Rotor bearing system FEA analysis for misalignment

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ABSTRACT

Rotating machine have common problem of bearing misalignment. In this paper present rotor bearing system model which is used for prediction and solving of real time problem in rotating machine. This technique reduces losses due to failure of machine. The rotor based model design used to monitoring vibration for both offset and angular misalignment of rotating bearing. The result obtain from model fault analysis compared with actual machine, so from the fault analysis result we can predict offset and angular misalignment in actual machine. This technique in industries will increase efficiency of machine by reducing dead time for repaired it.

Keyword: - *Rotor-Bearing system, vibration analysis, misalignment, Condition monitoring*

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1. INTRODUCTION

Misalignment is probably the most common cause of machinery malfunction. Considering the importance of alignment, the vibration spectra of alignment is not well documented. Experience has shown that diagnosis of misalignment through vibration analysis can be extremely difficult due in large part to the observed substantial variability in the character of machinery vibration even when apparently identical alignment states exist.
Misalignment is the most cause of machine vibration. Understanding and practicing the fundamentals of rotating shaft parameters is the first step in reducing unnecessary vibration, reducing maintenance costs and increasing machine. In industry 30% of the machine's down time is due to the poorly aligned machine. The experimental studies has carried out to generate data and discussed the development of neural network simulator for prediction of faults like mass unbalance, bearing cap loose, play in spider coupling and rotor with both mass unbalance and misalignment and health machine network, also numerically evaluated the effect of coupling misalignment on vibration response of the rotor, and obtained various vibration responses as characteristic signature of misaligned shafts. A model for the flexible coupling–rotor–ball bearing system is derived, including reaction loads from deformations of rolling elements of bearing and coupling elements as the misalignment effects. From orbital analysis, anisotropy of bearing stiffness was suggested as the misalignment indicator. Also comprehensive experimental studies in the rotating machinery has done for the faults like misalignment (parallel and angular misalignment), unbalance, mechanical looseness, rotor rub, bearing clearance and crack at the mid span of the shaft. Combination of faults like combined parallel and angular misalignment, combination of faults like crack and unbalance in rotors has also been carried. The reliable diagnostic methodologies have proposed for automatic fault diagnosis. Including statistical, polynomial, neural network, fuzzy and neuro fuzzy techniques offer a framework for analysis. Also build functional fuzzy models and fully transparent fuzzy systems. The described fuzzy model and systems are useful for the diagnosis of faults in noisy and multiple fault scenarios like misalignment and unbalancing

2. LITERATURE SURVEY

Many research works has been done in this field Every researcher has done experimental investigation of misalignment on shafts for confirmation of the results obtained by theory with the experimental results. Following is the literature review of some papers giving more information about contribution of various researchers in defect analysis of rotating shaft.

Sunil Pandey, Prof.B.C.Nakra [1] studies the effect of RMS Vibration acceleration in vertical , horizontal and axial direction of rolling element bearing using piezoelectric accelerometer.

Ali Vaziri, Prof. M. J. Patil [2]done experimental process to obtain the RMS acceleration values of vibration of cracked shaft. Results shows that if rotating speed is increased there is increase in overall R.M.S. value of vibration acceleration depends on crack's depth of cracked shaft. The comparison of overall R.M.S. value of vibration acceleration in difference bearing and crack's depth and difference speed shows that the natural frequency changes substantially due to the presence of cracks and this changes depending upon the location and size of cracks. RMS value of acceleration is higher in bearing which is away from motor. The position of the cracks can be predicted from the deviation of the fundamental modes between the cracked and un-cracked shaft. The frequency of the cracked shaft increases with increase in the crack depth for the all modes of vibration.

