Vibration measurement approach to the bearing damage evolution study in the presence of electrostatic discharge machining currents

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Abstract: The high-frequency components inherent in frequency converter operation can cause additional high-frequency currents to flow within an electric machine. Such currents can cause electrical discharges within the bearings that may damage the bearing surfaces and eventually lead to mechanical overload and bearing faults. Multiple mitigation approaches have been recently suggested by different researchers. However, neither of them provides a universal and complete solution nor gives a deep explanation of the damage evolution mechanism. This paper studies the vibration data observed during an accelerated life testing using excessive high-frequency voltage applied between the rotor shaft and the bearing house. The purpose of the analysis is to define the criteria suitable for the detection of damage patterns resulting from electrostatic discharge machining (EDM) currents.

Keywords: Rolling element bearing, diagnosis, condition monitoring, variable speed drive, EDM current

1. INTRODUCTION

The high-frequency components inherent in frequency converter operation can cause additional high-frequency currents to flow within an electric machine (Von Jouanne et al., 1996; Chen et al. 1996). Such currents can cause electrical discharges within the bearings that may damage the bearing surfaces (Boyanton and Hodges, 2002; Tischmacher and Gattermann, 2010) and eventually lead to mechanical overload and bearing faults. Multiple mitigation approaches were suggested by different researchers (Link, 1999; Schiferl and Melfi, 2004; Muetze and Binder, 2006). However, neither of them provides a universal and complete solution nor gives an exhaustive explanation of the damage evolution mechanism. This paper studies the vibration data observed during an accelerated life testing using excessive high-frequency voltage applied between the rotor shaft and the bearing housing. The purpose of the analysis is to define the criteria suitable for the detection of damage patterns on the rolling element bearing resulting from electrostatic discharge machining (EDM) currents.

2. METHOD

2.1 Vibration sampling and equipment

The test setup comprises of two 3-phase 4-pole 50 Hz 15 kW squirrel cage induction motors on the same shaft, fitted with 2 new type 6309ZZ bearings, powered by an ABB ACS 400 frequency converter, function generator Hameg HM-8131-2, 4 EPCOS - B57560G104F 100 kΩ thermistors, a piezo-electric vibration measurement sensor (Kistler 8712ASM1, sensitivity 1V/g, ±5g, BW(±5%) = 0.5-8000 Hz) attached with a screw to the end-shield of the test motor (Figure 1) and a computer with LabVIEW logging application. The acceleration and temperature signals were sampled using 16-bit National Instruments digital acquisition units NI USB-6211.

Fig. 1. Accelerometer attachment view.

The sampling of the acceleration signal was performed at sampling frequency (f_s) of 20 kHz with 20 second long samples, every 5 minutes during the constant speed rotation mode (25 Hz). The temperature was measured at 3 vertices of equilateral triangle around the shaft opening 3 cm away from the shaft surface on the end-shield using EPCOS B57560G104F NTC thermistors. All calculations were performed using MATLAB.

2.2 Test run description

The first test run was performed according to the methodology presented in (Romanenko, 2014). In this test run, the machine was line-fed from a 3-phase 50 Hz power grid for 1280 hours after a pre-run period of approximately 3 days. The shaft was
rotating at a constant speed of 1500 rpm. This test run showed significant variation in electrostatic discharge intensity over the duration of the run, suggesting significant changes in the grease’s electrical properties resulting in its direct current conductivity which might have prevented further discharges.

After the bearing was replaced for the second test run the setup was pre-run for 4 days at 1500 rpm without any additional voltage applied to the shaft. Then, the drive was set to variable speed mode \( n(t) = \left[ 1425 + 75 \cdot \sin(0.05 t) \right] \) rpm to improve the stability of the grease’s electric properties by facilitating variation in grease thickness and thus improving its mixing in the bearing. The machine was driven by the frequency converter to provide variable rotational speed, while an additional voltage (sine 20 V peak to peak, 300 kHz) was applied to the shaft. This point is considered \( t = 0 \) for the rest of the experiment. After 240 hours of operation, the voltage supplied to the shaft was increased to 55 Vpp. At \( t = 509 \) h the drive speed control from the PC was added. Since then the drive operated in iterative cycles of the following two modes of operation:

- 4 minutes. \( n(t) = \left[ 1425 + 75 \cdot \sin(0.05 t) \right] \) rpm
- 1 minute. \( n(t) = 1500 \) rpm.

