RESEARCH ARTICLE   OPEN ACCESS

Vibration Analysis of deep groove ball bearing using Finite Element Analysis

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ABSTRACT
Rolling element bearing is essential part of machinery. 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. The rolling bearings dynamical behaviour analysis is an important condition to determine the machine vibration response. Modern trend of Dynamic analysis is useful in early prediction. 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. ANSYS software is the promising tools for the modelling. The result obtained from FEA are validated with experimental results.
Keywords - deep groove ball Bearing, Outer race defects, Inner race defect, FFT analysis.

I. INTRODUCTION

The bearing type used in this study is a single row deep groove ball bearing. They are the most popular of all rolling bearing because it is non-separable, capable of operating at high even very high speeds, and require little maintenance in service. In addition they have a price advantages. The bearing model 6204 from SKF is used in study. This bearing has a bore diameter of 20 mm and widely used for many applications.

Experimental modal analysis, structural dynamics modification and finite element analysis is 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 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.

Many research works had been done in the field of signal analysis in rolling element bearing. Amit R Patel [1] in this paper Test rig for vibration monitoring of bearing have been designed. The design of test rig includes design of Shaft, Bearing selection, pedestal selection, motor selection, and base preparation. A 3-D model have been developed using Pro-engineer modeling software. The vibration signals generated by the healthy/faulty bearing can be captured by the accelerometer mounted on the test bearing. The analysis of the vibration signal is useful for the condition monitoring of rolling element bearings. Abhay Utpat [2] in this paper, the study of failure analysis of ball bearing is discussed by creation of artificial cracks of different sizes on various elements and noting down its signatures. The work has been extended with Finite Element Analysis of bearing with artificial defects to study the peaks at its outer ring as well as inner ring defect frequencies. It is concluded that at constant defect size and constant load with different speeds of rotation, amplitudes of vibration varies with increase in speed. In this case also amplitudes of vibration are observed higher for outer ring defected bearings than inner ring defected bearings for same defect size. U. A. Patel, Shukla Rajkamal [3] in this paper modern trend of dynamic analysis is used for prediction. PRO-E and ANSYS software are used for the modeling and model 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. Defect size was 0.02 mm3 was studied and the different plots in terms of acceleration and displacement amplitude were generated both in the experiment and FEA software ANSYS 12. The result was almost same in both. M.S. Patil, Jose Mathew, [4] in this paper work is focused towards the development of a theoretical model to study the effect of defect size on bearing vibration. The model makes it possible to detect the frequency spectrum having peaks at the bearing defect

frequencies. The amplitudes at these frequencies are also predicted. He developed a mathematical model for the ball bearing vibrations due to defect on the bearing race. N. Tandon , A. Choudhury, [7] a review of vibration and acoustic measurement methods for the detection of defects in rolling element bearings is presented in this paper. This paper include detection of both localized and distributed categories of defect, explanation for the vibration and noise generation, Vibration measurement in both time and frequency domains along with signal processing techniques such as the high-frequency resonance technique, acoustic measurement techniques such as sound pressure, sound intensity and acoustic emission. Dennis H. Shreve [8] this paper is considers the various data collection setup parameters and tradeoffs in acquiring fast, meaningful vibration data to perform accurate analysis in the field of predictive maintenance. The intent of this paper is to focus on the internal signal processing path, and how it relates to the ultimate root-cause analysis of the original vibration problem. An understanding of these basic concepts in signal processing and data manipulation will enable one to select instrumentation and to understand its use.

This paper consists of finite element modeling and analysis of healthy bearing and defective bearing. Modeling is a complex task for designing a bearing because in the modeling of bearing various types of joints should be applied at the design stage which is very complex. In defective bearing, defect is created on inner race and outer race. Amplitudes of acceleration are obtained for different defect size by varying rotational speed. At the last, amplitude vs. frequency graphs are obtained to know the results. Also results of experimentation are benchmarked through finite element analysis.

II. FINITE ELEMENT ANALYSIS
2.1 Finite element modeling
FEA has been carried out by using ANSYS 14 package. Initially 6204 bearing model are drawn in CATIA software and these have been imported to ANSYS software. Defect at inner race and outer race are created in ANSYS itself.

