then rotated so that it corresponds to the actual motion of the shaft.



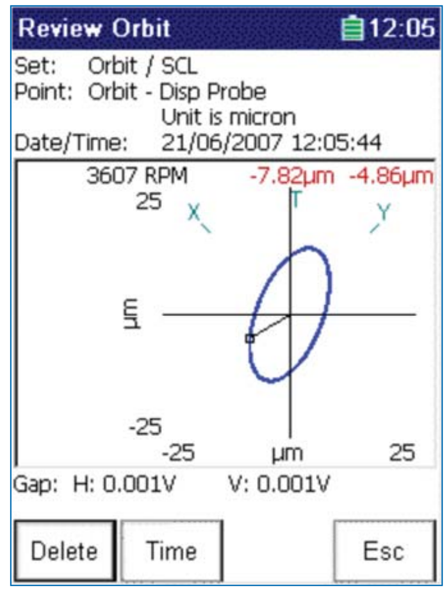
For NonRoute Orbit measurements, the sensors are assumed to be placed at 0° (X – CH1) and 90° (Y – CH2), and so the actual shaft motion may be rotated relative to that shown on the screen.

The orbit display shows a single trace which illustrates how the shaft moves within the bearing as it rotates, in the style of a Lissajous figure. Both filtered and unfiltered orbits can be viewed.

Filtered orbits

A filtered orbit excludes any frequencies not related to the 1X, 2X or 3X. As a result, a filtered orbit is always a smooth ellipse or circle.

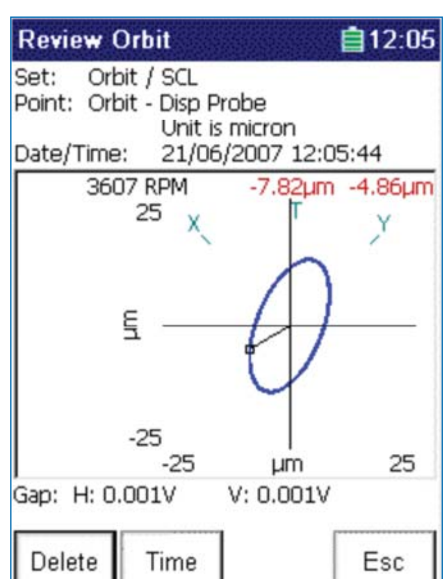
It is very important however to note that a filtered orbit does not tell the whole story, and non-1X related frequencies may be present which are missed if a filtered orbit is used.



Note that in the SKF Microlog Analyzer GX, filtered orbits are only supported in NonRoute, and any harmonic from 1X to 8X can be viewed. In SKF @ptitude Analyst, orbits are stored in unfiltered mode, but can optionally be viewed with a 1X, 2X or 3X filter applied.

Unfiltered orbits

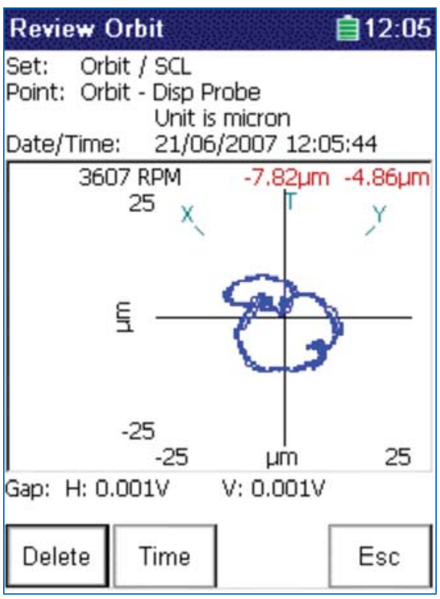
Unfiltered orbits show the raw time data collected from the two proximity probes, so they contain more information than a filtered orbit. In the SKF Microlog Analyzer GX, orbit measurements on the route are always displayed as unfiltered orbits, but may be viewed in filtered form after unloading to SKF @ptitude Analyst.



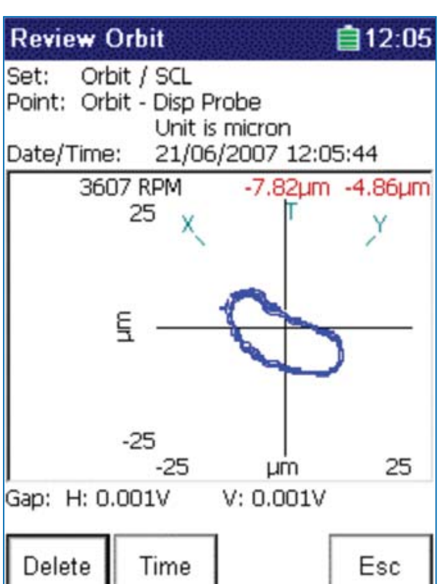
Some problems that might be detected using an unfiltered orbit are shown in the following:

- For a machine with only unbalance, the displayed orbit will be dominated by a once-per-revolution signal and have a circular or elliptical shape.

Orbits can exhibit one or more loops. Inner loops presented in an orbit plot will indicate a “hit and bounce” type condition. This occurs when the shaft contacts the bearing surface and bounces off. In early stages of contact, a “flat spot” in the orbit plot will be presented. As the condition becomes more severe, the number of loops present will increase and become tighter.

