Detection of Fault in Rolling Element Bearing using Condition Monitoring by Experimental Approach

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***Abstract*— Rolling element bearings are frequently encountered in rotating machine due to their load carrying capacity and low friction characteristics. The complexity of loading mechanism in bearing shows its effect in the form of distributed and local defects. Defects in bearing causes catastrophic failure and Hertzian contact stresses, which increases the vibration level in the machine. Efficient functioning of machine critically depends on good health of employed rolling bearings. Hence, health monitoring of rolling element bearing through their vibration responses is a vital issue. There are various techniques for vibration analysis such as wear debris analysis, oil analysis, temperature method and condition monitoring. Most of the work attempted in various papers was condition monitoring done by time domain and frequency domain method. Some of the work focused on time domain analysis, they compared stastical parameters of time domain signals with shaft speeds. In few papers the work focused on characteristic fault frequency of bearing in frequency domain analysis. In this dissertation work an experimental test rig is built to acquire vibration levels of defective bearing and also the frequency domain analysis is carried out by FFT analyzer to compare the vibration amplitudes on characteristic fault frequency of bearing. The effect of rotational speed, defect position and load on the diagnostics of rolling element bearing defects is investigated. Vibration signatures of these parameters are compared and sensitivity of vibration analysis is studied. Overall vibration increases in presence of local defects and dynamic radial loads.**

Keywords*— Condition monitoring, rolling element bearing, frequency domain analysis.

I. INTRODUCTION

Fault detection of the rolling element bearing, has been gaining importance in recent years because of its detrimental influence on the reliability of the machines. Vibration signature analysis is the most commonly used in fault-detection technique employed in bearing systems. Detection of the fault and its severity are two important steps for features of a condition monitoring system. Life of a machine component is determined by the severity of the fault in bearing. It is crucial, especially in critical systems, where continuous operation is generally unavoidable. In industrial applications, these bearings are considered as critical mechanical components and a defect in such a bearing causes malfunction and even lead to catastrophic failure of the machinery. Defects in bearings may arise during use or during the manufacturing process such as crack damage, spalling, corrosion, fatigue failure, etc. Therefore detection of these defects is important for condition monitoring as well as quality inspection of bearings. Bearing defects are classified into localized and distributed. The localized defects include cracks, pits and spalls caused by fatigue on rolling surfaces. The other categories of distributed defects include surface roughness, waviness, and misaligned races and off size rolling elements. These defects may result from manufacturing error and abrasive wear. Vibration monitoring of rolling element bearing is done by generally condition monitoring. Because condition monitoring is versatile technique, can detect almost all source of vibration. It includes use of FFT analyzer. Many researchers contributed in condition monitoring area. By reviewing some of research papers it was found that vibration analysis can be done by numerical and experimental approach. Mao Kunli, Wu Yunxin [1] focused on experimental approach to signal analysis. It describes the use of vibration measurements by a periodic monitoring system for monitoring the condition of rolling element bearing of the centrifugal machine. This paper proposed basic method of vibration diagnosis upon rolling bearing as frequency analysis. Different defect position such as outer race, inner race and ball has its corresponding fault characteristic frequency. Experimental set up consisted of motor driven shaft. That shaft was supported by two support bearing and one test bearing. Rotor was placed on motor driven shaft. Accelerometer to measure

