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MONITORING AND DETECTING BALL BEARING FAULTS

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This paper presents a ball bearing fault detection method using frequency domain analysis of vibration signals. Specifically, a method for production screening and an algorithm to detect ball bearing faults are presented which integrates tri-axial accelerometer sensors into a water pump and an air moving device. The detection algorithm analyzes the rotational machine by comparing the vibration data to the predetermined harmonics tables. Ball bearing faults manifest themselves in predetermined frequencies which are caused by ball revolution defects, inner ring dents, or outer ring dents. In this way, rotational faults can be detected in situ and the rotating machinery replaced prior to failure. The overall algorithm incorporates a peak detection algorithm which analyzes Fast Fourier Transform (FFT) processed accelerometer data to count the number of peaks within base and harmonic frequencies. The peak detection algorithm is used to eliminate other known frequencies, which are not related to bearing faults, and background noise. If a particular threshold is exceeded for one of the predetermined fault frequencies, then either a warning or an impending failure of the unit is reported to the system. Analysis is performed real-time and continuously compares incoming data to previously collected data to assure there has been no change in the ball bearing behaviour. In order to validate the algorithm, ball bearings were cross sectioned and analysed using an optical microscope to characterize the bearing wear. The bearings were pulled from rotating machinery with various states of operation, ranging from newly installed to having thousands of hours of operation. The results show that this method can be an effective tool for not only predicting ball bearing failure in the field, but can also be used to screen for ball bearing faults and other faults prior to shipping from manufacturing.

Keywords: ball bearing, fault detection, manufacturing screen.

1. Introduction

Rotating machinery, in the form of fans, blowers, pumps, and compressors provide active cooling to almost all Information Technology (IT) equipment. In order to provide continuous operation of this IT equipment for high reliability applications, sufficient cooling must be supplied nonstop 24/7/365. The requirement for continuous operation and zero downtime can put a lot of stress on the rotating machinery within these complex cooling systems. There are two types of failures that will be discussed here. The first is high vibration that can induce acoustical noise which can create concern for the reliability of the rotating machinery. This type of failure may not induce operating failure for a long time, but will induce higher than normal acoustical noise levels and can lead to a repair action which in turn leads to customer dissatisfaction.

The second is bearing failures that leads to operational failure. The bearings of the rotating machinery tend to be the weakest link in the cooling systems of IT equipment, and in our experience, they can account for up to 95% of all of the cooling system failures. To help prevent an outage due to the failure of a cooling component, an algorithm to predict ball bearing faults was developed.

High-end servers and mainframe computers use ball bearings in their rotating machinery throughout the cooling systems. All of the bearing systems are designed for high reliability and long service life. Even though the bearings are designed for long service life, damage to the bearings can occur during shipping and/or assembly which can cause premature wear out and subsequently early life failures.

This paper investigates the use of 6000z bearings in water pumps and 608Z bearings in a backward-curved centrifugal blower used to cool the main power supplies within high-end servers and mainframes. This blower uses a typical motor design and bearing structure incorporated in both forward and backward-curved centrifugal blowers designs which are used in high-end IT equipment. As many as 24 centrifugal blowers and 3 water pumps using this ball bearing system can be housed in a single mainframe computer. Detecting rotating machinery failure is critical to maintaining continuous operation of this type of high-end IT equipment. The publications on the analysis of bearing fault detection and predictions are listed in the references [1-3].

2. Vibration based production screening

To ensure high reliability in the IT equipment, all rotating assemblies were functionally tested at their normal operating speeds prior to shipment. In addition to the electrical and rotating machinery performance, the vibration data were collected and analyzed. The overall root-mean-square acceleration (Grms) values of the Power Spectral Density (PSD) curve from 10 to 1000 Hz represent the overall vibration of the tested samples. Water pumps, as shown in Figure 1, were installed one at a time in a tester and subjected to production screening.

A Brüel & Kjær triaxial accelerometer model 4524-B Cubic Triaxial Deltatron 100 mv/g was installed on the chassis of each water pump as shown in Figure 1. The nominal operating speed of the pump is 3600 RPM. Consequently, the recording and the spectrum analysis of vibration data were conducted at 3600 RPM using a Vibration Research VR9500.