In pre-processor menu, there are five cases taken into consideration for analysis in order to validate results of each case. Following are the cases of simulation,
1) Healthy bearing
2) (1×0.25) mm inner race defect
3) (1×0.5) mm inner race defect
4) (1×0.25) mm outer race defect
5) (1×0.5) mm outer race defect

While modeling each case, shell 163 element has been used with element edge length 2mm. Because shell element is suitable for curved shape component as well as rotating component. In order to get good results, shell element is used in rotating component.

In order to define bearing model assembly of three components is made those are, inner race, balls, outer race. These components are made of nodes. It means that each component is defined as a group of nodes.

Analysis has been carried in ANSYS EXPLICIT LS-DYNA. 3D dynamic analysis is done so as to get better response of the bearing. Results are obtained in frequency domain. Following figures shows meshed model of each case individually.
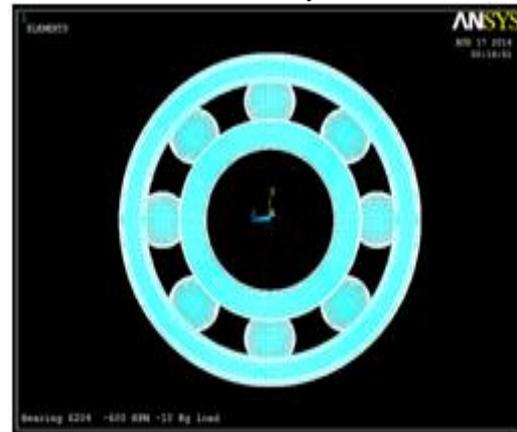

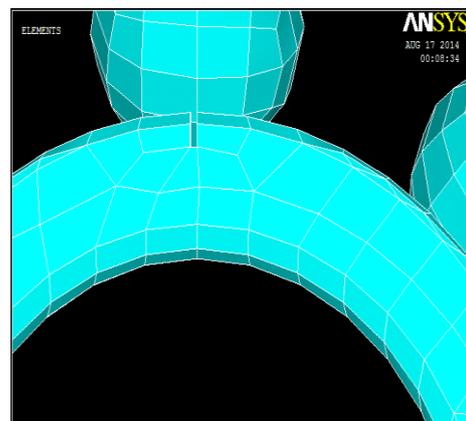Fig.1 Meshed models of healthy bearings
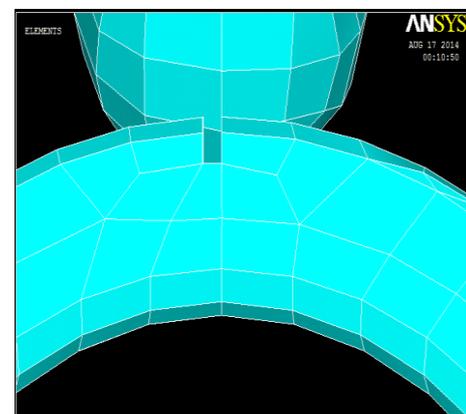

Fig. 2 Meshed model of 0.25mm inner race defect
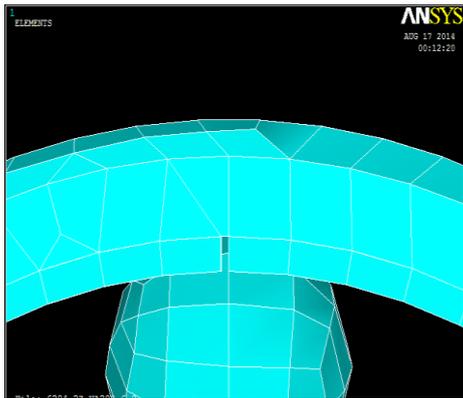

Fig. 3 Meshed model of 0.5mm inner race defect
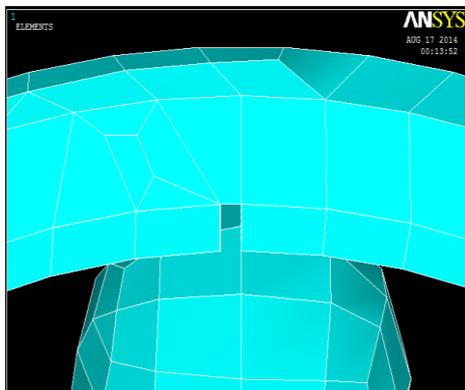
Fig. 4 Meshed model of 0.25mm outer race defect

Fig. 5 Meshed model of 0.5mm outer race defect

2.2 FFT Result

For analysis purpose, only one case is taken in to consideration i.e. 0.5mm defect on outer race and inner race defect with constant load of 10kg.

