Identification of Faults in Self-Aligned Double Row Ball Bearing.

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Abstract - Bearing is a mechanical Device used in industrial and domestic applications. Bearing plays an important role in rotating machinery parts so more care need to take to design and to maintain good condition of the bearing. Bearing should be changed before actual failure in the rotating parts in order to avoid accident. This paper gives an idea about condition monitoring of bearing and an identification of faults occurred in bearing elements. Faults in bearing are detected by frequency spectrum of the vibration of the bearing. In these paper theoretical mathematical equations of bearing faults has used, that mathematical equations are solved by malt lab and results are plotted and that results of mat lab simulations are validated by experiment.

Keywords - *Fault detection, bearing signature, FFT Analyzer, Defective Frequency.*

1. INTRODUCTION

Ball bearing is the most basic component used in machinery for various engineering applications. Most of the engineering applications such as textile industry, agricultural industry and food and beverage industry use self aligned ball bearings, which enable rotary motion of shafts apart from complex mechanisms in engineering such as power transmissions, rolling mills and aircraft gas turbines. Self-aligning ball bearings have two rows of balls and a common sphered raceway in the outer ring. The bearings are insensitive to angular misalignment of the shaft relative to the housing. Self-aligning ball bearings generate less friction than any other type of rolling bearing, which enables them to run cooler even at high speeds.

As ball bearing is most commonly used component in machinery, it has received a great attention in the field of condition monitoring. Even a newly manufactured bearing may also generate vibration due to components running at high speeds, 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 [7].

The distributed defects include surface roughness, waviness, 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/machinery [2]. In order to prevent bearing failure there are several techniques in use, such as, oil analysis, wear debris analysis, vibration analysis and acoustic emission analysis. Among them vibration and acoustic emission analysis [8] is most commonly accepted techniques due to their ease of application. The time domain and frequency domain analysis [3] are widely accepted for detecting malfunctions in bearings. The frequency domain analysis is more useful as it identifies the exact nature of defect in the bearings. The effect of vibration on perfect bearing can be considerably reduced by selecting the correct preload and number of balls [9]. The vibration monitoring technique is used to analyze various defects in bearing and it also provides early information in case of progressive defects [8]. Radial vibration measurement was used to capture the signals and it was found that defect bearing has a strong effect on the vibration spectra [10]. In case of defect on the fixed ring the frequency spectrums generated will appears at its multiples. If the defect is located on the inner ring or the ball, frequency spectrum is amplitude modulated. The more is the wear, higher are the amplitudes of the components. Low speed fault simulation tests were conducted with various defects on the bearing. This study gives the best frequency bandwidth for early detection of bearing defects running at lower speeds [11].

2. LITERATURE REVIEW

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3. THEORETICAL CALCULATION.

These frequencies of the ball bearing depend on the bearing characteristics and are calculated from the relations shown below [4].

Frequencies associated with defective bearings are

$$\text{Ball} - \text{Pass frequency outer} = \frac{n}{2}\left(\frac{N_s}{60}\right)\left(1-\frac{d}{D}\cos\propto\right) \quad (2)$$

$$\text{Ball} - \text{pass frequency inner} = \frac{n}{2}\left(\frac{N_s}{60}\right)\left(1+\frac{d}{D}\cos\propto\right) \quad (3)$$

$$\text{Rolling element defect frequency} = \frac{D}{d}\left(\frac{N_s}{60}\right)\left(1-\frac{d^2}{D^2}\cos\propto\right) \quad (4)$$

$$\text{Cage frequency} = \frac{n}{2}\left(\frac{N_s}{60}\right)\left(1-\frac{d}{D}\cos\propto\right) \quad (5)$$

where NS is the rpm of the shaft, d is the mean diameter of the rolling element, D is the pitch diameter of the bearing, n is the number of rolling element, and α is the contact angle.

3.1 TEST BEARING SPECIFICATIONS:

The test bearing used double row deep groove ball bearing DFM –85 a special purpose bearing manufactured at D. J. R. Deluxe Bearings Ltd, Sanaswadi Pune.

TABLE NO.1

DIMENSIONS OF BEARING

Inner Bore diameter (Di)	30 mm
Outer Diameter (Do)	62 mm
Raceway width (b)	16 mm
Pitch Diameter (Dp)	45.5 mm
Ball diameter (DB)	8 mm
Number of balls (n)	14
Static Load Carrying Capacity (Co)	4.65kN
Dynamic load carrying capacity (C1)	15.6KN

