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Rolling Element Bearing Analysis

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ABSTRACT

All rotating machinery uses bearings to support the load and maintain the clearances between stationary and rotating machinery elements. More than 90% of these machines have rolling element bearings. Unfortunately, rolling element bearings are prone to a myriad of premature failures. A mere 10% of rolling element bearings reach their L10 life, the expected life of 90% of similar bearings under similar operating conditions. Early failures are attributed to lubrication, load and design/application errors, and even pre-existing problems that were not detected during manufacture. A comprehensive condition-based maintenance program incorporating preventative maintenance and predictive maintenance should be in place to detect the onset of wear and deterioration of rolling element bearings. A mature program provides not only indications of wear in the bearings, but also an evaluation of the severity and recommendations for when corrective actions should be taken. The purpose of this paper is to briefly discuss how the high-frequency natural bearing resonance indicator, discrete frequency indicators (acceleration), acceleration time-waveform characteristics, acceleration time-waveform crest factor, and the velocity amplitude of bearing fault frequencies associated with rolling element bearings can be integrated to determine bearing health and the risk of catastrophic failure.

KEYWORDS: fundamental bearing faults, high-frequency natural bearing resonance indicator, discrete frequency indicators, time-waveform analysis, fast fourier transform.

Introduction

Rolling element bearings are used in more than 90% of the rotating machinery found in commercial and industrial applications. Regardless of the application, bearings are categorized according to size, speed and lubrication technique. It is not uncommon for the design load at the contact point to be over 3447 MPa (500 000 psi). Factors affecting contact pressure once a bearing is placed in service include lubrication issues, wear of the rollers and/or raceways (production of raised surfaces that reduce the design contact area), dynamic loading and temperature. The purpose of this paper is to introduce the multiple technologies and proprietary measurements used to determine the condition of rolling element bearings.

Failure of rolling element bearings often appears to be random in nature. Yet approximately 43% of these failures are attributed to lubrication. When the smooth rolling motion of the rollers along the raceways is degraded, the rollers will skid momentarily. Skidding is associated as a friction problem, and is normally monitored by high-frequency natural bearing resonance indicators (HFNBRIs). HFNBRIs range between 3 and 50 kHz. These resonances are in both the sonic (<20 kHz) and ultrasonic (>20 kHz) ranges. Shock and/or friction will excite these frequencies; therefore, HFNBRI detection is an effective tool for monitoring the earliest signs of bearing wear (Archambault, 2009).

Discrete frequency indicators are used to identify what is causing the HFNBRI to become excited. Often, the normal fast fourier transform (FFT) velocity spectrum will not indicate a bearing problem until later stages of wear. However, an acceleration spectrum (measured as units of gravity in g where 1 g = 9.8 m/s^2), which is divided into several

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bands, can effectively trend vibration from different sources. One band (the first bearing band) is used to watch fundamental bearing fault frequencies. Another band (the second bearing band) is used to watch harmonics of the fundamental bearing fault frequencies. Monitoring the increase in each band allows the analyst to determine the stage of bearing wear and rate of progression (Berry and Robinson, 2001).

Confirmation of bearing condition established with HFNBRI and discrete frequency indicators are accomplished with velocity FFT frequency analysis (measured in mm/s) and time-waveform characteristics. Spectral analysis includes harmonic families, their source and amplitude pattern. Time-waveform analysis includes pattern recognition, amplitude and crest factor. Crest factor is a unitless ratio derived from taking the maximum peak magnitude (positive or negative) and dividing it by the root-mean-square (RMS) value for the waveform. This provides a ratio of the impacting (that is, pulse-like transients) divided by the equivalent direct current power level of the same time-waveform. Early in the wear cycle, the crest factor may exceed 5 and will approach a value of 2.5 late in the wear cycle.