Alena Bilosova [4]studied the effect of loose rotating parts on the dynamic behavior of rotor bearing systems. An Experimental model along with a theoretical approach was used in the investigation. The experimental and analytical results show the existence of sub-synchronous self-excited vibrations with frequencies related to shaft rotating speed, also investigated the dynamic behavior of rotor-bearing-stator systems with stationary or rotational joint looseness. They used chaos theory for studying the machine behavior due to mechanical looseness and concluded that harmonic and sub-harmonic responses, as well as chaotic patterns of vibrations are the main characteristics of such systems.

Arun Kr.Jalan , A.R.Mohanty [5] describe the model based technique for fault diagnosis of rotor –bearing system.in this technique residual forces due to presence of fault are calculated .These residual forces are compared with the equivalent theoretical forces due to faults. The fault condition and location of faults are successfully detected by this model based technique.

Steve J.Rothebergn [6] presented a neural network predictor approach for analyzing vibration parameters of the ball bearing-shaft system. The system has been employed with two different working conditions. The simulation results obtained have supported the theory that the Feed-forward artificial neural networks (FNN) was able to represent different types of ball-bearing systems. The neural network was also trainable using the simple back propagation algorithm to analyze the ball-bearing system with two conditions. In future, this type of the neural networks could be used as a predictor of bearing system in experimental applications

Oliver Tonks, Qing Wang [7] present the potential temperature technique for use in wind turbine condition monitoring. Through this technique creation of misalignment , a temperature rise at the coupling mid-point was successfully detected

V. Hariharan and P.S.S. Srinivasan [11]verified and investigated rotor dynamic characteristics related to misalignment throughout the experimental and simulation works. Experimental predictions are good agreement with the numerical (ANSYS) results. Both the measured and numerical results spectra shows that misalignment can be characterized primarily two times shaft running speed (2X). However, misalignment (2X shaft running speed) effect is not close enough to one of the system natural frequencies to excite the system appreciably. Therefore in some case the misalignment response is hidden and does not show up in the vibration spectrum. On the other hand, if 2x shaft running speed is at or close to one of the system natural frequencies, the misalignment effect can be amplified and a high acceleration level at 2X shafts running speed is pronounced in the frequency spectrum.

Lutfi Arebi, Fengshou Gu and Andrew Ball[3] has used developed wireless accelerometer node which is based on a MEMS accelerometer ADXL202AE with a duty cycle output. This work presents a comparison between IAS waveform from conventional encoder and a newly designed wireless accelerometer waveform to investigate shaft misalignment. As the wireless sensor is mounted directly on the shaft, the true dynamics of the shaft can be recorded.

Mohsen Nakhaeinejad Suri Ganeriwala [8]has done experimental studies by using the Spectra Quest's Machinery Fault Simulator (MFS) was used for misalignment experiments. Results suggested vibrations and forces of a machine with rigid coupling to be more sensitive to the parallel misalignments than angular misalignments. Investigating axial forces in frequency-domain reveals significant 3X and 5X harmonics for angular and 3X and 6X harmonics for parallel misalignments.

Irvin Redmond [9] analyzed the vibration resulting from a simple mathematical misalignment model which would exhibit the basic characteristics of real rotor dynamic systems and thereby enable investigation of this common but complex phenomenon. The system equations show clearly that parallel misalignment introduces a static displacement in addition to fundamental-frequency (1X) lateral and torsional excitation components. A discrete second-harmonic (2X) torsional excitation term is also evident in the system force vector. The magnitude of this term is directly proportional to the support anisotropy and disappears for isotropic supports.

Adewusi [10] has explained the application of neural networks for rotor crack detection. The basic working principals of neural networks are presented. Experimental vibration signal of rotor with and without propagation crack were used to train the multi-layer feed-forward algorithm. The trained neural networks were tested with other set of vibration data, A simple two layer feed-forward neural network with two neurons in the input layer and one neuron in the output layer trained with the signals of cracked rotor and a normal rotor and a normal rotor without a crack. Trained three-layer network were able to detect both the propagation and non-propagation cracks. The FFT of the vibration signal showing variation in amplitude of the harmonics as time progresses are also presented for comparison.

