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Vibrational Analysis of Self Align Ball Bearing Having a Local defect through FEA and its Validation through Experiment

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Abstract: **The rolling bearings dynamical behaviour analysis is a important condition to determine the machine vibration response. The rolling bearing, with outer ring fixed, is a multi body mechanical system with rolling elements that transmit motion and load from the inner raceway to the outer raceway.Modern trend of Dynamic analysis is useful in early prediction; simulation of rotor bearing system as manufacturing of prototype is time consuming, costly, and required further analysis for fatigue failure. Dynamic analysis has become a very powerful tool for the betterment of the actual performance of the system. The methodology for prediction and validation of dynamic characteristics of bearing rotor system vibration is studied. PRO-E® and ANSYS software are the promising tools for the modeling and modal analysis of the bearing rotor system. Experiment result has been taken for the analysis of the signal that has been obtained through the use of FFT analyser. Matlab® program is generated for finding the BPFO and BPFI for bearing system; its graphical values are shown. The goal of this study is to prevent the system from the breakdown by continuous monitoring of vibration values.**

Keywords: ***BPFO, BPFI, vibration monitering, rollin element, frequency domain***

I. INTRODUCTION

A ball bearing is a type of rolling-element bearing that uses balls to maintain the separation between the moving parts of the bearing. The purpose of a ball bearing is to reduce rotational friction and support radial and axial loads. It achieves this by using at least two races to contain the balls and transmit the loads through the balls. Usually one of the races is held fixed. As one of the bearing races rotates it causes the balls to rotate as well.

In this paper Self-aligning ball bearings is considered whose bearing number is 1205k; it is constructed with the inner ring and ball assembly contained within an outer ring that has a spherical raceway. This construction allows the bearing to tolerate a small angular misalignment resulting from deflection or improper mounting.

Experimental modal analysis, structural dynamics modification and finite element analysis were used to analyze the dynamic properties of a test structure. Most noise, vibration or failure problems in mechanical structures and systems are caused by excessive dynamic behavior. In recent years, however, the implementation of the Fast Fourier Transform (FFT) in low cost computer-based signal analyzers has provided the environmental testing laboratory with a fast and more powerful tool for acquisition and analysis of vibration data.

The FFT spectrum analyzer samples the input signal, computes the magnitude of its sine and cosine components, and displays the spectrum of these measured frequency components. The FFT is simply a clever set of operations which implements Fourier's theorem. The resulting spectrum shows the frequency components of the input signal.

The big advantage of this technique is its speed. Because FFT spectrum analyzers measure all frequency components at the same time, the technique offers the possibility of being hundreds of times faster than traditional analogue spectrum analyzers. To measure the signal with higher resolution, the time record is increased. But again, all frequencies are examined simultaneously providing an enormous speed advantage. Today the ball bearing is used in numerous everyday applications. Ball bearings are used for dental and medical instruments. In dental and medical hand pieces, it is necessary for the pieces to withstand sterilization and corrosion. Because of this requirement, dental and medical hand pieces are made from 440C stainless steel, which allows smooth rotations at fast speeds.

II. LITERATURE SURVEY

Major contributors in the field of bearing analysis are Jones, Harris [9] Palmgren. Firstly Lundberg and Palmgren developed a theory to predict stress distribution at point of contact for normal loading. This theory was able to predict fatigue life of bearings to some extent with inclusion of empirical proportionality constant. Then Jones developed a general method, to obtain all forces and elastic deformations analytically in a redundant system like ball and roller bearing. This theory was a successful attempt to improve precision of Lundberg and Palmgren theory.

C.Zhang, T.Kurfess [5] did work on ball bearing which proposes a remaining life adaptation methodology based on mechanistic modeling and parameter turning through a defect propagation model and defect diagnostic model, an adaptive algorithm is developed to fine tune the parameter involved in the bearing. Antoniadis and G.Glossiotis [3] proposes an alternative frame work for analyzing bearing vibration signal with periodically varying statics, is better able to exhibit the underlying physical concepts of the modulation mechanism present in the vibration response of bearings. Sun-Min Kim and sun-Kyu Lee [4] investigates the effect of assembly tolerance on the spindle bearing compliance. In high speed spindle system, the bearing characteristics are significantly influenced by the initial assembly tolerance and the thermal deformation of the bearing support structure. Zeki kiral and Hira

Karagulle[6] done Simulation and analysis of vibration signals generated by rolling element bearing with defects in 2003, in this paper dynamic loading of a rolling element bearing structure was modeled by a computer program developed in Visual Basic programming language. Peter W. Tse in 2004 has done the Machine fault diagnosis through an effective exact wavelet analysis in which to minimize the effect of overlapping and to enhance the accuracy of fault detection, a novel wavelet transform, which was named as exact wavelet analysis, had been designed for use in vibration-based machine fault diagnosis. In 2009 Abhay Utpat, R.B.Ingle and M.R.Nandgaonkar[1] proposed a work in which vibration produced by a single point defect on various parts of the bearing under constant radial load are predicted by using a theoretical model. The model includes variation in the response due to the effect of bearing dimension, rotating frequency distribution of load. M.S.Patil, Jose Mathew, Sandeep desai in 2010 [2], they proposed an analytical model for predicting the effect of a localized defect on the ball bearing vibrations. In the analytical formulation, the contacts between the ball and the races are considered as non-linear springs. The contact force is calculated using the hertz contact deformation theory. A computer program was also developed to simulate time domain and frequency domain. The model yields both the frequency and the accelerations of vibration component of bearing.

