Rolling Element Bearing Methodology
Application Guide
Bently Nevada Machinery Condition Monitoring

Objective

This document is intended to be the complete source for understanding Bently Nevada’s methodology on monitoring/managing rolling element bearings. The intention is to reduce the number of sources/locations where such information is stored.

Deliverables and CTQs

This document describes in detail Bently Nevada's methodology for managing Rolling Element Bearings which has been proven over the past years using Trendmaster Pro, 1900/65A, vbX/SCOUT instruments along with Ascent software and System 1 Evolution. This methodology is accepted throughout Bently Nevada and multiple industries.

Introduction

This document is intended to be the master document for Bently Nevada's methodology for management of rolling element bearings (REB’s). To avoid the repetitive reworking and releasing of multiple documents, all other documents that require mention of Bently Nevada’s REB management methodology should simply refer to this document rather than duplicating parts of it.

Bently Nevada's REB management methodology includes the important aspects of transducer types/usage, transducer mounting, monitor and software configuration and processing techniques. It is based on both the principles of rotating machinery behavior and the vibration characteristics of rolling element bearings. It combines traditional strategies for detecting rotor-related problems (such as unbalance and misalignment) with specialized methodologies (such as enveloping or demodulation) for identifying rolling-element bearing defects. The monitoring system must provide data for detecting and preventing rotor and bearing related problems and give adequate advanced warning of these problems so corrective action may be initiated. The system must be capable of discriminating among bearing faults, and give an early warning when a lubrication problem is present. All this is accomplished using an accelerometer and presenting data in acceleration, velocity and acceleration enveloping (demodulation) units.

Note that in this document, both the American and European conventions for marking decimal places are used. Numerical values may employ either the "." or "," mark, as in 1.41 or 1,41.
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1. Rolling Element Bearing Basics

1.1 Rolling Element Bearing (REB) Principles

All bearings serve the primary function of keeping a rotating part from contacting another part that may be either rotating at a different speed or stationary. A bearing must provide stiffness to keep the rotor in position, and it must do so while minimizing friction. Fluid-film bearings do this by providing a thin layer of lubricating fluid between the rotor and journal. Rolling element bearings do this using a set of balls or rollers (also called elements), lubricated by grease or oil, that keep the inner and outer rings separated.

Rolling element bearings find many uses in today's machinery. They are used in applications where rotation speed is too low to create a reliable fluid-film support, where the cost or weight of a fluid-film bearing and its associated lubricant supply system cannot be justified, or in applications that require close tolerances between stationary and rotating parts. They can be found in motors, wind turbine gearboxes, gas turbines, pumps, and many other machines. Some of the reasons rolling element bearings are used are: low starting friction, low operating friction, ability to support loads at low (even zero) speed, simpler lubrication requirements, and the ability to support both radial and axial loads in the same bearing. When some of these factors are important, rolling element bearings may be in use. For example, industrial gas turbines use fluid-film bearings, whereas aircraft gas turbines use rolling element bearings because of weight and stiffness constraints. Large critical machinery in power plants have fluid-film bearings, while for cost reasons many smaller motors and other supporting machines have rolling element bearings.

Rolling element bearings are usually subjected to both static and dynamic radial loads in combination with axial or thrust loads. The radial loads are transmitted from the shaft, through the inner ring, shared by several rolling elements, and transmitted to the outer ring and into the housing. The number of rolling elements that share the load transmission will depend on the element spacing in the bearing and on the tolerance class of the bearing. High quality, close tolerance bearings will have more rolling elements involved simultaneously with supporting the load and load transmission than will a loose tolerance bearing.

The region of the bearing that is involved in the load transmission is referred to as the load zone. The load zone represents the area of the bearing involved in the transmission of radial loads. The angular
size of the load zone depends on the precision of the bearing. High precision bearings have wider load zones than low precision bearings. For the portion of time (rotation) that rolling elements are outside the load zone, they do not transmit radial load.

Rolling element bearings come in special designs for radial loads, thrust loads, or a combination of these loads. They differ in the shape and angle of contact of the elements, and in the geometry of the races (the groove or surface in the rings that the elements ride on). Most simple ball bearings, while primarily designed for radial loads, can handle a limited amount of thrust loading as well.

When more thrust must be accommodated, angular contact ball bearings are often used. These bearings have an extended race design that provides more support for the balls in thrust loading. Angular contact ball bearings can be installed in back-to-back pairs to provide thrust capability in both directions.

High loads require more contact area between elements and races to keep stresses within allowable limits. To accomplish this, cylindrical rollers are used instead of spherical balls. This geometry provides a much larger contact area, and lower stresses for the same loading.

For combinations of high radial and high thrust loads, tapered roller bearings are used. For example, tapered roller bearings are often used as automobile wheel bearings because of the need to handle high radial loads (loads from braking and bumping) and high thrust loads (loads from cornering).

For very high thrust loads, special thrust bearings are used where the rings are separated axially instead of radially. The thrust bearings can have balls, tapered rollers, or needles as rolling elements.

Most common rotating machine applications involve relatively simple radial or angular contact ball bearings. These kinds of bearings are commonly used in industrial machines such as electric motors, pumps, and fans. They are also the predominant type used in aeroderivative gas turbines.

1.2 Basic Bearing Geometry

Rolling element bearings come in a wide variety of shapes (geometries) that are optimized for different applications. Ball bearings are used for moderate loads and produce nearly “point” contact between the elements and races. This contact “point” is actually a small, elliptically-shaped contact area. Cylindrical and tapered roller bearings are used for relatively high loads and produce a much larger contact area. Some rollers are crowned or spherical and have shallow relative radii, producing an intermediately sized contact area for better load distribution and higher load carrying capacity than a simple ball bearing.

It is beyond the scope of this paper to explore the different geometries in detail. Consult a reference such as Harris [Ref. #1] for more information. See “References” on page 36. Also, most bearing manufacturers have websites containing a wealth of information.

A simple ball bearing serves to illustrate the geometry of a simple REB. Figure 1 shows a cross section of a radial ball bearing that is subjected to a simultaneous radial and thrust load. The clearance and axial displacement are exaggerated for clarity.
The clearance and axial displacement are exaggerated for clarity. Important parameters are the pitch diameter, \( D \), the contact angle, \( \alpha \) (alpha), and the element diameter, \( d \) (not shown). The cage serves to control the spacing of the elements.

The bearing consists of an outer ring and race, an inner ring and race, cage and a set of rolling elements. In most applications, the outer ring is installed with a slight interference fit in the machine structure and does not rotate. The inner ring carries the rotating shaft. In some cases, the inner ring is stationary and the outer ring rotates. In rare applications, both the outer and inner rings rotate at different angular velocities. The cage serves to keep the rolling elements separated and evenly spaced and, in roller bearings, to control the orientation of the rollers.

Four parameters are used to calculate characteristic bearing frequencies that are important for diagnostic work. See "Bearing Defect (Fault) Frequencies" on page 13. The pitch diameter, \( D \), is the distance across the bearing between element centers. The contact angle, \( \alpha \) (alpha), is a function of the ratio of the radial load to the axial load and the curvature of the elements and race surfaces. For purely radial loading and symmetric races, the contact angle is zero. The element diameter, \( d \), is the diameter of the rolling element, and \( n \) is the number of elements.

### 1.3 Load, Stresses, and Bearing Life

Properly designed and carefully lubricated fluid-film bearings can have virtually infinite life. However, given enough time all REBs will eventually fail through a fatigue mechanism that can be greatly accelerated by poor installation, overloading, improper lubrication, or contamination.