3. PROBLEM DEFINITION

Rotor bearing misalignment will influence on machine operation .Misalignment will lead to failure of machine, failure of cost and failure of time. The majority of misalignment studies in the past is theoretical whereas experimental investigations are relatively limited. The theoretical studies are often investigated with simplified assumptions. The outcome of these studies may not be accurate, since in practice there are many more sources of observed vibration characteristics in an actual rotor system.

There is a need for comprehensive experimental study in the rotating machinery faults. In this paper, effects of parallel and angular misalignment of rotating shaft will be carried out. Comprehensive experimentation will carried out monitor and analysis of vibrations for a rotating shaft supported by ball bearings. The RMS acceleration will be carried out for the vibration due to different faults like parallel misalignment, angular misalignment, and combined parallel and angular misalignment at bearing supports.

4. PROPOSED WORK

4.1 Objectives Of Project

To study numerical and experimental vibration characteristics of rotor bearing system due to misalignment the following objectives has to

- To study the effect of misalignments on the vibration of rotor bearing system.
- To study rotor bearing system numerically
- To Study vibration spectrum of bearing supports due to the misalignments of rotor bearing system.
- Comparison of numerical and experimental amplitudes of vibrations of rotor bearing system due to misalignments

4.2 Methodology

The following methodology is used for completion of the work:

1. To prepare a model in finite element method of rotating shaft by using FEA package (ANSYS-14), this method is used to study the effect of misalignment numerically.
2. To carry out Modeling and simulation of rotor bearing system with perfect alignment condition i.e. without misalignments and with misalignment, for numerical measurements of amplitudes of vibrations in terms of RMS values of acceleration using FEM (ANSYS-14).
3. Numerical measurements of amplitudes of vibration of rotor bearing system with angular misalignment, parallel misalignment and combination of angular and parallel misalignment.
4. Fabrication of rotor-bearing system with readily availability of the required components of rotor bearing system, and preparation of provisions to bearing supports for getting artificial misalignments.
5. Experimental measurements of amplitudes of vibration in terms of RMS values of acceleration of perfectly balanced rotor bearing system (without misalignments) by using 10Hz, 04 channels Fast Fourier Transform (F.F.T.) analyzer.
6. Experimental measurements of amplitudes of vibration of rotor bearing system with angular misalignment, parallel misalignment and combination of angular and parallel misalignment.
7. Finally compare the numerical and experimental amplitudes of vibrations of rotor bearing system for various misalignment conditions.

5. FINITE ELEMENT ANALYSIS OF ROTAR BEARING SYSTEM

Initially ANSYS LS-DYNA module selected for modeling and meshing of rotor bearing system model. Geometry drawn into Cad package that is in CATIA, then it is imported in Ansys module.Beam188 element is selected with circular orientation because it can tolerate irregular shape without loss of accuracy. Mapped meshing selected for better accuracy, in this meshing time consumption is maximum as compared to free meshing.

Table -5.1: Geometric and material properties

Material	EN 8
Length	700mm
Diameter	20mm
Density	7860kg/m^3
Poisson's Ratio	0.3
Young Modulus	2x10^5 MPa

Boundary condition and loading

The rotor shaft is supported by two identical pedestal bearing and has length of 700mm with bearing span of 300 mm, reason to select pedestal bearing is that it should be fully supported on a flat, rigid surface to avoid distortion of the pedestal or deflection under load. The bearing P204 type is represented by COMBIN 40 Element and stiffness of bearing is 1.5E4 N/mm. angles. 2900 RPM was applied over the nodes of shaft along X axis. While meshing the different models, types of finite elements are kept same but size of the element has been varied so as to obtain the effect of rotation of shaft. A Perfect aligned meshed model of rotor bearing system is as shown in fig. 5.1.