Bearing Fundamental Frequencies

The frequencies that rolling element bearings generate when rollers pass over a surface anomaly on either the roller or the raceway are called fundamental fault frequencies. These frequencies are a function of the bearing geometry (that is, pitch diameter and roller diameter) and the relative speed between the two raceways. When bearing geometry (Figure 1) is known, the fundamental fault frequencies (in Hz) can be calculated using Equation 1:

$$\text{(1)}\quad \begin{aligned}
\text{BPFI} &= \frac{N}{2}\times F\times\left(1+\frac{B}{P}\times\cos\Theta\right)\\
\text{BPFO} &= \frac{N}{2}\times F\times\left(1-\frac{B}{P}\times\cos\Theta\right)\\
\text{FTF} &= \frac{F}{2}\times\left(1-\frac{B}{P}\times\cos\Theta\right)\\
\text{BSF} &= \frac{P}{2B}\times F\times\left[1-\left(\frac{B}{P}\times\cos\Theta\right)^2\right]
\end{aligned}$$

where

BPFI = ball pass frequency inner race (Hz),
BPFO = ball pass frequency outer race (Hz),
FTF = fundamental train frequency (Hz),
BPF = ball pass frequency (Hz),
N = number of balls,
F = shaft frequency (Hz),
B = ball diameter (mm),
P = pitch diameter (mm),
Θ = contact angle.

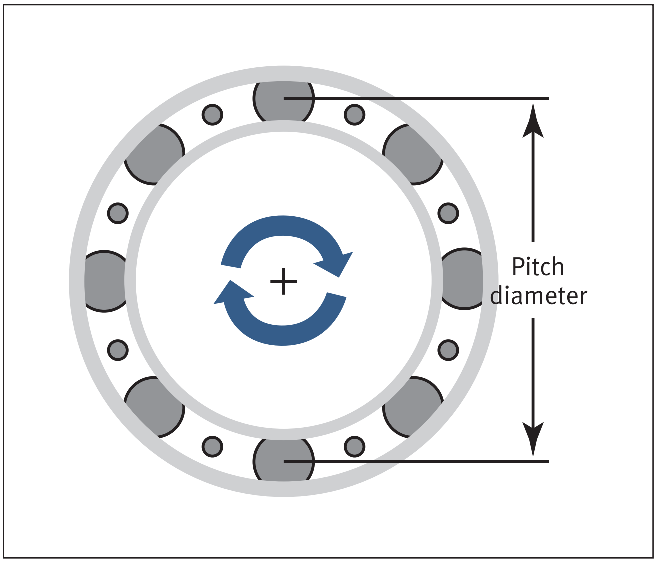

Figure 1. Overhead of bearing showing pitch diameter.

The value of FTF with respect to shaft rotational speed usually falls between 0.33 and 0.50 of the rotational speed, with a majority of bearings falling between 0.38 and 0.42. For this reason, fault frequencies may be estimated when bearing geometry is not known, as follows: BPFO is approximately $0.4 \times N \times F$; BPFI is approximately $0.6 \times N \times F$; FTF is approximately $0.4 \times F$; and BSF is approximately $0.2 \times N \times F$. The following relationships also apply: BPFO = FTF $\times N \times F$; BPFI = (1 – FTF) $\times N \times F$; and (BPFO + BPFI)/F = number of rollers.

High-frequency Natural Bearing Resonance Indicators

Fundamental bearing frequencies and their harmonics are generally observed in the velocity spectrum. In addition to those frequencies, there are high-frequency natural bearing resonances between 3 and 50 kHz. Testing was performed involving the vibratory response of a rolling element bearing, which was mounted on a high-speed spindle. The first stage of testing involved two-channel impact testing to identify the bearing's natural frequencies. The second stage comprised running the spindle through a broad sweep of operating speeds. As expected, rotational speed of the spindle excited the natural frequencies noted in the first stage. Also of note was indication that early bearing wear caused by impacting from wear, and friction caused by lubrication problems, excited the same natural frequencies.

Realizing the benefit of HFNBRIs, many companies have developed their own proprietary measurements for detection of sonic and/or ultrasonic frequencies that produced a value (magnitude) which was a summation of the energy in the frequency band of interest. The measurements of HFNBRI include: spike energy; shock pulse at a resonant frequency of 32 kHz (measured in dBsv); high frequency (>5 kHz) detection band (measured as peak *g*); high frequency (1 to 20 kHz) band (measured in *g* RMS); general purpose ultrasonics in the frequency range of 20 to 50 kHz (measured in dB); spectral emitted energy between 250 and 350 kHz (measured in dB); and acoustic emission measurements in the frequency range of 20 kHz to 1 MHz (measured in dB).