III. MODELING OF THE SYSTEM

As a first step in investigating the vibrations characteristics of ball bearings, a model of a rotor bearing assembly can be considered as a spring-mass system, where the rotor acts as a mass and the raceways and balls act as mass less nonlinear contact springs. In the model, the outer race of the bearing is fixed in a rigid support and the inner race is fixed rigidly with the rotor. A constant radial vertical force acts on the bearing. Therefore, the system undergoes nonlinear vibrations under dynamic conditions. Elastic deformation between the race and ball gives a non-linear force deformation relation, which is obtained by using the Hertzian theory. Other sources of stiffness variation are the positive internal radial clearance, the finite number of balls whose position changes periodically and waviness at the inner and outer race. They cause periodic changes in stiffness of the bearing assembly.

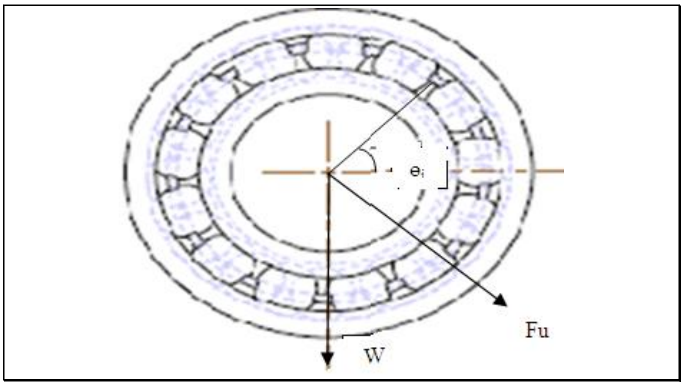


Figure 1: A schematic diagram of a rolling element bearing.

3.1 Ball passage frequency

When the shaft is rotating, applied loads are supported by a few balls restricted to a narrow load region and the radial position of the inner race with respect to the outer race depends on the elastic deflections at the ball to raceways contacts. Balls are deformed as they enter the loaded zone where the mutual convergence of the bearing races takes place and the balls rebound as they move to the unloaded region. The time taken by the shaft to regain its initial position is

T = time for completion rotation of cage/N_b

As the time needed for a complete rotation of the cage is 2π/ωc the shaft will be excited at the frequency of (Nb * □c known as the ball passage frequency. Here □*c* is the speed of the cage.

$$\omega_c = \frac{\omega inner}{2}\left(1-\frac{db}{dm}\right)+\frac{\omega outer}{2}\left(1+\frac{db}{dm}\right) \qquad (1)$$

Hence, ball passage frequency (ω*bp*) is

$$\omega_{bp}=\frac{1}{2}N_b\frac{\omega inner}{2}\left(1-\frac{db}{dm}\right)+\frac{1}{2}Nb\frac{\omega outer}{2}\left(1+\frac{db}{dm}\right) \qquad (2)$$

Since outer is assumed to be constant, the ball passage frequency is

$$\omega_{bp}=\frac{1}{2}N_b\frac{\omega inner}{2}\left(1-\frac{db}{dm}\right) \qquad (3)$$

III A. STRUTURE DEFECT INDUCED VIBRATION

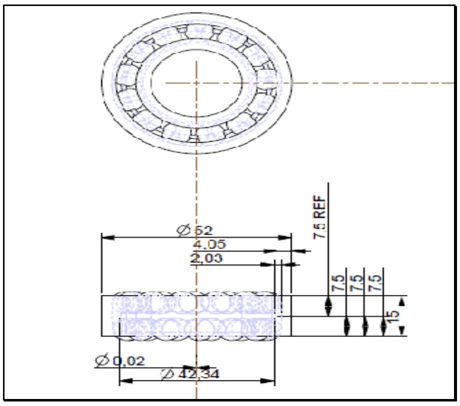


Figure 2: Geometry of self aligning bearing 1205k

The load distribution on a rolling element bearing is given by

q (□ □ □ □ q_{max} [1- (1-cos□ □□^{n}

(4)

Where q_{max}-Maximum load
Y-limiting angle
e - Load distribution factor
n= 3/2 for roller bearings
n=10/9 for ball bearings
In a bearing with nominal diametral clearance, Q_{max} can be approximated as,

Qmax = (5)

Where Fr – Applied radial Load
Z – Number of rolling elements
a - Mounted contact angle

Table I
Parameters for the self aligning bearing

self aligning bearing 1205k	Values
Outer diameter	52mm
Inner diameter	25mm
Thickness	15mm
Mean diameter	38.5mm
Ball diameter	8.1mm
Number of balls	26

IV. EXPEMENTAL SETUP

An experimental test rig built to predict defects in self aligning bearings is shown in Figure3. The test rig consists of a shaft with central rotor, which is supported on two bearings. A motor coupled by a flexible coupling drives the shaft.

