Theoretical and Experimental Studies on Vibrations Produced by Defects in Taper Roller Bearing using Vibration Signature Analysis

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***Abstract*–Rolling element bearings are indispensable mechanical components and defects in bearing unless detected in time may lead to malfunctioning of the machinery where so many methods are available for detection and diagnosis of the bearing defects. The research work presented focuses upon the detection of a localized defect in a taper roller bearing using vibration analysis. For bearing analysis, frequency domain approach is taken followed by experimentation for the detection of vibration signal in which single point defects are artificially created on roller and outer race, then for vibration response these defected taper roller bearings are tested under different speed and loading conditions, and resulting vibration signals are then processed using considered vibration-based techniques in frequency domain. Bearing vibration signature analysis is used as a medium for fault detection. Finally, these results are then compared to healthy bearing vibration response. The identification of the bearing defect is obtained by extracting characteristic defect frequency from vibration signal of defective bearing. Conclusions are drawn about the effective vibration monitoring of bearings.**

Keywords– Taper Roller Bearing, Vibration, Bearing Fault, Vibration Analysis, Fault Diagnosis.

1. INTRODUCTION

Early detection of bearing damage has raised quite a lot of interest among researchers and a significant amount of research has been carried out to the task of bearing condition monitoring. Vibration response measurement is an important and effective technique for the detection of defects in rolling element bearings. Even a geometrically perfect bearing under radial load may generate vibration due to varying compliance or due to time varying contact forces which exist between the various components of the bearing. However, the nature of vibration response changes with the presence of defects in various bearing elements [1].

In the previous investigations, the experiments on the vibrations of the normal tapered roller bearing have not been explained thoroughly and hence important to study the defects of taper roller bearing so as to avoid failures during the machine operations [2]. This study presents the results of the application of considered frequency domain vibration-based technique to real bearing vibration signals representing healthy and faulty conditions, and draws conclusions about their responses.

McFadden and Smith [3] reviewed the vibration monitoring of rolling element bearing by High-Frequency Resonance Technique. A detailed review on vibration and acoustic monitoring techniques for detection of defect in rolling element bearing is presented by Tandon and Nakra [4]. Tandon and Choudhury [5] presented a review on vibration and acoustic measurement methods for the detection of defects in rolling element bearings.

Patil M.S. [6] presented a review on vibration measurement in both time domain and frequency domain and intended bearing vibration signature analysis as a medium for fault detection. McFadden and Smith [7, 8] developed a model for the high-frequency vibration produced by single as well as multi-point defects on the inner race of a rolling element bearing under radial load. Tandon and Choudhury [9] proposed an analytical model for predicting the vibration frequencies of rolling bearings and the amplitudes of significant frequency components due to a localized defect on outer race, inner race or on one of the rolling elements under radial and axial loads.

The aim of this study is to apply multivariable regression analysis to the taper roller bearings for detecting defects. This work provides an in-depth study of vibration responses of taper roller bearings under single and multiple defect conditions in both roller and outer races.

2. BEARING DEFECTS

These are classified into two categories; distributed defects and localized defects [1, 6].

A. Distributed defects

Manufacturing error, inadequate installation or mounting and abrasive wear of bearings cause to distributed defects such as surface roughness, waviness, misaligned races and unequal diameter of rolling elements [15].

B. Localized defects-Fatigue of roller bearing causes defects like cracks, pits and spalls on rolling surfaces[13].

3. FREQUENCY-DOMAIN APPROACH

The frequency domain refers to the display or analysis of the vibration data based on the frequency. The time domain vibration signal is typically processed into the frequency domain by the application of Fourier transform, usually in the form of Fast Fourier Transform algorithm. The principal advantage of the method is that the repetitive nature of the vibration signals is clearly displaced as peaks in the frequency spectrum at the frequency where the repetition takes place [11, 18].