Discrete Frequency Indicators

Bearing frequencies less than 10 kHz may be present in predictive maintenance (PdM) route data (velocity spectrum). Typically, route data is collected with a high pass filter of 2 Hz, a low pass filter of 1 to 2 kHz, and standard sampling (to prevent aliasing). Such data collection techniques will pick up bearing wear later in their deterioration cycle, but not in their initial progression.

Alternate signal processing techniques are needed in order to observe bearing frequencies early in their wear cycle. Discrete frequency indicators use a high pass filter of at least 500 Hz with a high rate of sampling and high-frequency low pass filters to obtain the short time durations of shock measurements. Many of these measurements utilize full wave rectification of the time waveform, then process an FFT for analysis. This will help give a root cause as to what the initial bearing problem may be.

Similar to HFNBRIs, the same companies and others developed additional proprietary measurements which provide discrete frequency indicators (that is, spectral data) for detection of sonic and/or ultrasonic frequencies, including: spike energy; shock pulse; spectral emitted energy; cepstral analysis (measured in *g* dB); and bearing demodulation (measured in *g*).

If using a sonic HFNBRI and discrete frequency indicator, special care should be taken with magnetically mounted accelerometers for route based PdM programs, as their frequency response ranges will be limited due to the change in stiffness (K) and added mass (M) (Robinson, 2009).

Four Stages of Bearing Wear

The most common configuration for rotating machinery is horizontally mounted (that is, axis of rotation perpendicular to gravity) with a rotating inner race. Given such a configuration, certain factors lead to a higher or lower probability for wear of the various bearing components.

The outer raceway has the highest probability for wear and is usually the first discrete bearing frequency detected. The outer race is stationary with a stationary load zone. Transmission of high frequencies generated by lubrication and/or wear is very high for outer race frequencies. The load zone is relatively small with respect to the pitch diameter. The progression of wear allows for trending.

The inner raceway experiences wear at a faster rate than the outer raceway. The fundamental bearing frequency for an inner race is greater than that of an outer race, meaning that for one revolution of the shaft, more damage producing events will occur for an inner race fault. The main reason why inner race problems are detected less frequently than outer race problems is because transmission of high frequencies generated by lubrication and/or wear is attenuated greatly through multiple transitions and lubrication boundaries. The poor transmission of vibratory energy results in more severe wear by the time a problem is detected. Therefore, trending is limited for inner raceway wear.

The rolling elements and cage are typically the last components to fail though roller spin frequencies, and are often observed in non-ball bearing configurations.

The indications associated with the four stages of bearing wear are general and may not be all-inclusive for each bearing wear event. The stages are provided as follows as a point of reference used in the evaluation of bearing severity for an estimate of wear in lieu of disassembly and inspection (Woodward, 1995).

Stage 1

HFNBRI is excited with possible fault frequencies (harmonics) in the second bearing band. Since BPFO is easily detectable, a discrete frequency indicator should be used to identify the fault problem progression.

When bearings are in Stage 1 of deterioration, the typical recommendation is to continue to monitor for deterioration at the normal monitoring interval. The risk assessment for catastrophic failure is low, and the assigned risk indicator (ARI) is typically less than two.

Stage 2

When bearing wear progresses to Stage 2, the damage to the raceways and/or rollers grows from having microscopic size to being visible to the naked eye. Stage 2 wear will result in a higher level HFNBRI amplitude. Stage 2 wear will also be accompanied by rising quantity and amplitude of fault frequency harmonics. The fundamental outer race fault frequency (BPFO) may appear in the first bearing band. Also, harmonics of the fundamental inner race fault frequency (BPFI) may be seen in the second bearing band. The discrete frequency indicator should have increasing amplitudes, and multiple bearing fault frequencies should be present. Harmonics of bearing fault frequencies may be present in a velocity spectrum. Side banding around fault frequencies may occur in the second bearing band if modulation of time-waveform is present.

When bearings are in Stage 2 of deterioration, the typical recommendation is to continue with scheduled repairs at the next normal preventative maintenance interval. The risk assessment for catastrophic failure is moderate (ARI is typically two to three).

