THE CASE OF THE MISSING BEARING FAULT FREQUENCY

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Abstract: During testing on a small gas turbine, frequencies related to the power turbine shaft and a bearing were observed. The frequencies of the fault were not associated with any known bearing on the power turbine shaft. This paper is an investigation of why the observed bearing fault frequency of 6% higher than anticipated. It will be shown that because the faulted bearing was a worn thrust bearing, the contact angle and pitch diameter of the roller element had changed. This is an infrequently observed phenomenon, which can lead to missed fault detection on a critical component. A mitigation strategy for this type of failure is discussed.

Keywords: Spectral Kurtosis; vibration diagnostics; resonance; performance testing; hardware testing

Introduction: Using two MEMS accelerometer-based smart sensors along with tachometer inputs to supply rotational speed, a blind, seeded fault test was conducted to generate condition indicators for 10 shafts, 12 gears and 26 bearings for the Rolls Royce M250-C47 gearbox. The seeded fault was an unknown bearing installed and run on Day 1, while a nominal bearing replaced the faulted bearing for a run on Day 2. The data acquisition system collected 31 acquisitions on day 1, and 32 acquisitions on day 2. Acquisitions included both processed (condition indicator data) and raw vibration and tachometer data.

The spectral response from the two sensors indicated content up to the bandwidth of the sensors (32 KHz). This far exceeded the estimated modal response of the gearbox (by rap test). Validation of the configuration was performed with known frequencies of interest (corresponding to shaft rates and gear mesh rates) to ensure Condition Indicator (CI) values matched spectral data.

Based on the observed spectral content, the envelope configuration was set from 18 to 25 KHz. A consistent frequency of 4029 Hz was then seen on day 1 and not on day 2. It is hypothesized that this frequency was associated with the power turbine shaft, and further, that the fault feature was from an ball element fault. The 4029 Hz frequency would correspond to an outer race rate of 7.58, which did not match any know bearing fault rate on the engine – the closest being an roller element rate of 7.05. The issue is: why is there such a large disparity between the measures and published bearing data?

Review of Bearing Envelope Analysis: Rotating equipment in general and equipment such as helicopters are dependent on a transmission to condition the power for useful work. In the case of helicopters, the power of a high speed, low torque power shaft is converted into a low speed, high torque main rotor shaft, which delivers lift. Integral to the reliable operation of this transmission, are bearings. Reliability and availability of the helicopters are improved if monitoring techniques are developed, which can detect when a degrading or faulty bearing requires maintenance. This is the essence of condition monitoring for bearings.

Because the vibration signals of a faulty bearing are small compared to shaft order and gear mesh frequency, detection of faults at the bearing rate frequencies using Fourier analysis is difficult. Fault detection at the baseband frequencies of the bearing rate is “Stage 1” fault detection. Bearing faults detected using these types of analysis are Late Stage; the bearing can be close to catastrophic failure. At the very least, a bearing in this state is generating metal, which can cause damage to other components within the gearbox.

Ultrasonic emission can detect bearing inner and outer race roughness (a "Stage 3" fault), but the remaining useful life of a bearing at this stage is relatively long compared to the overall life of the bearing. Bearing envelope analysis (BEA) can typically detect bearing faults 100s if not 1000s of hours before when it is appropriate to do maintenance. It is for this reason that many conditions monitoring systems manufacturers are using envelope analysis techniques.

