FAULT DETECTION IN DEEP GROOVE BALL BEARING USING FFT ANALYZER

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ABSTRACT

Bearing failure occurs due to heavy dynamic loads and also contact forces which exist between the bearing components. Bearing defects may be classified as localized and distributed. The localized defects include cracks, pits and spalls caused by fatigue on rolling surfaces. The distributed defects include surface roughness, waviness and misaligned races and off size rolling elements. The sources of defects may be due to either manufacturing error or abrasive wear. Hence, study of vibrations generated by these defects plays an important role in quality inspection as well as for condition monitoring of the ball bearing or machinery defects are developed into bearing, but unfortunately we cannot observe that defects by naked eyes in initial stage of failure. But when these faults are increased to large amount, it will lead to severe damage. So it is very necessary to detect faults in bearing at an earlier stage. But in large industries it is very difficult to remove bearings and check the faults. This problem can be resolved by equipment called as FFT analyzer. FFT analyzer is used to detect faults in various components without disturbing setting of that component.

***Keywords*: Rotating elements, Inner Race, Accelerometer, Envelope Analysis**

I. INTRODUCTION

Bearings are key machinery elements, whose failure without forewarning can damage the system to uncorrectable levels. In most cases, the cost of the bearing is not significant in comparison to the production losses caused due to unscheduled maintenance resulting from the bearing failure. This necessitates a robust diagnostic system for the bearings. A vibration-based method to detect and identify bearing damage is more common due to the ease in measurement, and the measured data can then be further processed in the time domain, frequency domain and time-frequency domain to extract useful information can be related to the severity and type of bearing damage [1]. Mechanical industries have gone through the significant changes in last decade. Competition, cost, and equipment complexity have increased while budgets, operating margins, and maintenance staffs have decreased. So, maintenance department must be able to show a positive effect on the "bottom line". Customers focus on product quality, product delivery time and cost of product. Because of these, a company has to develop or introduce quality and maintenance system. For medium scale industry annual maintenance cost is 2 to 3 crores.

If equipment's are not maintained properly then breakdown occur which results in different losses as production loss, loss due to accidents, parts replacement loss etc. Cost of these losses is more than 3 crores. So inline monitoring and offline tests of equipment's are necessary to maintain equipment properly. Vibration monitoring is generally the key component of most condition based maintenance programmers. Maintenance programmed must include other monitoring and diagnostic techniques. These techniques include: vibration analysis, corrosion analysis, lubricant analysis, process parameter monitoring, visual inspections. [2]

II. FFT ANALYSER

An FFT spectrum analyzer works in an entirely different way. The input signal is digitized at a high sampling rate, similar to a digitizing oscilloscope. The resulting digital time record is then mathematically transformed into a frequency spectrum using an algorithm known as the Fast Fourier Transform or FFT. The original digital time record comes from discrete samples taken at the sampling rate [2].

Fourier's basic theorem states that any waveform in the time domain can be represented by the weighted sum of pure sine waves of all frequencies. If the signal in the time domain (as viewed on an oscilloscope) is periodic, then its spectrum is probably dominated by a single frequency component. What the spectrum analyzer does is represent the time domain signal by its component frequencies. [3]

III. EXPERIMENTAL SETUP

3.1 Test Rig

The fully assembled test rig is initially placed with the AC three phase induction motor of Crompton Greaves having the maximum speed of 3000rpm and 0.5 HP. It is connected with the coupling having rubber bushes to avoid vibrations and shaft is joined and extended to 1meter and having a diameter of 20mm horizontally the power is transmitted. There are 2 disc of 15 Newton each and is inserted in shaft to achieve no load readings. Furthermore, two deep groove ball bearings are inserted in the shaft which is supported with the self-manufactured Split type housing, to minimize the efforts in mounting and dismounting of bearing. Moreover, the opposite side of shaft that is to which the pulley is connected to the dynamometer which is connected with the help of B-22 belt size.