Stage 3

When bearing wear progresses to Stage 3, the damage to the raceways and/or rollers grows substantially in all dimensions, as well as in multiple locations. Stage 3 wear will result in even higher levels of HFNBRI amplitude. Outer race fault frequency and harmonics will propagate to both first and second bearing band with side banding. Inner race fault frequency harmonics will now be present distinctly in the second bearing band with side bands. Possible ball spin frequencies (BSF) may appear in the first bearing band. The discrete frequency indicator should have increasing amplitudes along with multiple fault frequencies, and harmonics of bearing fault frequencies should be easily identified in a velocity spectrum.

When bearings are in Stage 3 of deterioration, the typical recommendation is to schedule repair in the next 30 to 45 days. The risk assessment for catastrophic failure is high (ARI is typically three to five).

Stage 4

At Stage 4, the change in bearing geometry with increased wear produces a drop in HFNBRI amplitude. Random vibration will increase, generating a raised broad noise floor that may mask low amplitude bearing fault frequencies. Looseness will appear at one, two and three times revolutions per minute (RPM). At this time, the overall spectral (velocity) levels will exceed alarm. If the discrete frequency indicator shows the presence of cage frequency (FTF), the bearing is very near the end of life. HFNBRI may dramatically increase and the amplitude of vibration at turning speed may become excessive due to out of specification clearances.

When bearings are in Stage 4 of deterioration, the typical recommendation is to schedule immediate repairs. The risk assessment for catastrophic failure is very high (ARI is typically equal to or greater than five).

Estimating Bearing Wear Stage

Stages of bearing wear can be estimated comparing available data as applicable to Tables 1–3. Assigning an approximate bearing wear stage can be accomplished with evaluation of HFNBRI values and Table 1. To do so, it is necessary to first determine which column to locate HFNBRI values, based

TABLE 1
Assigning a bearing wear stage for high-frequency natural bearing resonance indicator (HFNBRI) amplitude versus revolutions per minute (RPM)

RPM	High frequency band (*g*'s root-mean-square)	HFNBRI High-frequency detection (*g*'s peak)	Spike energy (*g*'s spike energy)	Shock pulse (dB shock value)
50	0.225/0.45/0.90/1.8	0.21/0.42/0.84/1.7	0.075/0.15/0.3/0.6	16/22/28/34
100	0.335/0.67/1.34/2.7	0.32/0.64/1.3/2.6	0.115/0.23/0.46/0.92	21/27/33/39
300	0.45/0.9/1.8/3.6	0.425/0.85/1.7/3.4	0.15/0.3/0.6/1.2	27/33/39/45
500	0.5/1.0/2.0/4.0	0.475/0.95/1.89/3.8	0.17/0.34/0.68/1.36	32/38/44/50
600	0.95/1.9/3.8/7.6	0.9/1.8/3.7/7.4	0.325/0.65/1.3/2.6	34/40/46/52
900	1.1/2.2/4.4/8.8	1.05/2.1/4.2/8.4	0.375/0.75/1.5/3.0	36/42/48/54
1200	1.3/2.6/5.2/10.4	1.2/2.4/4.8/9.6	0.44/0.87/1.7/3.4	39/45/51/57
1800	1.5/3.0/6.0/12.0	1.4/2.8/5.6/11.2	0.5/1.0/2.0/4.0	42/48/54/60
3600	1.75/3.5/7.0/14.0	1.6/3.2/6.5/13.0	0.6/1.2/2.4/4.8	50/56/62/68
7200	3.0/6.0/12.0/24.0	2.8/5.6/11.3/22.6	1.0/2.0/4.0/8.0	52/58/64/70

*Four stages of bearing wear 1/2/3/4

TABLE 2
Bearing wear stage for single highest bearing fault frequency harmonic amplitude (mm/s peak) versus shaft rotational speed

Revolutions per minute	Raceway wear (outer race or inner race) Stage 1	Stage 2	Stage 3	Stage 4
50	0.254	0.508	1.016	2.032
100	0.381	0.762	1.524	3.048
200	0.508	1.016	2.032	4.064
300	0.572	1.143	2.286	4.572
450	0.635	1.270	2.540	5.080
600	0.762	1.524	3.048	6.096
900	0.826	1.651	3.429	6.858
1200	0.889	1.778	3.556	7.112
1800	1.016	2.032	4.064	8.128
3600	1.270	2.540	5.080	10.160
4000	1.524	3.048	6.096	12.192

