Experimental Investigation of Rolling Element Bearing with Unbalanced Rotor

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***Abstract*— Failure of rolling element bearing occurs due to manufacturing errors, improper assembly, overloading operation, bearing faults, misalignment, coupling wear, unbalance, ring wear and because of too harsh an environment. However, even if a bearing is perfectly made, assembled, it will eventually fail due to fatigue of the bearing material. Unbalance is a common feature of all rotational elements, results vibrations in the whole system. This study differs from the others due to the load carried by the bearing. In practice, the load applied is with some eccentricity and produces unbalanced force on bearing. This paper deals with the experimental investigation of the ball bearing health with and without unbalancing load. The vibration response from ball bearing due to the change in unbalancing mass, eccentricity for healthy and defective bearing with varying speed is found experimentally by using FFT analyser. The study investigated the variation of statistical parameters of vibration signals acquired from Ball bearings with respect to speed. The RMS value & peak value analysis validates that the ball bearing health can be fairly monitored using Time Domain Analysis. The method proves to be a simple, quick & cost effective method in the condition monitoring of ball bearings.**

Keywords— Unbalancing; FFT analyser; statistical analysis; defective ball bearing.

I. Introduction

Now a day, maintenance cost is one of the major operating costs in manufacturing companies. Maintenance cost involves spare parts inventory and manpower costs. Unexpected breakdowns, replacement and repair expenses from catastrophic failure results in loss of output. Adoption of predictive and preventive maintenance procedures significantly reduces losses from these causes and also improves the maintenance management and increases the product quality and cost. Predictive maintenance is based on continuous measurement of some machine operating parts, like temperature, power consumption, vibration, noise and forces are commonly employed for this purpose.

Generally machine fails due to bearing faults, misalignment, coupling wear, unbalance, ring wear, leakage etc. Studies show that bearing failures account for about 40% of all machine failures [14]. This is shown in a Fig.1. Ball and roller bearings are called as rolling element bearings and they are essential parts of rotating machinery. They are used to support load and allow relative motion of shafts in simple commercial devices such as bicycles, roller shaft, and electric motors. They are also utilized in complex engineering mechanisms including automobiles, aircraft gas turbine, in sugar factories, rolling mills and power transmission.

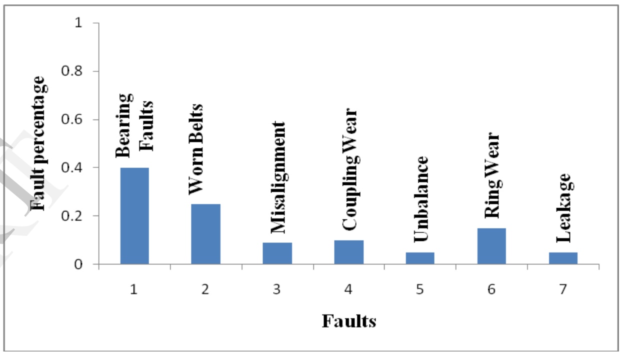

Fig. 1 Faults detected in a machine [14]