Following are the frequency domain graphs got from ANSYS at constant 10 kg load for 1×0.5mm outer race defect and inner race defect.
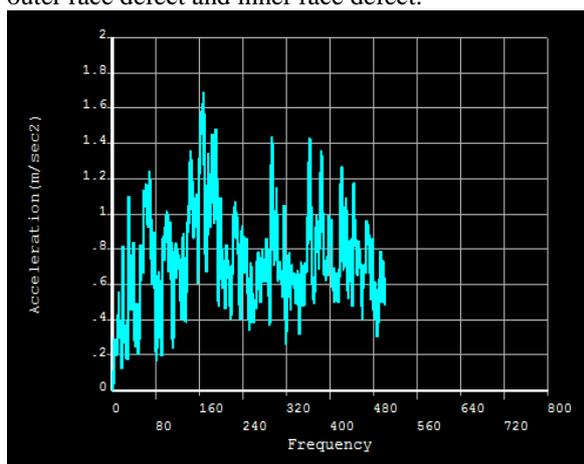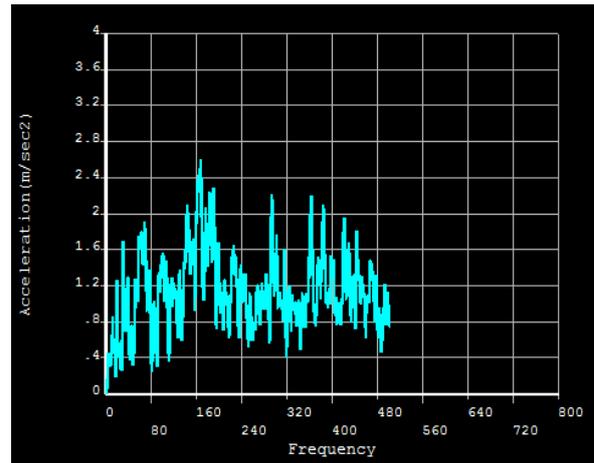

Fig.7 Amplitude of acceleration at 1500 rpm for 1×0.5mm outer race defect

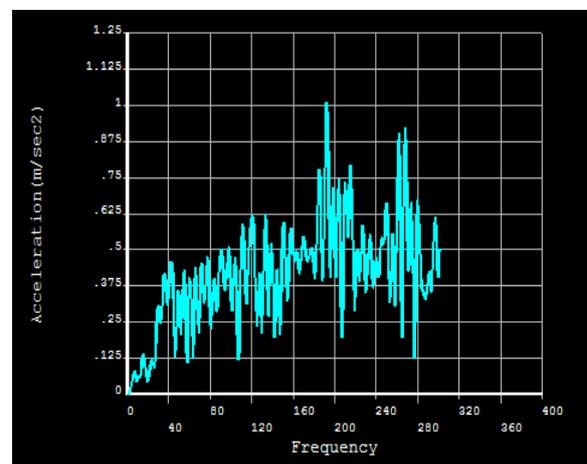Fig.8 Amplitude of acceleration at 1200 rpm for 1×0.5mm inner race defect

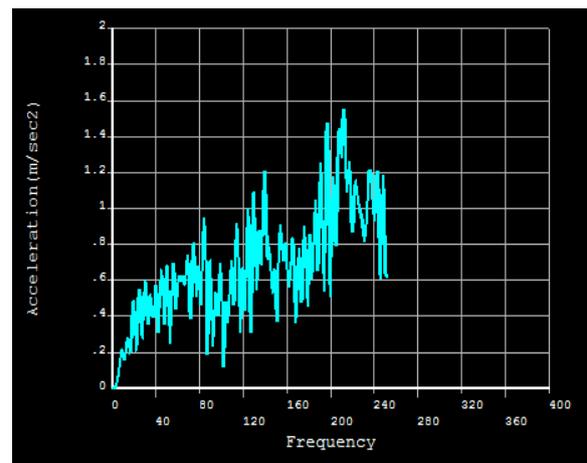Fig.6 Amplitude of acceleration at 1200 rpm for 1×0.5mm outer race defect

Fig.9 Amplitude of acceleration at 1500 rpm for 1×0.5mm inner race defect

Fig 6 and Fig 7 shows the frequency domain plot for 1×0.5mm defect on outer race at 1200rpm and 1500rpm. It is observed that amplitudes of vibration increases with increase in speed. In this case the

amplitudes of vibration are 1.77 m/s^2 and 2.59 m/s^2 at 1200rpm and 1500rpm respectively. The impulses are obtained at BPFO and multiple of BPFO.