Bearing Envelope Analysis: BEA is based on demodulation of high-frequency resonance associated with bearing element impacts. For rolling element bearings, when the rolling elements strike a local fault on the inner or outer race, or a fault on a rolling element strikes the inner or outer race, an impact is produced. These impacts modulate a signal at the associated bearing pass frequencies, such as associated with the Cage (FTF, fundamental train frequency),

$$FTF = \frac{S}{2}\left(1 - {}^{Bd}/_{D}\, cos(\phi)\right) \qquad \text{(eq 1)}$$

or the Ball Pass Frequency Inner Race (BPFI),

$$BPFI = \frac{Nb \times S}{2}\left(1 + {}^{Bd}/_{D}\, cos(\phi)\right) \qquad \text{(eq 2)}$$

or the Ball Pass Frequency Outer Race (BPFO),

$$BPFO = \frac{Nb \times S}{2}\left(1 - {}^{Bd}/_{D}\, cos(\phi)\right) \qquad \text{(eq 3)}$$

and the Ball Pass Spin Frequency (BSF),

$$BSF = \frac{Pd \times S}{Bd}\left(1 - \left({}^{Bd}/_{D}\right)^{2} cos(\phi)^{2}\right) \qquad \text{(eq 4)}$$

Where:

S is the shaft frequency,
Bd is the ball or roller element diameter
Nd is the number of balls or rollers
D is the pitch diameter and
Φ is the contact angle.

Note that the BSF is usually constructed as half of the rate (divided by two), but because a spall or damage to the roller hits both the outer and inner trace, the observed frequency is 2x, or as given in eq 2.

To illustrate BEA, Figure (1) is an example of an outer race fault, where the BPFO is approximately 80 Hz. Note that the modulation rate, T1, is approximately .0125 seconds (e.g., 1/80 Hz). The time T2, the period of the resonance, is approximately 1.12e-4 seconds, or about 9000 Hz. Note that the time domain representation is the superposition of many resonances of the bearing itself.

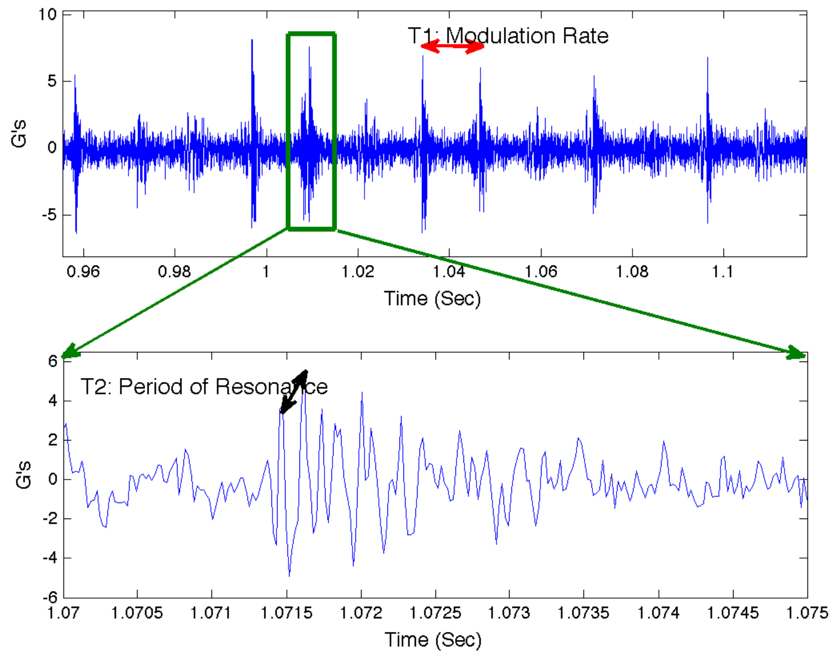

Figure 1 Example Outer Race Fault

Mathematically, the modulation is described as:

$$\cos(a) \text{ x } \cos(b) = \tfrac{1}{2} [\cos(a - b) \text{ x } \cos(a + b)] \qquad \text{(eq 5)}$$