Zeki Kiral et. al. [1] has proposed a method based on the finite element vibration analysis for defect detection in rolling element bearings with single and multiple defects on different components of the bearing structure using the time and frequency domain. Hamdi Taplak et.al. [2] has proposed their work on Experimental analysis on fault detection for a direct coupled rotor-bearing system. From the spectrum and waveform graphs he found that the highest amplitude of vibration is obtained at speeds between three and four-multiplies of running speed. Feiyun Cong et. al. [3] has presented a rolling element bearing fault model based on the dynamic load analysis of a rotor-bearing system. The test rig design of the rotor bearing system considered several operating conditions: (a) rotor bearing only; (b) rotor bearing with loader added; (c) rotor bearing with loader and rotor disk; and (d) bearing fault simulation without rotor influence. Rupendra Singh Tanwar et. al. [4] has proposed their work on Fault Diagnosis of ball Bearing through Vibration Analysis. An experimental test rig is built to predict defects in antifriction bearings. He concludes that Radial direction has a minimum vibration, axial direction has medium vibration and downward direction has maximum vibrations exist on the bearing either it is good or defective. M.S. Patil et. al. [5] has proposed their work on A theoretical model to predict the effect of localized defect on vibrations associated with ball bearing. Pawan Kumar Singh et. al [6] has done the work on Study of effect of imbalance in rotor-bearing system due change in radius of gyration. M. Tiwari, K. Gupta[7] has proposed their work on Dynamic response of an unbalanced rotor supported on ball bearings. He conducted Experiments to investigate the effect of radial internal clearance and the unbalance force on the dynamic response of a horizontal rotor. N.Tandon et al. [8] reviewed vibration and acoustic measurement methods for the detection of defects in rolling element bearings. They considered both localized and distributed categories of defects. The results indicate that bad bearing has a strong effect on the vibration spectra. The effect of the defect size and unbalancing force has been investigated. A computer program is developed to simulate the defect on the raceways with the results presented in the time domain and frequency domain. Sometimes Finite element Methods, computer programs are used to simulate the analysis. Therefore it is important to analyze the effect of unbalancing on the defective bearings at its initial stage in order to prevent sudden failure of the machine. It is necessary to study the same parameters considered by above authors with different approach for the better agreement

This paper deals with the experimental investigation of the ball bearing health with and without unbalancing load. The vibration response from ball bearing due to the change in unbalancing mass, eccentricity for healthy and defective bearing with varying speed is found by using experimental investigation method.

II. EXPERIMENTAL INVESTIGATION

A. *Experimental Model Description*

The experimentation is carried out on a developed rotor-bearing test rig as shown in Fig. 2. The test rig is devised to study the dynamic behavior of rotor supported on healthy & faulty bearing with different loading conditions. SKF 6204 Ball Bearing is used for the study. The test bearing placed of non drive end of shaft. The drive to test rig is provided with DC motor through flexible coupling as shown in Fig. A continuously variable speed of 600 rpm to 1500 rpm is achieved from control panel of DC motor. The signatures of the vibration will be collected for change in various running parameters using the FFT analyzer which is available at SKN COE, Korti, Pandharpur, and has an accelerometer as a sensor. The sensor will be directly attached to test bearing housing. Vibration signals for healthy bearings & bearings with faults were obtained for various Speeds with different amount of unbalance.

The readings will be obtained in the form of displacement, velocity and acceleration. The test rig as shown in Fig. 2 consists of rotor supported on ball bearings & driven by DC Motor. The shaft is made up of mild Steel of grade EN8. It is driven by DC motor whose operating Speed can be controlled. A circular disk with holes is attached at the center of the shaft in between two bearings. The eccentric mass is varied by attaching number of bolts to the disc. The eccentric masses considered for analysis are 0.010 kg and 0.020 kg. These masses are added on the disk holes which are located at various eccentricities as 0.030 m and 0.040 m. One pedestrul for healthy bearing and Bearing bracket for test bearing are used to hold & align the bearings.

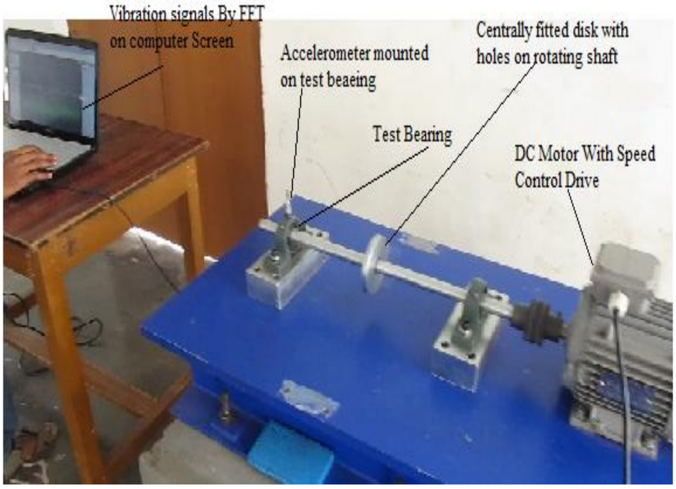

Fig. 2 Top and Side View of the Arrangement

B. *Introduction of a Defect in a Bearing*

The defect to the outer race of a bearing is produced by Wire Cut machining method. It consists of a 1 x 0.5 mm defect on the outer race of a bearing. In Wire Cut EDM, the conductive materials are machined with a series of electrical discharges (sparks) that are produced between an accurately positioned moving wire (the electrode) and the work piece. High frequency pulses of alternating or direct current is discharged from the wire to the work piece with a very small spark gap through an insulated dielectric fluid (water). The corresponding defect to a bearing is as shown in Fig. 3