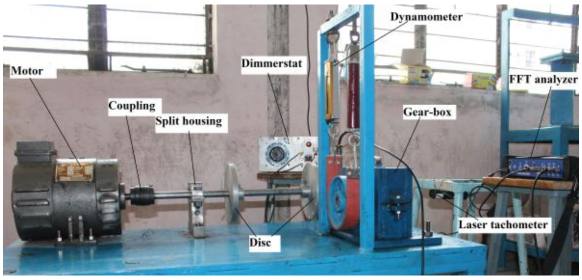


Fig. 3.1 Experimental setup

3.2 Instrumentation

Vibration signal from the tri axial accelerometer of sensitivity 5mV/g mounted on housing was passed to Fast Fourier Transform; the speed of the shaft is calculated by the laser Tachometer. Fast Fourier Transform is used for achieving desired graphs and results were of Time Domain and Frequency Domain. The readings and graphs of FFT were analyzed with the DDS software installed in PC. [1]

Fig. 3.2 Accelerometer

3.3 Measurement Condition:

The analysis of the vibrations was conducted on the various speeds of the motor that is 500, 750, 1000, 1250, 1500 rpm. From two of the bearing, the one nearest to the motor is kept unchanged that is good bearing and the other bearing placed forward was changed various times according to the defects taken(on inner race, outer race, ball) as shown in fig. 3.3 also the combination has been done(two defects on outer race, one outer race and one inner race). The different velocity and frequency graphs were obtained by making combinations of bearing and speeds.

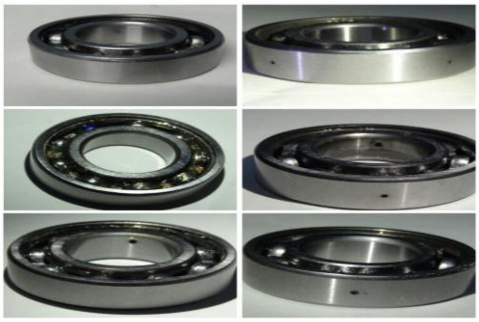


Fig. 3.3 Defects created on different bearings

IV. FREQUENCY DOMAIN ANALYSIS

The basic indicator is the characteristic defect frequencies in the frequency domain analysis. Spectral analysis of vibration signal is widely used in bearing diagnostics. It was found that frequency domain methods are generally more sensitive and reliable methods. The characteristic defect frequencies depend on the rotational speed and the location of the defect in a bearing. The existence of one of the defect frequencies in the direct or processed frequency spectrum is the main indicator of the fault. The interaction of defects in rolling element bearings produces pulses of very short duration whenever the defect strikes or is struck owing to the rotational motion of the system. These pulses excite the natural frequencies of bearing elements and housing structures. These frequencies depend on the bearing characteristics and are calculated according to the relations as shown below. [1]

Outer race frequency= $\frac{n}{2}\frac{N}{60}\left(1-\frac{d}{D}\cos\alpha\right)$

Inner race frequency= $\frac{n}{2}\frac{N}{60}\left(1+\frac{d}{D}\cos\alpha\right)$

Rolling element frequency = $\frac{D}{d}\frac{N}{60}\left[1-\left(\frac{d}{D}\right)^2\cos^2\alpha\right]$

Case frequency= $\frac{N}{120}\left[1-\frac{d}{D}\cos\alpha\right]$

n=Number of balls

α = Contact angle =0, for deep groove ball bearing.

D = pitch diameter

d = ball diameter

N= rotational speed in rpm [4]

4.1 Sample Analytical Calculations for of Characteristic Frequencies of Elements Deep Groove Ball Bearing

4.1.1 Specifications

Bearing number-6004

n=Number of balls=9

α = Contact angle =0 for deep groove ball bearing.