Figure 3: Experimental setup for bearing

Vibration characteristics are very important in the study of diagnostics for the system. In the experiment setup two bearing are considered in which one bearing was without defect and other bearing was with defect, both the bearing was attached to the system one by one for carry out the result. The dimension of the bearing are given in the Table 1.After taking the result it was seen that the values of amplitudes was more for the bearing with defect compare to bearing which has no defect. The natural frequency of the system was around the 34Hz,it was seen at the time of processing the data that for the defect bearing the values of amplitude was increasing when frequency of the was nearby natural frequency of the system. The various values of accerlation, velocity for the study of the vibration characteristics were taken and these are shown in the figure respectively.

To monitor load and vibration within the bearing structure, a sensor is embedded into a slot cut through the outer ring. The sensor has solid contact with both the top of the slot and the bearing housing. Each time a rolling element passes over the slot, the sensor generates an electrical charge output that is proportional to the load applied to the bearing F_r. Since the outer ring is structurally supported by the bearing housing, it can be assumed as rigid. The sensor is modelled as a spring with stiffness k that is related to its material composition. The section of the bearing outer ring where the slot is cut can be modelled as a beam of varying cross-section, with a spring support at the midpoint. Since the ends of the beam are solidly connected to the surrounding bearing structure, which is directly supported by a rigid housing, clamped boundary conditions are considered appropriate.

V. DESIGN AND ANALYSIS

In this segment modelling of bearing is done with the help of PRO-E software, modelling is a complex task for designing a bearing because in the modelling of bearing various types of joints should be applied at the design stage which is very complex.

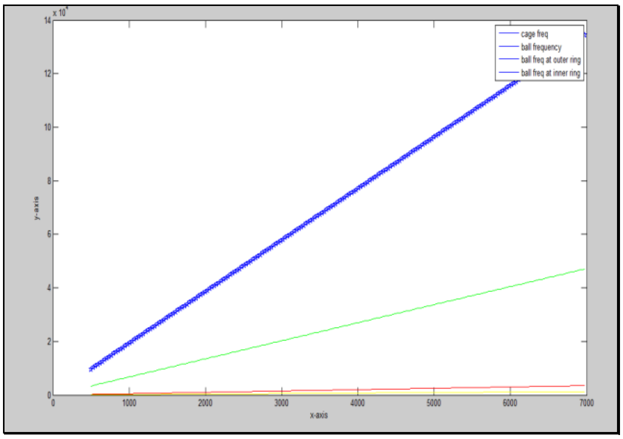


Figure 4: Bearing frequencies for shaft speed 0-7000rpm

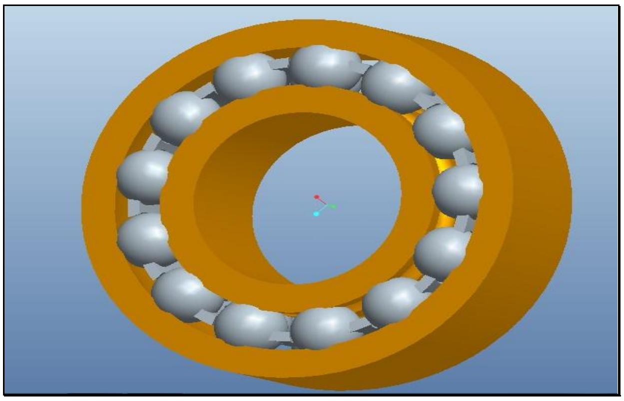


Figure 5: Self aligning bearing 1205k model with complete dimension, model in Pro-E software

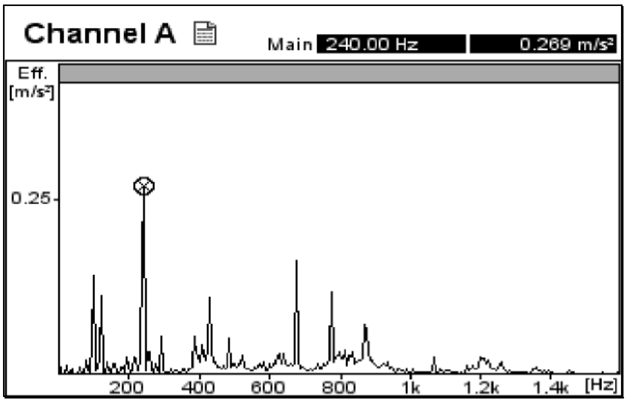
Combination of pin joint and cylindrical joint is applied in the model generation for the movement of balls with respect to inner ring, outer ring and cage part respectively. Matlab program is also prepared for finding different geometry parameters of the bearing with the help of formulae are which was found from different papers and books. For finding the frequency of ball, BPFI, BPFO and cage frequency, Matlab program is prepared and their results are mentioned in the figures. Figure 5 showing the model of self align bearing which is prepared in PRO-E software whose parameter is defined in Table1.Parameters for the bearing specifications were defined, so that they could be modified for any type of bearing that was to be analyzed. A defect in the outer ring was modelled by a cylindrical hole. Hence the parameters defined included:

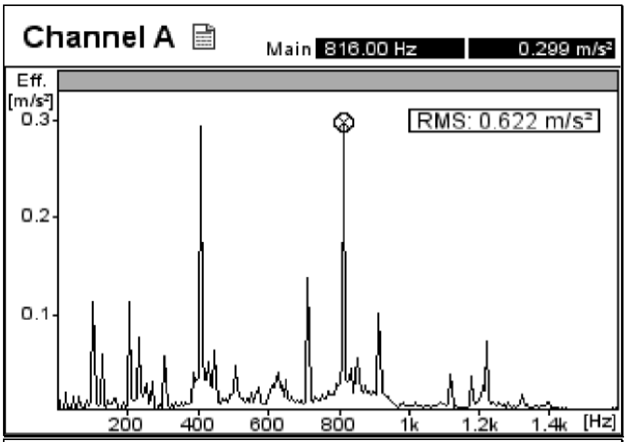
outer raceway diameter, outer ring diameter, thickness of the outer ring, raceway radius, defect depth and the defect radius.