This is an amplitude modulation of the bearing rate (a) with the high-frequency carrier signal (resonant frequency (b)). This causes sidebands in the spectrum surrounding the resonant frequency. It is sometimes difficult to distinguish the exact frequency of the resonance. It is usually not known *a priori* and cannot be determined easily without a faulted component. However, demodulation techniques typically do not need to know the exact frequency. One method for the BEA involves multiplying the vibration signal by a resonant frequency (example, 9 kHz). This is then low pass filtered to remove the high-frequency image, decimated, and the spectral power density is estimated (Eq 6)

$$\cos(b) \text{ x } \cos(a + b) = \tfrac{1}{2}[\cos(a+b-b)+\cos(a+b+b)] \text{ ->} H(\omega\Downarrow)\text{->} \cos(a) \qquad \text{(eq 6)}$$

An alternate way to generate the envelope is to take the FFT (fast Fourier transform) of the time domain signal and calculate the frequency of each bin (e.g., sample rate/length of time series). Calculate the "window" indexes (e.g., the range of frequencies in which the bearing is resonating, say 8 to 9 kHz). Copy the Fourier coefficients from the desired window (e.g., 8kHz to 9 kHz) into the baseband (e.g., 0 to 1 kHz) indexes (this is a Heterodyne operation). Then copy zeros into the remaining Fourier coefficients (Fourier filtering and Hilbert transform, recalling that the Hilbert transform is defined at the real frequency of the Fourier domain, see Ref 1). Finally, take the absolute value of inverse FFT Fourier coefficients. A Matlab® function is given as:

```
function [env,dty] = envelope( data, dt, lowf, highf)
% [env,dty] = envelope1(data,dt,nfilt,lowf,highf);
%Inputs:
% data       :data vector, time domain
% dt         :sampling time interval
% lowf       :low frequency limit of the bandpass filter
% highf      :high frequency limit of the bandpass filter
%Outputs:
% env        :Envelope of data
% dty        :decimated sample rate

n = length(data);
dfq = 1/dt/n;
idxLow = floor(lowf/dfq);     % get the bin for the window low
idxHi = ceil(highf/dfq);      % get the bin for the window high
D = fft(data);
idx = idxHi-idxLow + 1;

D(1:idx) = 2*D(idxLow:idxHi); % this is the definition of the
D(idx+1:end) = 0;             % Hilbert Transform
data = abs(ifft(D));
bw = highf - lowf;
r = fix(1/(bw*2*dt));         % Decimating after low pass
filtering
env =  data(1:r:n);           % crop first nfilt elements
dty = dt*r;                   % New sample rate after decimation
```

The bearing components have many vibration modes, which will correspondingly generate resonance at various frequencies throughout the spectrum. The selection of the frequency range used to demodulate the bearing rate signal (e.g., the window center frequency) should take into account some issues: First, the gearbox spectrum contains many high-energy frequencies from shaft and gear harmonics, which would mask analysis at lower bearing frequencies. Second, there are a number of accelerometers with natural resonance at frequencies that are similar to the bearing modes. Using a higher frequency window close to the accelerometer resonance can amplify the bearing fault signal, increasing the probability of fault detection.

BEA should be performed at frequencies higher than the shaft and gear mesh frequencies. This ensures that the demodulated bearing frequencies are not masked by the other rotating sources, such as shaft and gear mesh, which are present at FTF, BPFO, BPFI, and BSF frequencies. Typical shaft order amplitudes of 0.1 G's and gear mesh amplitudes of 10s of G's are typical. Damaged bearing amplitudes are 0.001 G's.

A technique used to identify an appropriate envelope window is Spectral Kurtosis (ref 2), where spectral kurtosis (SK) is used to determine the best window to determine the condition of a test gearbox with seeded faults. Kurtosis is a non-dimensional quantity that measures the relative “peakedness" of a distribution relative to the Gaussian distribution. Spectral kurtosis (SK) is a statistical parameter indicating how the impulsiveness of a signal varies with frequency. As noted, faults associated with rolling element bearings give rise to short impulse. The SK will be significant in frequency bands where the fault signal is dominant and small where the spectrum is dominated by stationary signals. Antoni developed the kurtogram, which is a map indicating the optimum center frequency and bandwidth combination.