D = pitch diameter=31.496mm

d = ball diameter=6.35

N= rotational speed in rpm

4.1.2 AT 500 RPM

Outer race defect frequency= $\frac{n}{2}\frac{N}{60}\left(1-\frac{d}{D}\cos\alpha\right)$

$$=\frac{9}{2}\frac{500}{60}\left(1-\frac{6.35}{31.496}\cos 0\right)=29.93 \text{ Hz}$$

Inner race defect frequency = $\frac{n}{2}\frac{N}{60}\left(1+\frac{d}{D}\cos\alpha\right)$

$$=\frac{9}{2}\frac{500}{60}\left(1+\frac{6.35}{31.496}\cos 0\right)=45.06 \text{ Hz}$$

Rolling element defect frequency = $\frac{D}{d}\frac{N}{60}\left[1-\left(\frac{d}{D}\right)^2\cos^2\alpha\right]$

$$=\frac{31.496}{6.35}\frac{500}{60}\left[1-\left(\frac{6.35}{31.496}\right)^2\cos^2 0\right]=19.82 \text{ Hz}$$

Fundamental train frequency = $\frac{N}{120}\left[1-\frac{d}{D}\cos\alpha\right]$

$$=\frac{500}{120}\left[1-\frac{6.35}{31.496}\cos 0\right]=3.32 \text{ Hz}$$

Similarly we can calculate characteristic frequencies for different RPM like 750, 1000, 1250 and 1500.

There are two types of frequency domain analysis

1. Velocity spectrum analysis
2. Envelope spectrum analysis

4.2 Velocity Spectrum Analysis

This is one of the frequency domain analysis in which we are plotting the velocity v/s frequency graphs obtained by FFT transformer. The bearings, when defective, present characteristic frequencies depending on the localization of the defect. Defects in rolling bearings can be foreseen by be the analysis of vibrations, detecting spectral components with the frequencies (and their harmonics) typical for the fault. There are five Characteristic frequencies at which faults can occur. They are the shaft rotational frequency f_s, fundamental train or cage frequency FTF, ball pass Frequency inner race BPFI, ball pass frequency outer race BPFO, and the ball spin frequency BSF. The characteristic fault frequencies, for a bearing with stationary outer race, can be calculated by the above described formulas.

Fundamental defect frequencies depend upon the bearing geometry and shaft speed. Once we identify the type of bearing installed we can calculate the defect frequency ourselves. There is also a bearing database available in the form of commercial software that readily provides the value upon entering the requisite bearing number. There will be occasions when the calculated defect frequencies do not exactly match the bearing defect frequencies that appear on the vibration spectra. This is due to higher than normal thrust loads which cause the bearings to run at a different contact angle. These abnormal thrust loads can be caused by sources such as misalignment. Also not all bearing manufacturers use the same number of rolling element in a particular bearing size. The most common bearing problem is the outer race defect then inner race defect. It is very rare to see a fault at the bearings ball spin frequency or BSF [5].