Fig 8 and Fig 9 shows the frequency domain plot for 1×0.5mm defect on inner race at 1200rpm and 1500rpm. It is observed that amplitudes of vibration increases with increase in speed. In this case the amplitudes of vibration are 1.02 m/s^2 and 1.55 m/s^2 at 1200rpm and 1500rpm respectively. The impulses are obtained at BPFI and multiple of BPFI.

III. EXPERIMENTAL TEST RIG

Experimental setup shown in Fig.10 consists of a shaft supported on two bearing and driven by a variable speed motor. The test bearing is place to a non-drive end of shaft with radial load arrangement, capable of changing load on test bearing. A continuously variable speed from 600 rpm to 1500 rpm is obtained from control panel of DC motor. The developed test rig is mounted on C-channel frame of size 480cm x 990cm with height of 150 cm. The mass of set up is 500kg approximately and special care is taken while design and fabrication to reduce the shocks and vibrations produced due to electric motor and rotating components. The signatures of the vibration will be collected for various running parameters using the FFT analyzer.
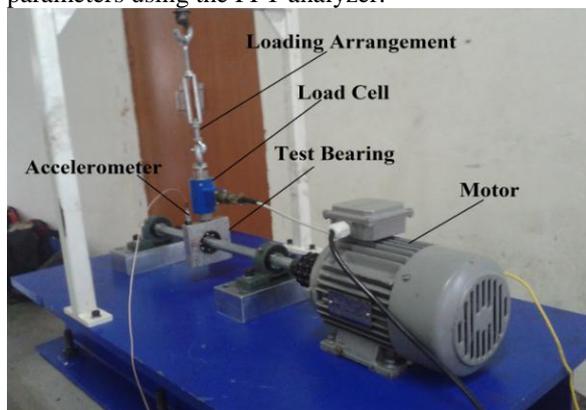

Fig.10 Experimental test rig

3.1 Experimental Result

The same case is taken in to consideration for experimentation i.e. 0.5mm defect on outer race and inner race defect with constant load of 10kg.

Following are the frequency domain graphs got from experiment at constant 10 kg load for 1×0.5mm outer race defect and inner race defect.

Fig 11 and Fig 12 shows the frequency domain plot for 1×0.5mm defect on outer race at 1200rpm and 1500rpm. It is observed that amplitudes of vibration increases with increase in speed. In this case the amplitudes of vibration are 1.79 m/s^2 and 2.63 m/s^2 at 1200rpm and 1500rpm respectively. The impulses are obtained at BPFO and multiple of BPFO.
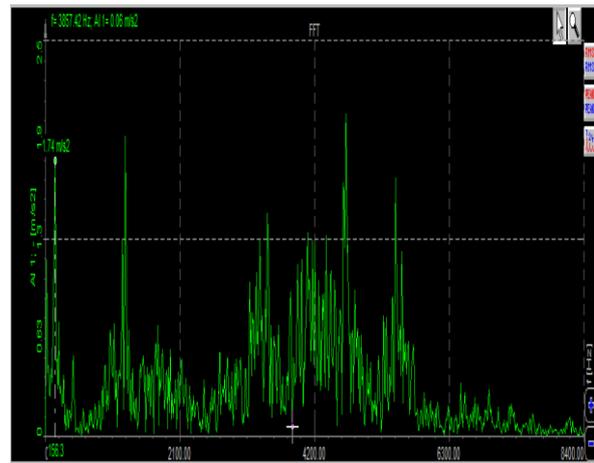