At around 580 total hours of runtime a power failure occurred that caused the system to stop for 5 days (\( t = 578 \) - 696 h). Later on the system suffered 2 more power failures at \( t = 1870 \) - 1897 h and \( t = 2090 \) - 2109 h.

3. RESULTS AND ANALYSIS

3.1 Visual inspection

Fig. 2. Scanning electron microscope picturing of a part of the dark trace area on the outer ring surface of the drive-end bearing.

The surface of the bearing was examined after a 2500 hours long test run. Both the outer and inner ring had a visible dark trace, approximately 3 mm wide, in the area where the contact with rolling elements is expected to occur. The surface of the trace was pictured using a scanning electron microscope. The scan result is presented in Figure 2.

3.2 Temperature monitoring

The average of the measured bearing temperatures is presented in Figures 3 and 4. During both the runs the bearing temperature varied between 28 and 34 degrees Celsius. The main difference in the bearing temperature between these two test runs is due to the variable speed mode of operation. This setting resulted in changes in friction and frictional loss in the bearings. This variation in dissipated energy was sufficient to cause temperature oscillations with the same frequency as the speed reference was changing. Another notable oscillation appears in the second test run and has a period of 24 hours. This oscillation might be caused by laboratory room temperature variations which were not monitored.

![Fig. 3. The hourly average temperature of the drive-end bearing during the first test run, which was 1280 hours long (Romanienko, 2014).](image)

![Fig. 4. The hourly average temperature of the drive-end bearing during the second test run, which was 2500 hours long.](image)

3.3 Discharge activity

The discharge activity plots are presented in Figures 5 and 6. The highest discharge activity observed during the first run was \( 4.4 \cdot 10^7 \) discharges per hour. The total number of discharges recorded over the duration of the experiment (1280 hours) was \( 3.8 \cdot 10^7 \). The highest activity for the second run (2500 hours) was \( 1.49 \cdot 10^6 \) discharges per hour while the total number of discharges was \( 5.28 \cdot 10^8 \). It can be noted that the discharge activity during the first test run increased rapidly and then declined to low levels after every stop of the system. In contrast, the discharge activity during the second run increased over time as the rotational speed was constantly changing between a constant and varying speed. To analyse the observed pattern, the lowest detected discharge activities over 20 consecutive measurements (10 minutes) were computed (Figure 7). The discharge activity grew continuously from 1000 to 2200 hours and decreased at the end of the monitored period.
3.4 Measurement range and fault frequencies

The linearity of the accelerometer allows performing measurements up to 8 kHz. The sampling frequency $f_s = 20$ kHz limits the upper cut-off frequency as follows: $20$ kHz / 2.56 = 7.81 kHz. The factor 2.56 is commonly used in FFT-analyzers. Based on these facts, the upper cut-off frequency of 7.8 kHz was chosen. A Butterworth anti-aliasing filter with the order $n = 8$ was used when calculating the spectra. The number of samples $N$ was 4096.

3.5 Time-domain vibration analysis

The most common time-domain quantities used in modern bearing diagnostics are the peak and root-mean-square (rms) values of the acceleration signal ($x^{(2)}$) and its first (jerk) and second (snap) time derivatives. The third useful quantity is the maximum absolute value of the signal divided by the rms value. It is known as the crest factor. These features were computed for the vibration samples. The jerk signal was obtained from the acceleration signal by computing the difference between consecutive samples without scaling for the time interval size. The snap signal was computed from the jerk signal in the same manner. The resulting rms values that were obtained using downsampling to 500 minutes are presented in Figure 8. When the acceleration signal is used, the changes of rms in the frequency range 1 - 7800 Hz are not as clear as in the case of jerk or snap (Figure 8).
The curve in Figure 4 has quite a similar shape with the curves in Figures 8b and 8c. This is to be expected because mechanical vibration transforms into thermal energy in the bearing. Figures 7 and 8 have some similarities as well.

The time domain signals in the frequency range of 1 - 7500 Hz (Figure 9) show that there can be high peaks in the acceleration signal. The peak values of the signals 9a and 9b are 0.2008 g and 0.608 g, respectively.

Fig. 9. Acceleration signals in the frequency range of 1 - 7500 Hz, when (a) t = 9.4 h and (b) t = 2188 h.