A load that originates in a rotor shaft is transmitted in a rolling element bearing from the inner ring and race, through the rolling element, and through the outer race and ring into the surrounding structure. Because of the geometry of the bearing parts, contact between the rolling elements and the races occurs in extremely small physical dimensions. Theoretically, an infinitely stiff spherical ball will contact an infinitely stiff race only at a point. Given any load at all, this would theoretically produce an infinite stress.

In reality, mutual elastic deformation of the ball and race will produce an elliptically shaped zone of contact over which the load is distributed. Similarly, a roller in contact with a race of exactly the same length will produce a rectangular contact zone. The deformation in the “point” contact zone produces
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a compressive stress distribution that was first analyzed by Hertz in 1881. Because of this, these stresses are referred to today as Hertzian contact stresses.

The maximum compressive stress occurs at the center of the contact zone and can be quite large in modern bearing designs. Harris stated that rolling element bearings often operate continuously with compressive stresses exceeding 200 ksi (1400 N/mm²)(Ref.#1). See "References" on page 36. The microscopic surface roughness of the contacting elements further complicates the picture, as does the presence of any kind of debris that may become trapped between the rolling element and the race.

It is, however, not the surface compressive stress that leads directly to failure. According to the distortion energy theory, the effective (Von Mises) stress is responsible for material yielding. Surprisingly, for the type of loading that occurs in rolling element bearings, the maximum effective stress occurs at some depth beneath the surface of the contacting elements. (Ref #1) See "References" on page 36.

As a rolling element passes over a location on a race, the material beneath the surface of the race is subjected to a high effective stress for a short time during element passage. This cycling of stress occurs once per element passage at the location. Simultaneously, a location on the rolling element itself is also subjected to a cycle of subsurface stress twice per revolution of the element. As the stress cycling occurs, the high subsurface stresses can produce subsurface micro-yielding and phase transformations that favor the development of a fatigue crack. Because of the nature of the element loading and the stresses that exist in rolling element bearings, they often fail due to the propagation of fatigue cracks that originate below the surface.

The cleanliness of bearing lubricant can also have a strong influence on bearing life. Loewenthal has shown that ultra-filtration can significantly prolong bearing life (Ref#2). See "References" on page 36. Even micron-sized debris that becomes trapped between a rolling element and a race can cause local plastic yielding and the formation of a small pit in either or both surfaces. The relatively small radius of the pit boundary acts as a stress concentration factor and greatly increases the likelihood of initiation of a fatigue crack.

Because of this complexity, rolling element-bearing failure is very much a statistical process. For any individual bearing, the timing of fatigue crack origin can depend on the details of metallurgical microstructure such as the location and size of inclusions, phase changes, and discontinuities. It can also depend on deviations of surface geometry or surface smoothness.

Because of the complex and concentrated loading in the various parts of a rolling element bearing, bearing lifetime is limited and is very sensitive to load. Typically, rolling element bearing life, \( L \), can be expressed as:

\[
L = L_R \left( \frac{F_R}{F} \right)^{3.33}
\]

**Figure 1 - 2: Equation #1**

Where \( L_R \) is the lifetime at the rated load, \( F_R \), and \( F \) is the actual load applied to the bearing. The exponent of 3.33 applies to roller bearings, whereas an exponent of 3 replaces it for ball bearings. \( L_R \) is usually expressed in terms of a large number of revolutions, such as 90 million. (While this seems like a large number, 90 million revolutions will be achieved by a 3600 rpm machine in a little over 17 days.) See "References" on page 36.
The lifetime, $L$, from the above equation is usually stated in terms of a reliability number such as 90% or 50% for a group of apparently identical rolling element bearings operating under identical loads and speeds. For example, the $L_{10}$ (sometimes called $B_{10}$) life is defined as the rated life where 10% of the group will show evidence of fatigue failure and 90% of the group will show no evidence of failure (Ref#3). See “References” on page 36. Rated lifetimes are based on bearing operation under ideal conditions at steady load in clean environments with clean lubricating oils.

Note that from Equation #1 above, bearing life is related to about the 3rd power of the load. Thus, a load increase of 50% above rated load results in a 70% reduction in expected lifetime. This computation assumes a steady load on the bearing. Dynamic or shock loading of rolling element bearings can drastically reduce bearing life.

### 1.4 Causes of Failure

Most real-world bearings do not operate under ideal conditions and they fail well before reaching their theoretical design life. Final rolling element bearing failure is almost always a result of fatigue, but premature failure can be related to a number of root causes. When analyzing premature rolling element bearing failures, it is important not only to detect that the bearing is failing, but also to determine the underlying cause of that failure. Most premature bearing failures can be attributed to one or more of the following causes:

- Excessive loading, either static or dynamic
- Improper lubrication
- External contamination
- Improper installation
- Incorrect bearing selection for application
- Exposure to vibration while not rotating
- Passage of electric current through the bearing

This list includes the major causes of premature bearing failure and can be used as an initial guide to determine the reason for a bearing failure. To ensure success, elimination of premature bearing failures must be a major goal of any predictive maintenance program.

**Excessive Loading**

As shown above, bearing life is very sensitive to loading. Bearing loads can be classified into static loads, which have constant magnitude and direction, and dynamic loads, which can vary in both magnitude and direction. To properly size a bearing at the design stage, a machine designer must consider both types of loading and their influence on the predicted life of the bearing. If during service the loads in a machine deviate significantly from design values, bearing life will be affected.

Misalignment is an important source of excessive static load. Properly designed and sized couplings will accommodate misalignment to some degree, but if misalignment exceeds the capability of the coupling, then excessive loads may appear at the bearing. Another important source of static load is a belt drive.

Dynamic loads are introduced by many rotor malfunctions, but may also result from the work the machine is doing. Unbalance, rub, or aerodynamic or fluid-induced instability can produce dynamic
loads that exceed design limits and reduce bearing life. Also, coupling malfunctions may introduce excessive static and dynamic loads.

**Improper Lubrication**

Rolling element bearings depend on an extremely thin layer of lubricant to keep rolling element/race contact to a minimum. The rolling elements actually skid to a degree, and the lubricant forms a type of elastohydrodynamic wedge that keeps the elements separated. The separation of elements is on the order of a few times the typical surface roughness, less than a micron (or a few micro inches).

The lubricating fluid also serves the purpose of removing the heat that is generated by shearing in this very small gap between elements and races. If the lubrication supply is insufficient, then metal-to-metal contact will occur, creating wear and possible smearing of races. The lubricant must be clean and of the proper type for the load and operating temperature. Rolling element bearings can survive for long periods of time quite well with relatively small amounts of lubricant. Excessive lubrication can be harmful by causing high temperatures from the lubricant shearing stresses and can actually lead to bearing failure.

**External Contamination**

Cleanliness of lubricant is vital to long bearing life (Ref #2). See "References" on page 36. If dirt particles get into the bearing either directly or through the oil system, then the particles can become trapped between the rolling element and the race. If the particle has sufficient hardness, then the extreme stress produced at the particle contact will cause local plastic deformation of the race and/or element material. This will produce a permanent pit that will act as a potential crack nucleation site. It is also possible for a large particle to jam in such a way as to reduce element rolling action and increase sliding motion. During sliding motion, the dirt particle can act like a plow, creating a permanent groove in the race surface.