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vibration signals were mounted at 4 different positions. Output of accelerometer sent to data acquisition system (DAQ). FFT analyzer processes data from DAQ in frequency domain. Bearing with defect present on outer race was analyzed. Graphs were plotted in frequency domain. After analysis of graph and comparing it with theoretical outer race pass frequency, peaks were observed. So vibration analysis can be able to detect location of fault. Using vibration diagnosis, such common bearing faults as crushing, crack, indentation, wear can be detected effectively. When ball passes through defect creates impact signal. By this impact source of fault can be detected. Bearing race pass frequency calculated by theoretical formula. K. Yavanarani1, G. S. Simon Sundara Raj[2] proposed wavelet-based vibration analysis for defense applications. Vibration signal recording was carried out on bearings of MSL engine test bed. The Integrated Circuit Piezoelectric (ICP) accelerometer was used to record data related bearing. Experimental set up consisted eight accelerometers (five uni-axial and three tri-axial) possessing high sensitivity and wide range of measurement. The Discrete Wavelet Transform (DWT) of the vibration signal computed using the Fast Wavelet Transform (FWT). Time domain and frequency domain data were recorded and compared. The Spectral Analysis using FFT provides information about the frequency content of the raw data but fails to provide time localization of the spectral components. So to overcome this drawback short term Fourier transform (STFT) and wavelet transform was employed and found that it can clearly detect multi resolution defect. Paper reported, it is possible to diagnose the state of operation or the evolution of faults in the bearing by observing the amplitudes of vibration and the frequency spectrum under different loading conditions. Prof. Dr. Zahari Taha, Nguyen Trung Dung[3] did signal analysis of single point defect by finite element analysis. To monitor the condition of a bearing, a SKF 1206 ETN9 self-aligning ball bearing is used in this study. The finite element method is adopted to observe the dynamic response of the structure. The housing and outer raceway structures are discredited into 71846 and 14091 4-node linear tetrahedron elements, respectively. The experimental setup consisted of lathe spindle on which test bearing was mounted. A defect is created intentionally on the outer raceway of the bearing with the size of 2 mm in diameter and depth using Electrical Discharge Machining (EDM). Accelerometer was mounted on test bearing and output signal given to DAQ system. Signals recorded in time and frequency domain analysis. If there is a peak at outer race pass frequency, it can be concluded that there is a defect on the outer raceway of the bearing. It is observed that in both simulation and experiment, the defect on the bearing can be detected. However, the squared magnitudes are different and the one in simulation is slightly larger. M Amarnath, R Shrinidhi, A Ramachandra, S B Kandagal[4] predicted vibration signal analysis of antifriction bearing. The test rig consists of a shaft with central rotor, which is supported on two bearings. An induction motor coupled by a flexible coupling drives the shaft. The cylindrical roller bearing is tested at constant speed of 1400 rpm with radial load of 230 N. Cylindrical roller bearing type with outer race and roller defects. Two-channel FFT analyzer was used to monitor vibration signals from good and defective bearings. Three types of bearing defects on inner race, outer race and roller defects were studied. Initially good bearing was fixed in the test rig and signals were recorded using FFT analyzer, shock pulse meter and spike energy analyzer. The good bearing was replaced by defective bearing and signals were recorded for each one of the case separately under the same standard condition. Time domain analysis and frequency domain analysis were carried out. Time waveform indicates severity of vibration in defective bearings. Frequency domain spectrum identifies amplitudes corresponding to defect frequencies and enables to predict presence of defects on inner race, outer race and rollers of antifriction bearings. R B Sharma, Yogesh Sharma [5] compared experimental results and CAE results of defective and defect free bearings. The experimental investigation for the mode shapes and the critical frequencies of the cracked bearing have been obtained from the experimental test rig setup and the mode shapes experimental step up. The bearing used for this work is of single cylinder SI engine of Herohonda 100 cc. the material of bearing was stainless steel. Experimental set up consisted of motor driven shaft o which test bearing was mounted. The bearing is mounted in between the DC motor and Eddy Current Dynamometer. DAQ card was provided for further signal analysis and Lab- View software was used to interpret signal in time domain analysis. A numerical simulation of the bearing is performed using ANSYS, and practical investigations were carried out to verify the proposed measurement approach. First mode shape and second mode shape were plotted and their behavior was observed. Defect type was triangular notch of 12mm each side and 14mm base side, depth of crack was 27mm approximately. The model is evaluated for the FEM analysis under the modal analysis head, to know the resulting fundamental frequency of the cracked crankshaft using ANSYS 14.0. The comparison is drawn in between the experimental investigation and the modal analysis using CAE methodology of the bearing with & without crack. The results obtained from the simulation are approximately validates by the experimental ones. This study is useful to provide the safety features against the catastrophic failures of bearing because the analysis of condition of bearing may be evaluated with the help of CAE based modal analysis using modeling the same model of bearing with defect as comparison with the size & nature of defect of bearing in practical real ground. The experimental and the simulation results were quite very closer to each other. Abhay Utpat [6] compared vibration signal by experimental and numerical approach. ANSYS is used for analyzing signals by numerical method. In numerical approach, whole bearing was modeled and imported in ANSYS. Analysis is carried out by assuming bearing as a mass spring model. Graphs were plotted in time and frequency domain analysis for defected and defect free bearings. In experimental approach, experimental set up developed with the help of belt drive arrangement. Shaft carrying test bearing and support bearing is driven by motor by belt and pulley arrangement. Support bearing were kept defect free. Only radial load is applied with the help of load cell. Vibration signal was acquired by piezo-electric accelerometer and output was fed to FFT analyzer for frequency domain analysis of signal. Defect free bearing, bearing with inner race defect and bearing with outer race defect were compared by experimentally and numerically. Further comparison is done by comparing FEA results and experimental results. Result analysis was carried out for various different factors such as comparison between defect