3.6 Frequency-domain vibration analysis

The most important fault frequencies in the low frequency range for the test rig are:

<table>
<thead>
<tr>
<th>Rotational frequency, n</th>
<th>Line frequency, f_l</th>
<th>Ball spin frequency, BSF</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 x n = 25.0 Hz</td>
<td>1 x f_l = 50 Hz</td>
<td>1 x BSF = 48.77 Hz</td>
</tr>
<tr>
<td>2 x n = 50.0 Hz</td>
<td>2 x f_l = 100 Hz</td>
<td>2 x BSF = 97.54 Hz</td>
</tr>
<tr>
<td>3 x n = 75.0 Hz</td>
<td>3 x f_l = 150 Hz</td>
<td>3 x BSF = 146.31 Hz</td>
</tr>
<tr>
<td>4 x n = 100.0 Hz</td>
<td>4 x f_l = 200 Hz</td>
<td>4 x BSF = 195.08 Hz</td>
</tr>
<tr>
<td>5 x n = 125.0 Hz</td>
<td>5 x f_l = 250 Hz</td>
<td>6 x BSF = 292.62 Hz</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Ball pass frequency of outer ring, BPFO</th>
<th>Ball pass frequency of inner ring, BPFI</th>
</tr>
</thead>
<tbody>
<tr>
<td>1 x BPFO = 75.86 Hz</td>
<td>1 x BPFI = 124.14 Hz</td>
</tr>
<tr>
<td>2 x BPFO = 151.72 Hz</td>
<td>2 x BPFI = 248.28 Hz</td>
</tr>
<tr>
<td>3 x BPFO = 227.58 Hz</td>
<td>3 x BPFI = 372.42 Hz</td>
</tr>
<tr>
<td>4 x BPFO = 303.44 Hz</td>
<td>4 x BPFI = 496.56 Hz</td>
</tr>
</tbody>
</table>

The Fast Fourier Transform (FFT) was applied to each sample to separate the frequency components from the vibration signal, resulting in a frequency spectrum from 1 Hz to 7.5 kHz. These spectra were downsampled by averaging to steps of 5 Hz and 500 minutes (8 hours 20 minutes). The magnitudes of the resulting spectra are represented as a spectrogram against frequency and runtime in Figure 10.

Fig. 10. Spectrogram of acceleration (g) over the whole frequency range.

It can be seen from Figure 10 that the interesting frequency ranges are around 2.5 kHz and in the low frequencies. Based on the spectrogram in Figure 11, the components below 400 Hz should be examined more closely.

Fig. 11. Spectrogram of acceleration (g) up to 1000 Hz.

An interesting fact is that frequencies 2 x n and 1 x BSF are close to each other. Similar is the case with 3 x n and 1 x BPFO as well as with 5 x n and 1 x BPFI. Vibrations in the frequencies of 25 Hz, 75 Hz, 100 Hz and 125 Hz can be seen in the high resolution spectrum in Figure 12b.

Fig. 12. The amplitude spectrum of acceleration (a) up to 600 Hz, when t = 1488 h and (b) up to 200 Hz, when t = 9.4 h.
The frequency analysis reveals that the vibrations around 2.5 kHz vary. This is apparent from the spectra in Figure 13. Furthermore, changes of its second harmonic (5kHz) can be noticed. Figure 14 shows the trends of the components at 75 Hz, 125 Hz and 2.5 kHz.

Fig. 13. The amplitude spectra of acceleration up to 7500 Hz, when (a) \( t = 339.8 \) h and (b) \( t = 456.8 \) h.

Fig. 14. The trend of rms acceleration in the range of (a) 70.8 - 80.6 Hz, (b) 119.6 - 129.4 Hz and (c) 2495.1 - 2504.9 Hz.

Practical experience has shown that bearing faults cause vibrations in the high frequency range of 2 - 4 kHz (Kowal, 1999).

Sidebands of either BPFO, BPFI or a combination of both can also occur. From this perspective the spectrum in the Figure 15 is interesting, because it has a peak at the frequency of 2500 Hz - 2 x BPFI.

Fig. 15. The amplitude spectrum of acceleration up to 7500 Hz, when \( t = 1380 \) h.