In this kurtogram example, filtering was limited between 1 kHz and 24.4 kHz (Nyquist). The lower limit was chosen because typically, below 1 kHz, there are significant gear mesh frequencies. The kurtogram frequency space spans eight octaves, such that frequency band is approximately halved with each increase in octave.

TABLE I. KURTOGRAM FREQUENCY MAP

Octave	Bandwidth	Number of Bands	PSD Window
1	12.2 kHz	2	32768
2	8.1 kHz	3	16384
3	6.1 kHz	4	8192
4	4.1 kHz	6	4096
5	3.05 kHz	8	4096
6	2.03 kHz	12	2048
7	1.5 kHz	16	1024
8	0.76 kHz	32	512

Figure 2 shows the raw spectrum and the Kurtogram. The largest kurtosis value was in Octave 5, or a window of 1 to 4 kHz. This is compared to the second example window in Octave 8, from 2.51 kHz to 3.266 kHz.

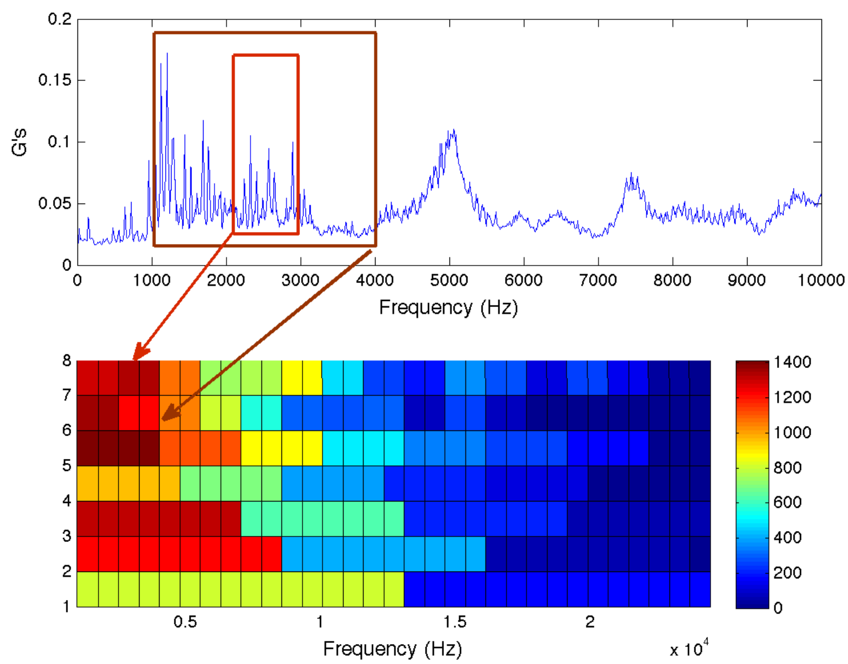

Figure 2 Kurtogram example

The envelope analysis spectrum for Window 1 and Window 2 is plotted in Figure 3. Note that a 3rd envelope analysis is also demonstrated for a poor window: 15 to 18 kHz (band 5, Octave 4). The SNR for the Window 1 was 23 dB, Window 2: 15 dB, and Window 3: 10 dB. There have been many reported cases where poor window selection has resulted in a missed detection of a faulted bearing.