Fig. 3 A 1 x 0.5 mm defect on the Outer Race

C. *Measurements of vibration amplitudes*

Initially the measurement of vibration amplitudes of healthy (defect free) test bearing is carried out for reference. Afterwards the defective test bearing (Defect size= 1x0.25) has been incorporated and amplitudes of vibrations are measured. Experimental readings were taken for different shaft Speeds, eccentricities and unbalance masses on the shaft, for test bearing. High frequency range accelerometer of 22.6 KHz capacity and sensitivity 10.1972 V/(m/s^2) is mounted on the test bearing housing. The sampling rate of 1500 Hz per channel is considered during the experimentation. For the same speed, load and defect size, the amplitudes of vibrations are confirmed after measuring repeatedly for 3 to 5 times. Following combination of experimental conditions i.e. cases are possible with healthy and defective bearing which are represented in a below table.

TABLE 1: CASES FOR EXPERIMENTAL RESULTS

Case	Description
I	Bearing Without unbalancing mass
II	Bearing With unbalancing mass(m)=0.010kg and eccentricity(e)=0.030 m.
III	Bearing With unbalancing mass(m)=0.010kg and eccentricity(e)=0.040 m.
IV	Bearing With unbalancing mass(m)=0.020kg and eccentricity(e)=0.030 m.
V	Bearing With unbalancing mass(m)=0.020kg and eccentricity(e)=0.040 m.

D. *Characteristic Defect Frequencies of Rolling Element Bearings*

When the rolling elements & the cage rotate with a constant rotational frequency of cage (Fc), a parametrically excited vibration is generated & transmitted through outer race. Depending on the working conditions, a localized defect may appear on different components in a rolling element bearing. Defect in the outer race, inner race & rolling element generate vibrations at distinct frequencies. Assuming no slip & outer race to be stationary, the general form of bearing defect frequency equations are given below [15].

Defect on the outer race:

$$F_{or} = \frac{ZN_s}{2x60}\left(1 - \frac{d_b}{d_p}\cos\alpha\right) \quad \ldots (1)$$

Defect on the inner race:

$$F_{ir} = \frac{ZN_s}{2x60}\left(1 + \frac{d_b}{d_p}\cos\alpha\right) \quad \ldots (2)$$

Defect on the rolling element:

$$F_{re} = \frac{N_s}{60}\frac{d_p}{d_b}\left(1 + \left(\frac{d_b}{d_p}\right)^2\cos^2\alpha\right) \quad \ldots (3)$$

Cage frequency:

$$F_c = \frac{N_s}{2x60}\left(1 - \frac{d_b}{d_p}\cos\alpha\right) \quad \ldots (4)$$

Theoretical calculation of the above listed frequencies at different Speeds (RPM) is shown in TABLE 2.

TABLE 2: THEORETICAL CALCULATION OF THE FREQUENCIES AT DIFFERENT SPEEDS

Speed (RPM)	Rotational frequency(Hz)			Defect frequency (Hz)		
	Fs or Fi	Fo	Fc	BPFO (F_{or})	BPFI (F_{ir})	F_{re}
600	10	0	3.8059	30.4477	49.5522	39.4869
900	15	0	5.7089	45.6716	74.3283	59.2304
1200	20	0	7.6119	60.8955	99.1044	78.9738
1500	25	0	9.5149	76.1194	123.8805	98.7173

III. RESULT AND DISCUSSIONS

The experimental results at different speeds, unbalancing masses, eccentricities and defect sizes are plotted. The acceleration spectra and results for amplitudes of vibration for each of the five cases for different shaft speeds are analysed. The statistical analysis of the time domain results is also carried out.