4.2.1 Velocity Spectrum Analysis At 500 Rpm

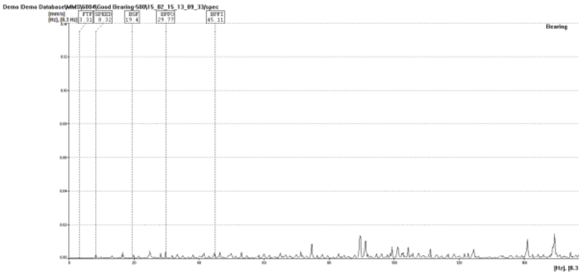


Fig. 4.1 Velocity v/s Frequency Spectrum for Non -Defective Bearing

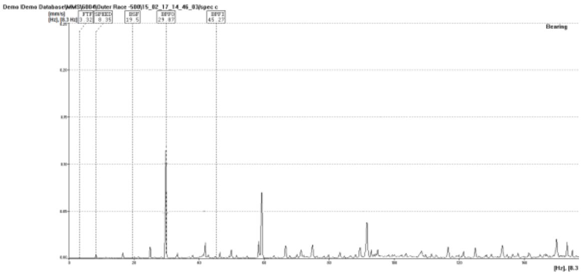


Fig. 4.2 Velocity v/s Frequency Spectrum for Defective Outer Race Bearing

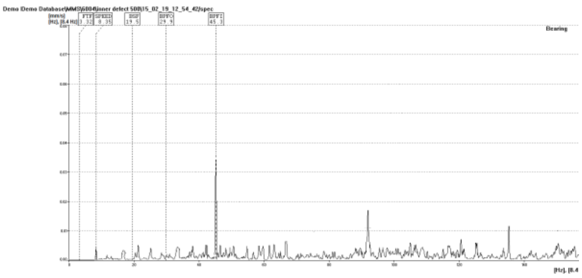


Fig. 4.3 Velocity v/s Frequency Spectrum for Defective Inner Race Bearing

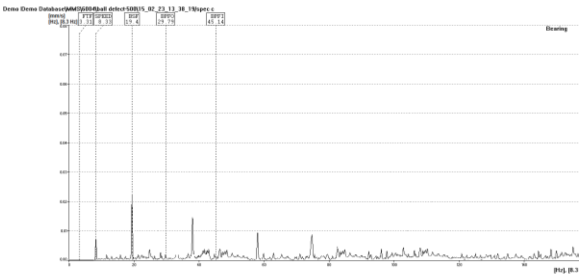


Fig. 4.4 Velocity v/s Frequency Spectrum for Bearing with Defective Ball

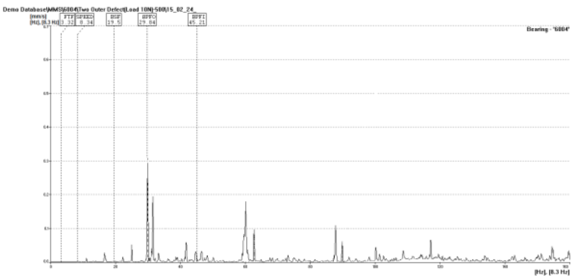


Fig. 4.5 Velocity v/s Frequency Spectrum for Two Outer Race Defects Bearing at External Load 10N

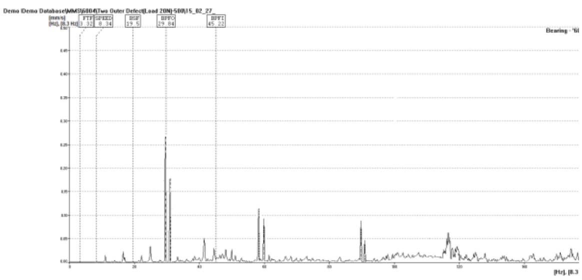


Fig. 4.6 Velocity v/s Frequency Spectrum for Two Outer Race Defects at External Load 20N

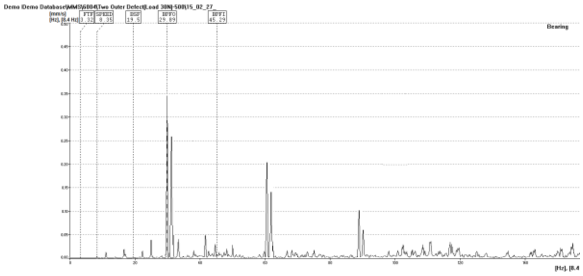


Fig. 4.7 Velocity v/s Frequency Spectrum for Two Outer Race Defects Bearing at External Load 30N

4.2.2 Result and Discussion

As we have already calculated the characteristic frequencies of all bearing element at 500RPM. In velocity spectrum analysis we have to just cross-check whether peaks are matching with marked values of frequencies in velocity spectrum. Fig. 4.1 shows that there is no any peak in the range of marked frequencies, so we can say that there is no any defect in bearing under consideration. From fig. 4.2 we can clearly observe that there is a peak at 30 Hz which is the characteristic frequency of outer race (BPFO) and its 2^{nd} harmonic i.e. at 60 Hz, so bearing under consideration have defect on its outer race. From fig. 4.3 we can see that peak is available at 45 Hz which is characteristic frequency of the inner race (BPFI) and its harmonic at 90 Hz, so we can conclude that there is defect at inner race. Similarly, in fig. 4.4 i.e. characteristic frequency of ball and its harmonic is matching with the BSF. Fig. 4.5, 4.6 and 4.7 shows the velocity spectrum of two outer race defects in which amplitude of the vibration is increasing continuously with increase in load so if we want to monitor the condition of bearing, we should apply some load on the bearing to get clear results.. So velocity domain analysis is very useful technique for finding out the location and severity of defect.