Whereas the defect in the rolling element strikes the rotation motion of the system, produces pulses of very short duration which ultimately excite the natural frequency of the bearing elements, resulting in the increase in the vibration energy at these high frequencies [5, 16]. With a defect on a particular bearing element, an increase in the vibration energy at this element rotational frequency calculated from the geometry of the bearing and element rotational speed [6].

i. $\mathrm{FTF} = \frac{1}{2} S \left(1 - \frac{B_d}{P_d} \cos\theta\right) \frac{rpm}{60}$

ii. $\mathrm{BPFI} = \frac{N_b}{2} S \left(1 + \frac{B_d}{P_d} \cos\theta\right) \frac{rpm}{60}$

iii. $\mathrm{BPFO} = \frac{N_b}{2} S \left(1 - \frac{B_d}{P_d} \cos\theta\right) \frac{rpm}{60}$

iv. $\mathrm{BSF} = \frac{P_b}{2B_d} S \left(1 - \left(\frac{B_d}{P_d}\right)^2 (\cos\theta)^2\right) \frac{rpm}{60}$

v. $\mathrm{RDF} = \frac{P_b}{B_d} S \left(1 - \left(\frac{B_d}{P_d}\right)^2 (\cos\theta)^2\right) \frac{rpm}{60}$

Where, S is the rotational speed of the shaft in rpm, $N_b$are number of rolling elements, B_d is Roller diameter in mm, P_d is Bearing pitch diameter and θ is Contact angle. Frequency domain techniques have ability to identify the location of fault(s) in bearing. Vibration peaks generates in spectrum at the bearing characteristic frequencies from which the defected bearing element can be detected easily [14, 17].

4. EXPERIMENTAL STUDY

In the following, the vibration signals of tapered roller bearings (SKF type 30208) are studied. The electrical-discharge machining method is applied to produce artificial defects on the surface of bearing components which are roller are outer race. The defect sizes and shapes are described in Table 1 and the description of the test rig is shown in Fig. 1. The vibration signals are measured on the housing of the test bearing by mounting an accelerometer has sensitivity 105.5 mV/g.

Fig. 1 Schematic of the test rig

Table 1 Defect in bearings

Defect type	Defect shape	Defect size Area mm^2 × depth mm
R_d	Circular	0.7853 × 1
		3.1415 × 2
		7.0685 × 3
	Square	9.0000 × 3
O_d	Circular	0.7853 × 1
		3.1415 × 2
		7.0685 × 3
	Square	9.0000 × 3

The measured direction is radial to the shaft. The tested bearings run at 800 and 1400 rev/min. The sampling rate of vibration signal is 25 kHz. According to the spectra of bearing vibrations, the frequency band from 3 to 5 kHz is chosen. According to the dimension of tested bearings, the characteristic frequencies for roller defect, outer- and inner-race defect are shown in Table 2.

Table 2 Characteristic frequencies for thebearing

Running Speed (rev/min)	T_f		E_f	
	O_f (Hz)	R_f (Hz)	O_f (Hz)	R_f (Hz)
800	96.33	86.88	95.21	92.77
1100	132.44	119.46	129.3	117.1
1400	168.56	152.04	168.4	141.6

5. MULTI VARIABLE REGRESSION ANALYSIS

Due to failure of traditional research for indicating the compounding effects of multiple defects and their interactions with various test parameters of vibration responses, such as speed, load, etc., a more holistic approach demands a shift from single to multiple factor analysis capable of assimilating data from multiple parameters. Multi Variable Regression Analysis (MVRA)[12] is accepted as a sound method for drawing inferences from observations, when inferences are not exact but subject to variation.

The present study aims at identifying the effect of various multiple defects. MVRA enables the identification of the effects of various localized bearing defects on vibration responses and their interactions between them. Severe vibration conditions are determined by approximating vibration amplitudes using a series of polynomials of the

MVRA. In Table 3, the values of these parameters along with some more parameters are listed.