Case – 1

Perfect Aligned Condition of Rotor Bearing System

- Offset distance: 0
- RPM: 2900 RPM
- Boundary Condition: 2900 RPM was applied over the nodes of shaft along X axis.
- Type of analysis in ANSYS: Transient dynamic
- ANSYS Module: LS-DYNA
- Number of nodes:-18400

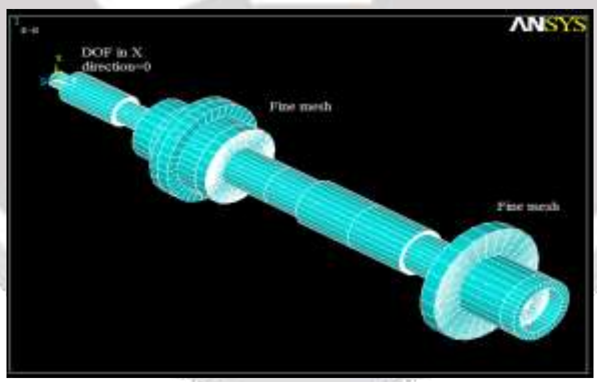

Fig -5.1: A perfect aligned meshed model of rotor bearing system (nodes 18,400)

Case – 2

Parallel Misalignment at Bearing No: 01 & 02.

- Offset distance:-0 to 1 mm (interval 0.2)
- RPM: 2900 RPM
- Boundary Condition: 2900 RPM was applied over the nodes of shaft along X axis. Horizontal movement of shaft is constrained at one end of shaft
- Type of analysis in ANSYS: Transient dynamic
- ANSYS Module: LS-DYNA
- Number of nodes:-18400

Maximum deflection values of parallel misalignments 0.205 mm and 0.638 mm in first and second step respectively have been obtained. Their contour plot is as shown below in fig. 5.2 for bearing support-1 and fig. 5.3 for bearing support-2.

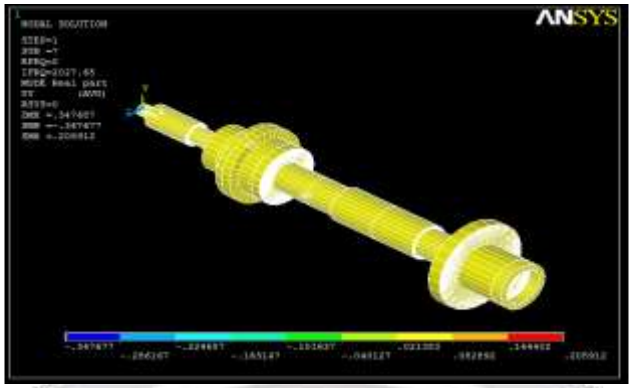

Fig -5.2: Maximum deflection values of parallel misalignments at bearing 1

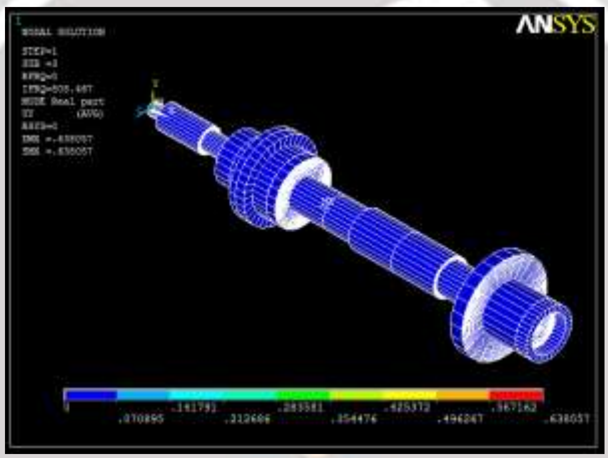

Fig -5.3: Maximum deflection values of parallel misalignments at bearing 2

Case – 3

Angular Misalignment at Bearing No: 01& 02

- Angular misalignment:- :-0 to 2.250 (interval 0.500)
- RPM : 2900 RPM
- Boundary Conditions: 2900 RPM was applied over the nodes of shaft along X axis. Y and Z axis movement of shaft is constrained at one end of shaft .X axis movement and rotation is free so as get effect of rotation.
- Type of analysis in ANSYS : Transient dynamic
- ANSYS Module: LS-DYNA
- Number of nodes:-18400

In the first step angular misalignment was done at bearing number 1 and angular misalignment at bearing number 2 was done in second step. Maximum deflection values are 0.6mm and 0.8 mm in first and second step respectively have been obtained. Their contour plot is as shown below in fig. 5.4. and fig. 5.5 respectively.