**Improper Installation**

Many bearings have been doomed during installation. Bearings must be handled carefully and installed in accordance with manufacturer’s specifications. If interference fits are used on the bearing rings, care must be taken to avoid deformation or cocking of the rings during installation. Cocked or misaligned bearing rings will produce abnormal loads in the bearing. Incorrect fits can also subject the bearing to abnormal loads. And although this may seem obvious, bearings should never be pressed on in such a way that the press load is delivered across the rolling elements. The high press loads can cause the balls to permanently dimple (brinell) the races.

**Incorrect Bearing for Application**

If the designer fails to consider all the possible loads that can occur in service, the bearing selected for the application could be incorrect. If the service loads exceed the design intention, then the bearing will fail prematurely. It is important for the designer to be aware of the static and dynamic loading that may be present in service.

On the other end of the load spectrum is the situation where the bearing is very lightly loaded. In this condition the roller skids rather than rolls, which is also not desired.

It is also possible that during assembly or repair a wrong bearing was installed.
Exposure to Vibration While Not Rotating

Brinelling is a form of non-rotating load damage. When a rotor is stopped, the rolling elements in bearings remain in one position. If the machine is subjected to a shock load, the elements (especially balls) can plastically deform the races, creating a permanent set of pits. This has happened to new machines being shipped by rail or truck. The machines arrive at the plant only to find that the bearings are ruined. To prevent this kind of damage, vulnerable machinery is usually rotated on turning gear during shipment.

It is normal that a spare (stand-by) machine is exposed to some level of vibration while not operating. Its static load will press lubrication away from the bearing load zone, allowing metal to metal contact between elements and races. If the vibration is high enough, when the primary machine fails after a long period of service the spare machine's bearings may fail due to brinelling damage shortly after it is put on-line and before the primary machine is repaired. A preventative measure is to run the backup machine periodically.

Electrical Current (EDM)

Improper electrical grounding can cause a current to flow through rolling element bearings. If arcing occurs, small quantities of bearing material are vaporized in the small, high temperature electric arc. This vaporization effectively removes material from the rolling element or bearing race or both. The resulting pit acts as a stress concentrator that is capable of nucleating a fatigue crack, leading to eventual failure. The damage caused by the electrical arc is similar to electrical discharge machining (EDM), and takes place on motors with a variable frequency drive. On rare cases this can also be due to buildup of static charge from the process fluids and materials.

1.5 Fatigue Failure in Rolling Element Bearings

Assuming that a bearing is properly loaded, clean, and supplied with a correct amount of clean lubricant of the proper type; failure of a typical rolling element bearing is still inevitable because of fatigue. Rolling element-bearing loading differs substantially from the type of loading that is found in other engineering structures where an endurance limit can be used to guarantee freedom from fatigue failure. Unfortunately, no such endurance limit exists for rolling element bearings, and fatigue failure of these bearings will eventually occur.

Failure of the bearing is most likely to begin with the initiation of a subsurface fatigue micro crack in an area of high effective, or Von Mises stress. Analysis has shown that the maximum of this stress occurs some distance below the surface of the race or rolling element (Ref#1). See "References" on page 36. Crack initiation is likely to be associated with some sort of discontinuity or phase change in the metal microstructure that acts as a local stress concentrator. Alternatively, surface damage due to debris can also produce a plastic deformation that will act as a stress concentrator. Barring surface damage, the location of the crack will be related to the subsurface stress distribution combined with the statistical distribution and size of material micro flaws.

As element passage produces stress cycling, the fatigue crack will begin to grow. The direction of growth may start to spread beneath the surface, but eventually the crack will migrate to the surface of the race or the rotating element. When the remaining material is sufficiently weakened, a spall will break away from the surface, leaving an irregularly shaped hole, or pit. The spall itself is a metal chip that becomes free to migrate around inside the bearing and cause additional damage. Spalls can range in size from very small pieces to large chunks nearly as long as the width of the race.
The initial surface failure leaves a sharp-edged hole in the race or rolling element (for this discussion, assume that the hole is in the race and that the rolling element is a ball). When the rolling element encounters such a hole in the load zone of the bearing, the ball will be forced into the hole. This will produce a sudden unloading of the ball and a corresponding unloading of the race. The race will elastically rebound slightly in the vicinity of the hole. When the ball encounters the sharp edge at the opposite side of the hole, it will be suddenly reloaded, and the race will return to its normal, loaded position. The sudden encounter with the edge of the spall pit causes an impact to be transmitted to the rotor and machine structure that will generate high frequency vibration.

When the hole is fresh, the edges of the hole have relatively sharp corners, and the sudden encounter of ball with the hole edge produces a step impulse to the entire rotor system similar to striking the race with a hammer. The step impulse has wide frequency content and will excite the rotor system, the housing, the casing and the transducer natural frequencies. These natural frequencies can extend to high frequencies and produce an increased high frequency vibration that is detected with accelerometers. The step impulse repeats when the next ball encounters the hole.

**Figure 1 - 3: Fatigue Damage Development Over 1200 Hours of Operation (Ref #6)**

The very high stresses at the contact point of the ball and the edge of the hole will usually exceed the yield strength of the material, and the edge of the hole will become peened with repeated passages of the balls. Thus, over a longer period, the hole edge will become smoothed, and the amount of high frequency vibration energy will decrease. The high stresses at the contact point also accelerate the fatigue of the balls, shortening the time it will take to develop fatigue cracks.

Meanwhile, the spall is free to move around inside the bearing where it can become trapped between a ball and a race, or between a ball and the cage. If the chip becomes trapped between the ball and race, the local load is sharply increased, causing the outer race to deflect farther than it would under normal loading. Plastic deformation of the race and ball is likely during this period because of the very high local stresses. Additionally, the very high stresses will accelerate the formation of more fatigue cracks.

Thus, the formation of a single defect will usually accelerate the formation of additional defects in the bearing. Damage will spread from races to rolling elements to the cage in a relatively short time. As the damage spreads and multiplies, the high frequency vibration energy will tend to increase until the damage becomes so extensive that gross peening of the nearly totally destroyed race reduces high frequency energy again. Thus, in the final stage of bearing failure, high frequency vibration may decrease. For this reason, high frequency vibration, like demodulation, should not be used as a sole bearing failure indicator.
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In the final stage of failure, gross wear of the rolling elements and the races will open the bearing clearances. This will reduce the constraint on the motion of the rotor shaft, and, because rolling element bearing machines are built with close internal clearances, a rotor to stator rub becomes a real possibility. Bearing temperatures are likely to increase because of locking and sliding of the rolling elements and the inability of the lubrication film to keep the elements separated. Noise and lower frequency vibration will also increase.

1.6 Failure Stages

There are four main failure stages for rolling element bearings and each stage is defined by a frequency range. These stages have been defined by Technical Associates of Charlotte, P.C. (Ref#5) and are described below. See "References" on page 36.

Stage 1

This stage of bearing failure appears at ultrasonic frequencies ranging from approximately 20kHz – 350kHz. This is the earliest failure stage and indicates slight defects that are not visible; subsurface fatigue micro cracks appear as described above. Bearings should not be replaced if the defect frequencies are shown only in this high frequency range, typically by use of enveloping. As wear progresses, the defects are manifested in lower frequency data.