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free and defected bearing, comparison between defects present on outer race and inner race in frequency domain. During comparison of defected and defective bearing, change in peak amplitude was observed. From graph it was noted that vibration amplitude varies with change in defect position. Rise in vibration amplitude was observed due to which outer raceway defect found to be more severe than inner raceway defect for constant load and constant rpm. Also vibration signal varies with defect size for constant load and constant rpm. Vibration signal for different radial load but constant defect size and rpm did not show drastic change. U.A.Patel, Shukla Rajkamal [7] studied FEA model of defect free and defective bearing. Author's focus was on study of vibration signals of defective bearing by numerical method in ANSYS. In numerical method whole bearing assembly is modeled in PRO-E. Model was meshed in ANSYS with the help of tetrahedron 4-node SOLID element. First modal analysis was performed for application of load and then dynamic analysis was carried out by assuming bearing as spring mass model. Cylindrical defect on outer raceway was modeled. Resulted signals were plotted in frequency domain i.e. graph of amplitude of vibration versus frequency. Defect size selected 0.2mm^3. The natural frequency of system was around 34Hz. Experimental set-up consisted of shaft driven by motor. Shaft supported a rotor carried two support bearing and one test bearing. Accelerometer was mounted on test bearing which supplies signal to FFT gives signal in frequency domain data. Results of both experimental data and numerical data for defected and defect free bearing were compared. Author reported that finite element method can used to detect presence of fault. Vibration analysis is sensitive to presence of defect. Small difference was observed between experimental and numerical result due to complexity of modeling bearing is involved. Author also noted different values of damping coefficients at different shaft speeds and graph of amplitude of vibration versus damping coefficients is plotted. From graph it was found that, with increase in damping coefficients and increase in damping ratio amplitude of vibration decreases. So by varying different values of damping ratio and damping coefficients jerks in bearing can be eliminated. While different speeds with constant radial load and defect size vibration amplitude found to be increased. Hence, vibration signal analysis is sensitive change in defect position, change in rpm, change in defect size, presence of defect. Also, whole bearing model can be meshed in ANSYS for detail signal analysis in numerical method. In numerical method, meshing, selection of element while meshing is very important part during signal analysis. Overall vibration analysis is versatile technique as it can detect source of vibration. Condition monitoring can be effectively used to differentiate between signals for different defect sizes, different defect positions, and change in load. There are various approaches to vibration based condition monitoring which are experimental, numerical, theoretical and analytical.

II. METHEDOLOGY OF VIBRATION ANALYSIS

Vibration analysis works on principle that, whenever a local defect on element interacts with its mating element abrupt changes in the contact stresses at interface which generates pulse of very short duration [11]. The periodic impacts occur at ball-passing frequency that is characteristic defect frequencies, which can be estimated from the bearing geometry and the rotational speed. The vibration signal of a defected bearing produces impact when ball passes through defect. These defect frequencies are calculated with the help of the Fast Fourier Transform (FFT) spectrum.

A. Time Domain Analysis

Time domain technique is easiest and simplest technique to analyze the vibration signal waveform. Peak-to-peak amplitude is measure from the top of the positive peak to the bottom of the negative peak. Root mean square (RMS), measures the overall level of a discrete signal [7]is given by,

$$\text{RMS(x)} = \frac{x^2}{N} \qquad (1)$$

Where, N is the number of discrete points and represents the signal from each sampled point. The resultant RMS values are compared with recommended values to determine the condition of a bearing however, this method is not sensitive to detect small or early-stage defects. The crest factor is the ratio of peak acceleration over RMS. Value of the crest factor for good bearing is approximately five. At advance stages of material wear, bearing damage propagates, RMS increases, and crest factor decreases. Kurtosis is another important parameter to indentify the health of bearing. The equation to calculate the value of kurtosis is given by:

$$K = \sum_{n=1}^{N} \frac{[x(n)-\mu]^\wedge 4}{N.(\sigma^2)^\wedge 2} \qquad (2)$$

Where, x (n) is the time series, μ is the mean value of the data, σ 2 is the variance of the data and N is the total number of data points. A good surface finish bearing has a theoretical kurtosis of 3, and when the surface finish deteriorates value of kurtosis increases, but kurtosis is insensitive to loads and speeds. Kurtosis value, Crest factor, Impulse factor and Clearance factor are non-dimensional statistical parameters. Impulse and Clearance factors have similar effects like Crest factor and Kurtosis value. Impulse factor, Kurtosis value, Crest factor and Clearance factor are all sensitive to incipient fatigue spalling. Peak-to-peak value, RMS, Crest factor and kurtosis are only shows the damage at the ball bearing but do not give information about the location of defect e.g. inner race, outer race, cage or the roller.

B. Time- Frequency Domain Analysis

Time-frequency domain techniques have capability to handle both, stationary and non- stationary vibration signals. This is the main advantage over frequency domain techniques. Time–frequency analysis can show the signal frequency components, reveals their time variant features. A number of time-frequency analysis methods, such as the Short-Time Fourier Transform (STFT), Wigner-Ville Distribution (WVD), and Wavelet Transform (WT), have been introduced. STFT method is used to diagnosis of rolling element bearing faults. Local defects or wear cause periodic impulses in acoustic emission and vibration signals. The amplitude and period of these impulses are determined by shaft rotational speed, fault location, and bearing dimensions. The frequencies of these impulses, considering different defect locations [1]. If the shaft rotational frequency per