The histogram of root mean square acceleration values over a month period, period 1, is shown in Figure 2. The mean values of acceleration on 114 pumps at period 1 is 0.5 Grms with standard deviation 0.8 Grms. There are two pumps with acceleration values higher than the mean value plus 3 standard deviations (3.9 Grms). Dismantled, reassembled, and realigned pump assemblies resulted in lower vibration overall. Continued emphasis on improving assembly process brought the overall vibration reduction as shown in Table 1. The histogram of the average root mean square acceleration values for periods 2, 3, and 4 trend downward as shown in Figure 3.

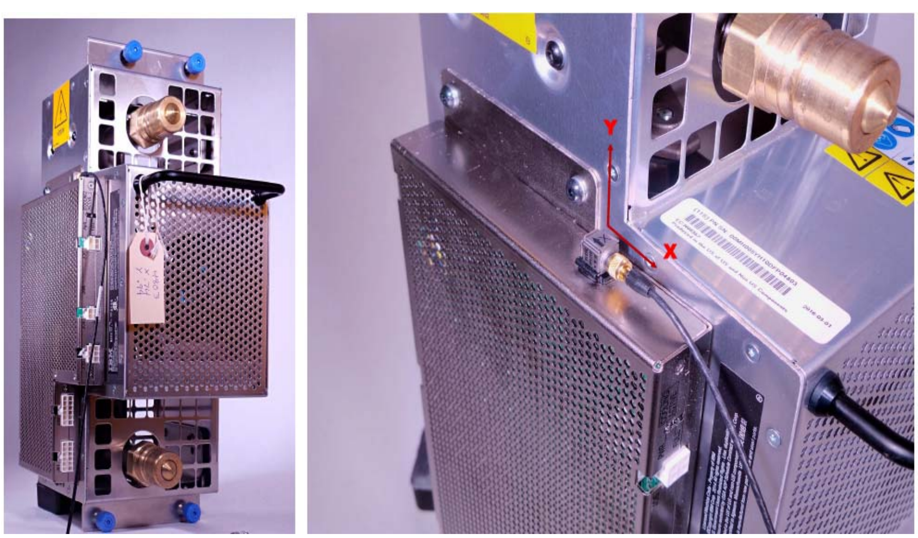

Figure 1: Water pumps.

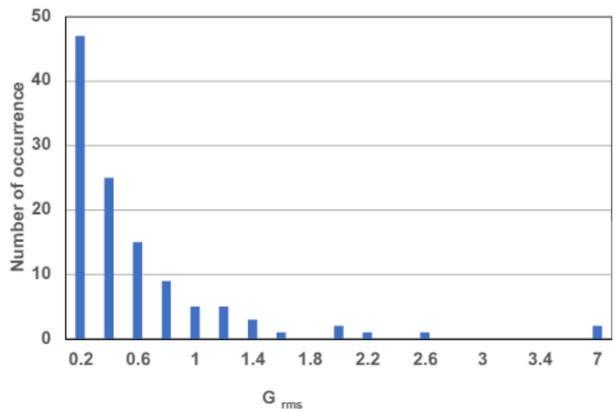

Figure 2: Histogram G_{rms} of pumps in period 1 running at 3600 rpm

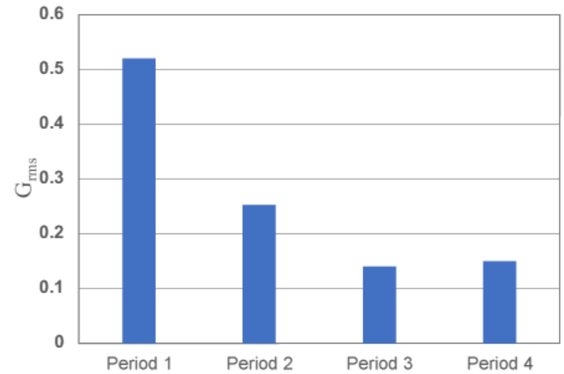

Figure 3: Monthly G_{rms} average

Table 1: Grms mean values of pumps at different periods.