Stage 2

As bearing defects become larger, they ring the natural frequency of the bearing components and sometimes the bearing support/machine casing. As a result, this stage occurs in the 500 – 2000 Hz frequency range. Sideband frequencies around the component/casing natural frequencies begin to show at the lower end of the frequency range due to progressive wear. Higher frequency information is also contained in the spectrum. Enveloping techniques are used to detect the defect frequencies, and the acceleration waveform and spectrum may have indications of the bearing problem.

Stage 3

During this stage, defects and wear become visible and the need for bearing replacement is imminent. Bearing defect frequencies appear now also on the velocity spectrum. As wear progresses, well-formed sidebands accompany the defect frequencies and harmonics. Sidebands also continue to grow around the bearing component natural frequencies. The high frequency content evident in stages 1 and 2 is now even higher in amplitude/energy.

Stage 4

This stage is comprised of mainly rotor-related frequencies (i.e., 1X, 2X, 3X, and 4X RPM). When bearing defects progress to this stage, increases in rotor vibration occur. This causes the bearing component natural frequencies and some of the bearing defect frequencies to decrease in amplitude. However, more random broadband high frequency vibration develops, resulting in an increasing noise floor. Just prior to total bearing failure, amplitudes in the highest frequency region defined in stage 1 may grow excessively. If a bearing reaches this stage, damage to other machine components (e.g., rotor, seal, coupling) is highly possible.
1.7 Bearing Defect (Fault) Frequencies

As the bearing evolves toward fatigue failure, spalling occurs on the outer race, inner race, or elements. As rolling elements encounter spall pits (or as damaged elements rotate), the impact/response vibration tends to repeat at characteristic frequencies, called bearing defect frequencies, often referred as fault frequencies. Each bearing has a set of defect frequencies associated with inner race, outer race, element, and cage defects that are determined by its particular geometry and operating speeds.

For an outer race defect (with the outer race fixed), the sequence of ball/hole impacts produces a series of more or less constant amplitude impulse/response vibration signatures that repeat at the ball pass frequency. For an inner race defect, the defect on the race will rotate in and out of the load zone. When the defect is in the load zone, the impulse/response vibration amplitude will be relatively large; when the defect is out of the load zone, the amplitude of the vibration will be relatively low or nonexistent. Thus, the amplitude of the impulse/response vibration is modulated at the 1X vibration frequency of the rotor. This modulation can be a clue to the location of the defect. Repeating impulse/response vibration is used in the acceleration enveloping (demodulation) algorithm to detect bearing defects. Modulation will also occur in the case of a stationary outer ring when the dynamic load (i.e. unbalance) becomes large relative to the static load. In those cases where the inner ring is stationary and the outer race rotates, modulation will also occur.
In all cases, knowledge of the existence and extent of bearing damage is vital. Once damage has occurred, the flaw on any element rapidly generates secondary flaws on the other elements. It is important to detect that a bearing is failing so action can be taken in time to prevent more extensive damage. Frequency analysis is helpful, but in most cases, precise knowledge of which part of a bearing is failing is not as important because the entire rolling element bearing is normally replaced and not just the damaged portion. However, when evaluating the safe remaining operating time for a REB it is important to know which part of the bearing is failing. The general order of tolerance to a failure is (highest to lowest): Outer ring – Inner ring – Ball – Cage. This means that with outer ring failure the replacement is typically less urgent than with ball failure.

In a complex system like a multistage gearbox, it is also important to identify which bearing is failing. In a system with different types of bearings, frequency analysis and high frequency detection methods can be very helpful in identifying the problem bearing. System 1 Evolution (S1 Evo) contains information on more than 30,000 bearings. If the needed information is not available in S1 Evo but the bearing geometry is known, the bearing fault frequency equations on Figure 5 can be used to estimate the defect frequencies. High frequency vibration is likely to decrease with distance, which can also help identify which bearing is failing.

The equations in Figure 5 below have been derived using the bearing geometry and assuming there is no slippage (sliding) of elements (Ref#1). See “References” on page 36. When using these equations, remember that slippage may occur and modify the frequencies. Also, contact angle may change depending on the axial load and result in a change in frequencies.

Note that these frequencies may, and often do, include sum and difference frequencies. These frequencies will appear when the amplitude of the fundamental defect frequency is modulated by some other rotating element - often rotor speed. Any combination of flaws can generate or be modulated by the cage frequency, i.e. the frequency at which the elements revolve as a set.

In most applications, the outer race is fixed and does not rotate. However, in some applications, just the outer race, or even both races rotate. We will first present the result of the typical case where the outer race does not rotate (is fixed) in Figure 4 below. Then we will present the general case where both races rotate (labeled free in the equations).

Finally, it is important to remember that when both races are free to rotate, they can rotate in either direction, depending on the machine design. With these equations, use a consistent approach to define the direction of motion. For example, use positive speeds to define rotation in a counterclockwise (X to Y) direction and use negative speeds to define clockwise (Y to X) rotation.

The frequency equations can produce positive or negative numbers. The sign represents the direction of the motion relative to the observer’s location given the direction convention used. The sign in the final result does not matter for frequency analysis.
The following symbols are used in the equations:

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<th>Symbol</th>
<th>Description</th>
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<td>( N_0 )</td>
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<tr>
<td>( N_i )</td>
<td>inner race angular speed in rpm</td>
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</tr>
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</table>

A common group of factors appears in all the frequency equations. To simplify the expressions, define \( \gamma \) (gamma) as:

\[
\gamma = \frac{d}{D} \cos \alpha
\]

Rolling element bearings have specific defect frequencies depending on the size of the bearing components and the speed of rotation. REBs are comprised of an inner race, balls or rollers (needles, etc.), a cage, and an outer race as shown the following figure.

![Rolling Element Bearing Structure](image)

**Figure 1 - 5: Rolling Element Bearing Structure**

When damaged, these components produce vibration at fundamental frequencies related to the bearing's basic geometry. These formulas assume a rotating inner race and a fixed outer race.
In all cases when identifying defect frequencies, it is very important to know the correct shaft speed. When bearing information is not known, one clue to identifying bearing defect frequencies in the spectrum is that they are non-integer multiples of the shaft speed. A review of the S1 Evo bearing database, where fault frequencies are expressed as multiples of shaft speed (see the following table), is a good way to investigate this.

**Table 1 - 1: Defect Frequencies of Number 22320 Bearings Shown in S1 Evo Bearing Database**

<table>
<thead>
<tr>
<th>Manufacturer</th>
<th>Part Number</th>
<th>FTI</th>
<th>FTO</th>
<th>BFS</th>
<th>BPFO</th>
<th>BPFI</th>
<th>Outer Race Diameter</th>
<th>Inner Race Diameter</th>
<th>Bearing Width</th>
<th>Number of Elements</th>
<th>Element Diameter</th>
<th>Pitch Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>TOR</td>
<td>22320</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>ZKL</td>
<td>22320</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>14</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>L/B</td>
<td>22320</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>30.90</td>
<td>159.74</td>
</tr>
<tr>
<td>NSK</td>
<td>22320AM</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>NTN</td>
<td>22320B</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>14</td>
<td>30.90</td>
<td>161.44</td>
</tr>
<tr>
<td>NTN</td>
<td>22320BK</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>14</td>
<td>30.90</td>
<td>161.44</td>
</tr>
<tr>
<td>TORR</td>
<td>22320BCC3</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>SKF</td>
<td>22320CC</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>SKF</td>
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<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
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<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TORR</td>
<td>22320CNN33</td>
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<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TORR</td>
<td>22320CN33C2</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TORR</td>
<td>22320CN33C3</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>TORR</td>
<td>22320CN33C4</td>
<td>0.44</td>
<td>0.58</td>
<td>2.84</td>
<td>6.15</td>
<td>8.94</td>
<td>215.00</td>
<td>100.00</td>
<td>73.00</td>
<td>15</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

**Figure 1 - 7: Defect Frequencies of Number 22320 Bearings Shown in S1 Evo Bearing Database**
It is also worth noting that bearings with the same number but from different manufacturers may produce different fault frequencies. A customer’s CMMS may or may not have accurate information identifying which bearing is actually in use, so one should never confine the identification of a bearing fault to one manufacturer’s defect frequencies.