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second is Fs, the number of balls/rollers in the bearing is ‘Z’, the ball/roller diameter is‘d’, the pitch diameter of the bearing is ‘D’, and the contact angle is ‘β’ (β = 0 for a radial bearing), then the rotational frequency of a rolling element, fr and the rotational frequency of the ball cage with a stationary outer race, Fbcsor are expressed as

1) The frequency of rolling element bearing making contact with a certain point on inner race

$$f_i = \frac{z}{2} f_r \left(1 + \frac{d}{D}\cos\beta\right)$$

2) The frequency of rolling elements making contact with a certain point on outer race

$$f_o = \frac{z}{2} f_r \left(1 + \frac{d}{D}\cos\beta\right)$$

3) The frequency of rolling elements spinning around their own axes

$$f_b = \frac{f_r}{2}\frac{D}{d\cos\beta}\left(1 - \left(\frac{d}{D}\cos\beta\right)^{\wedge}2\right)$$

4) The frequency of cage spinning

$$f_c = \frac{f_r}{2}\left(1 - \frac{d}{D}\cos\beta\right)$$

From above formulae bearing race pass frequencies for outer race and inner race is tabulated below

Speed (rpm)	Fs	BPFO (Fod)	BPFI (Fid)
1800	30	105	165
2100	35	122.5	192
2400	40	144	220
2700	45	156	247
3000	50	175	275

Table 1. Bearing race-pass frequencies

III. EXPERIMENTAL APPROACH

Experimental set up consists of motor driven spindle on which two support bearings and one test bearing are mounted. Defects on bearing are created by EDM. Fig. 1 shows experimental setup. Most of researchers used belt and pulley arrangement for driving spindle on which test bearing is mounted. But it was found that belt and pulley arrangement will act as another source of vibration due to slight misalignment and this will corrupt vibration signal. Hence in this set up, gear coupling is used to connect motor shaft and spindle carrying test bearing. Load is applied with the help of spring thimble nut arrangement. One revolution of thimble nut spring gets compressed in radial direction corresponds to 212N. Load arrangement is done with the help of spring and thimble nut because load application should be easy and compact arrangement. VFD is used to get variety of speeds. The setting on VFD is done in such a way that 50 Hz corresponds to 1410rpm and 140 Hz corresponds to 4000rpm.VFD is chosen as it possesses advantages that it is simple, compact and posses flexible ranges of speeds. Also recent trend in industry is to use VFD. Given experimental set up includes accelerometer mounted on bearing housing as shown in fig. This accelerometer converts vibrating force into electrical signal. This signal is given to DAQ (Data Acquisition System) which processes given signal and FFT analyzer interprets signal in frequency domain waveform.

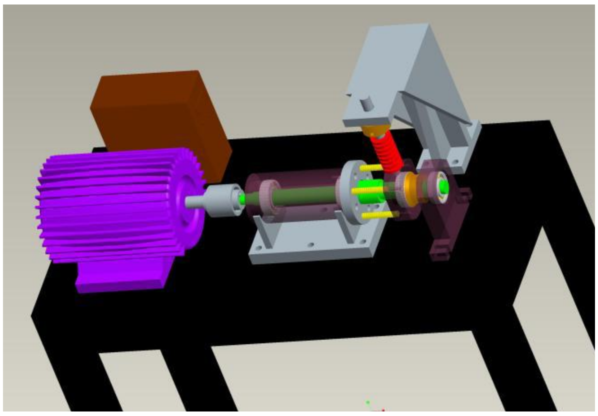

Fig. 1. Experimental set-up

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A set up consists of 1 HP at 1410rpm three-phase induction motor and output shaft which is mounted on table. Radial load is applied on test bearing by spring thimble nut arrangement. Test bearing is mounted in between two support bearing. The support bearings are defect free bearings and the test bearings are defective. A piezo-electric accelerometer with magnetic base is mounted on the housing of the test bearing. The accelerometer is connected to FFT Analyzer which processes the time signals. The output of analyzer is connected to computer which has the relevant hardware and the Lab-View software to acquire the data. Cylindrical defect created by EDM from 1mm to 3mm diameter sized defect is analyzed. Deep groove single row ball bearing is preferred. Load is varied from 424N to 724N while speed is varied from 1400, 1500 upto 2400 rpm. Focus of experimental approach was on frequency domain analysis.

IV. MEASUREMENTS OF VIBRATION AMPLITUDES

The support bearings are defect free bearings and the test bearings are defective. A piezo-electric accelerometer with magnetic base is mounted on the housing of the test bearing. The amplitudes of vibrations of test bearings are measured at different speeds, loads and defect sizes on outer ring as well as inner ring of bearings under following different conditions.

i) At constant speed & radial load with variations in defect sizes. ii) At constant defect size & radial load with variations in speed. iii) At constant defect size and speed with variations in radial load at test bearing.Signals are acquired in frequency domain and interpreted in proper format.