	Mean G_{rms}	Standard Deviation	Mean + 3 Standard Deviation
Period 1	0.5	0.8	2.9
Period 2	0.3	0.3	1.2
Period 3	0.2	0.1	0.5
Period 4	0.2	0.1	0.5

Table 2: Rolling Element bearing faults frequencies [1]

Ball revolution $= fa = \frac{1}{2}\left(1 - \frac{Dw}{Dpw}\cos\alpha\right) fr$ (1)

Retainer rotation $= fr = fa$ (2)

Ball rotation $= fc = \frac{1}{2}\left(\frac{Dpw}{Dw} - \frac{Dw}{Dpw}\ cos^{2}\alpha\right) fr$ (3)

where

fr	= rotational speed (Hz)	Z	= number of balls
Dw	= ball diameter (mm)	Dpw	= pitch cycle diameter (mm)
α	= nominal contact angle (degree)	n	= harmonic n = 1, 2, 3, 4, …..

Vibration caused by inner ring raceway dents or bumps (fe)

Vibration in axial direction = fet = n Z (fr – fa) (4)

Vibration in radial direction = fer = fet ± fr (5)

Vibration caused by outer ring raceway dents or bumps (ff) = n z fa (6)

Vibration caused by ball surface dents or bumps (fg) (7)

Vibration in axial direction = fgt = 2 n fc (8)

Vibration in radial direction = fgr = fgt ± fa (9)

3. Vibration based detection of failures

Not only can the overall vibration magnitude be evaluated, the recorded power spectrum density of pump assemblies can also be used to assess the bearing health. Most bearing faults can be detected prior to catastrophic failure using vibrational analysis. Well known equations for the frequencies and harmonics of the different fault modes are shown in Table 2. For the water pumps running at 3600 rpm, the bearing fault frequencies are shown in Table 3, rows 10 to 17. Columns E to I show the harmonics of bearing fault fundamental frequencies given in column C.

In addition to water pumps, the backward-curved centrifugal blower operating at 1900 rpm was analyzed to detect bearing faults. The centrifugal blowers, shown in Figure 4, have 608Z bearings. The bearing fault frequencies for the 608Z is shown in column C of Table 4. A Brüel & Kjær triaxial accelerometer model 4524-B Cubic Triaxial Deltatron 100 mv/g was installed on the blower chassis as shown in Figure 4. The location of the accelerometer was set as close to the bearing as possible in order to detect vibrations in all three axes. A Vibration Research VR9500 was used to record and analyse the vibration data at the operational speed of 1900 rpm.

Table 3: Water pump bearing faults frequencies at 3600 rpm.

A	B	C	D	E	F	G	H	I
1		**6000z**		**2**	**3**	**4**	**5**	**6**
2	Water pump	3600	RPM	Frequencies				Hz
3	Ball diameter (Dw)	4.8	mm	Inner ring rotational speed (fr)				60.0
4	Number of balls (Z)	7		Ball Revolution (fa)				22.1
5	Nominal contact angle (α)	0	rad	Retainer Rotation (fb)				22.1
6	Outer reference diameter (Lo)	26	mm	Ball Rotation (fc)				105.4
7	Inner reference diameter (Li)	10	mm	Ball Pass (fd)				0.6
8	Pitch circle diameter (Dpw)	18	mm					
9								
10	Inner Ring Raceway Dents (fe)		Hz					
11	Axial (fet)	266	Hz	531	797	1062	1328	1593
12	Radial (fer-)	206	Hz	411	617	822	1028	1233
13	Radial (fer+)	326	Hz	651	977	1302	1628	1953
14	Outer Ring Raceway Dents (ff)	154	Hz	309	463	618	772	927
15	Ball Surface Dents (fg)							
16	Axial (fgt)	211	Hz	422	633	844	1054	1265
17	Radial (fgr-)	189	Hz	378	567	755	944	1133
18	Radial (fgr+)	233	Hz	466	699	932	1165	1398

