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Vibration Signature analysis of defective deep groove ball bearings by Numerical and Experimental approach.

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Abstract: Bearing is crucial part of any rotary components and its failure causes disastrous failure of machinery. Vibration signature analysis is one of the most effective tools for monitoring the condition of ball bearings. Best method to study the failure analysis of ball bearing is by creation of artificial cracks of different sizes on various elements and noting down its signatures. It takes long time for life test where healthy bearings are rotated till initialization of crack. From the literature survey, it is observed that most of the work was carried out at one particular speed up to 1500 rpm. Therefore, author finds a scope for faulty bearing operation and measurement of amplitude of vibrations at different speeds from 1000 to 5000 rpm, different loads up to 200N and at various defect sizes ranging from 250 micron to 2000 micron on bearing races.

The model has been developed as spring mass system by assuming races as masses and balls as spring. 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. The actual measurements of vibration amplitudes of bearings with artificial local defects have been carried out for verification of the numerical results. It is noted that the numerical results show good agreement with the experimental results. Support bearings are standard during the entire experimentation.

Index Terms :- ball bearing, race defect, defect frequencies, vibration spectrum, amplitudes, time domain, local defect.

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1. Introduction

LIFE of any rolling element bearing depends upon some of the important parameters like operating conditions, quality of lubrication and presence of surface irregularity if any. These parameters are main sources of vibrations in the rotor bearing system. These vibrations affect the endurance strength of bearing leads to its failure in turns failure of machinery [1]. Presence of surface irregularity (local defects like pits, cracks and distributed defects like waviness, roughness etc) in bearings is one of the most commonly observed parameters in failure analysis. Some of the important diagnostic techniques such as vibration measurement, sound intensity, shock pulse method, stator current method and Acoustic emission technique are reviewed by few researchers [2]. Accordingly, it was noted that vibration measurement is one of the most effective and widely used techniques for bearing failure analysis. A detailed study of forces generated due to waviness in ball bearing is carried out by Wardle [3]. A model has prepared by Choudhury and Tandon [4] for study of waviness in the races of roller bearing. The model predicts that the amplitudes of spectral components due to outer race waviness were much higher as compared to those due to inner race waviness. If waviness is controlled during manufacturing then main source of vibrations are the local defects like pits, cracks produced due to unhealthy operating conditions. The effort has carried out by McFadden and Smith [5] and also by Choudhury and Tandon [6] in developing the model for local defect on the bearing races. Gunhee et al [7] presents an analytical model to investigate vibration due to ball bearing waviness in a rotating system taking account of the centrifugal force and gyroscopic moment of the ball. This research shows that the centrifugal force and gyroscopic moment of the ball plays the important role in determining the bearing frequencies.

Vibration monitoring of rolling bearing by High Frequency Resonance Technique was reviewed by McFadden and Smith [8]. Each stage of HFRT in the terms of success and failures was examined and approaches of each investigator were compared. Few authors [9] have studied two point defect models where the original model was extended to describe the vibration produced by multiple point defects, thereby enabling large defects to be modeled by treating them as the sum of a number of point defects. Some researchers [10] have also prepared model for prediction of local defects. Prediction of amplitude becomes difficult because of the complex nature of system resulting from assembly of bearing elements and mounting of the same on the shaft and housing. Root cause failure analysis of outer ring fracture of four-row cylindrical roller bearing was carried out by H. Hirani [11]. In his study, a visual examination of failed rolling surfaces was emphasized.

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The effect of radial clearance, number of rollers and viscosity on the bearing noise was examined by Byoung-Hoo Rho et al [12]. It was concluded from their study that the fundamental frequency of the noise components of the cylindrical roller bearing corresponds to the multiplication of the number of rollers and the whirling frequency of the roller center.

Few authors [13] have also compared the effectiveness of vibration analysis and Acoustic Emission techniques (AET) and concluded advantages and limitations of both the techniques. According to their study though AET gives early prediction of the defect, it is very difficult to process the acoustic signals. Vibration analysis is commonly adopted and easy process to monitor the condition of rolling bearing. With large defects and medium operating speeds, most methods of detection works satisfactorily but at low and high speeds, problems arise. Smith [14] describes work on measurements at low speeds and discusses the difficulties which arise. H. Arslan et al [15] have carried out an investigation of rolling element vibrations caused by local defects at higher speed. The model was prepared by assuming the races as a springs and balls as mass. These authors describe the possibility of identification of the defected element of ball bearing by studying ball vibrations using the simulation model.