A. *Observed Spectra from FFT analyser*

For each case, the spectrum was recorded for 4 different shaft Speeds. The sample experimental result for amplitudes of vibration at 1200rpm for each of the five cases for test bearing with 1x0.25 mm^2 defect (width x depth) on outer ring is as shown in below Figures (Time domain and frequency domain). Figures show an acceleration spectrum as well as time domain signals for above mentioned cases.

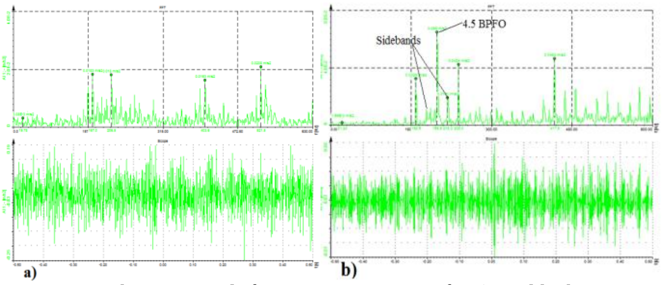

Fig. 4 Vibration signals for Case I at 1200 rpm for a) Healthy bearing b) Defective bearing.

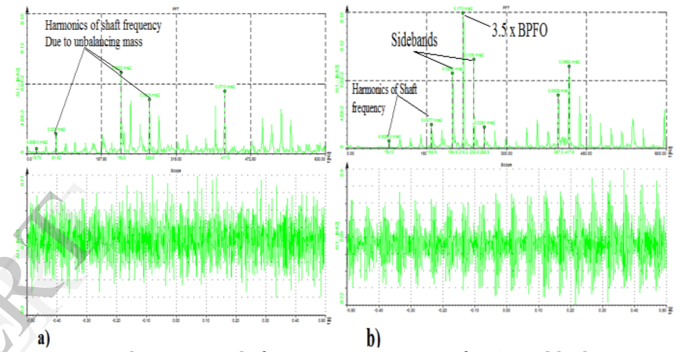

Fig. 5 Vibration signals for Case II at 1200 rpm for a) Healthy bearing b) Defective bearing.

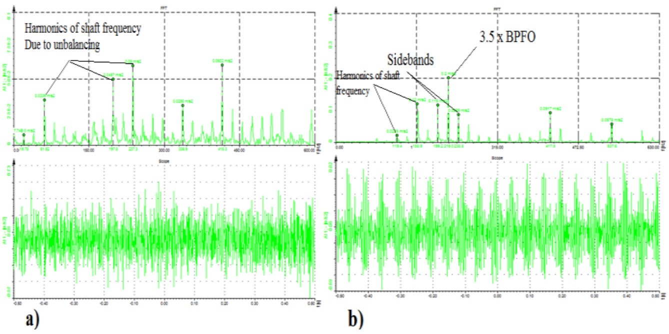

Fig. 6 Vibration signals for Case III at 1200 rpm for a) Healthy bearing b) Defective bearing.

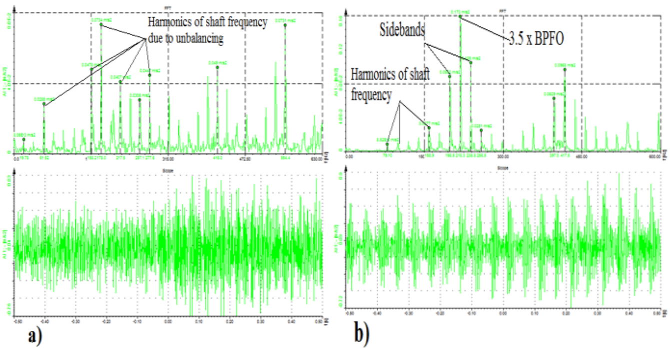

Fig. 7 Vibration signals for Case IV at 1200 rpm for a) Healthy bearing b) Defective bearing.

