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Beating Phenomenon of Multi-Harmonics Defect Frequencies in a Rolling Element Bearing: Case Study from Water Pumping Station

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***Abstract*—**Rolling element bearings are widely used in industry, especially where high load capacity is required. The diagnosis of their conditions is essential matter for downtime reduction and saving cost of maintenance. Therefore, an intensive analysis of frequency spectrum of their faults must be carried out in order to determine the main reason of the fault. This paper focus on a beating phenomena observed in the waveform (time domain) of a cylindrical rolling element bearing. The beating frequencies were not related to any sources nearby the machine nor any other malfunctions (unbalance, misalignment …etc). More investigation on the spike energy and the frequency spectrum indicated a problem with races of the bearing. Multi-harmonics of the fundamental defects frequencies were observed. Two of them were close to each other in magnitude those were the source of the beating phenomena.

***Keywords*—**Bearing, beating, spike energy, vibration.

I. Introduction

CONDITION monitoring of rotating machinery is a critical subject in most industries. It can prevent catastrophic failure of machines, production losses and reduces the downtime for maintenance. The vibration analysis technique consists mainly of vibration measurement and its interpretation. The vibration measurement is done by picking up signals from the machines by means of the vibration instruments, then, these signals are processed using an FFT analyzer to obtain the frequency spectrum. The interpretation of the results is mainly done by relating the measured frequencies with their relevant causes such as unbalance, misalignment, bearing defects, resonance…etc.

There are several researches and studies on the vibration analysis of the rotating machinery. Marangoni [1] investigated experimentally the characteristics of unbalanced and misaligned flexible coupling rotor system. The theoretical predictions were in good agreement with his results. Both of them showed that the unbalance and misalignment can be characterized by 1X and 2X shaft running speed, respectively. Haung [2] studied the torsional vibrations of a shaft with unbalance. In his work, a synchronous torsional vibration accompanying with small harmonic components were excited except when the speed of the rotor is near or equal to half the natural frequency of the torsional vibration.

Rolling element bearings vibrations have been covered widely in many studies, and it has been shown that vibration behavior of a good bearing is different from a defective one in the time domain, frequency spectrum and spike energy [3], [4]. Orhan and Nizami [5] presented comprehensive case studies on the defect diagnosis of rolling element bearings. They utilized the vibration analysis as a predictive maintenance tool to diagnose the faults in ball and cylindrical roller element bearings. It was proved that ball and cylindrical roller element bearing defects were evolved in similar manner regardless of the rolling element type.

According to Nizami [5], there are quite few case studies on rolling element bearings under real operating conditions. This work is trying to fill this gap by presenting the application of condition monitoring through vibration measurement and analysis in the industrial environment to investigate the problems associated with cylindrical rolling element bearings.

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II. Vibration Measurement

We collected the vibration data using SKF Microlog data analyzer CMXA 50 shown in Fig. 1. It has a built in FFT algorithm, however, computer software was used to study and analyze the signals. We used a general purpose low profile industrial accelerometer manufactured by SKF (CMSS2200) with a sensitivity of 100 mV/g and a typical frequency range from 1 Hz to 10 KHz – Fig. 1. The machine data and vibration measurement parameters we adopted are given in Table I.

We took the measurements in the three axes: axial, horizontal and vertical. The horizontal data was dominant relative to vertical and axial; therefore, the measured vibration data in the horizontal direction was adopted for the analysis of the machine

Fig. 1 Instruments used in this paper: SKF Microlog CMXA 50 and the accelerometer CMSS 2200

TABLE I
VIBRATION MEASUREMENT PARAMETERS

Machine speed	1480 rpm (24.67 Hz)
Machine power	405 KW
Frequency range	2-1000 Hz
Window type	Hanning
Number of spectral lines	800 lines for velocity 1600 lines for gE
Number of average	4

III. OBSERVATIONS

A double suction centrifugal pump running at 1480 rpm (24.67 Hz) is installed in West Riffa II water blending station in Bahrain water transmission network as shown in Fig. 2. The main duty of the pump is to pump water from a ground storage tanks to an elevated service reservoir (ESR). The last overhauling of the pump was on May 2008 after which the pump was running smoothly. Fig. 3 shows the vertical frequency spectrum of the Pump-drive end bearing which has an overall vibration of 0.6 mm/s only.