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V. RESULTS AND DISCUSSIONS

Experimental amplitudes of vibrations are measured at various speeds, loads and for different defect sizes on outer and inner ring of bearings. For defect present on outer race with defect size 2mm, 2.5mm and 3mm at 1800 rpm vibration amplitudes observed 0.005g, 0.0065g and 0.08g respectively. Fig. 1, fig. 2, fig. 3 shows vibration amplitudes when defect present on outer race defect present at load 424N with 1800rpm having defect sizes 2mm,2.5mm and 3mm. Like this different vibration amplitudes are measured by variation of defect sizes, defect position, speed of shaft which is tabulated in table 2. Graphs of some acquired signals are shown in fig.1-7. When defect is present on inner race with defect sizes of 2mm at 1500rpm, 1800rpm and 2400rpm vibration amplitude is noted as 0.0011g, 0.001g and 0.0005g as shown in fig. 6 and fig.7. While defect is present on inner race with defect size 2mm at 1800rpm load is varied from 424N, 530N, 636N and 742N and vibration amplitude noted as 0.001g, 0.0005g, 0.0004g and 0.00035g respectively. These frequencies are measured at characteristic fault frequencies or bearing race pass frequencies. Because principle behind vibration analysis is that , whenever a local defect on element interacts with its mating element abrupt changes in the contact stresses at interface which generates pulse of very short duration. The periodic impacts occur at ball-passing frequency that is characteristic defect frequencies, which can be estimated from theoretical formula mention in previous section. Efforts have been taken about measuring of above vibration signals with 2-3 different types of accelerometer. It was found that not much difference was there in vibration amplitudes. Slight changes may have occurred due to change in load zone.

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Sr. no.	Defect size Mm	Location of Defect Inner race/outer race	RPM	Defect Free New Bearing mm/s^2	
				Experimental	Theoretical
1	2	Outer race Force=424N	1800	Approx 104Hz with amplitude 0.005g	105Hz
2	2.5	Outer race Force=424N	1800	105Hz with amplitude 0.0065g	105Hz
3	3	Outer race Force=424N	1800	105Hz amplitude 0.08g	105Hz
4	2	Outer race Force=424N	2400	144Hz amplitude 0.08g	105Hz
5	2.5	Outer race Force=424N	2400	144Hz amplitude 0.012g	105Hz
6	3	Outer race Force=424N	2400	144Hz amplitude 0.082g	105Hz
7	2	Inner race Force=424N	1500	138-140Hz amplitude 0.0011g	137.5Hz
8	2	Inner race Force=424N	1800	165Hz amplitude 0.001g	165Hz
9	2	Inner race Force=424N	2400	220Hz amplitude 0.0005g	220Hz
10	2	Inner race Force=424N	1800	165Hz amplitude 0.001g	165Hz
11	2	Inner race Force=530N	1800	165Hz amplitude 0.0005g	165Hz
12	2	Inner race Force=636N	1800	165Hz amplitude 0.0004g	165Hz
13	2	Inner race Force=742N	1800	165Hz amplitude 0.00035g	165Hz
14	1	Outer race Force=424N	2400	144Hzamplitude 0.007g	144Hz

Table 2. Experimental vibration amplitude.

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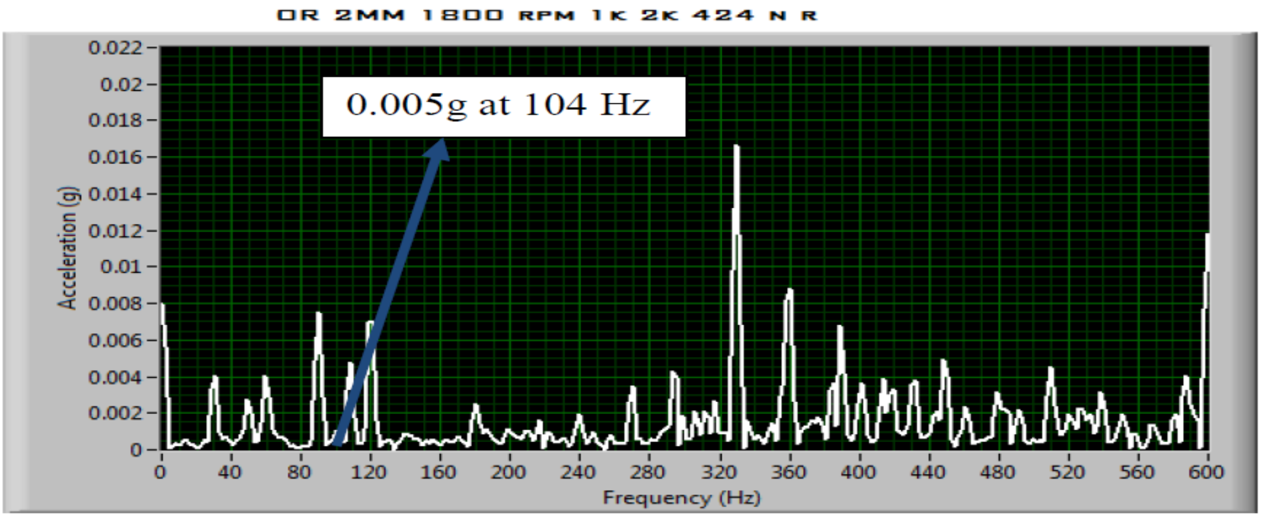