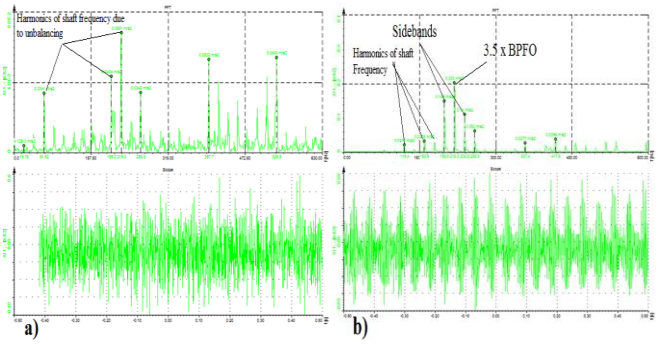

Fig. 8 Vibration signals for Case V at 1200 rpm for a) Healthy bearing b) Defective bearing.

It is seen from above FFT spectra that for defective bearing there is increased vibration amplitude at multiples of BPFO i.e. harmonics of BPFO than that of healthy bearing. These multiples of BPFO are decreasing with increase in speed i.e. for case II at 600 rpm it is (6 x BPFO), at 900 rpm it is (4.5 x BPFO), at 1200 rpm it is (3.5 x BPFO) and at 1500 rpm it is (3 x BPFO). The sidebands are also observed near the defect frequency which are located at (BPFP – Fs) and (BPFP – Fs) for respective shaft speed (Fs). It is also observed that there are increased vibration amplitudes at the harmonics of shaft speed (Fs) with increase in unbalancing mass from 0 kg to 0.02 kg and eccentricity from 0 m to 0.04 m.

B. *Inference from FFT vibration spectra*

It is observed from the time domain data collected experimentally, that it is very difficult to analyze the vibration signals. Therefore we converted these time domain vibration signals into frequency domain using FFT. Frequency domain signals shows peaks at various frequencies such as shaft frequency, BPFO, Cage frequency, Fundamental train frequency etc. These frequencies are different for different speeds and calculated in TABLE 3. Below table represents the peak amplitude at various speeds for defective bearing.

TABLE 3: MAXIMUM AMPLITUDES AT DIFFERENT SPEEDS

RPM	Maximum Vibration Amplitude (m/s2)				
	Case I	Case II	Case III	Case IV	Case V
600	0.0195	0.0484	0.0754	0.0491	0.0521
900	0.0478	0.128	0.142	0.136	0.143
1200	0.065	0.15	0.2	0.174	0.203
1500	0.116	0.201	0.224	0.221	0.267

Variation of Maximum vibration amplitude (m/s^2) with increase in Speed (RPM) and eccentricity (E) i.e. for the Case I, II and III are represented in Fig. 9.

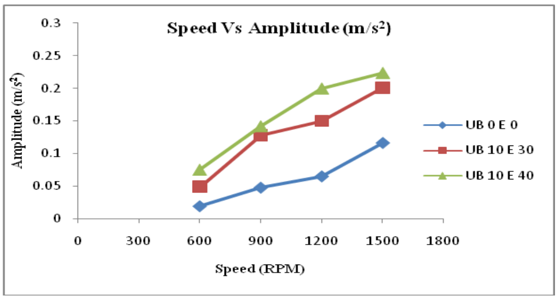

Fig. 9 Maximum Vibration amplitude Vs Speed for Case I, II and III

Variation of Maximum vibration amplitude (m/s^2) with increase in Speed (RPM) and eccentricity (E) i.e. for the Case I, IV and V are represented in Fig. 10

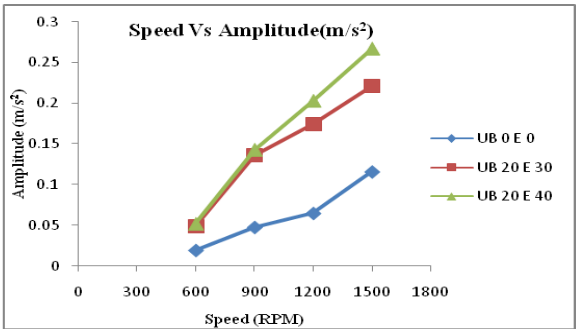

Fig. 10 Maximum Vibration amplitude Vs Speed for Case I, IV and V

It is observed from the above graphs that with increase in unbalancing mass and eccentricity (E) there is increase in vibration amplitudes.