Table 3 Factors and levels selected for the experiments

Parameters		Levels		
		L-1	L-2	L-3
Speed, N	*rpm*	800	1000	1400
Defect size	mm^3	1	2	3
Load, W	N	500	1000	1500

MINITAB-16 software is employed to design 9 experiments as listed in Table 4. Three independent variables (p = 3) are taken as,

Table 4 DOE and test results

Expt. No	*N*	*W*	*Od*	*Rd*	*Amplitude* (m/s^2)
1	800	500	1	1	0.0189
2	800	1000	2	2	0.0196
3	800	1500	3	3	0.0172
4	1100	500	2	3	0.1140
5	1100	1000	3	1	0.2180
6	1100	1500	1	2	0.0259
7	1400	500	3	2	0.2220
8	1400	1000	1	3	0.0507
9	1400	1500	2	1	0.0707

The final regression formula obtained for vibration amplitude may be expressed as,

$$\textbf{\textit{Amplitude}} = \mathbf{-0.093 + 0.000160N + 0.00115W - 0.0137Od - 0.0430Rd} \quad (1.1)$$

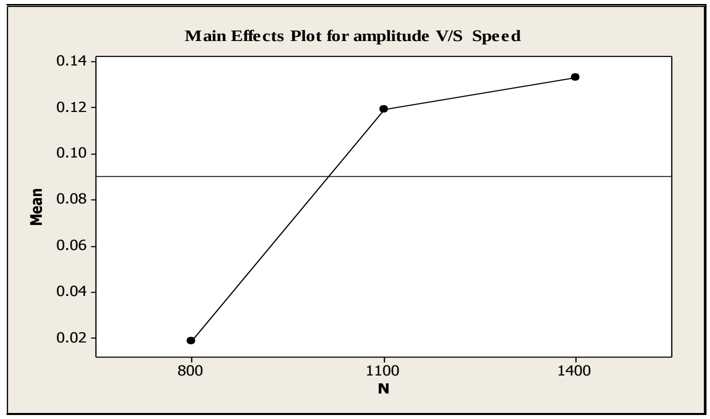
The F-value of 1.53 implies that the model is significant and values of p less than 0.05 indicate that the corresponding model terms are significant. In the present investigation, N, O_d, and W may be considered as significant terms. The values of p greater than 0.1000 indicate that the terms with those parameters may not be significant.

The significant terms in the model are found by an analysis of variance at 5 percent level of significance and a 95 percent confidence level. The goodness of the fit is indicated by the closeness of the value of R^2 to unity. The value of R^2 being 0.9720 implies that the model explains with 97.20 percent variability in vibration amplitude.

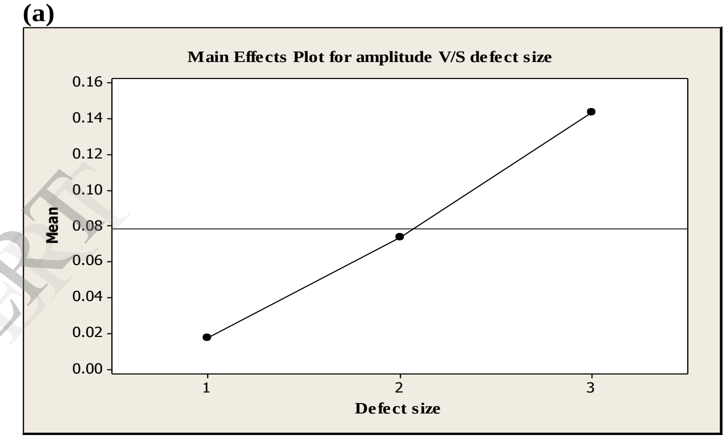
Table 5 Variance analysis for the model of the amplitude

Parameter	*DF*	*SST*	*MS*	*F*	*p*
N	2	0.019	0.0096	1.53	0.039
W	2	0.022	0.0110	1.75	0.036
O_d	2	0.001	0.0006	0.10	0.010
Error	2	0.012	0.0063		
Total	8	0.015			
$\mathbf{R^2}$=97.20					