Based on the obtained results, we decided to make an envelope analysis using a band-pass filtering over the frequency range of 2 - 5 kHz. The envelope spectra were down-sampled by averaging to steps of 5 Hz and 500 minutes. The spectrogram is presented in Figure 16. The main harmonics are multiples of 25 Hz and 100 Hz. The spectrum in Figure 12 reveals that there is vibration around 300 Hz at the sideband frequencies of 300 Hz - n, 300 Hz + n and also at the frequency of 315 Hz.

Fig. 16. Spectrogram of enveloped acceleration (g) using a band-pass filter from 2 to 5 kHz.

4. DISCUSSION

The visual representation of these spectra as a spectrogram provides a convenient representation of the frequency composition changes in the vibration signal as the bearing damage evolves. Among these frequencies and bands several criteria can be selected for the quantitative evaluation of the bearing damage. While most of them provide some insight into the bearings’ state of health, they do not allow predicting the failure reliably as a standalone criterion. Such criteria are usually implemented as part of a neural networks based, fuzzy decision-making or some other artificial intelligence system (Lahdelma and Juuso, 2011).

New criteria can be added to the system by the implementation of monitoring for other physical processes apart from vibration and electrical current monitoring. One of such processes is the monitoring of electromagnetic pulses emitted by the discharges taking place in the bearings. The discharge activity under certain conditions seems to have some correlation with the vibrations occurring during machine operation. This effect might be caused by electromagnetic coupling between the vibration measurement system and discharge sparks. However, such disruptions are expected to be wide-band and unlikely to
have the same characteristic frequencies as the mechanical parts of the system. On the one hand, the discharges are expected to depend on vibration and shaft rotational speed as both affect the thickness of the grease layer (Muetze et al., 2011). On the other hand, the process of discharging is also affected by the chemistry of the grease (Romanenko et al., 2015). Therefore, a universal relationship between the instantaneous discharge activity of a bearing and its state of health might be very difficult, if not impossible, to derive, given the current state of knowledge of the discharge and damage processes. The absence of such would make this method a good addition to other diagnosis criteria but not a complete replacement.

Comparing the two test runs it seems that the change in the rotational speed of the electrical machine is likely to affect the bearing health degradation process, which is in line with the influence of the shaft rotational speed on the discharge activity and the occurrence of high discharge activity during the “start-up test” observed by Muetze et al. (2011). The variable speed and load conditions promote the intensity of the discharging activity, which in some cases (i.e. in lifetime greased bearings) reduces the lifetime of the bearing.

The observed changes at some characteristic frequencies can be explained by the presence of melted trace areas in the bearing. A single area at the bottom of the bearing where the melted race starts to appear in first place should result in increased vibrations at the ball pass outer ring frequency in the same way it happens for a single damage point (Lindh, 2003). However, as the trace becomes larger it would probably disappear from this frequency as the track becomes sufficiently uniform and it is unclear what would be the characteristic frequencies for such complex damage pattern. Thus, further theoretical research and modelling is necessary to analyse the possible effects of fluting patterns and the presence of local melting areas on the vibration spectra.

The analysis showed that the levels of the vibration component at 2.5 kHz vary. Around this frequency there were also sidebands which correspond to the fault frequencies BPFI and BPFO. These observations support the results (Kowal, 1999) according to which EDM causes vibrations in the high frequency range of 2 - 4 kHz. There was also some variation in the vibration at the second harmonic of 2.5 kHz. The vibration level trend showed that the 2.5 kHz vibration was mostly low, but at times the levels were high. The vibrations at the rotational frequency were quite stable. The acceleration component of 75 Hz varied and was mainly higher than the 125 Hz component, although the latter had high values at times as well.

Some short time domain signals showed strong impacts. The connection between these impacts and the electric discharges is worth to be examined closer. There were similarities between the rms values of the vibrations and the temperature values, especially when jerk and snap were used. It is also worth paying attention to the connection between the logarithm of the lowest observed discharge activity and the rms values of vibrations, because there were similarities between these trends as well (Figure 7 and 8).

5. CONCLUSIONS

The initial studies in detecting the occurrence of the electrostatic discharge machine (EDM) currents have been encouraging. The rms of the first and second time derivatives of acceleration respond well to the changes in the high frequency vibrations. There are similarities between their rms values and the logarithm of the lowest detected level of observed discharge activity. Similarity was also quite clear between the temperature and the vibrations. In the future, more long-term tests could be performed, and real and complex order time derivatives and Ip norms could be used.

REFERENCES