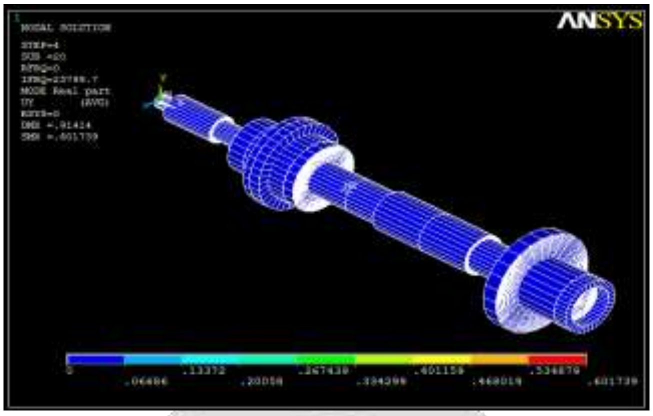

Fig -5.4: Maximum deflection values of angular misalignments at bearing 1

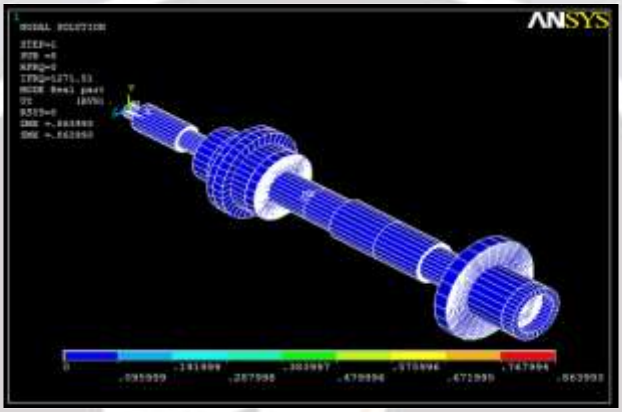

Fig -5.5: Maximum deflection values of angular misalignments at bearing 2

Case – 4

Parallel and Angular Misalignment at Bearing No: 01& 02

- Parallel and Angular misalignment:- 0 to 1.00 mm (interval 0.10), 0 to 2.00 (interval 0.250)
- RPM: 2900 RPM
- Boundary Conditions: 2900 RPM was applied over the nodes of shaft along X axis. Y and Z axis movement of shaft is constrained at one end of shaft. X axis movement and rotation is free so as get effect of rotation.
- Type of analysis in ANSYS: Transient dynamic
- ANSYS Module: LS-DYNA
- Number of nodes:-18400

Meshed model of this case is same as that of angular misalignment, because numbers of nodes are increased during angular misalignment. Maximum deflection values of combined parallel and angular misalignments are 0.791 mm and 0.687 mm in first and second step respectively have been obtained. Their contour plot is as shown in fig. 5.6 and fig. 5.7 respectively.