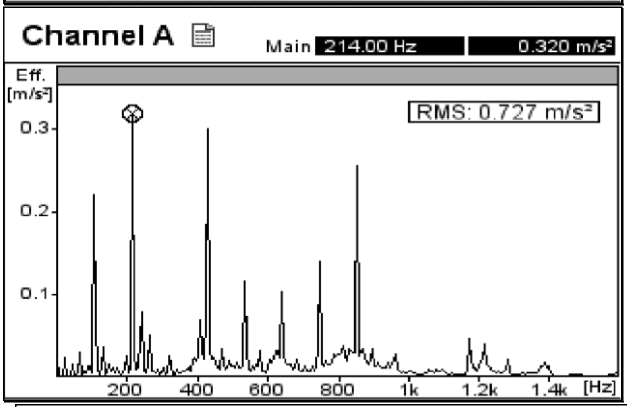
VI. RESULT AND DISCUSSION

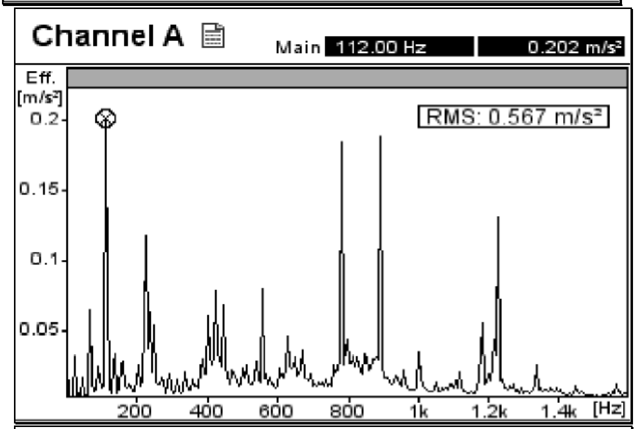
Vibration characteristics are very important in the study of diagnostics for the system. In the experiment setup two bearing are considered in which one bearing was without defect and other bearing was with defect, both the bearing was attached to the system one by one for carry out the result. The dimension of the bearing are given in the Table 1.After taking the result it was seen that the values of amplitudes was more for the bearing with defect compare to bearing which has no defect. The natural frequency of the system was around the 34hz,it was seen at the time of processing the data that for the defect bearing the values of amplitude was increasing when frequency of the was nearby natural frequency of the system. The various values of accerlation, velocity for the study of the vibration characteristics were taken and these are shown in the figure respectively.

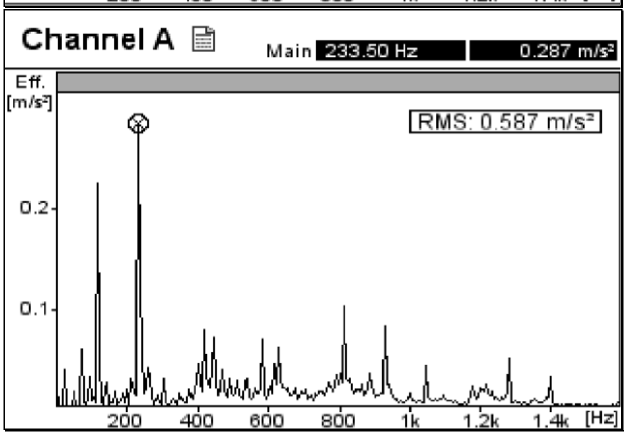
First setup is run for few minutes to settle down all minor vibration. After this Accelerometer along with the vibration analyzer is used to acquire the vibration signals. Vibration signals are measured at different speeds of the system for both defective and non defective bearing. Following are the few results which are taken through the help of FFT analyser. During performing the experiment shaft speed are vary from 1200 rpm to 2040 rpm, during these speed of the shaft amplitude values in terms of accerlation (m/s^2) and velocity (m/s) were taken for better understanding. For without defect and with defect bearing result were taken in time domain, correspondingly frequency domain result were also taken for defective bearing for better understanding of vibration amplitude values.