4.3 Envelope Analysis

The bearing frequencies are present throughout the spectrum (1/T line spectrum), but obscured at lower frequencies by other vibrations. However, there is a technique that makes it possible to extract the bearing frequencies from the part of the vibration spectrum where the 1/T line spectrum is dominant that is amplitude demodulation: A band-pass filter, with Centre frequency f_c, filters out the selected part of the spectrum; the output is shifted to low frequency and subjected to envelope detection. If the band-pass filter encompasses a range where the 1/T line spectrum is dominant, the resulting time history will be dominated by the envelope of the original pulse train. This envelope time history can now be subjected to FFT analysis for easy identification of Bearing Frequencies. [4]

Vibration signals become amplitude modulated due to periodic changes in forces. In case of rolling element bearing, periodic changes in forces occur due interaction of local defects on the surfaces of bearing elements. Modulated signals could be the product of the modulating signal with the carrier signal. The envelope or boundary of the amplitude modulated signal embeds the informative low frequency signal. The demodulation of the signal can be done using various Fourier transform techniques like Hilbert transform, short time Fourier transform etc.[2]

4.3.1 Envelope Spectrum Analysis at 1000 Rpm

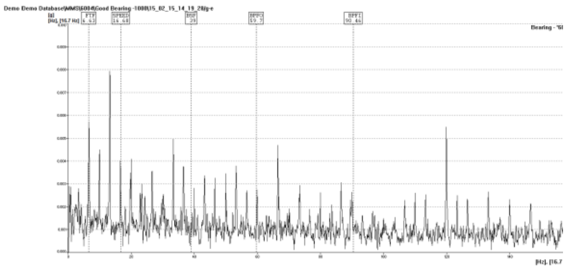


Fig. 4.8 Envelope spectrum for non-defective bearing

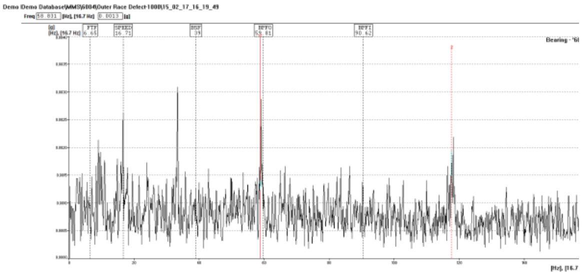


Fig. 4.9 Envelope spectrum for Outer race defect bearing

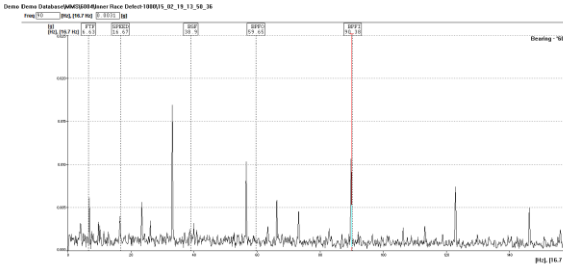


Fig. 4.10 Envelope Spectrum for Defective Inner Race Bearing

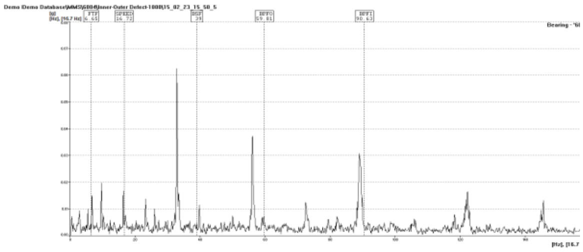


Fig. 4.11 Envelope Spectrum for Inner- Outer Race Defect Bearing

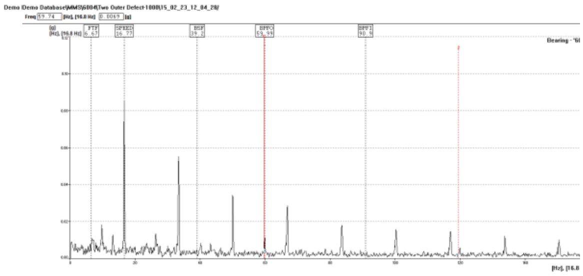


Fig. 4.12 Envelope Spectrum for Two Outer Race Defects Bearing

4.3.2 Result and Discussion