**Fundamental Train (Cage) Frequency (FTF)**

The cage frequency, $f_c$, is the frequency corresponding to one complete revolution of the bearing cage relative to a fixed reference. The cage frequency is often called the fundamental train frequency (FTF). System 1 Evo recognizes both FTFO and FTFI, cage frequencies relative to the outer and inner rings, respectively. Very often the latter is not taken into consideration.

FTF both races free (Hz)  
\[ f_{Cage} = \frac{1}{120} \left[ N_i (1-\gamma) + N_o (1+\gamma) \right] \]

FTF outer race fixed (Hz)  
\[ f_{Cage} = \frac{1}{120} N_i (1-\gamma) \]

**Outer Race Element Passage Frequency (BPFO, or ORBP, or EPx)**

This is the frequency at which the rolling elements pass a point on the outer race. When the outer race is free to move, the reference point moves with it.

ORBP both races free (Hz)  
\[ f_{Outer} = \frac{n}{120} \left[ (N_i - N_o)(1-\gamma) \right] \]

ORBP outer race fixed (Hz)  
\[ f_{Outer} = \frac{n}{120} N_i (1-\gamma) \]

**Inner Race Element Passage Frequency (BPFI, or IRBP)**

The BPFI is the frequency at which elements pass a point on the inner race.

BFPI both races free (Hz)  
\[ f_{Inner} = \frac{n}{120} \left[ (N_o - N_i)(1+\gamma) \right] \]

BFPI outer race fixed (Hz)  
\[ f_{Inner} = -\frac{n}{120} N_i (1+\gamma) \]

**Element Spin Frequency (BSF)**

The element spin frequency is the rotation speed of an individual rolling element. A defect on the element can produce a once per turn disturbance on either the inner or outer race at the Element Spin Frequency. It can also produce a disturbance on both the inner and outer race, producing a 2X Element Spin Frequency.
BSF both races free (Hz)

\[ f_{\text{Element}} = \frac{1}{120} \frac{D}{d} \left( N_o - N_i \right) \left( 1 - \gamma \right) \left( 1 + \gamma \right) \]

BSF outer race fixed (Hz)

\[ f_{\text{Element}} = -\frac{1}{120} \frac{D}{d} N_i \left( 1 - \gamma^2 \right) \]

**Example of defect frequencies**

The following figure is an example of how defect frequencies show up in a demodulation spectrum for a FAG 6004 bearing and a motor speed of 1970 rpm.

![Figure 1 - 8: Defect Frequencies (6004 BPFO) Shown with System 1 Evo](image_url)
2. Transducers Types and Usage

2.1 Transducer Location and Quantity

General

From an engineering principle standpoint, one would want to monitor machines with REB the same way as critical journal bearing machines are monitored (vibration transducers at each bearing, bearing temperature, Keyphasor sensor, online monitor). However, oftentimes it is economically difficult to justify that level of monitoring.

The best method for monitoring REB machines is online monitoring - using either a permanently installed continuous or scanning system. With an online system, a proper transducer suite is needed to detect when a machine is having a problem. See "Number of Sensors" below.

In the absence of an online monitoring system, a portable data collector (PDC) is the next best approach.

Number of Sensors

The recommended transducer suite for online monitoring of rolling element bearing machines is one vibration transducer mounted on each bearing. If the distance between outboard and inboard bearings is less than 1 meter (3 feet) and there is a solid metal connection between the bearings (as is the case with overhung pumps, electric motors and with many other machines), then the number of vibration sensors may be reduced to one per machine case. In rare cases more than one transducer per bearing is justified.

The transducer should be mounted on or near each of the bearings in a location that provides the best reading; there should be minimal number of interfaces between transducer and bearing. Please refer to the following figure.
For a portable data collector, the data collection points are chosen to accurately display and predict the condition of the machine. Typically, measurements are taken in 3 directions (horizontal, vertical, axial) for each bearing. But after a minimum of 10 surveys, the number of measurement directions per bearing can usually be reduced to one (horizontal), with one additional axial measurement from the bearing on either side of the coupling (see below).

**NOTE**
The above machinery measurement recommendations are guidelines only. Due to machine construction and environmental operating conditions, the monitoring package needs to be uniquely addressed for each machine or group of machines. In many situations these guidelines will be adequate.
**Speed Information**

As mentioned before, knowing the correct shaft speed is very important when identifying bearings fault frequencies.

For each variable speed machine monitored by an online system the speed needs to be available either by a Keyphasor or as an OPC tag from control systems. For constant speed machines there is no need to have online speed data; the speed needs only to be entered into the condition monitoring system during configuration phase.

When a PDC is used with variable speed machines then the speed at the time of data collection needs to be identified using a tachometer, stroboscope or by other means. With System 1 Evo it is also possible afterwards to manually enter the correct speed info.

### 2.2 Sensor selection

An accelerometer is the recommended sensor for condition monitoring of rolling element bearings. There are several reasons for this:

- Accelerometers give the earliest indication of a bearing problem (see the following figure). A lubrication problem will often lead to a bearing failure and, with a properly installed acceleration measurement, it can be detected early. See "Stage 1" on page 12. If a lubrication problem is corrected at this stage and before actual bearing deterioration has started, the maintenance work and operational interruption related to a bearing failure and replacement can be avoided. When symptoms of a bearing problem are detected with a Velomitor, Motor Current or Temperature probe, the physical damage may have already begun and the bearing change cannot be avoided. Refer to stages 3 and 4. See "Stage 3" on page 12.
- Demodulation (enveloping) is done using an acceleration signal.
- Alarm levels are based on acceleration readings. See "Alarm Levels" on page 29.
- The acceleration signal can be integrated to velocity to provide additional diagnostics. Thus, machinery problems like unbalance and misalignment that may lead to bearing failure can be detected and corrected. The use of the velocity signal for protection will be dependent upon the application and OEM recommendations.
- The acceleration signal integrated to velocity can be used to evaluate the failure indicators in stages 3 and 4.
- The acceleration waveform will give additional information about which part is failing and the severity of the problem.
**Figure 2-3: REB P-F Curve with Earliest Failure Detection Point for Different Sensors**

There may be other machine monitoring and protection reasons that require the use of sensors other than an accelerometer, but in those cases, failure stages 1 and 2 cannot be detected.

A brief description of different transducer types is provided below.

**Accelerometer**

Accelerometers measure the machine vibration in acceleration units and are mounted on the bearing housing. Bently Nevada has several accelerometers than can be considered. All these sensors are piezoelectric type with an integral charge amplifier. A summary of the most commonly used Bently Nevada accelerometers for REB applications are shown in the following table.