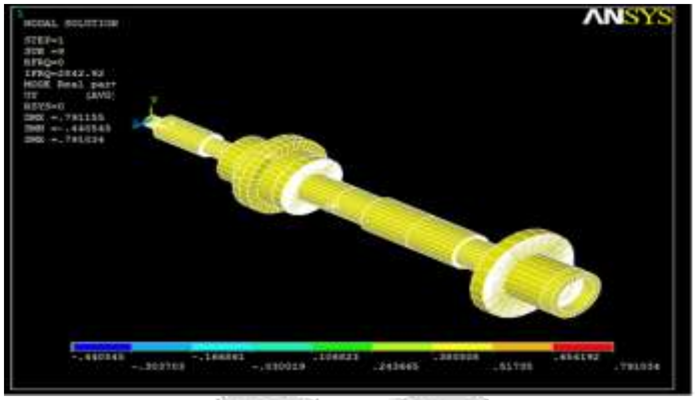

Fig -5.6: Maximum deflection values of combined parallel and angular misalignments at bearing 1

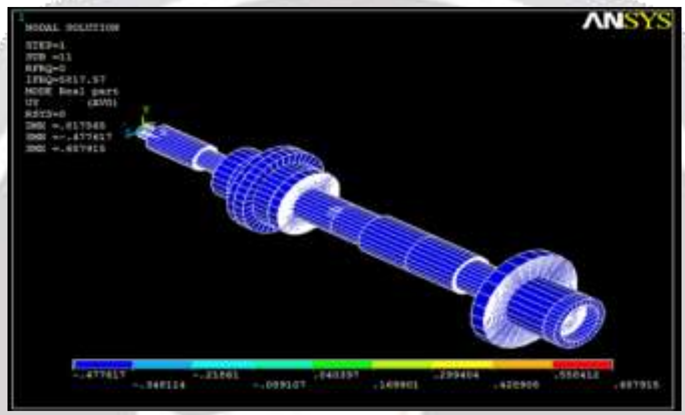

Fig -5.7: Maximum deflection values of combined parallel and angular misalignments at bearing 2

6. FEA RESULTS AND DISCUSSION

The results obtained by FEA analysis are as follows: The obtained RMS acceleration values of vibration in m/s2 by FEA method for parallel misalignment at bearing support-2 are tabulated in the table 6.1.

Table -6.1: RMS Acceleration for parallel misalignment at bearing no: 02

Offset distance(in mm) from the centre of rotation of		FEA Results RMS Acceleration in m/s^2	
Bearing 01	Bearing 02	Vertical	Horizontal
0	0	3.111	4.11
0.2	0.2	4.917	5.111
0.4	0.4	6.129	6.192
0.6	0.6	7.211	8.101
0.8	0.8	7.793	8.901
1	1	8.511	9.413

Chart 6.1 shows the graph of RMS acceleration in m/s2 verses offset distance of bearing support 2 from the center of rotation of the shaft. This graph is plotted by taking offset distance on X-axis and by taking FEA RMS values of acceleration in m/s2 on Y-axis.

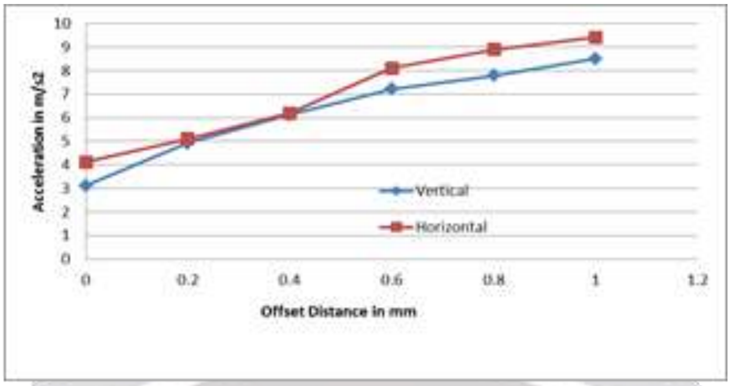

Chart 6.1: RMS Acceleration Vs Offset distance from the centre of rotation at bearing no: 02

The RMS acceleration values of vibration in m/s2 by FEA method for parallel misalignment at bearing support-1 in vertical and horizontal direction are tabulated in the table 6.2.