We have all theoretical characteristic frequencies of all elements. In envelope spectrum analysis also, we have to check whether we are getting peaks at marked frequencies. If peak is present, then there will be defect in that respective element. In envelope analysis we are getting harmonics of speed unlike in velocity spectrum analysis. Also harmonics of respective defects are also repeating frequently as compare to velocity spectrum analysis. From fig. 4.8 we observe that there is no any peak in range of marked theoretical characteristic frequencies so we can easily conclude that there is no any defect in the bearing under consideration. From fig. 4.9 we can clearly observe that peak is available at 60 Hz which is the BPFI at 1000 RPM. Then from fig. 4.10, we observed that there are peaks available at the marked frequencies of inner race. In fig. 4.11 we observed the peaks at BPFI, as well as at BFFO so we conclude that there are defects present at both inner race as well as outer race. From 4.12 we observe that there are 2 peaks in range of the BPFO, so it's clear that there are 2 defects present on outer race. So envelope analysis is an effective tool for finding out the location of defects in ball bearing.

V. SIGNATURE ANALYSIS

The condition of machines may be determined by measuring physical parameters like Vibration, Noise, Temperature, Wear etc. Change in these parameters called as signatures. Signature indicates change in machine condition.

If f = vibration frequency [cycles/min] or [Hz]

n = Bearing rotation speed [rpm]

There are different causes of vibration as below

When f = n

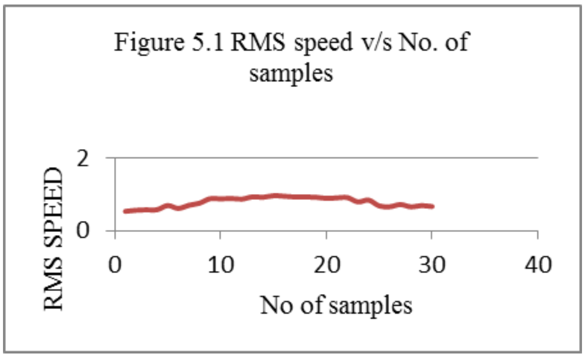
Unbalances in rotating bodies - Intensity proportional to unbalance, mainly in the radial

Direction, increases with speed.

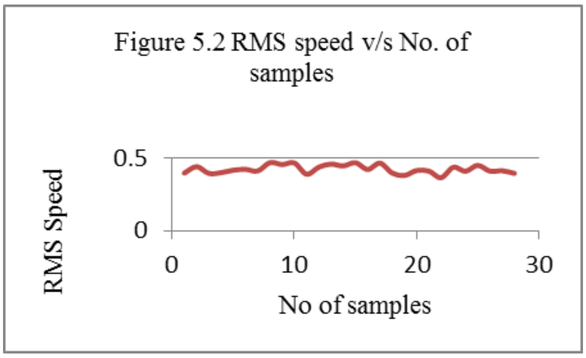
When f = n

Misalignment in the rotating bodies

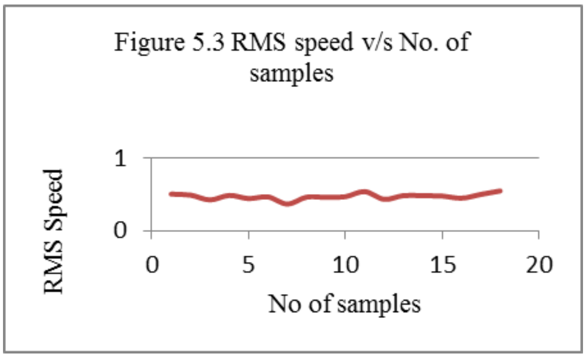
5.1 Rms Speed V/S No. of Samples at 750 Rpm