**Table 2-1: Most Commonly Used Bently Nevada Accelerometers**
An accelerometer is the only sensor that can be used to monitor the condition of rolling element bearings and predict failure over all four failure stages. An accelerometer is also needed when enveloping is required, a technique that has become an industry standard for REB monitoring. The lower frequency content of the acceleration signal can better be analyzed when integrated to velocity. Therefore, all instruments used with an accelerometer should support acceleration, acceleration enveloping and velocity readings from the accelerometer sensor.

Accelerometers are relatively easy to install and involve minimal resources. However, accelerometer performance is very sensitive to mounting location and technique, and a poor installation can very easily degrade the performance and frequency response of the accelerometer. Signal attenuation occurs with multiple material interfaces between a bearing and the transducer. See “Number of Sensors” on page 19. Recommended mounting techniques will be discussed later.

Long accelerometer cable runs will degrade the high frequency signal quality. Verify the effect of field cabling on the measurements in advance. Also, hazardous area requirements may influence field cabling and sensor selection. See “Cables” on page 28.

When using a PDC, a single accelerometer can easily be used on multiple machines using quick connection mounts or Mag-Base mounts. However, trending accuracy of high frequency data is sensitive to small variations in mounting location or technique. Consistent measurements must be taken, or trended values may be invalid and/or misleading. See “Mag-base” on page 28.

**Velomitor Seismic Transducer**

Velomitor piezo-velocity sensors are designed to measure absolute bearing housing, casing or structural vibration velocities. Velomitor sensors incorporate a solid-state design and are specialized piezoelectric accelerometers with embedded integration electronics. A summary of the most commonly used Bently Nevada Velomitor sensors for REB applications are shown in the following table.

| Table 2 - Most Commonly Used Bently Nevada Velomitor Sensors |
|------------------|------------------|------------------|
| **Sensor**       | **Freq Range**   | **Notes**        |
| 330500/525       | 4.5 – 5 000 Hz   | Recommended Velomitor |
| 190501           | 1.5 – 1 000 Hz   | Typically used for slow speed applications when protection is required |

Velomitor sensors can be used to monitor the condition of rolling element bearings and typically predict failure when the bearings are likely to have visible damage (stage 3). In order to get the earliest indication of a failure it is recommended that the measurement has the highest frequency range as possible, with peak detection used instead of rms.

The Velomitor sensor is not recommended to be used with a PDC.
Motor Current

Motor current monitoring using AnomAlert or another instrument will detect bearing problems when the bearing failure is affecting the motor rotor / stator relationship. Thus, it is much less sensitive to detection of early phase bearing problems than other REB vibration systems, especially if the bearing problem is on the driven machine. It can detect bearing faults in stage 3, when the bearing fault frequencies are detectable in the motor current spectrum.

Temperature

Thermocouples and Resistive Temperature Devices (RTD’s) are often employed for bearing temperature monitoring. Temperature measurement cannot be relied on to independently identify bearing faults. It may be difficult to mount the sensor close to the bearing due to the thickness of the casing or bearing pedestal. Many factors outside the machine can also influence the temperature readings, including ambient temperature and ambient airflow. The operator needs to be aware of all factors influencing the temperature reading and take them into consideration when a temperature change is noted. Nonetheless, temperature measurement should be considered if bearing protection is required, as the temperature typically increases as the bearing enters stage 4, nearing failure. See "REB P-F Curve with Earliest Failure Detection Point for Different Sensors" on page 22.

Thermocouples and RTD’s are much less complex in design than vibration transducers, and as such they are less expensive and generally very reliable. Also, their small physical size allows them to be employed when space is limited. Finally, they require almost no power to operate.

Bently Nevada does not manufacture temperature transducers. Temperature sensor specifications are manufacturer and model specific. See www.omega.com or www.minco.com for examples.

Roller Element Bearing Activity Monitor (REBAM)

In the early 1980’s Bently Nevada developed the REBAM methodology as a method of direct measurement of roller element bearing analysis [7]. This technique was first demonstrated by G. J. Phillips using fiber optic sensors observing deflections in the outer race of roller element bearings [8]. See "References" on page 36. Bently Nevada developed high sensitivity eddy current sensors to measure the micro-inch deflections in the outer race of a REB. REBAM can provide significant improvement in sensitivity and noise reduction [9], but REBAM was not widely adopted by industry due to the need to modify the bearing housing for installation of the REBAM probe. The REBAM probe and associated monitoring hardware are obsolete and no longer offered by Bently Nevada. However, there may be applications where the REBAM technology is still being used, so the following is offered for the purpose of awareness only. Refer to References 7, 8 and 9 See "References" on page 36.

Fault Detection

REBAM probes are used to monitor the condition of rolling element bearings and predict failure only when the bearings are likely to have visible damage. Refer to stage 3 See "Stage 3" on page 12.

Drawbacks

Since REBAM probes must view the outer ring, the bearing housing must be drilled and tapped to accommodate the probe. Installation is time-intensive and expensive when compared to casing-mounted transducers. Sometimes a relief hole around the probe tip must be used and if it exceeds the maximum allowable for the bearing, it can contribute to a cyclic stress failure of the outer ring. Also,
REBAM probes will not work if the outer race rotates or if there is a loose fit between the bearing and its housing, such as with squeeze-film damper bearings. Neither will REBAM probes work properly with triple ring bearings.

Predicting bearing failure after there is visible damage, as REBAM probes do, can be considered either a drawback or an advantage depending on customer preference. Although there is less advanced warning with REBAM than with other methods, when a bearing is indicated as defective, the defects are generally serious enough to warrant quick replacement. Poor lubrication or excessive wear is indicated by high frequency signals that are not likely to be picked up by displacement probes until considerable damage has been done. Also, inner race defects and corresponding signals are attenuated since the REBAM probe looks at the outer ring. The signal must pass from the inner race, through a rolling element, and to the outer race.

**Benefits**

REBAM may have been employed when casing mounted transducers had proven not to work for the application.

Signal-to-noise ratio (SNR) improvement was obtained using REBAM (Ref #7). See "References" on page 36. The REBAM vibration signal is separated into two main regions: Rotor Vibration Region and Prime Spike Region. In addition, because each REBAM probe looks directly at one bearing, there is no confusion as to which bearing has a problem.
2.3 Accelerometer Vibration Transducer Mounting

There are three main mounting methods used:

- Stud
- Quick connection adapter
- Magnet

Occasionally other methods are used, like an extension tip (stinger), adhesive, and even beeswax.

The mounting method has a significant effect on vibration signal linear response (see following figure).

**Frequency Response**

![Diagram showing the effect of different mounting methods on transducer sensitivity](https://via.placeholder.com/150)

*Figure 2 - 4: Transducer Mounting Effect on Sensitivity*
**Stud-Mount**

This type of mount gives the best coupling between a transducer and the casing. It requires a drilled and tapped hole in the machine casing into which the transducer or adapter stud is threaded (see the following figure). This mounting gives the full frequency response range for the accelerometers and Velomitor sensors shown in Tables 2-1 and 2-2 above.

![Dimensions and Tolerances for Stud Mounting](image)

**Figure 2 - 5: Dimensions and Tolerances for Stud Mounting**

**A Quick Connection Adapter**

This mounting type is preferred when a PDC is used. The mounting technique requires either a drilled and tapped hole in the machine casing into which a stud is threaded, or an adhesive to directly bond the quick connection adapter to the machine case. The quick connection stud should be mounted on a flat surface and is usually bonded with a quick setting epoxy like J-B Weld. This mount style has a +3dB point at approximately 6-7 kHz when the quick connection stud is attached. Both pieces of a quick connection adapter - the stud and base shown in the following figure - is threaded on two sides (i.e., two of the four quadrants are threaded). Due to cost this solution is not widely used.