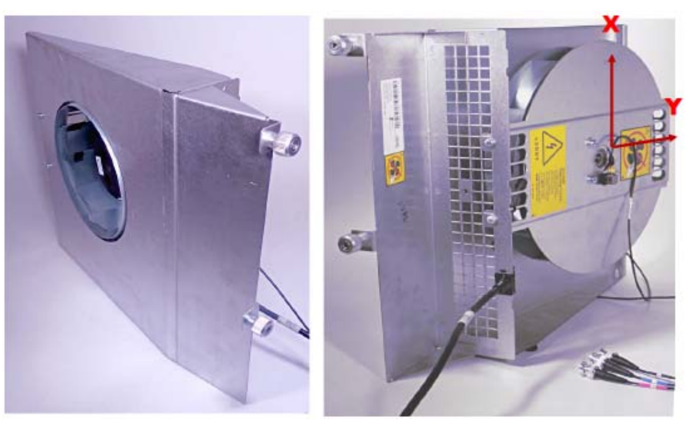

Figure 4: Blower with triaxial accelerometers Vertical (X), Radial (Y) and Axial (Z).

Table 4: Blower bearing faults frequencies at 1900 rpm.

A	B	C	D	E	F	G	H	I
1		6008Z		2	3	4	5	6
2	Blower speed	1900	RPM	Frequencies				Hz
3	Ball diameter (Dw)	4	mm	Inner ring rotational speed (fr)				31.7
4	Number of balls (Z)	7		Ball Revolution (fa)				11.6
5	Nominal contact angle (α)	0	rad	Retainer Rotation (fb)				11.5
6	Outer reference diameter (Lo)	19.1	mm	Ball Rotation (fc)				54.9
7	Inner reference diameter (Li)	10.8	mm	Ball Pass (fd)				0.6
8	Pitch circle diameter (Dpw)	14.95	mm					
9								
10	Inner Ring Raceway Dents (fe)		Hz					
11	Axial (fet)	140	Hz	281	421	562	702	843
12	Radial (fer-)	109	Hz	218	326	435	544	653
13	Radial (fer+)	172	Hz	344	516	689	861	1033
14	Outer Ring Raceway Dents (ff)	81	Hz	162	244	325	406	487
15	Ball Surface Dents (fg)							
16	Axial (fgt)	110	Hz	220	330	440	549	659
17	Radial (fgr-)	98	Hz	197	295	393	491	590
18	Radial (fgr+)	121	Hz	243	364	486	607	729

4. Criteria of bearing failures and prediction algorithm

Bearing faults will induce one or more peaks on Power Spectral Density (PSD) curves. Those peaks will occur at frequencies related to bearing faults frequencies and their harmonic frequencies as shown in Tables 3 and 4. Those peaks will also typically be found in at least two directions, such as the radial and vertical directions. An algorithm was developed to be able to detect bad bearings in situ. The algorithm consists of the following aspects:

1. Collect vibration data in all three directions.
2. Calculate PSD from the vibration data.
3. Find the peak frequencies: A peak frequency is defined as a frequency which is at least 1.75 times the average of a 10 Hz (±5 Hz) window around that frequency. (The average signal strength in the 10 Hz window is used to eliminate the background signal).
4. The 10 largest peaks are tabulated in the peak table.
5. Peak check function: The peak check function determines if each of the bearing fault harmonic frequencies exists within the peak table. The peak frequencies (recorded in step 4) need to be within ±5 Hz of a bearing fault harmonic in order to be considered a match, and subsequently recorded as a failure for this harmonic.
6. Report: If two or more different harmonic frequencies are determined to be failed, then report the bearing as being bad.

In order to analyse a number of different rotating-machinery products, software was written which automates the above algorithm and allows for the processing of large amounts of power spectrum density data.

5. Analysis of vibration data

Typical PSD plot of a good water pump is shown in Figure 5. The magnitude of acceleration was 0.29. The peaks in the plot show the harmonics of the 3600 RPM operating speed, namely 60

Hz, 120 Hz, and 180 Hz. When comparing the peaks of the PSD plot to the bearing fault frequencies in Table 3, no correlation in peaks can be identified.