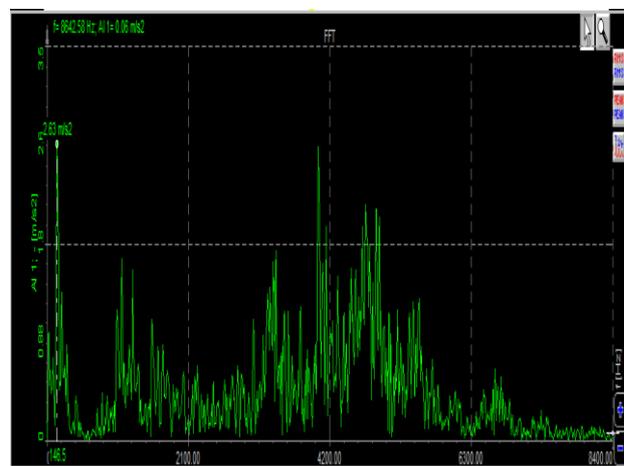Fig.11 Amplitude of acceleration at 1200 rpm for 1×0.5mm outer race defect

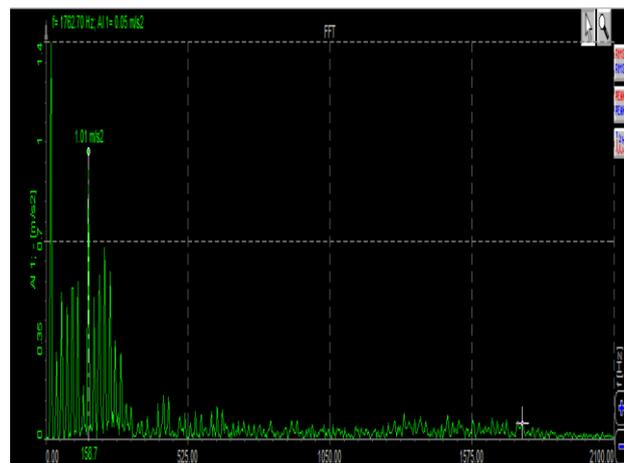Fig.12 Amplitude of acceleration at 1500 rpm for 1×0.5mm outer race defect

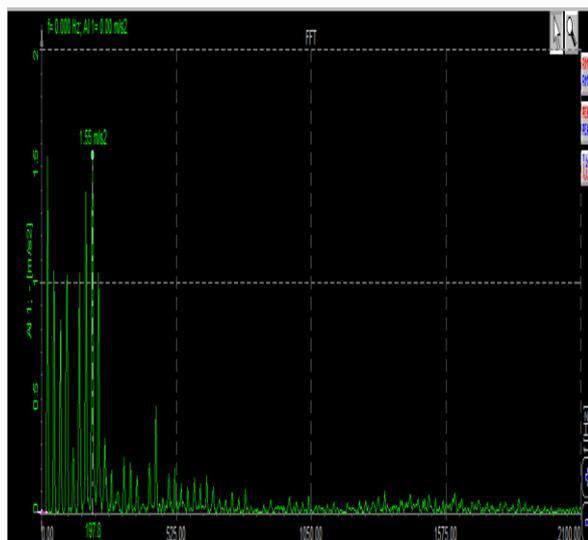Fig.13 Amplitude of acceleration at 1200 rpm for 1×0.5mm inner race defect

Fig.14 Amplitude of acceleration at 1500 rpm for 1×0.5mm inner race defect

Fig 13 and Fig 14 shows the frequency domain plot for 1×0.5mm defect on inner race at 1200rpm and 1500rpm. It is observed that amplitudes of vibration increases with increase in speed. In this case the amplitudes of vibration are 1.01 m/s² and 1.55 m/s² at 1200rpm and 1500rpm respectively. The impulses are obtained at BPFI and multiple of BPFI.

IV. RESULT AND DISCUSSION

The FEA and experimental results at different speeds, loads and defect sizes are plotted and comparison is made for following cases.

Table -1: Cases of experimentation

Cases	Type of Bearing	R.P.M	Radial Load in Kg.
1	H.B.	600,900, 120,01,500	0
			5
			10
			15
2	D.B. (Defect 1×0.25 mm²)	600,900, 120,01,500	0
			5
			10
			15
3	D.B. (Defect 1×0.5 mm²)	600,900, 120,01,500	0
			5
			10
			15

4.1 Comparison of Result

Obtained FEA result is compared with experimentation as follow. For validation purpose,

two cases are considering i.e. 0.5mm defect on outer race and inner race with constant load of 10 kg.