Table -6.2: RMS Acceleration for parallel misalignment at bearing no: 01

Offset distance (in mm) from the Centre of rotation of		FEA Results RMS Acceleration in m/s^2	
Bearing 01	Bearing 02	Vertical	Horizontal
0	0	4.121	4.121
0.2	0.2	4.182	5.121
0.4	0.4	4.999	5.819
0.6	0.6	6.191	5.918
0.8	0.8	6.712	7.121
1	1	7.121	6.102

Chart 6.2 shows the graph of RMS acceleration in m/s2 verses offset distance of bearing support-1 from the centre of rotation of the shaft. This graph is plotted by taking offset distance in millimeter of bearing support-1 from centre of rotation of the shaft on X-axis and FEA results and experimental results for RMS values of acceleration in m/s2 are taken on Y-axis.

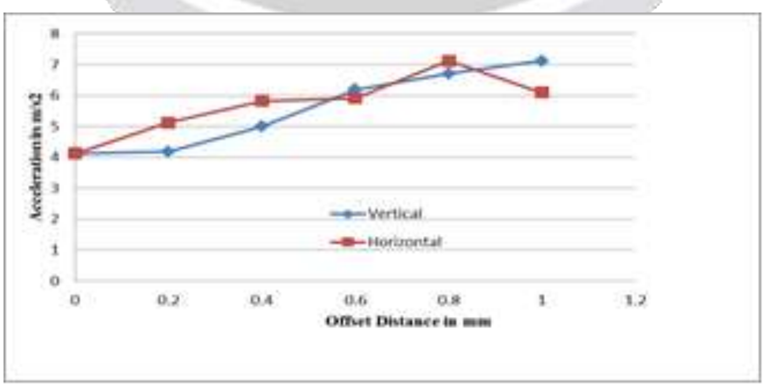

Chart 6.2: RMS Acceleration Vs offset distance from the center of rotation at bearing no: 01

From Chart 6.1 it is observed, the amplitude of vibration in RMS acceleration is higher in horizontal direction than that of amplitude in vertical direction in case of parallel misalignment at bearing no-02 which is away from the motor, From above Chart. 6.2 it is observed that in case of parallel misalignments the RMS values of acceleration are lesser at bearing support-1 (near to the motor) in both horizontal and vertical directions. The highest value of overall RMS accelerations is obtained at bearing support-2. It is also observed that the amplitudes are higher in horizontal direction in both the bearings and increase when misalignments increased.