TABLE 3
Bearing wear stage for maximum time-waveform peak-to-peak magnitude versus shaft rotational speed

Revolutions per minute	Time waveform (g's peak-to-peak) Stage 1	Stage 2	Stage 3	Stage 4
50	0.16	0.32	0.64	1.28
100	0.26	0.52	1.02	2.04
200	0.50	1.00	2.00	4.00
300	0.75	1.50	3.00	6.00
450	0.88	1.75	3.50	7.00
600	1.00	2.00	4.00	8.00
900	1.70	3.40	6.90	13.60
1200	2.40	4.80	9.60	19.20
1800	4.10	8.20	16.40	32.80
3600	10.20	20.40	40.80	81.60
7200	20.00	40.00	80.00	160.00

upon the vendor specific method of measurement. Next, one should select an appropriate shaft RPM. Finally, one needs to compare the measured HFNBRI value versus the values shown for Stages 1–4.

Assigning an approximate bearing wear stage can be accomplished with evaluation of the single largest bearing spectral fault frequency harmonic (velocity), and comparison to the values assigned in Table 2 (Woodward, 1995). First, it is necessary to select an appropriate shaft RPM, and then compare the value of the single largest bearing fault frequency harmonic versus the values shown for Stages 1–4.

Assigning an approximate bearing wear stage can be accomplished with evaluation of the maximum peak-to-peak value in the acceleration time-waveform, and comparison to the values assigned in Table 3. As before, one needs to select an appropriate shaft RPM and then compare the value of peak-to-peak time-waveform versus the values shown for Stages 1–4.

The highest wear stage determined will be designated as the bearing indicator, when calculating a final ARI.

Dynamic Force Versus Bearing Life

As previously mentioned, increasing either speed or load decreases bearing life. The following equation calculates the bearing life in operating hours:

$$H = \left(\frac{C}{L}\right)^3 \times \frac{16667}{\text{RPM}} \tag{2}$$

***note: the exponential value of 3 is used for ball bearings; a value of 10/3 is used for all other roller bearings.**

where
H = hours,
C = rated capacity (kN),
L = dynamic load (kN),
RPM = machine speed (revolutions per minute).

Excessive vibration also contributes to the dynamic load the machine experiences during operation. Vibration increases the load with the following equation:

$$H = \left(\frac{C}{L + 4.14016 \times 10^{-6}\, MVF}\right)^3 \times \frac{16667}{\text{RPM}} \tag{3}$$

***note: the exponential value of 3 is used for ball bearings; a value of 10/3 is used for all other roller bearings.**

where
H = hours,
C = rated capacity (kN),
L = dynamic load (kN),
RPM = machine speed (revolutions per minute),
M = mass (kg),
V = velocity (mm/s),
F = frequency (Hz).

Machines with imbalance or misalignment will have increased loads, thus reducing the life of the bearing (Graney, 2008). Dynamic load calculations should use 80–90% of overall vibration velocity alarm for turning speed vibration and 50–60% of overall vibration velocity alarm for two to three times turning speed vibration. *ISO 10816-1:1995* provides guidelines for overall vibration levels (ISO, 1995).

Review of Equation 3 for bearing life shows that bearing lifetime is proportional to L^{-3} and is also proportional to the reciprocal of speed, RPM^{-1}. Therefore, it is possible to evaluate the risk of failure with respect to increased speed and/or load. Doubling only load will reduce the expected life to 12.5%, whereas doubling only speed reduces the expected life to 50%. Increasing the speed above the original design is normally accomplished with an engineering process change. Load may be increased above the original design in the same manner. However, the dynamic load may increase due to excess vibration (inertia loading). Velocity vibration amplitude is used to assign a dynamic force indicator, which will be used to calculate a risk assessment value.

Assigning Risk Indicator to a Rolling Element Bearing

To assign risk to a rolling element bearing, it is necessary to first evaluate the velocity spectrum for conditions that would contribute to an increase in dynamic loading (that is, imbalance, misalignment and/or looseness). Then, a dynamic force indicator, DFI, is assigned as appropriate for the actual vibration characteristics with respect to the lower alarm level (often referred to as an alert value):

- DFI = 1 for vibration less than the lower alarm value.
- DFI = 2 for vibration approximately equal to the lower alarm value.
- DFI = 3 for vibration of approximately 1.5 times the lower alarm value.
- DFI = 4 for vibration in excess of 2 times the lower alarm value.