a. Non -defective bearing



b. Outer Race Defect



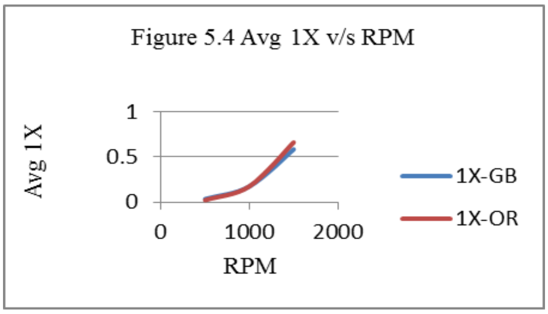
c. Inner Race Defect

5.1.1 Result and Discussion

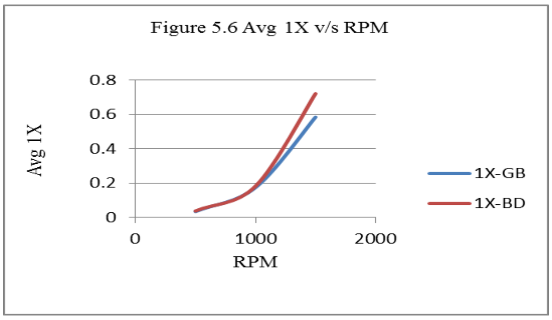
We observe from above graphs that at same speed the RMS values of velocity are changing due to unbalance in loads caused by vibrations or some other factors like external vibrations, misalignment etc. So we concluded that for condition monitoring of any bearing at any shaft speed should not be judged on single reading. Instead, we should take number of reading to normalize and get proper result. Moreover, care should be taken of component such as dimmer-stat which is abruptly fluctuating and other unknown disturbing parameters such as unbalancing, misalignment etc.