After an exhaustive literature survey, the scope for the work is as follows;

It is observed from the exhaustive literature survey that, most of the work was carried out for the shaft speed from 60 to 1500 rpm. The previous work was focused on the cantilever arrangement for loading on the bearing and also not validated by numerical approximation methods i.e. using ANSYS.

The present work is focused on development of mathematical model for deep groove ball bearing with simply supported arrangement for loading on test bearing. The efforts have been carried out by measurement of amplitudes of vibration of test bearing by FEM using ANSYS/LS-DYNA.

Further the work is also focused on measurement of amplitudes of vibration of test bearings experimentally with operational speed range 1000 - 5000 rpm with the defect size ranging from 0.25 mm to 2.00 mm at the outer and inner rings.

The present research work involves numerical and experimental approach to study the effect of defect size as well as speed of rotation on amplitudes of vibration and defect frequencies.

2. Numerical Approach using ANSYS

Finite element method is one of the several methods for solving complicated engineering problems. Scope of finite element method is to find the solution of a complicated problem by replacing it by a simpler one. Since the actual problem is replaced by a simpler one in finding the solution, it is able to find only an approximate solution rather than the exact solution. Thus Finite Element Method (FEM) is a numerical approximation method. The present work deals with, modeling and meshing of test bearing and solution is obtained for number of defect sizes and at different speeds using LS-DYNA solver. Separate meshing is used for each bearing elements. When defect is on outer ring, Tetrahedron (Solid 168) free meshing is used for outer ring as well as balls. This meshing is very suitable for irregular and complicated parts. For the bearing with defect on outer ring, free meshing is used which is as shown in Fig 2b. For inner ring without defect, mapped meshing Hexahedron (Solid 164) is preferred. But in case of inner ring defected bearings, inner ring is meshed with Solid 168. Here for inner ring defected bearings, outer ring and balls are kept defect free hence mapped meshing Hexahedron (solid 164) element is used for outer ring and tetrahedron is used for ball. Fig 2a and Fig 2b depicts meshed model of test bearing and model of defect on outer ring of bearing. Total numbers of solid elements used are 33971 where as nodes are 54801. Each element of bearing is meshed separately. Bearing is very complicated geometry, even though the author has tried at the best to obtain the close results.

When defect is created on outer ring of bearing, excitation force $Q_2 \sin\omega t$ is responsible for excitation of mass m2. Under this circumstance the excitation force $Q_1(t)$ is zero. Hence for the steady state solution, vibration of mass m2 is written as, $x = X_2 \sin\omega t$. Thus, X_2 gives amplitude of vibration of mass m_2 due to excitation of force Q_2 which is produced by striking of balls with defect on outer ring.

When defect is created on inner ring of bearing, excitation force $Q_1 \sin\omega t$ is responsible for excitation of mass m_1. Under this circumstance the excitation force $Q_2(t)$ is zero. Here, in this case also X_2 gives amplitude of vibration of mass m_2 due to excitation of force Q_1 which is produced by striking of balls with defect on inner ring. It is observed that the amplitude X_2 changes with the excitation force, defect size on outer ring and inner ring and RPM of the shaft. Hence from Fig -1 for two degree of freedom system subjected to excitation forces Q1 and Q2, the , the differential equations of motion are given as [18,19];

$$m_1\ddot{x}_1 + (k_1 + k_2)x_1 - k_2x_2 = Q_1(t)$$
$$m_2\ddot{x}_2 + (k_2 + k_3)x_2 - k_2x_1 = Q_2(t)$$

Therefore, the excitation force Q1 obtained when pulses are generated on the inner race. Similarly excitation force Q2 obtained when pulses are generated on the outer race. These excitation forces are proportional to rolling element load (P) which is function of bearing deflection δ and distribution factor. Fig 1a shows cross section of test bearing and physical model of bearing system.