TABLE NO.2

THEORETICAL BEARING DEFECT FREQUENCIES

Shaft Speed	Shaft Rotational Frequency	Defect Frequencies
N1 = 1090 rpm	(FR)1 = 18.66 Hz	(Fcage)1 = 7.48 Hz
		(FIRD)1 = 149.55Hz
		(FORD)1 = 104.83 Hz
		(FB)1 = 100.15 Hz
N2 = 1809 rpm	(FR)2 = 30.15 Hz	(Fcage)2 = 12.42 Hz
		(FIRD)2 = 248.16 Hz
		(FORD)2 = 173.94 Hz
		(FB)2 = 166.18 Hz
N3 = 3015 rpm	(FR)2 = 50.25 Hz	(Fcage)3 = 20.70 Hz
		(FIRD)3 = 413.60 Hz
		(FORD)3 = 289.90 Hz
		(FB)3= 276.96Hz

4. EXPERIMENTAL SETUP

The experimental bearing test rig is designed and fabricated to identify the presence of defects on a radially loaded self aligned ball bearing by vibration analysis technique is shown in Figure 1. The test rig consists of a 1.5 Hp motor connected with step pulley which is connected with another step pulley. Test bearing is mounted on shaft supported by two support bearings. Test bearing is loaded with dead weight.

Fig. 1. Experimental Setup

4.1 EXPERIMENTAL PROCEDURE.

Four Self Aligned Ball Bearings 1206 are used in the present study. One is a healthy bearing (assumed to be free from defects). Then defects through cracks were created on the inner race of second bearing, outer race of third bearing and on ball of forth bearing respectively as given in Table 6.3. The defect (crack) is 08 mm, 0.5 mm and 0.5 mm in width respectively for inner race, outer race and ball. Defect depth has varied as 1.2mm, 1.5mm, and 08 mm respectively for inner race, outer race and ball. These defects are created by laser machining in Asia Leaser Bhosari, Pune.

The tests were conducted for various load and speed conditions of 35 N, 45 N and 55 N. Speeds are being 1090 rpm, 1809 rpm and 3015 rpm.

This load and speed conditions are achieved by simple loading and step pulley.

Table No 3: Dimensions of Faults created on bearing

Fault	Length (mm)	Width (mm)	Depth (mm)
Inner Race	2	0.8	1.2
Outer Race	1.6	0.5	1.5

4.2 FAULT CREATED ON BEARINGS.

Fig. 2 Fault on inner race

Fig. 3 Fault created on outer race

5. RESULT AND DISCUSSION

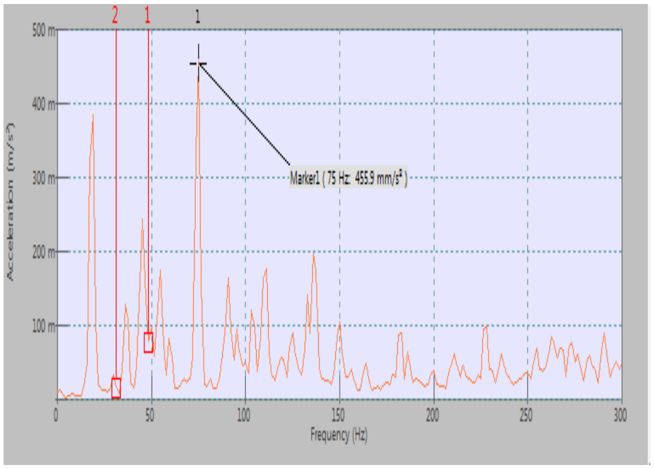

Fig .4 Spectrum of healthy bearing with 45 N load at 1090 rpm

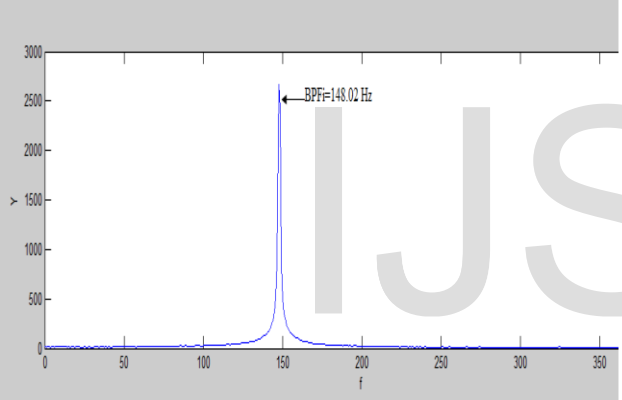

Fig. 5 Spectrum of MATLAB output of bearing with inner race defect with 45 N load at 1090 rpm

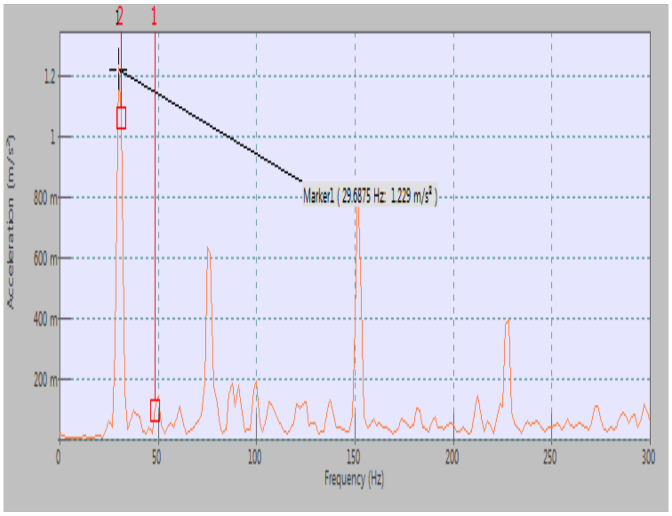

Fig. 6. Spectrum of bearing with inner race defect with 45 N load at 1090 rpm.