Fig. 2 West Riffa centrifugal pump

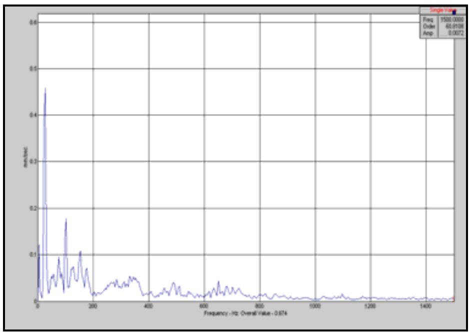

Fig. 3 Vibration readings on May 2008

At the end of the same year, we took the discharge butterfly valve for complete overhauling and rehabilitation. The valve was scheduled to be ready within weeks; however, due to special seat and rubber requirement for the valve and the material delivery, it took five months to set the valve back in the system. During the valve outage period, the pump was idle along with its motor.

In May 2009, we installed the butterfly valve back in the system and the pump has been started to run. We commissioned the pump and took vibration measurements to ensure the smooth running of it. The measured data showed abnormality in the pump, several high peaks appeared in the spectrum associated with noise. These observations indicated a problem with the pump, and we started our investigation and analysis.

IV. ANALYSIS

The first readings of vibrations shows interesting peaks on the velocity spectrum at the pump drive end bearing. Two major peaks at around 597.5 Hz and 622.5 Hz were clearly appeared along with some other higher frequencies peaks. Fig. 4 shows the frequency spectrum of the pump drive end bearing. The running speed of the pump is 1480 rpm (24.67 Hz) and non of the above mentioned frequencies are related to it (24.22X and 25.23X) respectively.

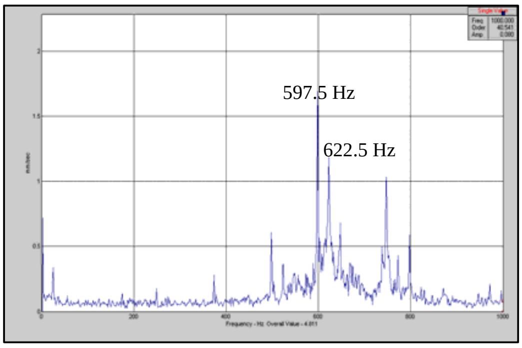

Fig. 4 Frequency spectrum of the pump drive end bearing

We carried out an analysis to the time domain (waveform) for more specific symptoms. A first look on it shows abnormality in the general layout as seen in Fig.5. It is clearly indicating a beating phenomenon in the machine.

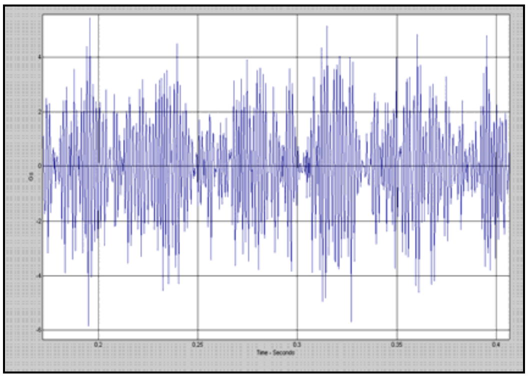

Fig. 5 Time domain (waveform) showing the beating phenomena

The first source one might think about is the two frequencies (597.5 Hz and 622.5 Hz), since they are only 25 Hz apart. Therefore, a beating computation analysis was carried by our tem to figure out and confirm this conclusion.

Fig. 6 shows the time domain of several beating cycles and the period of one complete cycle (beating period) can be computed as follows:

$$\text{Beating period} = \frac{0.38 - 0.22}{4} = 0.04 \text{ sec} \quad (1)$$

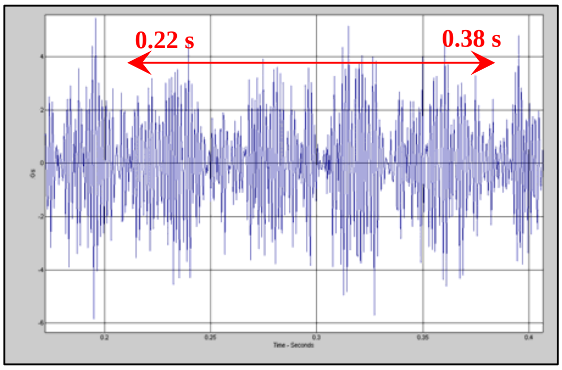

Fig. 6 Beating cycles in the time domain

In Fig. 7, the time domain for one complete beating cycle is shown, and the oscillation period can be computed as:

$$\text{Oscillation period} = \frac{0.33 - 0.315}{9} = 0.001667 \text{ sec} \quad (2)$$