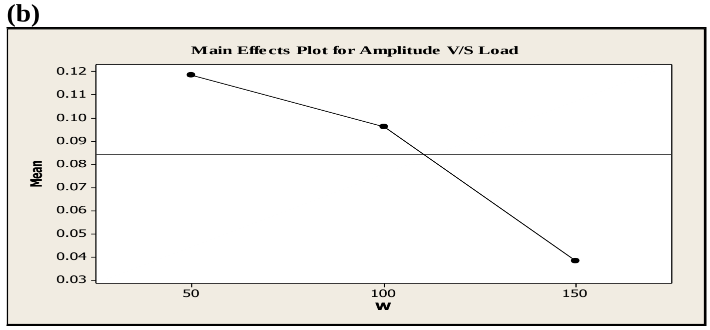
It is to be observed from Fig. 2 (a) and (b) that the amplitude is having an increasing trend with the speed and defects size as well as from Fig. 2 (c) seen that the amplitude is showing a decreasing trend with increase in load. All these are agreeing with physical expectations.



(a)



(b)

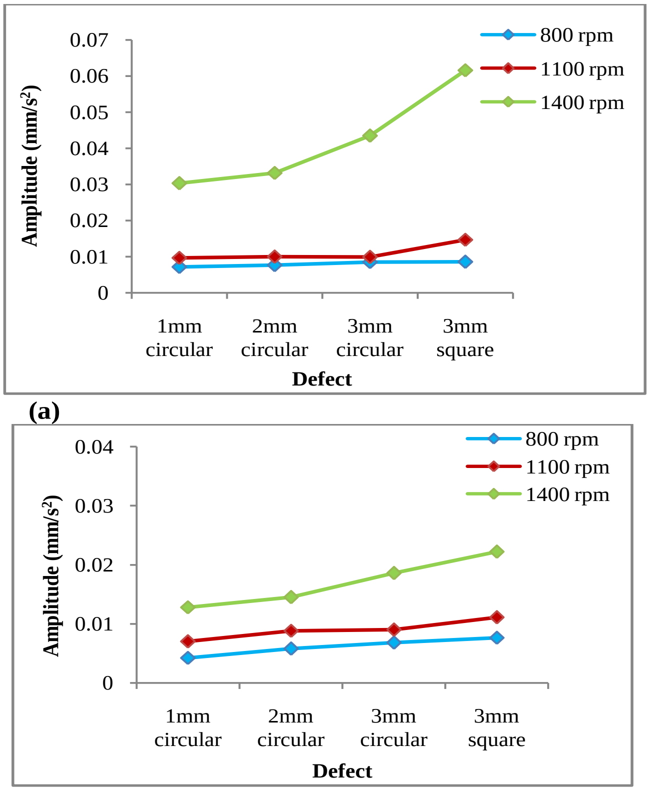


(c)