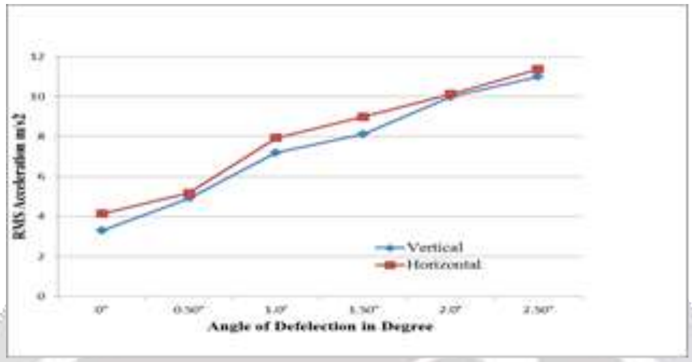

Chart 6.3: RMS Acceleration Vs Angle of deflection at bearing no: 02

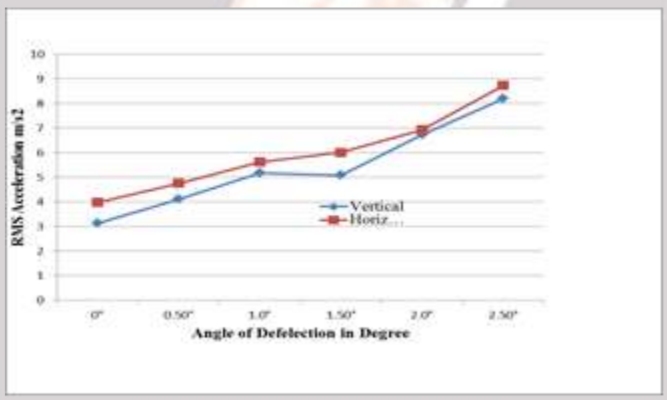

Chart 6.4:RMS Acceleration Vs Angle of deflection at bearing no: 01

Chart 6.3 and Chart 6.4 shows the graph of RMS acceleration in m/s2 verses angular deflection in degree for bearing support-2 and bearing support-1from the center of rotation of the shaft respectively. This graph is plotted by taking angular deflection in degree for bearing support-2 and bearing support-1 from center of rotation of the shaft on X-axis and RMS values of acceleration in m/s2 by FEA.

Chart 6.5 and Chart 6.6 shows the graph of RMS acceleration in m/s2 verses degree of combined parallel (in mm) and angular (in degree) misalignment for bearing support-1and bearing support-2 respectively. The graph is plotted by taking misaligned distance in millimeter of both supports from the center of rotation and angular deflection in degree, from center of rotation of the shaft on X-axis and RMS values of acceleration in m/s2 by FEA as well as by experimental are taken on Y-axis. From Chart 6.5 and Chart 6.6 it is observed that in case of combined parallel and Angular misalignments the amplitude of vibration in RMS acceleration is higher in horizontal direction than that of amplitude in vertical direction at bearing support-1 and bearing support-2

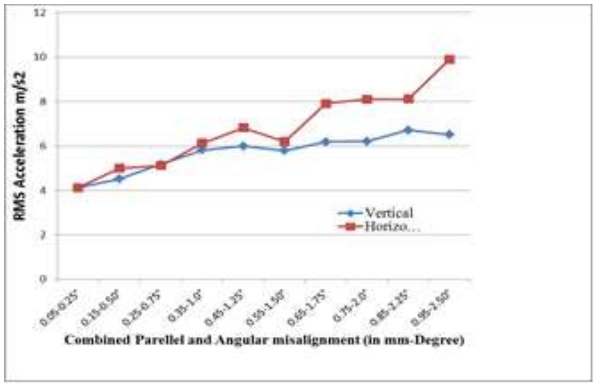

Chart 6.5: RMS Acceleration Vs combined Parallel and Angle of deflection at bearing no: 01

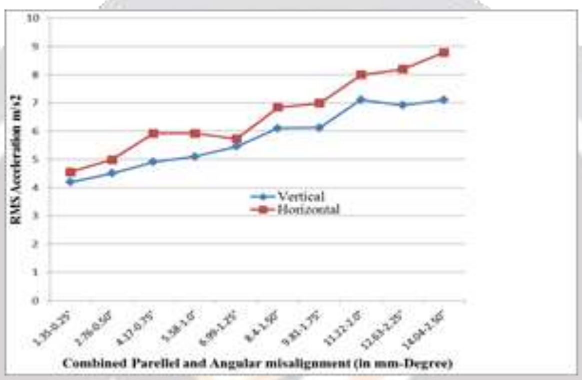

Chart 6.6: RMS Acceleration Vs combined Parallel and Angle of deflection at bearing no: 02

7. CONCLUSIONS

ANSYS simulation is carried out for present work. The following conclusions can be drawn:

1. This Rotor bearing Model based analysis method used to monitor vibration in both offset and angular misalignment. This method has advantage that, systems like Automobile Gear Box, Wheel Bearing, Turbine Rotor where actual fault analysis is difficult, so by Model Based analysis we can predict the type of fault and location.
2. Overall magnitude of vibration in RMS values of acceleration for various misalignment conditions like parallel misalignment, angular misalignment and combination of parallel and angular misalignment are found higher at bearing support-2 (away from the motor) and lesser at bearing support-1(near to the motor) in each the horizontal and vertical directions.
3. Conjointly it's found that the highest value of overall RMS accelerations for vibration is higher in horizontal direction and found less in vertical direction just in case of each bearing supports, and will increase when misalignments exaggerated.

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