HFNBRI, discrete frequency indicators, time-waveform characteristics, velocity spectrum and crest factor are then evaluated to obtain the worst case stage of bearing wear. A bearing indicator, *B*, which corresponds to the stage of wear, is then assigned (Graney, 1998).

The ARI can then be calculated.

$$\text{ARI} = \sqrt{\text{DFI}^2 + B^2} \quad (4)$$

Note that the ARI may be further multiplied to an asset criticality number for a more detailed risk factor.

An ARI less than two indicates a low risk of catastrophic failure; an ARI of two to three indicates a moderate risk of catastrophic failure; an ARI of three to four indicates a high risk of catastrophic failure; and an ARI greater than four indicates a very high risk of catastrophic failure.

Example 1: Ballast and Scrubber Pump

In the case of a ballast and scrubber pump, operating conditions were as follows: turning speed was 1760 RPM; type 6420 rolling element bearings were installed with overall velocity alarms of 7.62 mm/s peak (alert) and 11.43 mm/s peak (alarm). The high-frequency band measured 58.21 m/s^2 (5.94 *g*) RMS when spectral data was collected. Figure 2 shows the harmonics of the outer race fault frequency (3.28 × turning speed). Overall vibration level was 20.32 mm/s peak.

Stage of Bearing Wear Identification for Calculation of Assigned Risk Indicator Value

The first step of calculating an ARI requires assessment of bearing wear. Table 1 shows that a shaft speed of 1800 RPM and an HFNBRI of 58.21 m/s^2 (5.94 *g*) RMS (for high-frequency band), indicates bearing wear Stage 2. Review of the spectral data in Figure 2 has the highest fault frequency harmonic at 6.604 mm/s peak. Using Table 2 for an 1800 RPM shaft speed, the vibration level is greater than the value for Stage 3, and less than the value for Stage 4, indicating bearing wear Stage 3. Review of the time waveform data in Figure 2 shows an amplitude of time-waveform of 367.3 m/s^2 (37.48 *g*) peak to peak. Using Table 3 for an 1800 RPM shaft speed, the time-waveform value is greater than the value for Stage 4, indicating bearing wear Stage 4.

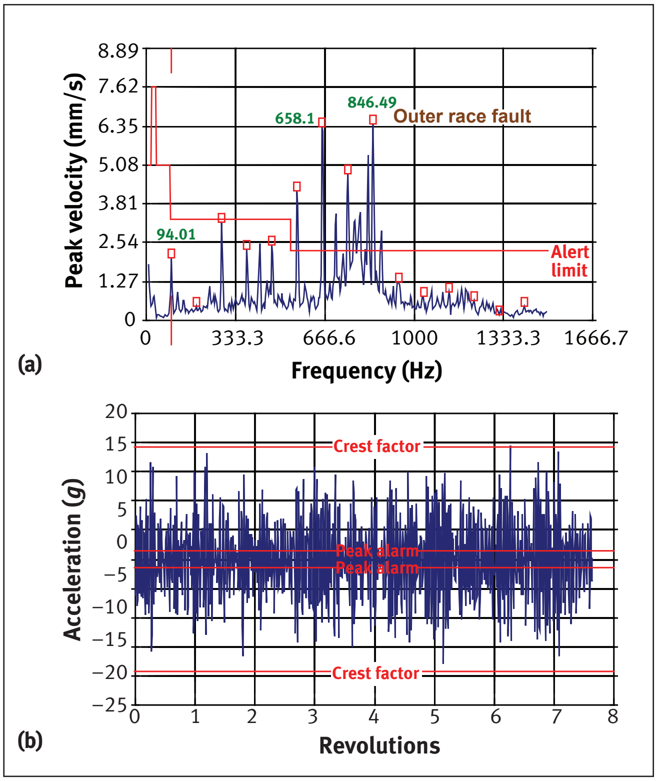

Figure 2. Example 1, information collected from a ballast scrubber pump showing: (a) spectrum; (b) time-waveform.