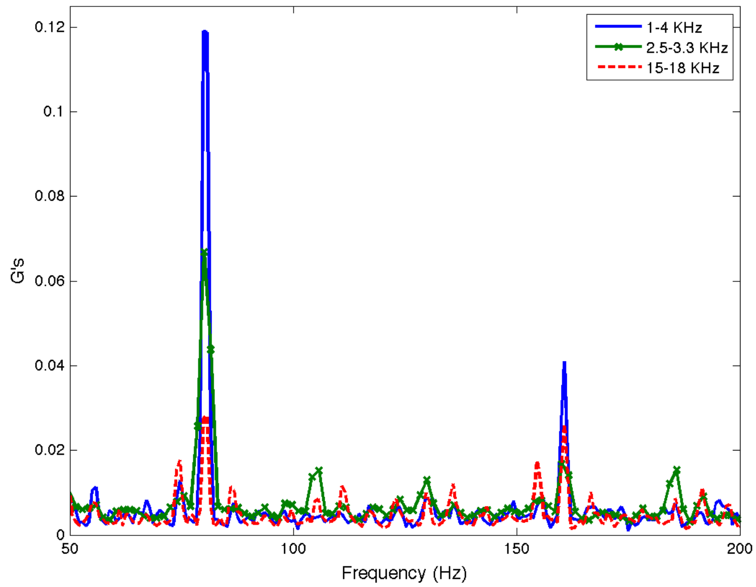

Figure 3 Example of the effect of the Envelope Window on the Spectra

The Test Article: The test article was a small, gas turbine engine used to power a helicopter. The power turbine shaft rate was: 32,184 rpm or 536.4 Hz at 100%. This type of engine typically runs at 55% for idle, and at 100% for flight. Power is delivered from the turbine shaft via a gearbox to the power output drive via a gearbox integrated into the engine. Four bearings support the power turbine shaft. The bearing of interest has the following rates: 231, 3779, 5782, and 4409 Hz for the *FTF, BSF, BFPI,* and *BPFO*.

The envelope analysis frequencies were chosen as 18 to 25 kHz, while the spectral density window as 8192. For a typical run, the envelope response was seen as in Figure 4. Note that there is a great deal spectral content, with energy associated with both the shaft (at 536 Hz) and cage at 230 Hz. However, the other spectral content at 3493, 4029 and 4565 Hz does not match any of the calculated bearing fault frequency. Curiously, it was noted that that 3493 and 4565 is -/+ 536 Hz, which was the shaft rate. This seems to indicate sidebands of the bearing fault modulated by the shaft, but 4029 Hz is almost 6.5% higher than the ball fault frequency, as calculated in eq 2 (see Figure 4).

Further investigation showed that these frequencies were associated with the power turbine, that they were present only when the engine was above 65% torque. The frequency of 4029 Hz was too high a frequency to be associated with any other bearing other than the bearing of interest. The problem was, the fault frequency did not match the published bearing data. What is the cause of this discrepancy? One clue is that this is a thrust bearing.

From (ref 3) it is seen that for rigidly mounted bearings incapable of radial deformation, the contract angel β due to a thrust load can be written as:

$$\beta = cos^{-1}\left({}^{D - P_d/2}\big/_{D + \delta}\right) \qquad \text{(eq 7)}$$

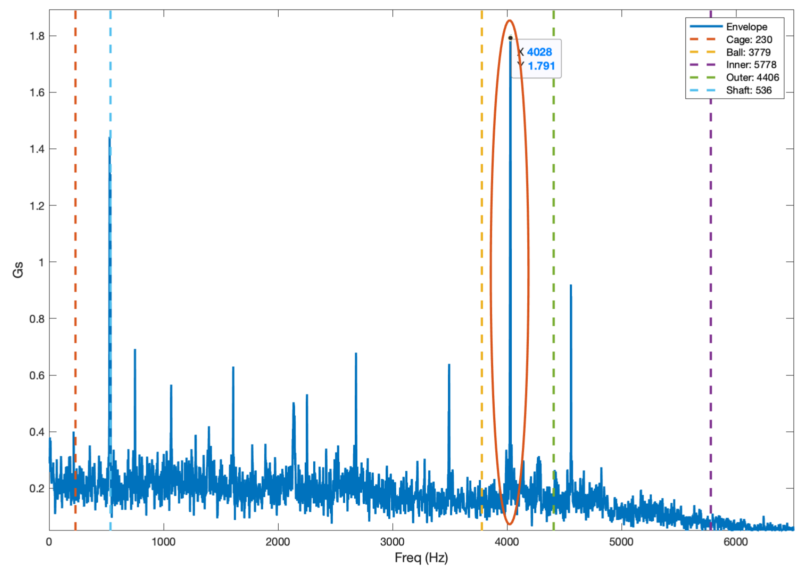

Figure 4 Envelope Spectrum of a Turbine Engine

Where *D* is the pitch diameter, P_d is the diametral clearance, and δ is total elastic deformation or wear. From (ref 4) it is given that:

$$\cos(\beta_f) = {D - P_d/2}\Big/{D} \qquad \text{(eq 8)}$$

which is the manufactures contract angle β_f of 14.7 degrees. This allows one to calculate P_d as 0.0183. Rearranging terms gives:

$$\delta = cos^{-1}\left({cos(\beta_f)}\Big/{cos(\beta)}\right) \qquad \text{(eq 9)}$$

The thrust loads axial deflection, δ_t, give in in 3, is then:

$$\delta_t = (D + \delta)sin(\beta) - Dsin(\beta_f) \qquad \text{(eq 10)}$$

and finally:

$$\delta_t = {D\ sin(\beta - \beta_f)}\Big/{cos(\beta)} \qquad \text{(eq 11)}$$

Given that this is a thrust bearing, one can hypothesize that the turbine axial load is causing a displacement δ_t, which is both changing the contact angle of the bearing from β_f to β and the pitch diameter from *D* to $D+\delta_t$. This change in contact angle and apparent pitch diameter then *increases* the observed frequency. In our case, from 3779 Hz to 4028 Hz, as per Figure 5.

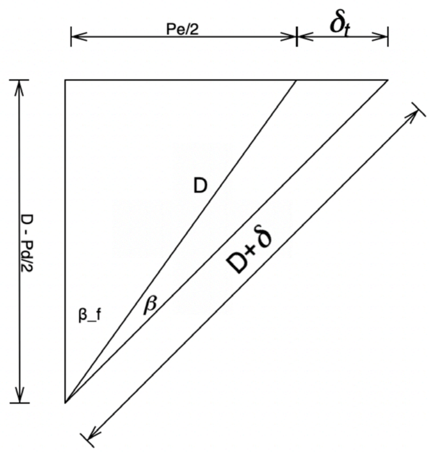

Figure 5 Change in Pitch Diameter and Contact Angle due to thrust load

If the contact angle changed from 14.7 to 18.16 degrees, the δ_t becomes 0.128. This intern suggests the apparent pitch diameter under load is: 2.129. The calculated *BSF* becomes the desired 4029 Hz.

A Mitigation Strategy: These phenomena will only be observed on thrust bearings, and generally only under load. Further, it is suspected that due to the age of the test article, seen only on system that are worn or towards end of life. While a rare event, a mitigation strategy is needed to allow for fault identification.

Typically, diagnostic systems, which rely on RMS or other, gross time domain identification, will not work on these complex systems. For example, the peak-to-peak measured G on the test article was close to 250 Gs. This is because there are a number of gear mesh interactions at different rates, and in time domain, the measured signal is the superposition of all vibration sources. While each gear mesh may be 5 or 10 Gs, there are 12 gears in the gearbox, and the measured, time domain peak to peak is the superposition of all signals.

Instead, the mechanization for diagnostics in complex gearboxes is based on configuration for a known fault frequency. Because bearings have non-Herztian contact, there tends to be some slip. As such, the calculated frequencies (eq 1, 2, 3 and 4) tend to be 0.5 to 1 percent low. Hence, a system does a local, directed search around the anticipated fault frequencies and picks the peak energy in search window. This becomes the condition indicator.