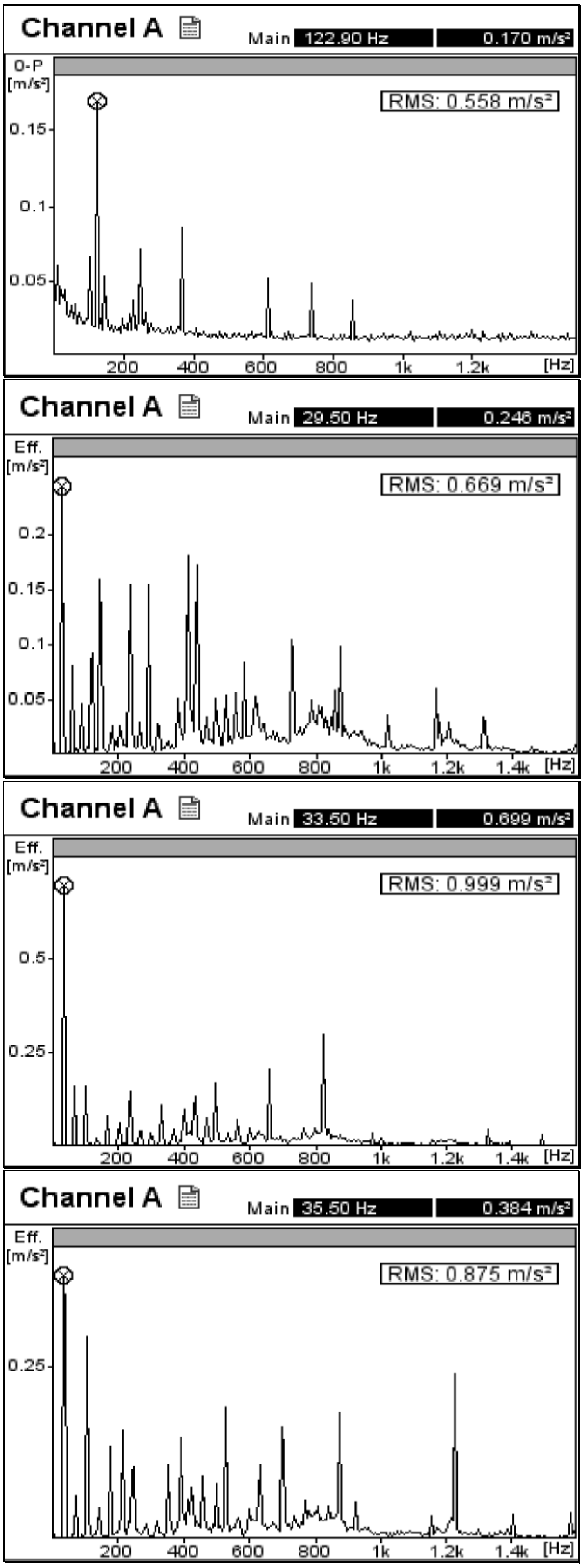












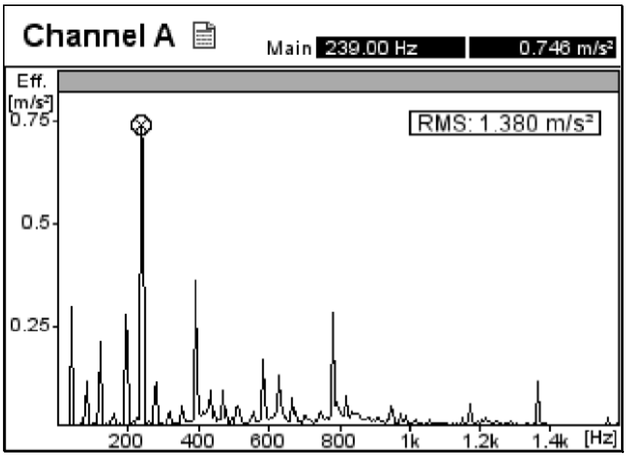


Figure 6: Accerlation plots for different speed of the shaft

Table II
Amplitude values with respect to shaft speed

Speed of the shaft Rpm	Non-defective bearing Maximum amplitude		Defective Bearing Maximum amplitude	
	mm/s	µm	mm/s	µm
1200	1.212	29.035	2.465	46.945
1260	1.362	26.775	4.090	50.114
1320	2.267	42.570	7.954	94.750
1380	2.906	63.241	5.716	71.036
1500	3.601	44.590	3.616	44.810
Speed of the shaft Rpm	**Maximum amplitude values** m/s^2			**RMS** m/s^2
1200	0.269			0.522
1260	0.299			0.622
1320	**0.727**			**0.727**
1380	0.202			0.567
1440	0.287			0.587
1500	0.170			0.558
1800	0.246			0.669
2040	0.699			0.999
2160	0.384			0.875
2400	0.746			0.980

For the finite element analysis, ANSYS software were used for comparing the result with the experiment, at the beginning parasolid file of whole assembly was transported from the PROE software in ANSYS module, in this module firstly modal analysis was done, then the dynamic analysis was performed on the system, in the analysis two element was defined one for the bearing and another for the shaft and plumber block. These two elements were tetrahedron 4-node and tetrahedron 10-node element both are solid element. Refinement was done at ball and cage as well as at the both outer and inner ring for better result and accuracy.