Diagnosis (Crosscheck)

Though high-frequency indicators point towards bearing wear at a worst condition of Stage 4, it is a good practice to cross-check the diagnosis with spectral conditions. Outer race harmonics with sidebands are present in the first and second bearing bands, indicating at least a Stage 3 condition. Bearing harmonic amplitudes of greater than 4.064 mm/s peak but less than 8.128 mm/s peak indicate a Stage 3 condition. The raised noise floor is an indication of vibration from more random sources such as would be caused by degrading bearing geometry, indicating a Stage 4 condition. Crest factor is low, near 2.5, indicating a Stage 4 condition.

Dynamic Force Indicator Identification for Calculation of Assigned Risk Indicator Value

Vibration at turning speed is low, at <1.27 mm/s peak. Since the vibration caused by turning speed and misalignment is less than the alarm value of 7.62 mm/s peak, the dynamic factor indicator is assigned a value of 1.

Using the worst-case bearing wear stage for bearing indicator (Equation 4), ARI = 4.123. An ARI of 4.123 is considered a very high risk of catastrophic failure. The appropriate recommendation should be to replace bearings immediately (as can be scheduled).

Example 2: Boiler Feed Pump

On a second case, a boiler feed pump, operating conditions were as follows: turning speed 3525 RPM; type 5308 rolling element bearings with overall velocity alarms of 7.62 mm/s peak (alert) and 11.43 mm/s peak (alarm). High-frequency band measured 161.5 m/s^2 (16.5 *g*) RMS when spectral data was collected. Figure 3 has harmonics of the inner race fault frequency (4.805 × turning speed), and Figure 4 has the addition of outer race harmonics (3.195 × turning speed). The overall vibration level was 22.22 mm/s peak.

As previously show in Example 1, the first step of calculating an ARI requires assessment of bearing wear. Table 1 shows that a shaft speed of 3600 RPM and an HFNBRI of 161.5 m/s^2 (16.5 *g*) RMS (for a high-frequency band), indicates bearing wear Stage 4. Review of the spectral data in Figure 3 has the highest fault frequency harmonic at 4.34 mm/s peak. Using Table 2 for a 3600 RPM shaft speed, the vibration level was greater than the value for Stage 2, and less than the value for Stage 3, indicating bearing wear Stage 2.

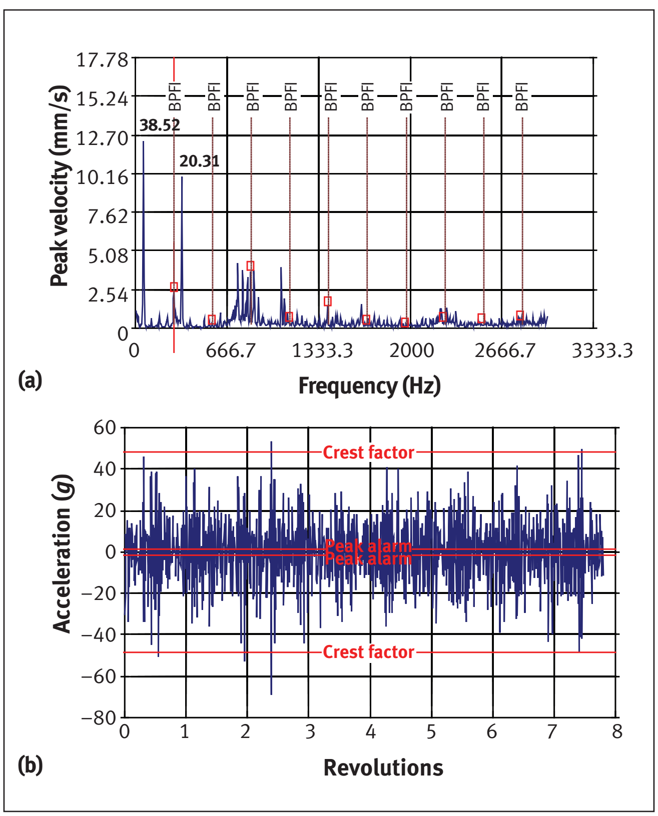

Figure 3. Example 2, information collected from a boiler feed water pump showing: (a) spectrum; (b) time-waveform. Ball pass frequency inner race is indicated as BPFI.