Fig.1 vibration amplitude when defect at outer race of 2mm with load 424N and speed of 1800rpm

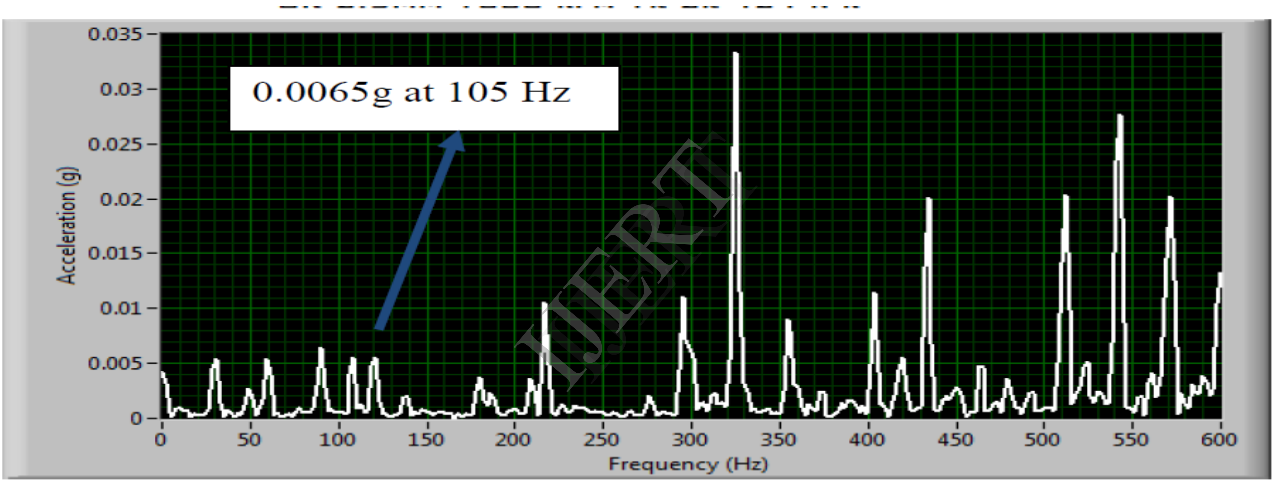

Fig. 2. vibration amplitude when defect at outer race of 2.5mm with load 424N and speed of 1800rpm

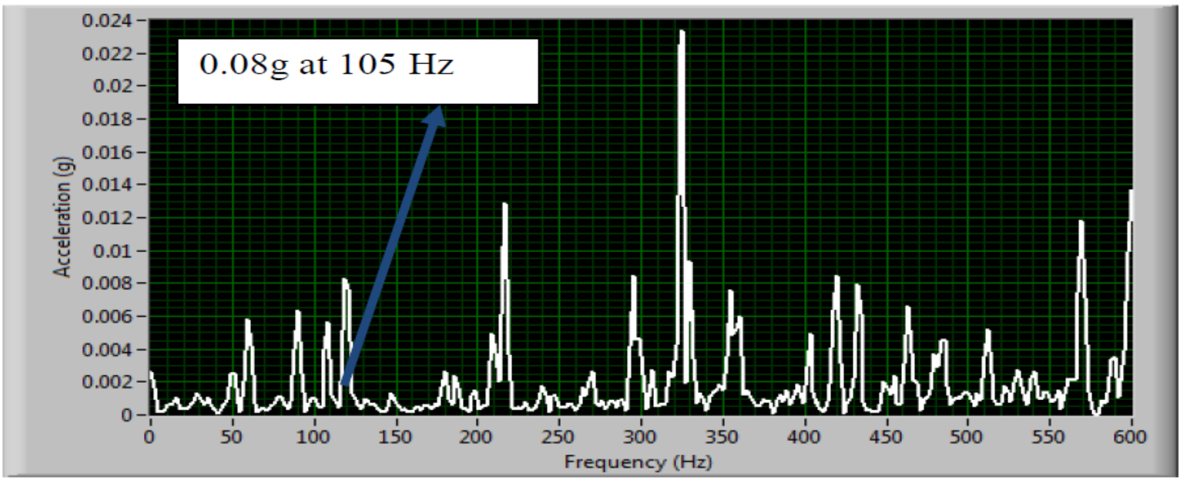

Fig.3 vibration amplitude when defect of 3mm at outer race with load 424N and speed of 1800rpm.