For comparison, a pump with known bearing faults and noticeable audible noise also has peaks at 60 Hz, 120 Hz, and 180 Hz. In addition, there are high peaks at 155 Hz and 466 Hz in all three axes (Y, X, and Z) as shown in Figures 6, 7, and 8. These fault frequencies match the outer ring raceway dent base frequency of 154 Hz, and the third harmonic frequency of 154 Hz which is 463 Hz (see Table 3).

The typical spectrum of a blower running at 1900 RPM with a good bearing is shown in Figure 9. There are three peaks at 32 Hz in all directions and harmonic frequencies such as 256 Hz and 768 Hz. When comparing these peaks to the frequencies related to the bearing defects shown in Table 4, no correlation can be found. In contrast, the vibration power spectral density plot of a failed blower with a bad bearing and noticeable audible noise is shown in Figure 10. There are high peaks at 125 Hz and 240 Hz in all three directions, namely vertical, radial, and axial. These frequencies are within the frequency window of 121 Hz and 243 Hz of the radial ball surface dents failure mechanism.

The bearings from the blower were carefully removed from the assembly without introducing additional damage, and both bearing raceways and bearing balls were inspected by Keyence VHX Digital Microscope at 200x. Figure 11 shows some dents on the outer raceway and severe ball damage on 5 out of the 7 balls.

The above bearing fault detection can also be applied to newly installed blowers. The power spectral density of a suspect blower is shown in Figure 12. Peaks related to the operating speed are shown as the base frequency of 32 Hz and the harmonic frequencies such as 96 Hz, 384 Hz, and 768 Hz. However, the high peak at 84 Hz is due to outer ring race way dents as indicated in Table 4. Figure 13 shows false brinelling on the outer raceway. There are 7 false brinelling marks on each raceway which correspond to the 7 balls (as shown in Figure 11).

False brinelling is damage caused by fretting and is characterized by material wear, or removal, and occurs over an extended time in the presence of a small load. It is generally a result of vibration while the bearing is non-operational (e.g. during shipping). As the ball elements vibrate between the races, the lubricant is forced out of the contact area. Without lubricant between the balls and races, the small oscillatory movements cause the formation of a fine reddish iron oxide powder. When the powder is combined with the lubricant, it creates an abrasive compound which further accelerates the wear.

False brinelling appears as elliptical impressions in the axial direction at each ball/roller position as shown in Figure 14. The indentations from false brinelling will result in a rough and noisy bearing which will lead to failure. However, the above described bearing failure algorithm can detect this failure mechanism well before catastrophic failure occurs.

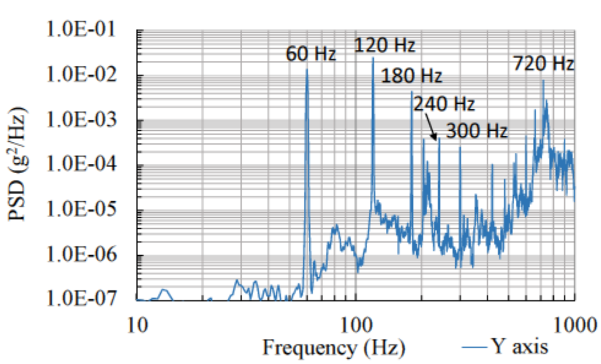

Figure 5: Good Pump running at 3600 rpm (60 Hz)