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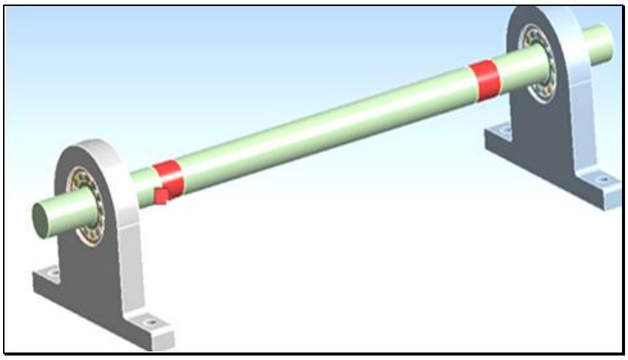
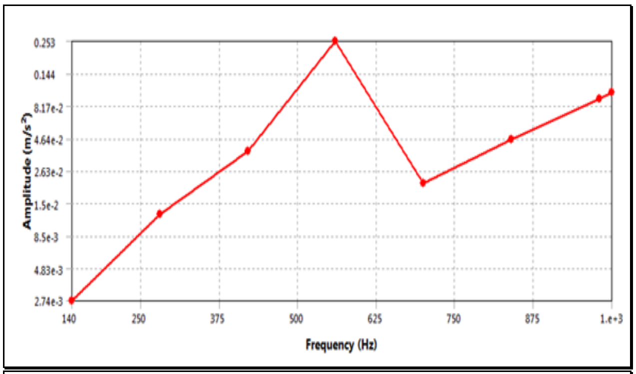
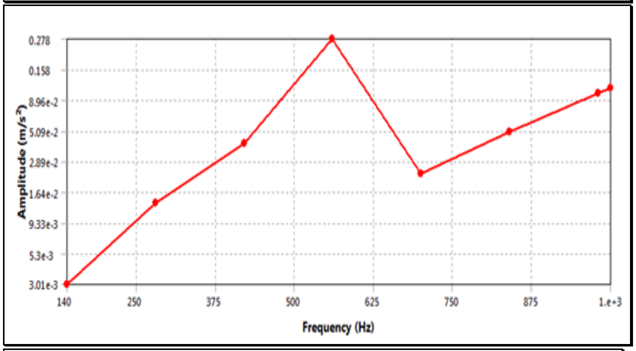
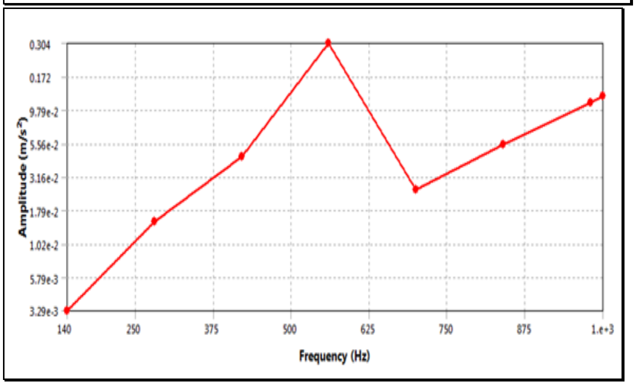
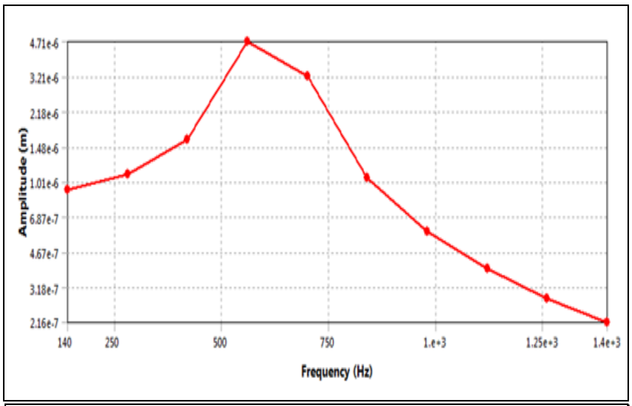


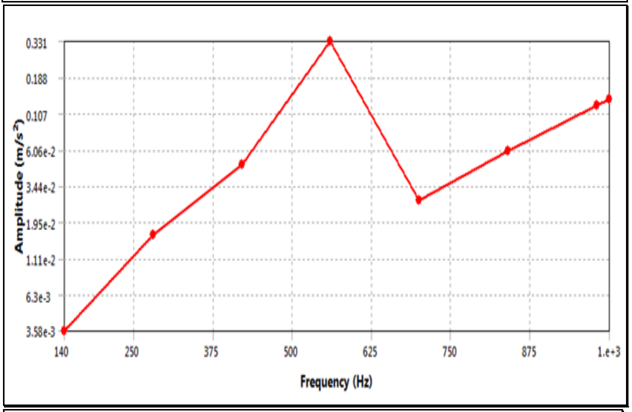
After meshing properly and defining the loading condition in the system analysis were run for number of time for getting the result in FEA software ANSYS12.whose result are shown in the figure 7.these result almost matching with the maximum amplitude values of experiment result which were shown in the Table II.