Table-2: Show Result for 0.5mm outer race defect at 10kg load

Speed	Experimentation		FEA		Deviation in %
	O 1×0.5mm		O 1×0.5mm		
	Frequency Hz	Amplitude m/s²	Frequency Hz	Amplitude m/s²	
600	273.4	0.859	279	0.839	2
900	263.7	1.06	258	1.04	2
1200	156.3	1.79	164	1.77	2
1500	146.5	2.63	165	2.59	4

Table 2 shows % deviation in amplitude of acceleration. During experimentation, it is observed that maximum 3 times increase in amplitude of acceleration when speed has been varied from 600 rpm to 1500 rpm at constant load of 10 kg.

Similarly, during finite element analysis, it is observed that maximum 3 times increase in amplitude of acceleration when Speed has been varied from 600 rpm to 1500 rpm at constant load of 10 kg.
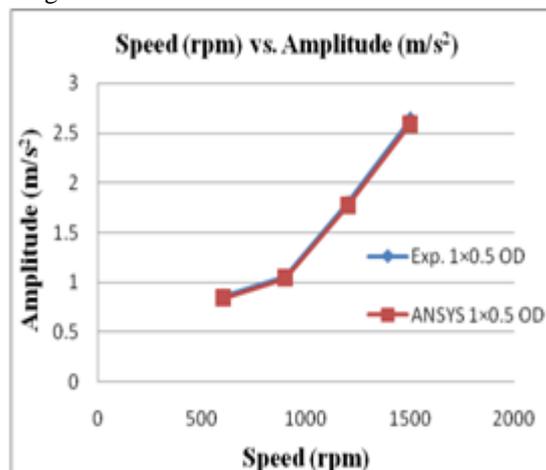

Fig. 15 Graph of Speed vs. amplitude for 1×0.5 outer race defect at 10kg

Below table 3 shows % deviation in amplitude of acceleration. During experimentation, it is observed that maximum 3.87 times increase in amplitude of acceleration when speed has been varied from 600 rpm to 1500 rpm at constant load of 10 kg.

Similarly, during finite element analysis, it is observed that maximum 3.87 times increase in amplitude of acceleration when Speed has been varied from 600 rpm to 1500 rpm at constant load of 10 kg.

Table-3: Show Result for 0.5mm outer race defect at 10kg load

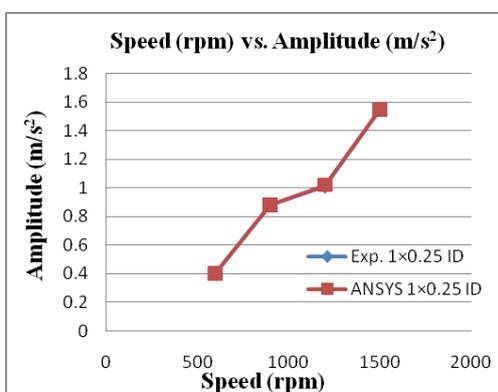| Speed	Experimentation		FEA		Deviation in %
	I 1×0.5mm		I 1×0.5mm		
	Frequency Hz	Amplitude m/s^2	Frequency Hz	Amplitude m/s^2	
600	201	0.4	201	0.4	0
900	127	0.88	127	0.88	0
1200	158.7	1.01	192	1.02	1
1500	197.8	1.55	212	1.55	0

Fig. 16 Graph of Speed vs. amplitude for 1×0.5 inner race defect at 10kg

From table 2 and 3 it is observed that amplitude of outer race defect is more than that of inner race defect. Also deviation in frequency occurs due to uncontrolled parameters during experimentation. As amplitude changes with respect to frequency also it changes from node to node. Here, nodes on the surface of bearing are picked up to get the result. Also due to birth death of element, large deviation occurs in frequency.

V. CONCLUSION

The proposed simulation method is used to determine the vibration signal response for various shaft speeds and loading condition, which is compared with experimental result. It is found simulated vibration pattern has similar characteristic compare to experimental results. The deviation in amplitude of acceleration is may be due to variation of mesh density in the region near to defect and also deviation in frequency occurs due to uncontrolled parameters during experimentation.

It is observed that as the speed increases the peak amplitude values (m/s^2) gradually increases for constant load. Also the peak amplitude values (m/s^2) for healthy bearing are smaller than defective bearing.

It is found that the amplitude values for the case of outer race defect are more than that for the inner race defect. It is because of defect present on the outer race is remained in the load zone at maximum position as in second case, inner race moves in and out of the load zone during each revolution of the shaft. The strong fault vibration spectrum produced while the defect is in the load zone and weaker fault vibration spectrum produced while the defect is outside the load zone.

Acknowledgements

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