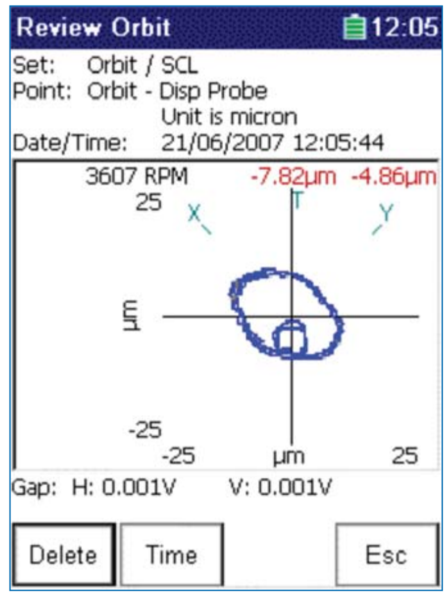


Misalignment will add significant levels of vibration to the shaft at once-per-revolution signal, and will show vibration at 1X harmonics. The once-per-revolution component from misalignment will not be in phase with the unbalance component. The effect is to make the orbit plot less circular and more elliptical or even non-elliptical (i.e., banana-shape or figure-eight pattern).



The orbit of a shaft experiencing oil whirl will show a secondary loop that moves around independently of the unbalance loop, as it is not synchronous with rotation.

The orbit of a shaft with rub (i.e. where a shaft contacts the bearing surface) has a shape similar to whirl except that the internal loop remains steady and does not rotate, as it is occurring at an exact sub-multiple of the shaft rotation speed. As a rub becomes more severe or continuous, then the orbit becomes more complex and possibly erratic, with bearing structural resonances being excited and multiples of subsynchronous frequencies showing up.



On the SKF Microlog Analyzer GX, the precession of the orbit loops can be examined by using continuous acquisition mode. Alternatively, after unloading to SKF @ptitude Analyst, the orbit can be viewed one rotation at a time to see how the loops change.

Mechanical looseness caused by excessive bearing clearance will tend to produce a rub orbit. Sub-synchronous effects show up as secondary loops. However, in this case, shaft movement may be in a forward direction, rather than the reverse precession that is typical of a rub.

Resonance and excessive bearing wear will be indicated when the orbit changes noticeably with changes in running speed.

Cross channel phase

Cross channel phase measurements can be used to determine how a shaft is moving, by taking measurements at bearings at each end of the shaft.

Appendices

The following tables are an overview of the typical causes and symptoms of journal bearing vibration, together with a summary of the bearing design features and operating conditions that may affect that vibration.

The tables are a only a guide and should be used in conjunction with the machine's maintenance and operating manuals, and vibration analysis guidelines such as those contained in ISO 18436 Part 2.

Appendix 1 – Typical causes of bearing vibration

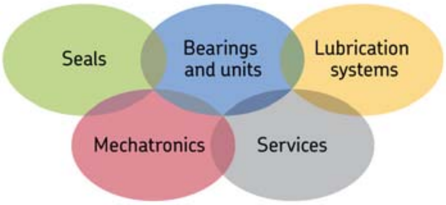
Source of vibration	Character and frequency relative to 1X	Conditions of occurrence	Suggested remedy	Remarks
Out of balance	Steady 1X	• Rotor out of balance • Journals misaligned (with three or more bearings)	• Re-balance • Check journal alignment	Some vibration usually present
Out of balance thermal effect	Varying amplitude, 1X	Thermal distortion of rotor	Improve starting and operating techniques	Mainly on rotors with high temperature inlet
Bearing – light load instability	Irregular, less than 1X	Light bearing load, e.g. turbine at 3 000 RPM with bearing loading less than 0.4 MN/m^2 (60 lbf/in^2)	• Vary oil supply condition • Use stabilised bearing	Mainly on small turbines
Bearing – half speed oil whirl	Whirl at 0.42X – 0.46X	Within narrow speed range close to twice critical speed	Change critical speed of rotor	
Bearing – low frequency oil whirl	Whirl at lowest critical speed, below 0.5X	Too wide speed range	• Vary oil supply condition • Use elliptical or dam wall bearing	Greatest risk when critical speed is below 0.4X
Steam force	Whirl at lowest critical speed, below 0.65X	Instability above certain turbine load	• Vary oil supply condition • Use elliptical or tilting pad bearing	Mainly on high pressure turbine of high-rated turbine set
Synchronous whirl	Very slow buildup of amplitude at 1X	May occur during starting or on change of load condition	• Vary oil supply condition • Shorten bearing • For elliptical bearings, increase vertical clearance	• Intermittent on certain sets • Sometimes mistaken for thermal wander

Appendix 2 – Design features affecting bearing vibration

Design feature	Requirements
Critical speeds	• No critical speed near normal running speed. • No critical speed close to half running speed. • Lowest critical speed above 40 % of running speed when possible.
Bearing design	• Adopt special bearing design. • Appendix 3 lists typical causes of bearing vibration, and some of the design features and operating conditions which affect bearing vibration. • See Appendix 1 for typical causes of bearing vibration, where there is risk of any type of unstable vibration. Use same bearing design at both ends of each rotor.
Maintenance of bearing alignment	• Design plant so there is little or no distortion of stationary parts, either thermally or under load, which may change bearing alignment.

Appendix 3 – Operating conditions affecting bearing vibration

Condition	Observed vibration	Means of improvement
Starting	Responds to speed or temperature change	• Follow instructions provided in the machine operating manual. • Operate machine so it passes rapidly through critical speed ranges: elsewhere change running conditions slowly and steadily.
Loading	Responds to load change	• Follow instructions provided in the machine operating manual. • Change load conditions slowly except in an emergency.
Journal bearing alignment	Responds to alignment change	• Examine operating conditions affecting alignment. Reduce variation if possible. • During overhaul, reset alignment so it reduces variation during operation.
Oil supply to bearings	Responds to change in lubricant condition	• Follow instructions provided in the machine operating manual. • Change oil supply pressure and temperature. • Watch oil outlet temperature and bearing temperature.



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Publication **CM3114 EN** • March 2009