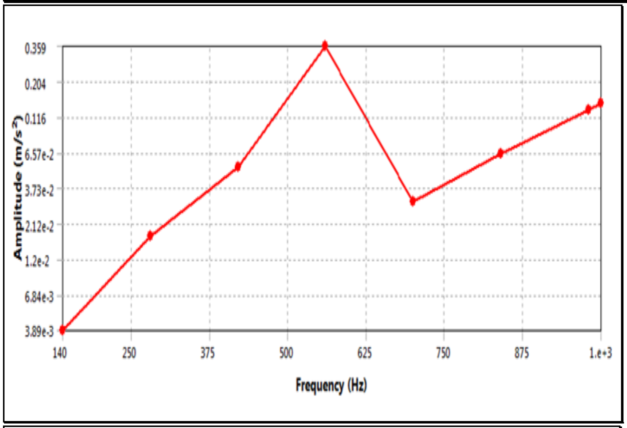


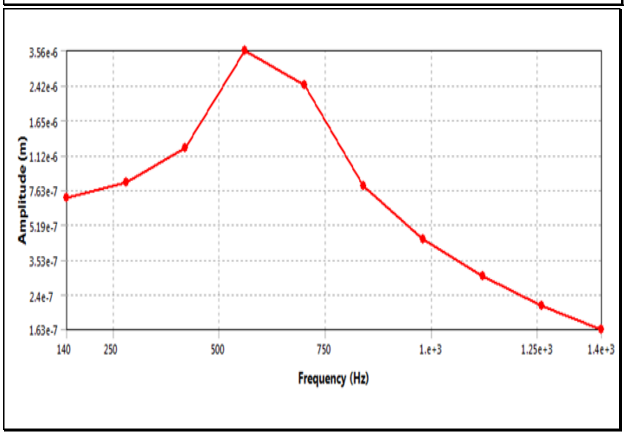












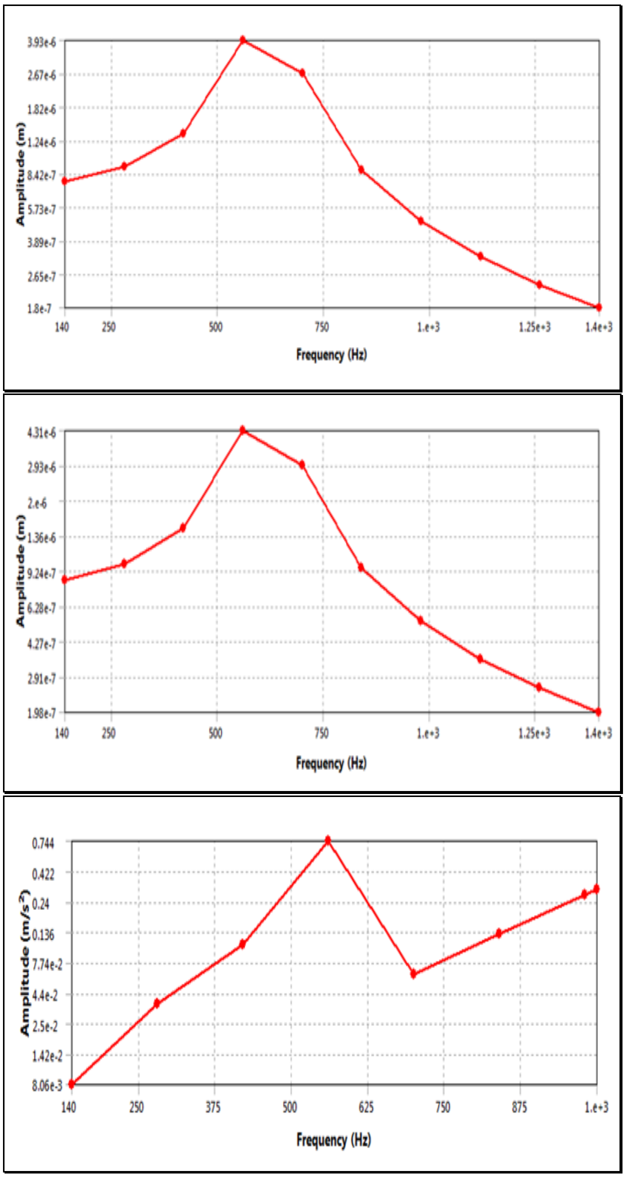


Figure 7: Amplitude values in terms of accerlation and displacement with respect to shaft speed in ANSYS 12 software.

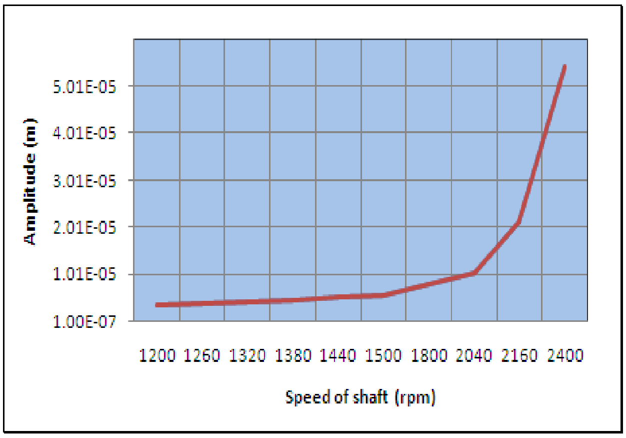


Figure 8: Amplitude values in terms of displacement wrt shaft speed.

Figure 8 is showing the amplitude values in terms of displacement with respect to the shaft speed, in the analysis the probe was set at non drive end the above result is almost same compare to the Experiment result, also for the further analysis in finite element software different damping coefficient(Nm/s) and damping ratio was changed and correspond to that various amplitude in terms of displacement were recorded and for these values graphs were plotted by seeing these graphs, it is clear that with the increasing value of damping ratio and damping coefficient the maximum amplitude was decreasing linearly, so the fact is that if the values of vibration amplitude decrease by this approach then it is beneficial for its life of the bearing and jerks can be eliminated at a great extent. Table III is showing the all values which were obtained at the time of analysis.