C. *Statistical Analysis*

Vibration response from ball bearing due to the change in unbalancing mass (0.01 and 0.02 Kg), eccentricity (0.03 and 0.04 m) for healthy and defective bearings (defect size = 1x0.25 and 1x0.50 mm^2) with varying speed (600, 900, 1200, and 1500 rpm) are found by using experimental setup. The statistical analysis method was used because of its simplicity & quick computation. The following statistical variables with respect to Speed were used for analysis. [1]

1. *RMS Value Analysis*

For a dispersed data having N number of data points & X_m as an arithmetic mean, a root mean square value is defined as square root of sum of squares of all deviation values divided by Number of samples, where X_i is i^{th} data point.

$$\text{RMS} = \sqrt{\frac{\sum_{i=1}^{n}(xi - xm)^2}{N}} \qquad \dots (5)$$

2. *Peak Value Analysis*

For a dispersed data having N number of data points, a peak value is defined as half of the difference between maximum & minimum ordinate values.

$$\text{Peak value} = \frac{\text{Maximum ordinate} - \text{Minimum ordinate}}{2} \qquad \dots (6)$$

Change in statistical variables with respect to Speed for above considered five cases for healthy and defective bearing-

Case I: Without Unbalancing mass

In this case the unbalancing force considered is zero. The following statistical parameters are calculated for the readings got from experimentation. The data points of Peak Value, RMS value of vibration amplitudes (m/s^2) with Speed (RPM) for the Case I are given in TABLE 4.

TABLE 4: THE DATA POINTS FOR PEAK VALUE, RMS VALUE OF VIBRATION AMPLITUDE (m/s^2) WITH SPEED (RPM) FOR CASE I:

RPM	PEAK VALUE		RMS	
	Healthy	D 1x0.25	Healthy	D 1x0.25
600	0.2615	0.4349	0.0753	0.1408
900	0.3422	0.529	0.1038	0.132
1200	0.5227	0.9686	0.1629	0.3057
1500	0.8945	1.4904	0.2673	0.5644

Variation of Peak Value and RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case I are represented in Fig. 11.

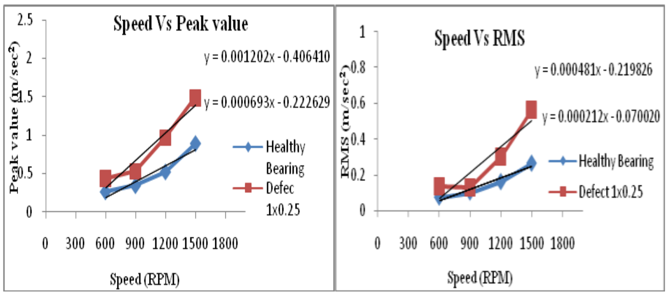

Fig. 11 Variation of Peak value and RMS value with Speed & their "Best Fit" for Case I

Case II: Bearings with unbalancing mass=0.010kg and eccentricity= 0.030 m.

In this case the unbalancing mass (m) =0.010 kg at eccentricity (e) =0.030 m is applied by using nut and bolt. The following statistical parameters are calculated for readings got from experimentation. The data points of Peak Value, RMS value of amplitude (m/s^2) with Speed (RPM) for the Case II are given in TABLE 5.

TABLE 5: THE DATA POINTS FOR PEAK VALUE, RMS VALUE OF VIBRATION AMPLITUDE (m/s^2) WITH SPEED (RPM) FOR CASE II:

RPM	PEAK VALUE		RMS	
	Healthy	D 1x0.25	Healthy	D 1x0.25
600	0.29851	0.39699	0.09292	0.1341
900	0.46486	0.55056	0.12325	0.251
1200	0.70312	0.86006	0.2447	0.3977
1500	0.9406	1.36526	0.25551	0.4886

Variation of Peak Value and RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case II are represented in Fig. 12

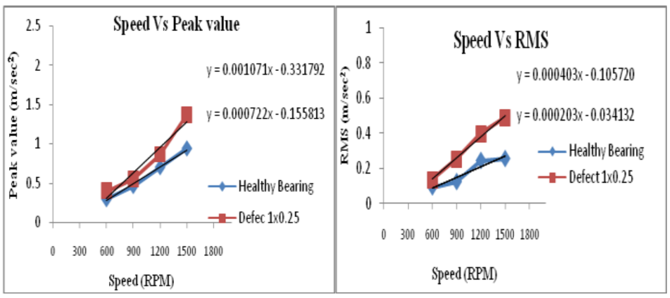

Fig. 12 Variation of Peak value and RMS value with Speed & their "Best Fit" for Case II

Case III: Bearings with unbalancing mass=0.010kg and eccentricity= 0.040 m.