Fig. 2 Effect of various factors on vibration amplitude

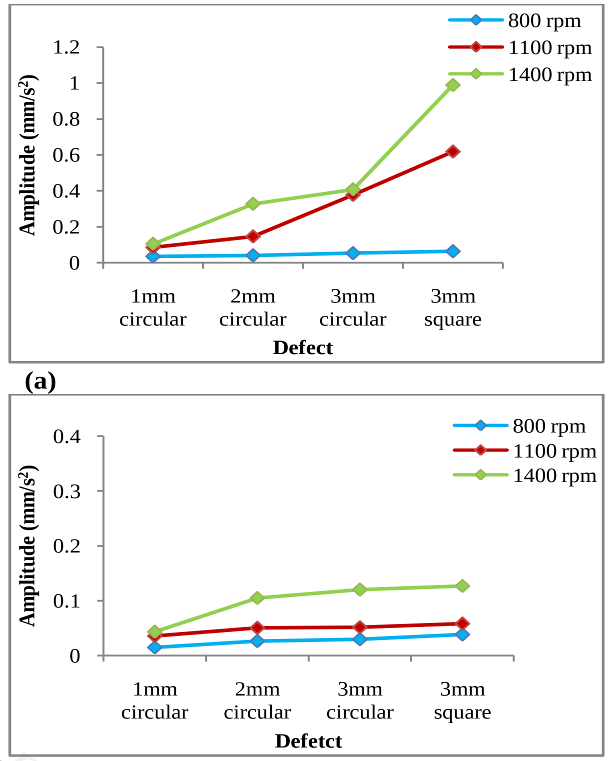
6. RESULTS AND DISCUSSION

In case studies, the first is done on a healthy bearing of the rotor-bearing system and recorded vibration spectra showed the peaks much smaller than the acceptable limits which enable to identify whether a bearing is healthy or not. In the same case study three other bearings, each with different defects are used and vibration spectra are obtained. By studying the frequency spectra, attempt is made to guess the number of defects.



(b)

Fig. 3 Effect of load and defect on vibration amplitude of outer-race (a) for 500N (b) for 1500N



(b)

Fig. 4 Effect of load and defect on vibration amplitude of roller (a) for 500N (b) for 1500N

At NDE four faulty bearings with 1.0mm, 2.0mm, 3.0mm circular and 3.0mm square defects are tried. The experimental frequency values nearly similar to all defects of different size and shape are around 1.01 time's theoretical frequency values, $T_f$listed in Table 2. This shows that there could be around 1 defect onouter ring race and roller.

In Fig. 3 (a) the vibration amplitude for 1mm circular shaped defect at 800rpm is 0.00714mm/s^2 which increases up to 0.0303mm/s^2 at 1400 rpm. But for 3mm circular shaped defect, increase in amplitude value from 800rpm to 1400 rpm is 0.03498mm/s^2 which indicates change in vibration amplitude is rapidly increased for larger defect size. This change is again rapid and more (0.05303mm/s^2) from 800rpm to 1400rpm for 3mm square shaped defect among all of other. Though this change for various size of defects at 1100rpm is not considearable compared with change at 800rpm, it is near about 0.02mm/s^2 more for each defect compared with change at 1400rpm.

As indicated in Fig. 3 (b), increase in load from 500N to 1500N lead to decrease in vibration amplitude by nearly 0.0180mm/s^2 for each defect. This decrease in vibration amplitude due to increase in load is rapid and continuous for increase in defect size as well as more for square shaped defect compared to circular and this change is considerable at all rotational speeds.

In Fig. 4 (a) the vibration amplitude for at 800rpm is not considerable for various defects. But, change in vibration amplitude is rapidly increased for 1100rpm and 1400rpm for various defects. This change is more for 3mm square shaped defect compared to circular shaped defect. The change in vibration amplitude of roller defect is more than chage for outer race defect. It is because of fault on roller does not make any continuous contact with outer race. Whenever there is complete contact, vibration amplitude increases rapidly and it slows down at partial contact of fault on roller with outer race.

As indicated in Fig. 4(b), increase in load from 500N to 1500N lead to decrease in vibration amplitude for each defect. This decrease in vibration amplitude due to increase in load is constant for each defect, but considerable at all rotational speeds.

7. CONCLUSION

In this study experimental analysis is done on the self-align bearings with different fault conditions. Experimental results are presented, so that a clear insight can be obtained about the signals involved. Multi variable regression analysis is then performed and a linear relation between amplitude of vibrations and various other parameters is developed. The values of vibration amplitudes measured in experiments are compared with the values predicted by MVRA.For outer race as well as roller defect, increases in speed, defect size cause to increase in vibration signal whereas increase in load decreases it rapidly at high speed. Change in defect shape increases the vibration signal whereas influence of defect size is more significant.

Nomenclature

T_f	= Theoretical defect frequency
E_f	= Experimental defect frequency
FTF	= Fundamental Train Frequency
BPFI	= Ball Pass Frequency Inner
BPFO	= Ball Pass Frequency Outer
BSF	= Ball Spin Frequency
Hz	= Hertz
O_f	= Outer race defect frequency
R_f	= Roller defect frequency
O_d	= Outer race defect
R_d	= Roller defect
FFT	= Fast Fourier Transform

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