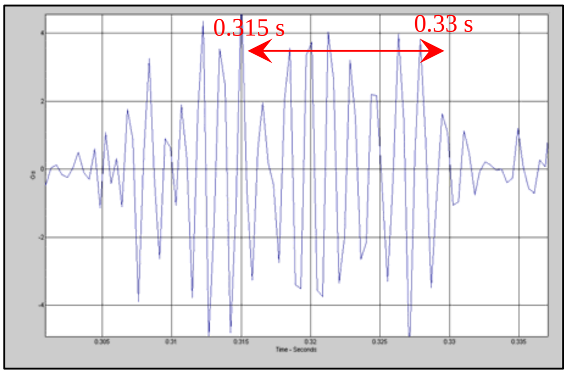

Fig. 7 One complete beating cycle in the time domain

Since, both beating period and oscillation period have been evaluated, one can calculate the frequencies of the two signals. Denoting the frequencies of them by ω_1 and ω_2, we can write:

$$\text{Beating period} = \frac{2\pi}{|\omega_1 - \omega_2|} = \frac{2\pi}{\omega_1 - \omega_2} = 0.04 \quad (3)$$

assuming $\omega_1 > \omega_2$

$$\text{Oscillation period} = \frac{4\pi}{\omega_1 + \omega_2} = 0.001667 \quad (4)$$

This leads to the system of two equations:

$$\omega_1 - \omega_2 = 157 \quad (5)$$

$$\omega_1 + \omega_2 = 7538 \quad (6)$$

Upon solving them, one gets:

ω_1 = 3847.5 rad/s = 612 Hz

ω_2 = 3690.5 rad/s = 587 Hz

The frequency spectrum shows frequencies at 597.5 Hz and 622.5 Hz. The agreement between the calculated frequencies and those obtained from the FFT analyzer is quiet excellent. Therefore, we proved our claim of the beating frequencies, but we haven't found their source(s) yet.

Now, our aim is targeted to figure out the source of these frequencies. To do this, we looked at the spike energy spectrum to find out any possibility of failure in the bearings.

Fig. 8 shows the spike energy spectrum in term of gE. Surprisingly, high and sharp peaks appeared in the plot. A conclusion of bearing problem is expected, since the spikes are clear and strong.

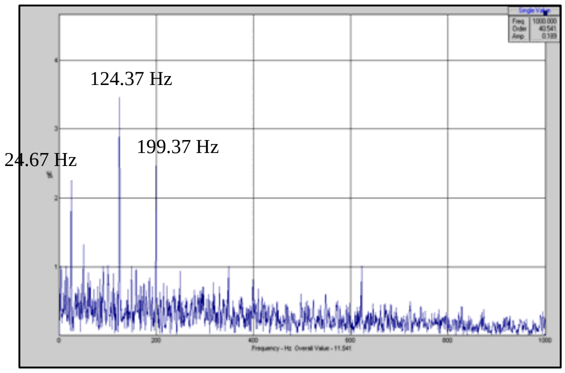

Fig. 8 Spike energy spectrum at the pump non drive end

From Fig. 8 three major peaks (spikes) are appeared, those are: 24.67 Hz, 124.37 Hz and 199.37 Hz. The running speed of the pump is 1480rpm (N = 24.67 Hz), therefore, the first peak is the running frequency. To clarify the reasons of the other frequencies, we evaluated the bearing fault frequencies.

V. Bearing Defect Frequencies

At the drive end, a cylindrical roller bearing (Rollway U1313E) was installed as shown in Fig. 9. The main geometric dimensions are listed in Table II.

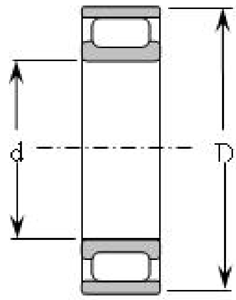

Fig. 9 The cylindrical roller bearing U1313E

The main three defects frequencies in the bearings are: cage defect frequency or train frequency FTF, outer race frequency ORF and inner race frequency IRF. They can be calculated as following:

Fundamental train frequency (cage frequency) FTF:

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$$FTF = \frac{n}{2}\left[1-\frac{B_d}{P_d}Cos\phi\right] = \frac{24.67}{2}\left[1-\frac{20}{115}Cos0\right] = 10.19 \text{ Hz}$$

Outer race frequency ORF:

ORF = N.FTF = 13×10.19 = 132.47 Hz

Inner race frequency IRF:

IRF = N.(n-FTF) = 13.(24.67-10.19) = 188.24 Hz

TABLE II
GEOMETRIC DIMENSION OF THE BEARING U1313E

D – Outer diameter	140 mm
d – Inner diameter	65 mm
B – width	33 mm
Pitch diameter P_d	115 mm
Roller diameter B_d	20 mm
Contact angle ϕ	0^o
No. of rollers n	13
Pump speed N	1480 rpm = 24.67 Hz

The measured bearing defects frequencies are normally deviated from the calculated ones, and this deviation can reaches several hertz in some cases [1]. As observed from the above calculations, the fundamental train frequency FTF doesn't appear on the gE frequency spectrum and it is rare to observe it. The outer race frequency ORF is 132.47 Hz and it is matched with the second peak at 124.37 Hz with difference of 8.1 Hz only. Hence, we concluded that there is a defect in the outer race of the bearing causing this spike. The inner race frequency IRF which is 188.24 Hz is close to the third peak of 199.37 Hz with difference of 11 Hz. Therefore, a conclusion of inner race defect also is claimed. Fig. 10 shows these frequencies with their relevant causes.