In this case the unbalancing mass (m) =0.010 kg at eccentricity (e) =0.040 m is applied by using nut and bolt. The following statistical parameters are calculated for readings got from experimentation. The data points of Peak Value, RMS value of amplitude (m/s^2) with Speed (RPM) for the Case IV are given in TABLE 6

TABLE 6: THE DATA POINTS FOR PEAK VALUE, RMS VALUE OF VIBRATION AMPLITUDE (m/s^2) WITH SPEED (RPM) FOR CASE III:

RPM	PEAK VALUE		RMS	
	Healthy	D 1x0.25	Healthy	D 1x0.25
600	0.28139	0.4366	0.09104	0.1445
900	0.44003	0.89	0.13522	0.29078
1200	0.69266	1.4001	0.20896	0.45769
1500	0.98594	1.7537	0.29894	0.51914

Variation of Peak Value and RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case III is represented in Fig.13.

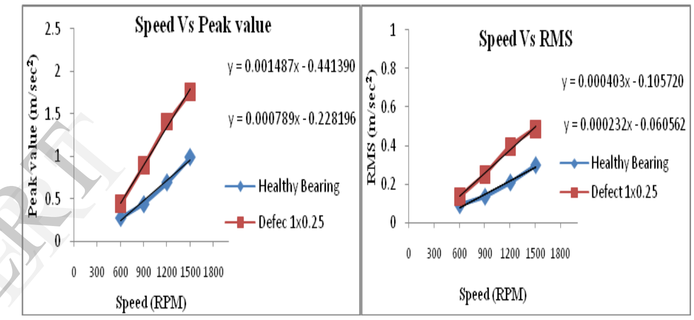

Fig. 13 Variation of Peak value and RMS value with Speed & their "Best Fit" for Case III

Case IV: Bearings with unbalancing mass=0.020kg and eccentricity= 0.030 m.

In this case the unbalancing mass (m) =0.020 kg at eccentricity (e) =0.030 m is applied by using nut and bolt. The following statistical parameters are calculated for readings got from experimentation. The data points of Peak Value, RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case IV are given in TABLE 7

TABLE 7: THE DATA POINTS FOR PEAK VALUE, RMS VALUE OF VIBRATION AMPLITUDE (m/s^2) WITH SPEED (RPM) FOR CASE IV:

RPM	PEAK VALUE		RMS	
	Healthy	D 1x0.25	Healthy	D 1x0.25
600	0.2818	0.37461	0.08162	0.09323
900	0.54354	0.6328	0.15479	0.18986
1200	0.96434	0.89039	0.31385	0.27698
1500	0.9848	1.4919	0.29599	0.49759

Variation of Peak Value and RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case IV are represented in Fig. 14

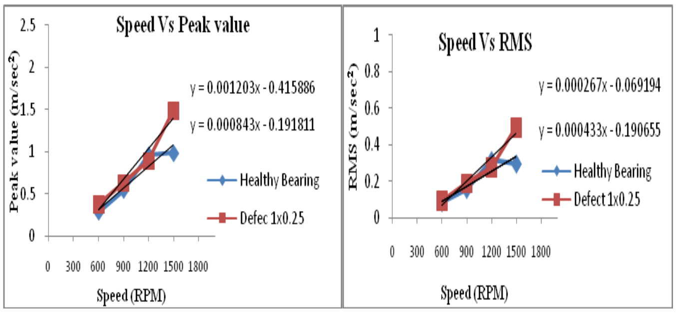

Fig. 14 Variation of Peak value and RMS value with Speed & their "Best Fit" for Case IV

Case V: Bearings with unbalancing mass=0.020kg and eccentricity= 0.040 m.