Review of the time waveform data in Figure 3 shows an amplitude of time-waveform of 1185.8 m/s^2 (121 *g*) peak to peak. Using Table 3 for a 3600 RPM shaft speed, the time waveform value is greater than the value for Stage 4, indicating bearing wear Stage 4.

Diagnosis (Crosscheck)

Though high-frequency indicators point towards bearing wear at a worst condition of Stage 4, it is a good practice to crosscheck the diagnosis with spectral conditions. Inner race harmonics with sidebands are present in the first and second bearing bands, indicating at least a Stage 3 condition. Multiple problems are present in Figure 4, including outer race, inner race and FTF, indicating at least a Stage 3 condition. The time-waveform in Figure 3 has high-energy impacting with extremely high levels, and a slight raised noise floor in the spectrum. If the frequencies of interest were related to the outer race, this would indicate a Stage 3 condition. However, inner race vibration transmission will always be degraded; therefore, the indications present are that of bearing wear Stage 4.

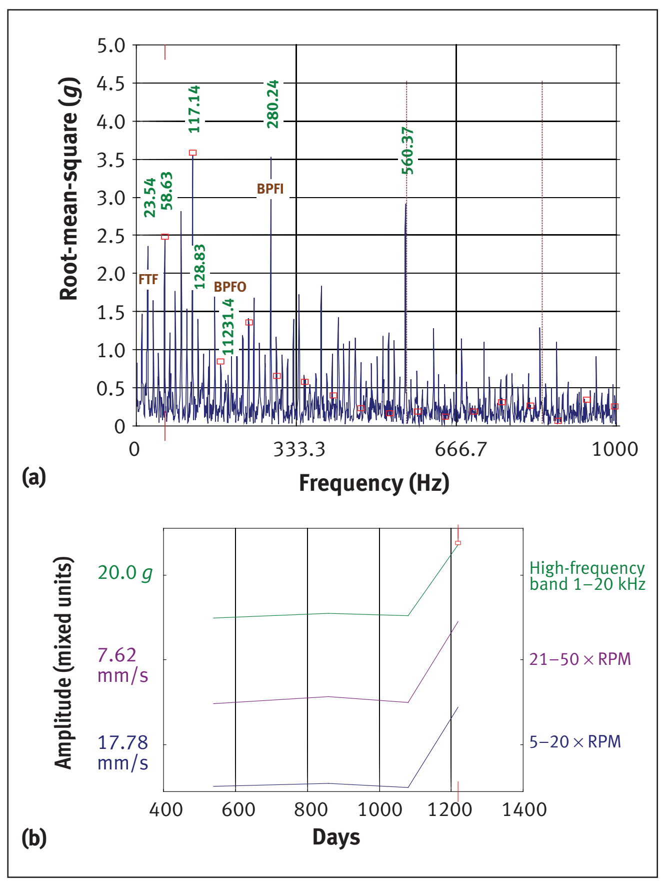

Figure 4. Example 2, information collected from a boiler feed water pump showing: (a) the high-frequency natural bearing resonance indicator; (b) discrete frequency indicator and trends. Fundamental train frequency (FTF), ball pass frequency outer race (BPFO) and ball pass frequency inner race (BPFI) are indicated.

Dynamic Force Indicator Identification for Calculation of Assigned Risk Indicator Value

Vibration at turning speed is high, at 12.34 mm/s peak. Since the vibration caused by turning speed and misalignment is approximately equal to 1.5 times the alarm value of 7.62 mm/s peak, the dynamic factor indicator is assigned a value of 3.

Using the worst-case bearing wear stage for bearing indicator (Equation 4), ARI = 5. An ARI of 5 is considered a very high risk of catastrophic failure. Therefore, the appropriate recommendation should be to replace bearings immediately (as can be scheduled).

Conclusion

Bearing failures occur randomly due to bearing wear and may be accelerated by dynamic forces and lubrication problems. Rolling element bearings require multiple parameters to be monitored and analyzed to effectively assess the bearing condition in order to take corrective actions before failures occur. An effective PdM program can be enhanced through use of an ARI value to determine potential risk.

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