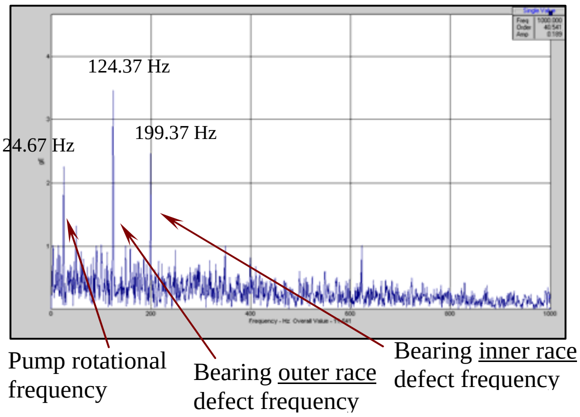

Fig. 10 Spike frequencies with their sources

Yet, we haven't achieved any conclusion for the beating phenomena, and still there are un-clarified peaks of 597.5 Hz and 622.5 Hz in the velocity frequency spectrum. A suspected harmonics of the races defects frequencies were most close possible reason. Table III represents the main two bearing defects frequencies and their multiple harmonics.

TABLE III
ORF & IRF DEFECT FREQUENCY AND THEIR HARMONICS

Defect	Fundamental frequency (Hz)	2X	3X	4X	5X	6X
ORF	124	248	372	496	620	744
IRF	199	398	597	796	995	1194

A quick inspection of the Table III shows that the 597.5 Hz is equivalent to the third harmonic (3X) of the inner race frequency, while the 622.5 Hz is equivalent to the fifth harmonic (5X) of the outer race frequency.

Moreover, several harmonics are appeared in the frequency spectrum. Fig. 11 shows the two frequencies of beating signal along with other harmonics. 4X and 6X of the outer race frequency are appeared at 497.5 Hz and 747.5 Hz respectively. The inner race frequency shows 4X at 797.5 Hz as well as the 3X at 597.5 Hz.

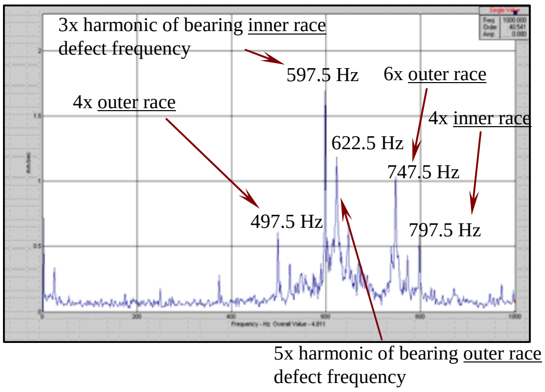

Fig. 11 Multi-harmonics of bearing defect frequencies

Hence, a recommendation of dismounting the pump and replace the bearings were sent to the maintenance group. Accordingly, the action was taken and the pump overhauled. We inspected the dismounted bearing to confirm the concluded results. Fig. 12 shows the bearing after dismounting. Clearly, dark marks appear inside the outer race and the rollers. A closer view on the roller is shown in Fig. 13. Several rollers were damaged and worn out.

Both outer and inner races were defective as shown in Fig. 14 and Fig. 15. The outer race has a lot of scratches and wear. The same thing could be said about the inner race.

Fig. 12 The bearing after removal. Note the marks in the rollers and the outer race

Fig. 13 The defects (dark marks) in the rolling elements of the bearing

Fig. 14 The defects (dark marks) in the outer race of the bearing

Fig. 15 The defects (dark marks) in the inner race of the bearing

VI. Conclusion

In this paper, we analyzed two cases, the beating phenomenon and the bearing defect frequencies. The former was observed in the time domain due to two close frequencies. They appeared in the frequency spectrum and calculations showed a consistent between the frequency domain and the time domain. The later – bearing defect frequencies – considered according to high spikes appeared in the frequency spectrum. The bearing defect frequencies were evaluated and compared with the measured data, which indicated defects in both inner and outer races of the bearing. Examining the multi harmonics of the fundamental bearing defects frequencies showed two frequencies close to each other, one is coming from the inner race, the other came from the outer race. These frequencies have the same magnitude of those generated the beating signal in the first case.

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