5.2 Comparison of 1x of Non -Defective Bearing

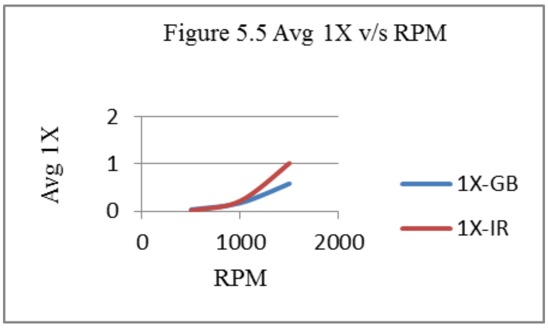
With Defective Bearing



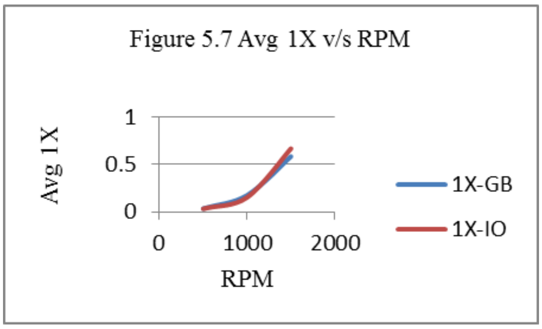
a. Non -Defective Bearing and OR Defect



b. Non -Defective Bearing and IR Defect



c. Non -Defective Bearing and Ball Defect

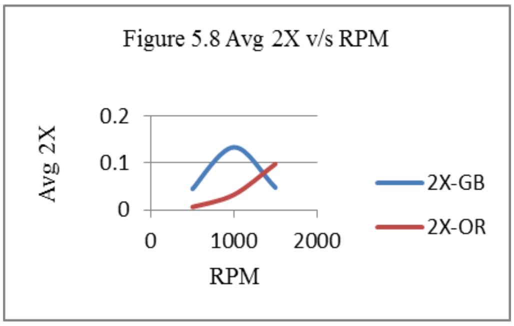


d. Non -Defective Bearing and Inner-Outer Defect

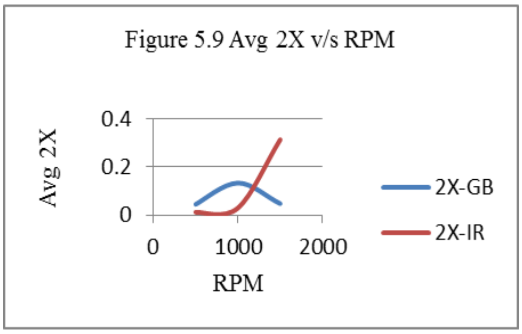
5.2.1 Result and Discussion

From above graphs we observe that 1X of all defective bearings is more than of non-defective bearing. So in any practical application, where we know the 1X v/s speed graph of non-defective bearing, we can easily identify the state of bearing by comparing with non-defective graph to check whether it is defective or non-defective without knowing specifications of bearing unlike in frequency domain analysis where we can't recognize defect without knowing bearing specifications. This method doesn't help to find the location of defect in ball bearing without knowing specifications of bearing and shaft speed.

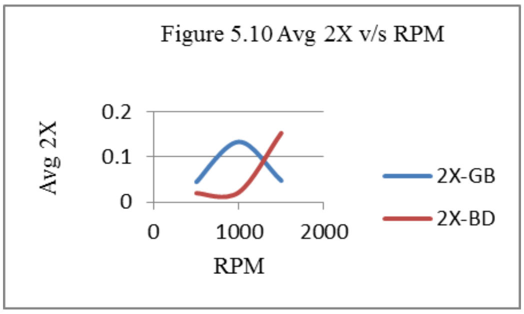
5.3 Comparison of 2x of Non-Defective Bearing with Defective Bearing



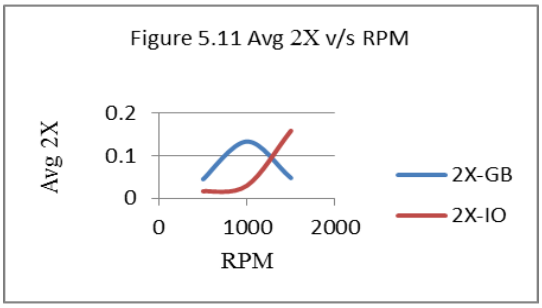
a. Outer race defect



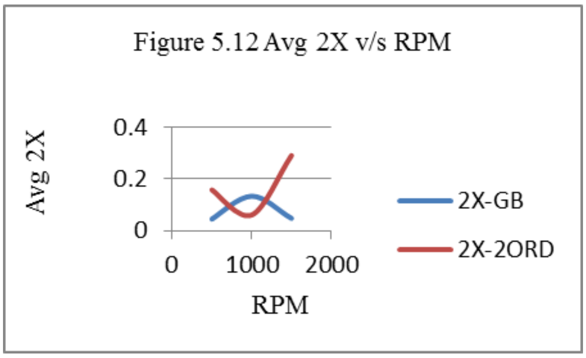
b. Inner race defect



c. Ball Defect



d. Inner-Outer Race Defect



e. Two-Outer Race Defect

5.3.1 Result and Discussion

From above graphs we can observe that 2X of all defective bearings are increasing continuously while for non-defective bearing this value is increasing and then decreasing. So in any practical application, where we know 2X v/s RPM graph of non-defective bearing, we can easily identify the state of bearing by comparing with non-defective graph to check whether it is defective or non-defective without knowing any specification of bearing unlike in frequency domain analysis where we can't recognize defect without knowing bearing specifications. This method doesn't help to find the location of defect in ball bearing without knowing the specifications of bearing and shaft speed.

VI. CONCLUSION

Velocity spectrum analysis is the effective tool for finding location of defect in ball bearing, also as radial load on bearing increases, the amplitude of the vibration increases. So by applying heavy loads we can easily find location of defect in ball bearing as compare to light loads. Envelope analysis is also very useful tool for detection of location of defect in ball bearing. The signature analysis is very useful tool for observing the change in vibration parameters like RMS velocity, 1X, 2X etc. with time. This analysis not useful to find location of defect in ball bearing. It's useful to find state of bearing whether it is defective or non-defective.

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