The frequency spectrum of the vibration signals with FFT analyzer from the inner race defect bearing is shown in Fig.6. It shows peaks at 29.68 Hz and 152.12 Hz. Amongst them first one is shaft rotational frequency (FR1) & second one is inner race defect frequency (FIRD1). The difference between estimated inner race defect frequency i.e. 149.55 Hz and experimental inner race defect frequency i.e. 152.12 is only 2.57 Hz, since the measured bearing defects frequencies are normally deviated from the calculated ones, and this deviation can reaches several hertz in some cases.

The MATLAB output of same defect shows this defect frequency (FIRD1) at 148.2 Hz, which varies by 1.35 Hz from theoretical and by only 3.92 Hz from measured one.

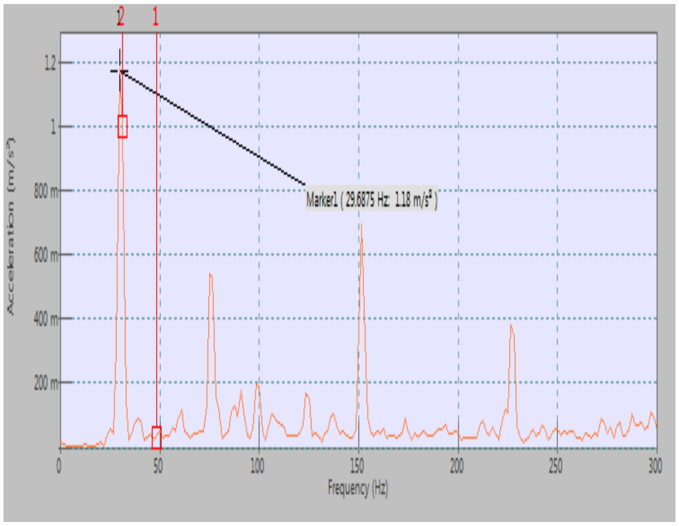

Fig. 7 Spectrum of bearing with outer race defect with 55 N load at 1809 rpm

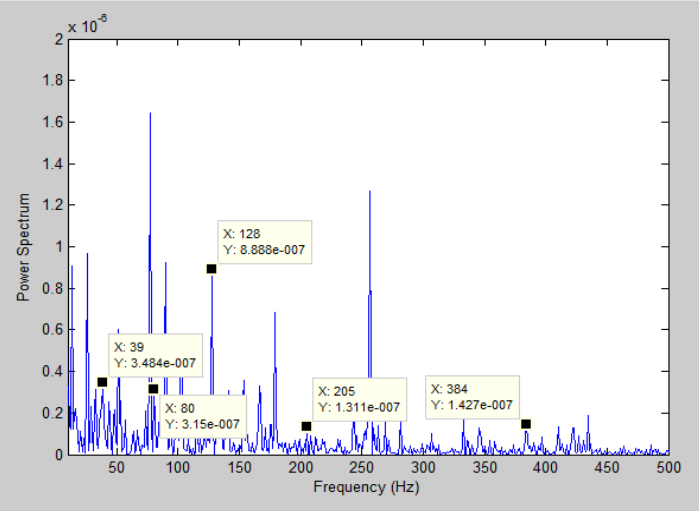

Fig. 8 Spectrum of MATLAB output of bearing with outer race defect with 55N load at 1809 rpm

The spectrum of the vibration signals from the outer race defect bearing at speed 1809 rpm is shown in Fig No 7. It shows peaks at 29.75 Hz and 239.92 Hz. Amongst them first one is shaft rotational frequency (FR1) and second one is outer race defect frequency (FORD1). There is difference 8.24 Hz between estimated (248.16 Hz) and experimental (239.92 Hz) for outer race defect frequency. The MATLAB output of same defect shows this defect frequency (FORD1) at 255.18 Hz, which varies by 7.02 Hz from theoretical and 15.26 Hz from measured one.

6. CONCLUSION-

As per experimental results, it has observed that peaks are generated at the characteristic frequencies. It is been observed that from the acquired graphs there is increase in amplitude as the defect size level increases For some faulty bearings

from Figure 6 and 7 the values computed from the frequency domain signals and amplitude of vibrations for new and defect bearings shows the location of the fault and severity of the defect. The value shows that as the load increases, the magnitude of vibration response also increases.

7. REFERENCES

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