In this case the unbalancing mass (m) =0.020 kg at eccentricity (e) =0.040 m is applied by using nut and bolt. The following statistical parameters are calculated for readings got from experimentation. The data points of Peak Value, RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case V are given in TABLE 8.

TABLE 8: THE DATA POINTS FOR PEAK VALUE, RMS VALUE OF VIBRATION AMPLITUDE (m/s^2) WITH SPEED (RPM) FOR CASE V:

RPM	PEAK VALUE		RMS	
	Healthy	D 1x0.25	Healthy	D 1x0.25
600	0.29787	0.31838	0.0713	0.09379
900	0.41904	0.56721	0.11696	0.16358
1200	0.67804	0.93476	0.18579	0.2967
1500	1.00397	1.63092	0.23275	0.53697

Variation of Peak Value and RMS value of vibration amplitude (m/s^2) with Speed (RPM) for the Case IV are represented in Fig. 15

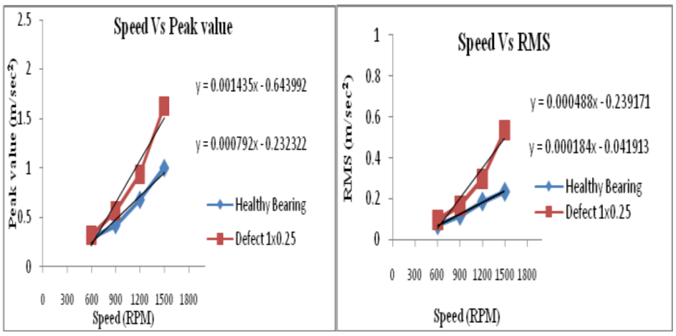

Fig. 15 Variation of Peak value and RMS value with Speed & their "Best Fit" for Case V

D. Inference from statistical analysis

It is seen from the graphs of Peak value Vs Speed (600 to 1500 rpm) and RMS value Vs Speed (600 to 1500 rpm) for all five cases for healthy as well as defective bearing that, as the speed increases the peak values gradually increases. The combined plot of the healthy and defective bearing shows that for defective bearing the statistical parameters like peak and RMS value are higher than the healthy bearing. The slope of the best fit lines drawn for defective bearing curves are greater than the healthy bearing and these slopes are gradually increasing with increase in unbalancing mass and eccentricity.

IV. CONCLUSION

From the experimental and statistical analysis carried out in the present work, following conclusions are drawn.

1. At constant speed and constant unbalancing load with introduction of defect on outer ring, amplitude of vibration increase and maximum values of amplitudes are observed at corresponding characteristics defect frequencies
2. At constant defect size and constant unbalancing load with different speeds of rotation, amplitudes of vibration increases with increase in speed. While the harmonics of BPFO of maximum amplitude of vibration are decreasing with increase in speed (RPM).
3. It is noticed that at constant defect size, constant speed and eccentricity of unbalancing mass with different unbalancing masses, the amplitudes of vibration slightly increases with increase in unbalancing load.
4. It is also noticed that at constant defect size, constant speed and unbalancing mass with different unbalancing mass eccentricities, the amplitudes of vibration slightly increases with increase in unbalancing eccentricity.
5. The study investigated the variation of statistical parameters of vibration signals acquired from Ball bearings with respect to speed. The RMS value & peak value analysis validates that the ball bearing health can be fairly monitored using Time Domain Analysis. The method proves to be a simple, quick & cost effective method in the condition monitoring of ball bearings.

During the experimentation, some non zero amplitudes are observed for unknown frequencies which may be due to influence of other source of vibrations in the test set up at higher speed.

The scope of this work is limited to measurement of amplitudes of vibration at the outer ring defects with unbalancing mass. Focus is not on the defects on the inner ring and balls in the bearing as vibrations of balls are controlled by misalignment, lubrication and ball slip.

ACKNOWLEDGMENT

The authors would like to acknowledge the Kuuality Engineers Ltd, Pune (India) for providing the test bearing samples required for the experimentation. The gratitude is also extended to Prof. S.S. kulkarni, Pandharpur and Prof. B.S. Gandhare, Pandharpur (India) for their valuable guidance during this work.

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