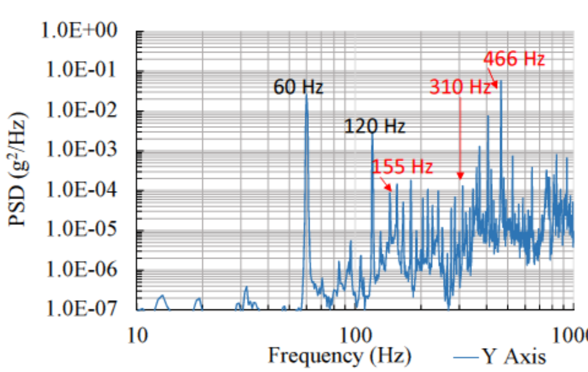

Figure 6 : Pump with failed bearing – 3600 rpm

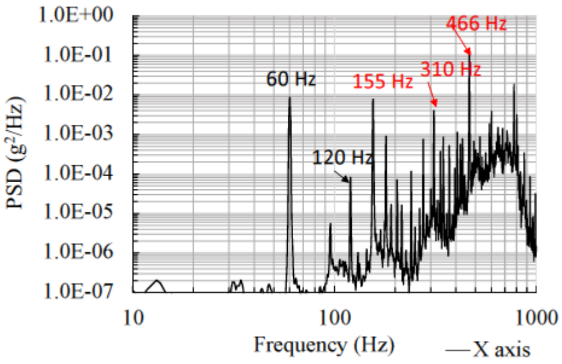

Figure 7: Pump with failed bearing running at 3600 rpm

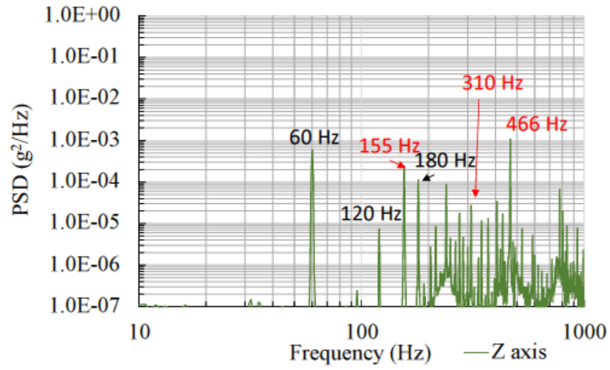

Figure 8: Pump with failed bearing

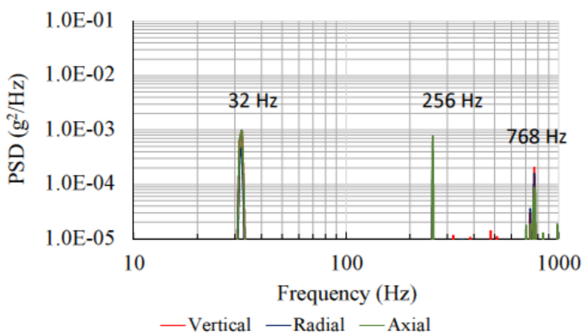

Figure 9: Good Blower running at 1900 RPM

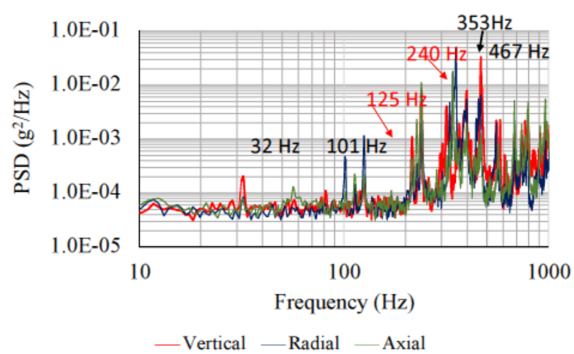

Figure 10: Blower with failed bearing

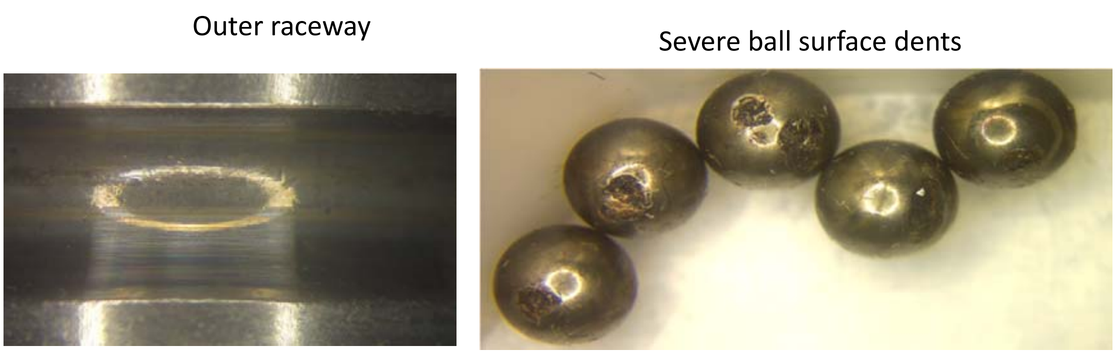

Figure 11: Outer raceway dents and ball surface dents

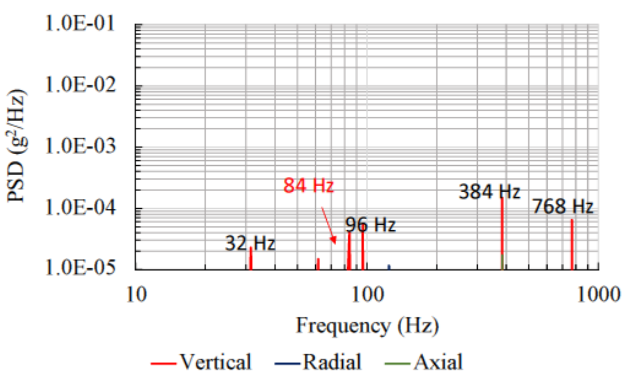

Figure 12: Suspect blower running at 1900 rpm

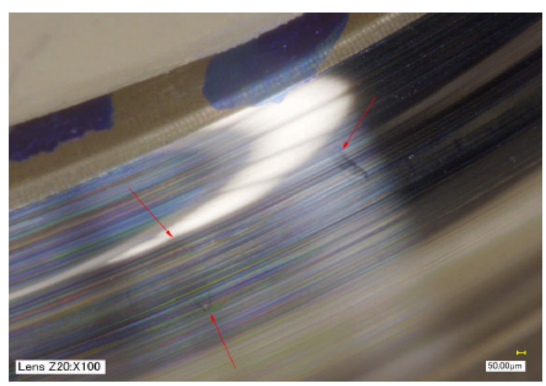

Figure 13: Outer raceway of suspected blower

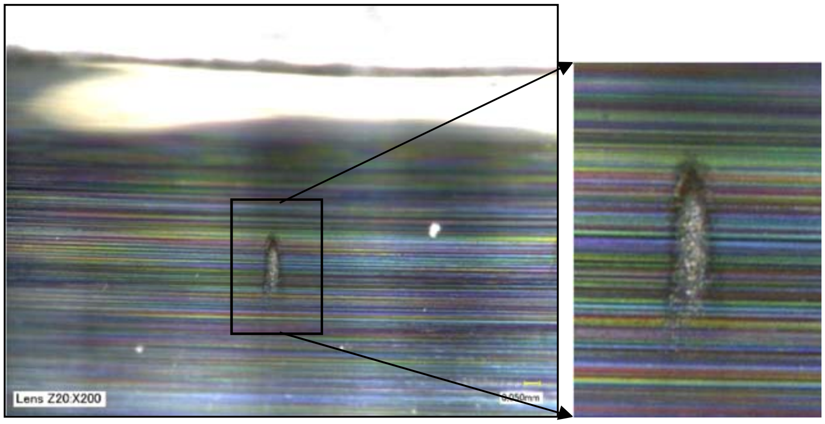

Figure 14: Suspected blower with false brinelling

6. Summary

This paper shows a method for production screening and in situ bearing failure detection using frequency domain analysis of vibration signals. Vibration data was analyzed on new production pumps running at their nominal operating speed. Any pump with an overall vibration magnitude (Grms) higher than the average plus three sigma will be subjected to failure analysis and reassembled. A majority of these rejects are due to unbalanced rotors or misalignments. Improving production quality brought down the Grms significantly. This method has been implemented for 524 pump samples.

Also, an algorithm was developed to detect ball bearing defects. This algorithm has been used to evaluate more than 50 blowers and has been successful in detecting both known failures and initial defects. This algorithm can also be used in situ to accurately report bearing health of rotating machinery in the field.

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