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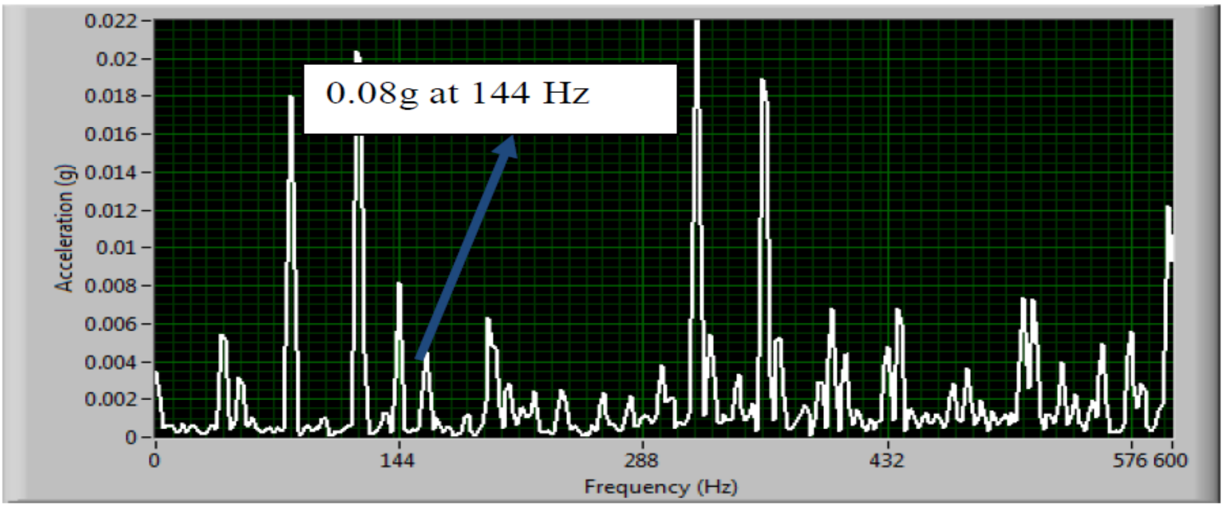

Fig.4 vibration amplitude when defect present at outer race of 2mm with load 424N and speed of 2400 rpm

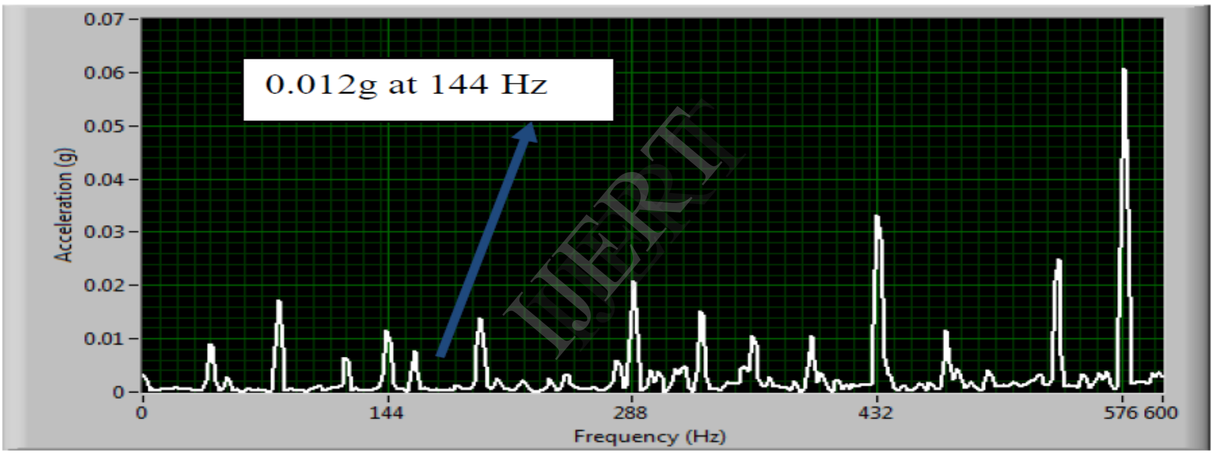

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Fig. 5 vibration amplitude when defect present at outer race of 2.5mm with load 424N and speed of 2400rpm