![A Quick Connection Adapter Stud and Base](image)

**Figure 2 - 6: A Quick Connection Adapter Stud and Base, Bently Nevada Part # 46122**
**Mag-base**

Magnetic (Mag) base mounts (see following figure) are commonly used in walk-around programs due to the ease of installation and the variety of casing locations that can be tested with minimal mounting preparation. Depending on the level of coupling between the magnet and the casing, the +3dB point can be anywhere between 1 to 8 kHz. The latter requires a very smooth surface and a strong flat magnet. Most common is a two-pole magnet, which under good conditions gives a +3dB point around 2 kHz. When a magnetic base mount is used for routine data collection, the measurement point should be marked in some manner to consistently locate the sensor during each measurement. Paint markings around a mag-base mounting location, or a small punch mark, will promote measurement repeatability.

![Sensor on Magnetic Mount](image)

**Figure 2 - 7: Sensor on Magnetic Mount**

**Cables**

Sensor cables between an accelerometer and the charge amplifier are susceptible to creating their own acceleration signal due to the triboelectric effect if they vibrate. When installing an accelerometer that requires an external charge amplifier, route the accelerometer cable away from the moving components of the machine and avoid small bending radii in order to extend the cable life.

Long cable runs will reduce the overall system’s capability to measure high frequency readings. Use the Bently Nevada field wiring calculation tool to define the system maximum frequency for a given field cable length and capacitance.
3. Alarm Levels

Since the REB is always an integral part of the machine in which it is used, there is no industry standard for alarm levels associated with a particular REB. The machine OEM’s recommendations should always be followed for Alert and Danger alarm levels.

Bently Nevada does not make recommendations for machinery protection Alert and Danger levels, but can work with end users and OEMs to acquire and evaluate engineering data so the end user can set preliminary alarm levels.

In the absence of OEM recommendations or engineering data on a machine, some of the factors that influence vibration levels are:

- Machine speeds
- Bearing types
- Load on machine and bearings
- Environment temperature
- Signal path from bearing to sensor
- Sensor connection method
- Failing part in the bearing

ISO 13373-3

A draft standard currently under evaluation is “ISO 13373-3 Condition monitoring and diagnostics of machines - Vibration condition monitoring. Part 3: Guidelines for vibration diagnosis”. Annex D titled “Diagnosis of rolling element bearings” contains proposed alarm levels for acceleration measurements from 10 Hz – 10 kHz, both in pk and rms values. The relationship of these values (very much like Crest – factor, see 4.2) is the basis of the alarm levels. These values are not recommended to be used for protection. Note that in order to get reliable measurements up to 10 kHz the accelerometer must be stud mounted. This standard may be accessed at http://edms.pw.ge.com/trslibrary/

Crest factor

Crest factor is defined as the ratio of zero-to-peak amplitude of a waveform to the rms value of the same waveform (see following formula ). It can be described as a measure of the "spikiness" of the waveform. The rms value of a purely sinusoidal waveform is \( A/\sqrt{2} \), so the crest factor would be \( A/ (A/\sqrt{2}) \), which equals to 1.414. In general, a high crest factor means the waveform contains impact type events. In the context of rolling element bearings, a high crest factor is an indication of a bearing defect. In a gearbox application, it may be an indication of a cracked tooth.

The crest factor measurement has been used to detect impact symptoms in vibration signals for many years. It has been used in Bently Nevada ADAPT.Wind and in S1 Evo for any waveform signal. For REB purposes it is calculated from an acceleration waveform with a frequency setting of \( F_{\text{max}} \sim 50 \times \) using formula:

\[
Cr = \frac{Pk}{Rms}
\]
Based on field experience, and in the absence of OEM recommendations or other engineering values, a crest factor level of 4.0 can be considered as an initial alarm level. It is not recommended to use crest factor for protection, as it is mainly targeted for early warning purposes and typically will decrease as a bearing failure moves into its final stages.

Crest factor can also be calculated from enveloping (demodulation) waveforms, but since the signal processing will remove lower frequency components, the crest factor will be much higher than in an acceleration waveform, and a higher alarm levels should be applied.

**Enveloping (Demodulation)**

Enveloping is an early indication method and as such should not be used for protection. Bently Nevada does not promote any Enveloping alarm levels. Other vendors publish Enveloping alarm levels, but due to different filter settings, peak detection techniques and other signal processing related issues those cannot be directly applied.

In general, Enveloping alarm levels are normalized by shaft speed and diameter. This means that the larger the bearing diameter and the higher the speed, the higher the alarm level.

The enveloped peak-value usually gives a very early indication of bearing problems. On the other hand, the enveloped rms-value, sometimes referred as carpet value, has proven to be a good indicator of a bearing lubrication problem. If the bearing lubrication was perfect, there would be no metal-to-metal contact within a bearing and the enveloped rms-value would be close to zero. Because there is always some level of metal-to-metal contact, all bearings will have a non-zero enveloped rms-value. An increase of the enveloped rms-value for a particular bearing might indicate under-lubrication, improper lubrication, or even increased bearing loading due to misalignment of the rotor or an unbalance. For both peak and rms values to be useful during trending, a baseline must be taken when the bearing is known to operate correctly.

For a better understanding of the enveloping process, refer to Bently Nevada Basic Vibration Analysis training, Document 176105, Signal Processing.
4. Vibration Settings for Bently Nevada Monitors in REB Applications

This section gives guidelines for optimizing hardware settings to collect the best possible data for REB. It must be noted that other monitoring and protection priorities may prevent these settings from being implemented in full.

4.1 3500 Monitor

If the preferred sensor for REB, an accelerometer, is installed, it is recommended that a channel pair is reserved for a single input by enabling “Take Input from Channel 1 Transducer” in the Channel 2 or 4 Transducer Selection option. One channel will measure integrated Velocity, thus monitoring rotor related issues and REB failures at stage 3 and 4. The other channel measures the acceleration pk value, thus giving an earlier warning of bearing failure.

If a Velomitor is installed, then only a single channel is needed. Fmax should be set to 50X and pk – units should be used.

4.2 3701 Monitor

The 3701 monitor processes all the data needed for good REB condition monitoring from a single accelerometer: velocity, acceleration and acceleration enveloping. All measurements derived from waveforms and spectrums can be individually configured using suitable HP and LP filters and spectral lines. Recommended settings are:

- Velocity spectrum with Fmax 50X
- Acceleration waveform Fmax 50X (primary)
- Acceleration waveform Fmax 10 kHz (secondary) (use a stud mounted accelerometer)
- Enveloping spectrum

Note that 3701 processing power is limited, so not necessarily all recommended measurements can be assigned to all channels. In that case, a secondary acceleration waveform is the first one that should be removed.

4.3 2300 Monitor

The 2300 monitor provides all the data from a single accelerometer that is needed for good REB condition monitoring: velocity and acceleration trended values, and acceleration and demodulated (enveloping) waveforms.

The 2300 monitor comes with five (5) predefined static variables:

- Acceleration pk
- Acceleration rms
- Acceleration derived pk (rms x 1.41)
These five (5) predefined variables all have the same HP and LP filters. For bearing monitoring purposes the LP filter should be set to 10 kHz.

The user can also define two (2) additional independent bandpass variables for each input channel with independently configurable HP and LP filter settings. Each additional variable can be in acceleration or velocity units.