It is recommended that for thrust bearings, the search window be set from 1% low to 7% high for the anticipated frequency bear fault frequency. For example, in case the anticipated frequency was 3778 Hz. The search window for the fault feature would then be 3740 Hz to 4042 Hz. Were these a non-thrust bearing application, the window would be: 3740 to 3816 Hz.

Note: On a separate, operational turbine engine installation, the same thrust bearing phenomena was seen, albeit with a change of frequency of just 2.36% change. This newer engine had less wear. As can been seen in Figure 6, were the ball rate is calculated at 3775 Hz, the observed frequency was 3864 Hz. Note the side bands at both 3328 Hz and 4399 Hz, or +/- the shaft rate of 535, again suggested a ball fault modulated by shaft.

Conclusion: During a test of a light turbine engine, indications of a bearing fault were found in the envelope spectrum. However, the fault frequency was not associated with any know bearing. The fault frequency had side bands at the power shaft rate (536 Hz), which indicated an inner race fault, or roller/ball element fault: side bands are commonly observed when the shaft rate modulates the bearing fault frequency. The observed fault frequency was 6% higher than would be expected for a roller/ball fault, whereas the 8% lower for the next know frequency (outer race frequency).

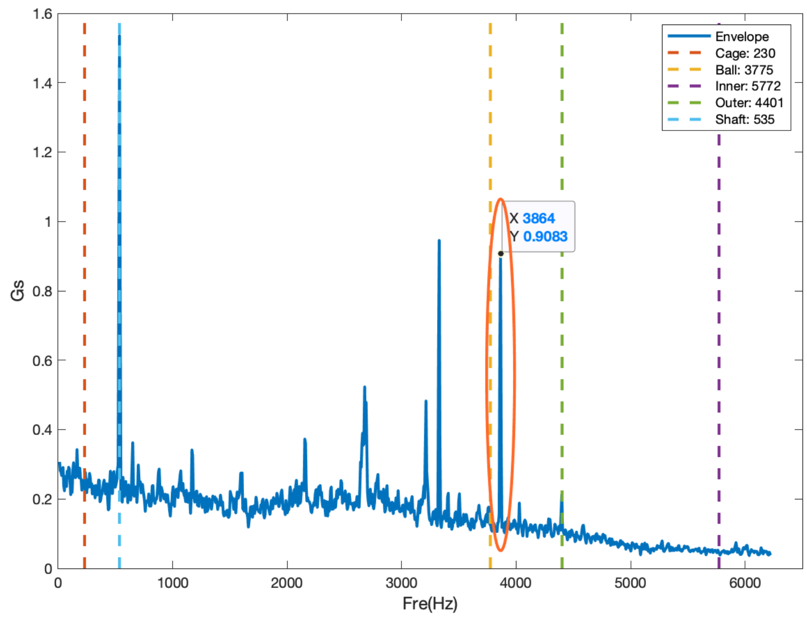

Figure 6 Ball Fault on Turbine, 2.6% error

The fault was observed when the engine output was greater than 65% rated power. Further, it was observed that the bearing with fault frequencies that was closes to the observed feature was a thrust bearing. Thrust bearing under axial loads can change their contact angle. A change in the contact angle additionally changes the axial displacement of the bearing. This change in axial displacement increases the observed pitch diameter, which increases the observed fault frequency. As mitigation, it is recommended that for thrust bearings, the frequency bound should be set from one percent below to seven percent above the specified fault frequency.

References:

[1] Randal, R. B., *Vibration-based Condition Monitoring*, John Wiley & Sons, 2011.
[2] Antoni, J., Randall, RB., "The spectral Kurtosis: application to the vibratory surveillance and diagnostics of rotating machines," Mechanical systems and signal processing 20(2), 2006
[3] Hamrock, B., Dowson, D., "Ball Bearing Mechanics," NASA Technical Memorandum 81691, 1981
[4] Hamrock, B., Dowson, D., "Introduction to Ball Bearings," NASA Technical Memorandum 81690, 1981

Bibliography:

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