Table III
Amplitude values wrt various damping coefficient.

Damping Coefficient (Nm/s)	Amplitude at1500 rpm	Amplitude At 1440	Amplitude At 1380
0	5.37E-06	5.13E-06	4.71E-06
50	3.32E-07	3.26E-07	2.83E-07
100	1.40E-08	1.35E-07	9.80E-08
150	3.54E-08	3.42E-08	2.99E-08
200	1.28E-08	1.19E-07	8.90E-08
250	1.76E-09	1.56E-09	1.10E-09
350	2.06E-09	1.98E-09	1.43E-09
400	3.98E-10	3.45E-10	3.02E-10
500	4.23E-11	4.12E-11	3.76E-11
600	4.35E-12	3.19E-12	2.76E-12

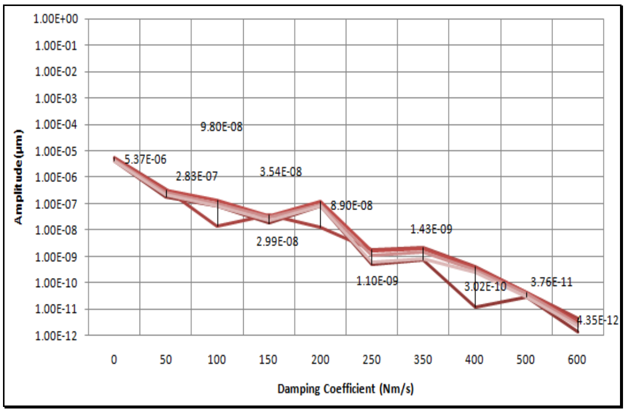
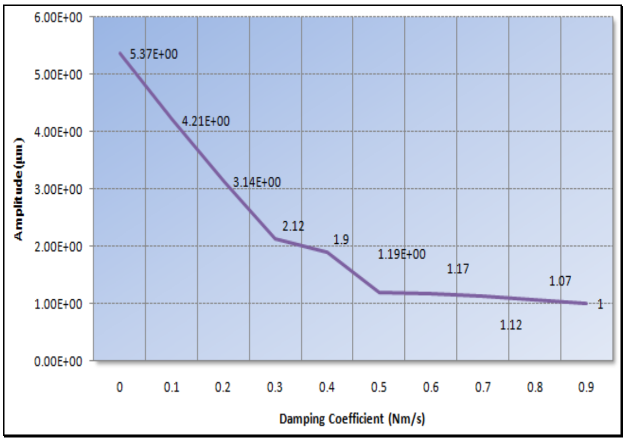


Figure 9: Comparison of maximum amplitude with respect to damping coefficient at different shaft speed.



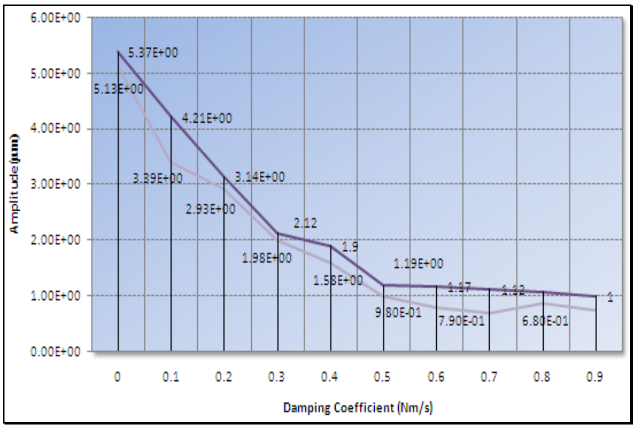


Figure 10: Comparison of maximum amplitude with respect to damping ratio at different shaft speed.

VII. CONCLUSION

It has been shown that finite element modelling can be effectively used to differentiate between vibration signatures for defects of different sizes in the bearing. Assumptions have been made for the variation of forces exerted by the rolling element on the outer ring in the vicinity of the defect. The main aim has been to understand the trend of vibration signatures for the local defect in the bearing through finite element analysis as well with the experiment that has been done with the help of FFT analyser. The natural frequency of the system was around 34Hz because for both the system the amplitude that was obtained during the study of the experimental result was maximum than any other values. Defect size was 0.02 mm^3 was studied and the different plots in terms of accerlation and displacement amplitude were generated both in the experiment and FEA software ANSYS 12. The result was almost same in both. A more detailed analysis based on this project was also done by changing the value of damping coefficient and damping ratio in the finite element analysis whose plots are shown in the paper. With the increasing damping coefficient as well as damping ratio the amplitude was constantly decreasing.

IIX. FUTURE WORK

It is important to be able to precisely understand the variation of forces due to the rolling element passing over a defect in a bearing. A detailed analysis using experiments on a bearing test rig should be performed. The finite element model can then be iteratively adjusted so as to conform to the vibration signature that is arrived at by experimentation. Matlab programming and other codes would be used for better exercise to improve the accuracy of the finite element modelling results and for reducing the time consumption.

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