**Recommended settings for dynamic data are:**

- Velocity waveform with Fmax 50X
- Acceleration waveform Fmax 50X
- Demodulation waveform

Note that the 2300 monitor does not calculate any demodulation-based static variables: all these variables need to be calculated in S1 Evo, where the alarming also takes place.

### 4.4 1900/65A Monitor

The 1900/65A monitor allows three (3) independent static variables to be defined for each accelerometer sensor. The recommended settings for these variables are:

**Table 4 - 1: 1900/65A Settings for REB**

<table>
<thead>
<tr>
<th>Variables</th>
<th>Quantity</th>
<th>Detection</th>
<th>Filter settings</th>
<th>Alarms</th>
</tr>
</thead>
<tbody>
<tr>
<td>#1</td>
<td>Velocity</td>
<td>Pk or RMS</td>
<td>0, 5X – 50X</td>
<td>OEM or ISO</td>
</tr>
<tr>
<td>#2</td>
<td>Acc Enveloping</td>
<td>Pk</td>
<td>Enveloping hp filter 1000, Enveloped Variable Filter to cover 3 x BPF</td>
<td>See 4.3</td>
</tr>
<tr>
<td>#3</td>
<td>Acceleration</td>
<td>Pk</td>
<td>Fmax 10 kHz</td>
<td></td>
</tr>
</tbody>
</table>

If only one variable can be specified for REB monitoring, then either #2 or #3 should be selected depending user preference.

### 4.5 VbOnlinePro Monitor

The VbOnlinePro monitor provides all the data from a single accelerometer that is needed for good REB condition monitoring: velocity, acceleration and acceleration enveloping. All measurements derived from waveforms and spectrums can be individually configured using suitable HP and LP filters and spectral lines. Recommended settings are:

- Velocity spectrum with Fmax 50X
- Acceleration waveform Fmax 50X
- Enveloping waveform
If the Fmax setting is above 10 kHz, the performance of VbOnlinePro monitor will be affected (see product user guide, document 115M4367 for details).

### 4.6 Trendmaster Pro Dynamic Scanning Module (DSM) TIM-line

The recommended transducer is the Bently Nevada 200157 accelerometer, and the corresponding TIM-module part number is 200200-06-06. This combination provides the ability to monitor both velocity and acceleration enveloping data. These settings are recommended:

- **Transducer Tab**: pk detection. This setting enables both rms and pk static values.
- **Waveforms Tab**: Enable all return waveforms, both Velocity waveforms and Acceleration Enveloping. Set Frequency Span on waveform 1 to 50X and on waveform 2 to at least 5 kHz. The enveloping filter is fixed, and the frequency span should be high enough to cover 3 x BPFI.
- **Variables Tab**: Enable all that can be used; some require the Keyphasor sensor to become active.
- **Filters Tab**: Enable all filtered variables and set the frequency span according to the machine monitoring needs.

### 4.7 RangerPro (Wireless)

The configuration of RangerPro sensors is done with RangerPro software. The recommended settings are:

- **Static values**:
  - Acceleration HP 5 Hz, LP 5 kHz on X and Y directions, 10 kHz on Z direction.
  - Velocity Hp 5 Hz, LP 1000 Hz on machine RPM < 1800, 2000 Hz on machines with RPM > 1800 rpm.
  - Peak Demod enabled
  - Temperature enabled.
  - Pk and RMS selection based on detection needs.

- **Dynamic waveform settings**:
  - Acceleration X and Y directions 5 kHz, 10 kHz on Z direction.
  - PeakDemod waveform enabled
  - Waveform point count selected based on resolution and data transfer speed requirements.

Note that System 1 Evo allows additional spectral band and waveform static value calculations from the dynamic data.
4.8 SCOUT Portable Data Collector (PDC)

The SCOUT PDC provides the user all required REB condition monitoring data from a single accelerometer: velocity, acceleration and acceleration enveloping (demodulation). This data can be collected in series, or in parallel using 6Pack functionality to reduce collection time. When System 1 Evo Quick Configuration is used, all the recommended settings are implemented automatically. Alternatively, all dynamic data can be configured individually using suitable HP and LP filters and spectral lines to provide all desired spectral and waveform variables. Recommended settings are:

<table>
<thead>
<tr>
<th>Waveform/Spectrum</th>
<th>Detection</th>
<th>Filter settings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Velocity spectrum</td>
<td>Pk or RMS</td>
<td>Fmax = 4 x BPFI x rpm</td>
</tr>
<tr>
<td>Acc Enveloping waveform</td>
<td>Pk</td>
<td>Case dependent, usually enveloping filter 1 – 10 kHz (stud mounted), Enveloping spectrum to cover 4 x BPFI</td>
</tr>
<tr>
<td>Acceleration waveform</td>
<td>Pk</td>
<td>Fmax = 4 x BPFI x rpm</td>
</tr>
</tbody>
</table>

Considerations for specific machine types:

- If the machine is slow and big (as in food plants, pulp & paper, mining, etc.), then the demodulation filter should be lower because the local natural frequencies are lower. Use a 0.5-10 kHz or even a 0.25-10 kHz enveloping filter.
- If the machine has a gearbox it is desirable to see the effect of the gear mesh frequency in the enveloped spectrum. In this case, the minimum frequency of the demodulation filter should be increased (especially for small machines).

Data collection interval

An important part of establishing a PDC route is defining how often the data is taken. The more often the data is collected, the more likely an early bearing failure detection will be made. But as a consequence, the cost of the condition monitoring will be higher due to more frequent data collection and analysis. There are no industry standards to guide this process. The route creation and timing depend on the machine operating hours, running speed, load, process and environmental conditions, and so on. The following general guidelines may be useful in establishing the initial PDC route:

- A Four (4) week interval is generally recommended for the initial period
- If the interval is longer than eight (8) weeks, advanced bearing failure predictions may no longer be feasible
- If the interval ends up being less than two (2) weeks, an online system may be justified

Spared machines should also be monitored. Sometimes this requires that machines are started just for data collection purposes. In that case the machine needs to run long enough so that normal operating conditions and temperatures are reached before data collection is done.
5. Monitoring Slow Speed Machines

In slow shaft speed situations (below 100 rpm), monitoring rolling element bearings becomes challenging. Low energy levels and a relatively long time between bearing failure-based impacts will drive two changes: Velocity data becomes less important, and acceleration enveloping becomes more important. Although the acceleration waveform is still important, it must be sufficiently long and the sampling frequency high enough to assure that the impacting is not filtered away.

The slow speed also effects how the analysis is performed: Less value is gained from the spectral data, and more emphasis is placed on waveforms and static value trends. When dealing with very slow speed machines (10 rpm and below), it is likely that acceleration enveloping waveform and trending of enveloping pk and rms values are the most important tools.
6. Conclusion

This document provides key information regarding rolling element bearings - their construction and failure modes, and the methods, equipment, and settings used to detect or avoid those failures in typical industrial applications. Although much of it applies universally to all REB applications, there are some unique applications (aeroderivative gas turbines, wind turbine generators, other) that are beyond the scope of detailed treatment in this document. In those cases, or for clarification or further information regarding anything herein, please contact a Bently Nevada Application & Solution Architect (ASA) or Machinery Diagnostic Services (MDS) engineer.

6.1 References


6. FAG Publ. No. WL82102/2EA, “Rolling Bearing Damage Recognition of damage and bearing inspection”