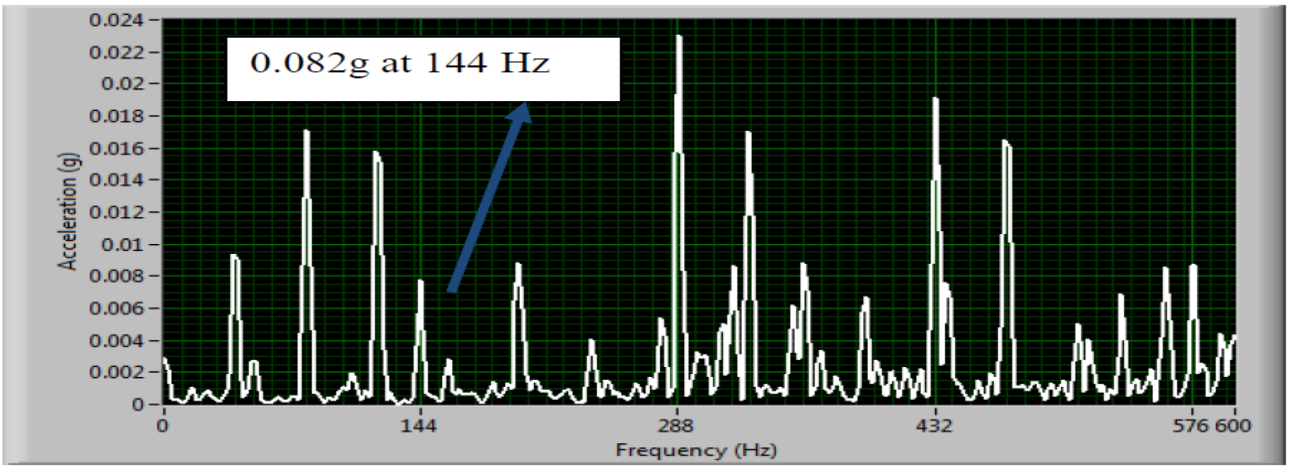

Fig. 6 vibration amplitude when defect present at outer race of 3mm with load 424N and speed of 2400rpm

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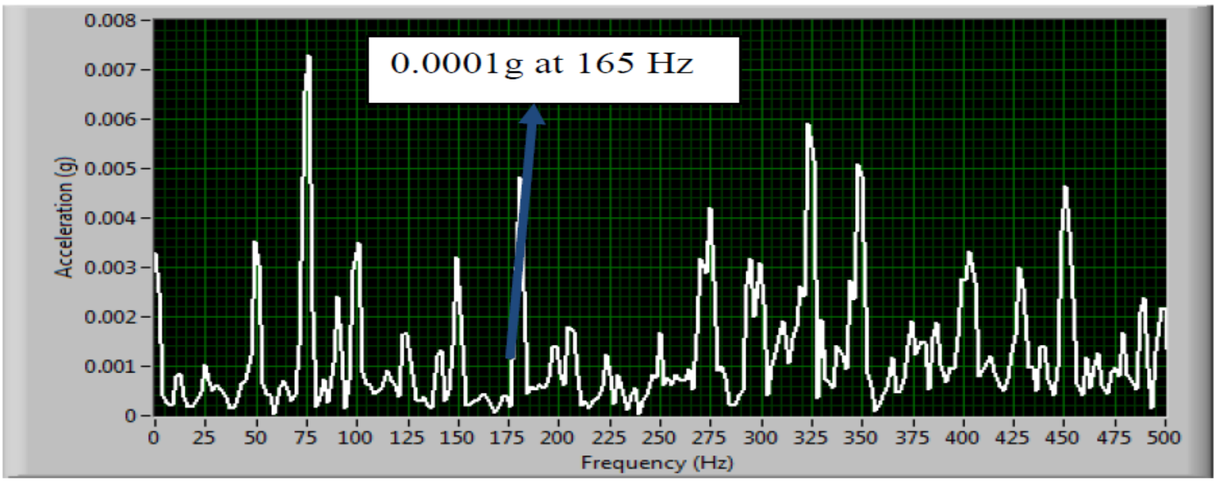

Fig. 7 vibration amplitude when defect present at outer race of 2mm with load 424N and speed of 1500rpm

CONCLUSION

When defect is present on outer race at constant force 424N and 1800 rpm vibration level varies with increase in defect size. It is observed that, vibration amplitude increases with increase in size of defect. When defect is present on inner race at constant speed 1800 rpm and defect size 2mm vibration level varies with change in load. . It is observed that, vibration amplitude decreases with increase in load. When at constant speed and constant load but defect positions are compared that is vibration amplitudes of defect present on inner race and outer race is compared, it is found that vibration amplitude at inner race is slightly less than that of outer race. Hence, outer race defect is more severe. Condition monitoring can be done by time domain and frequency domain technique. Time data graphs are acquired and observed. It was found that, data extraction from time domain signals is very difficult because it can detect presence of fault but unable to detect exact location. In frequency domain analysis vibration amplitudes can be compared at bearing race pass frequency. It was observed that, frequency domain analysis can be effectively used to detect the various sources of fault. Numerical approach of finite element analysis of bearing can successfully detect change in defect size of bearing by comparing values of energy dissipated. In some cases, finite element analysis is unable to provide rotation frequency to the bearing due to complex dynamic behavior. In experimental approach, proposed test rig effectively helps to compute sensitivity of vibration signatures for different source of vibration in bearing.

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