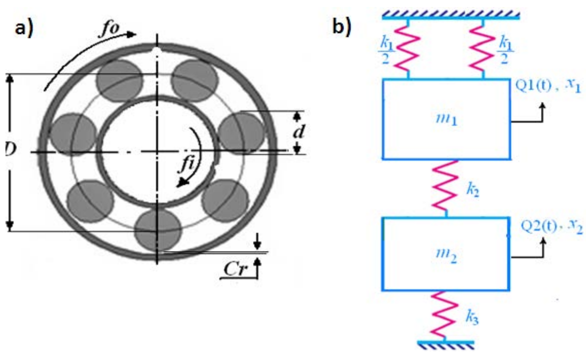

Fig 1:- a) Cross section of test bearing b) Physical model of bearing system

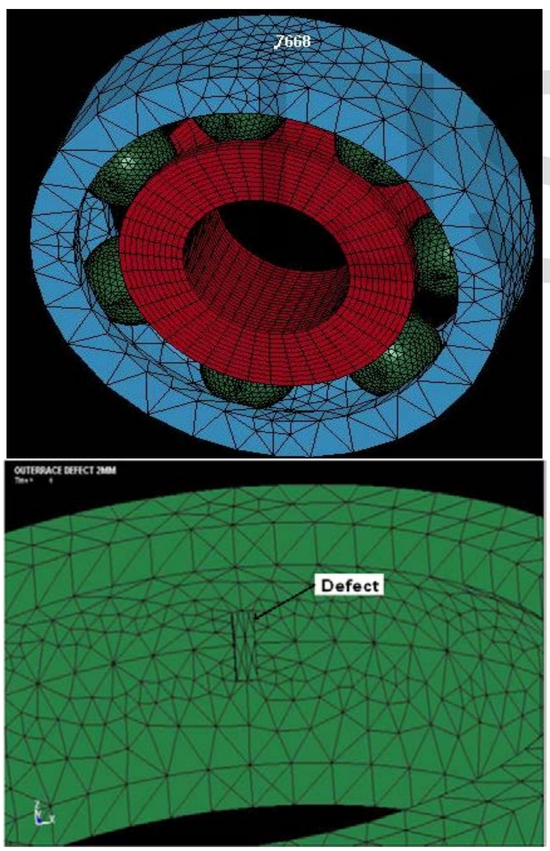

Fig 2:a) Meshed model of bearing with outer ring defect b) Defect on outer ring of Bearing

3. Experimental Approach

The test rig has been developed for confirmation of the predicted results after referring the results obtained from the trial set-up. The developed test rig is as shown in Fig-3. Mounting table of size 100cm x 100cm with mass 800 kg approximately is designed and fabricated to reduce the shocks and vibrations produced due to electric motor and rotating components etc. A set up consists of 3HP/2880 rpm three-phase induction motor and output shaft which is mounted on table. Radial load is applied on test bearing by tension rod and measured by load cell. Test bearing is mounted in between two support bearing. The support bearings are healthy (Defect free) bearings and the test bearings are healthy as well as defective. Split type housing is designed for mounting and dismounting ease of test bearing. A special purpose bearing DFM 85 supplied by DJR Deluxe Bearings Ltd, Pune is used as test bearing which is single row deep groove ball bearing with contact angle zero and normally used in four wheeler engines. A piezo-electric accelerometer of capacity 10 kHz with magnetic base is mounted on the housing of the test bearing. The accelerometer is connected to OROS made 4 channels FFT Analyzer which processes the time signals. The output of analyzer is connected to computer which has the relevant hardware and the software to acquire the data. The data has been stored and displayed in the form of time domain signal. A defect size of 250 micron to 2000 micron width with depth of 100 micron is created on outer and inner ring of the bearings and operated at different speeds ranging from 1000 to 5000rpm and with load varies from of 5kg to 20kg. Sensor has mounted at the maximum position in the load zone that is at the top of bearing. Proper cleaning of bearings was carried out before application of grease to make them free from contaminants, if any. Support bearings are healthy during the entire experimentation and there vibration levels are also measured time to time for confirmation. In experimental approach a focus is on Time domain and frequency domain analysis for defected outer ring and inner ring.

3.1 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 has been incorporated and amplitudes of vibrations are measured. High frequency range accelerometer of 10 kHz capacity and sensitivity 97.5mV/g is mounted on the test bearing housing. The sampling rate of 25.6 kilo samples per second 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. The percentage error of 0.2 to 0.6 only has been observed in the measurement of amplitude of vibration.

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The amplitudes of vibrations of test bearings are measured at different speeds, loads and defect sizes on outer ring as well as inner ring of bearings under following different conditions.
i) At constant speed & radial load with variations in defect sizes. ii) At constant defect size & radial load with variations in speed. iii) At constant defect size and speed with variations in radial load at test bearing.
In case of outer ring defected bearings, defect is stationary and is always located at maximum position in the load zone. The effort has been carried out in measuring the amplitudes of vibrations of outer ring defected bearings with changing the position of defect in the load zone. As explained earlier, support bearings are healthy for entire experimental work. The efforts have also been carried out by measuring the vibrations of test bearing when support bearings are replaced by defective bearings.

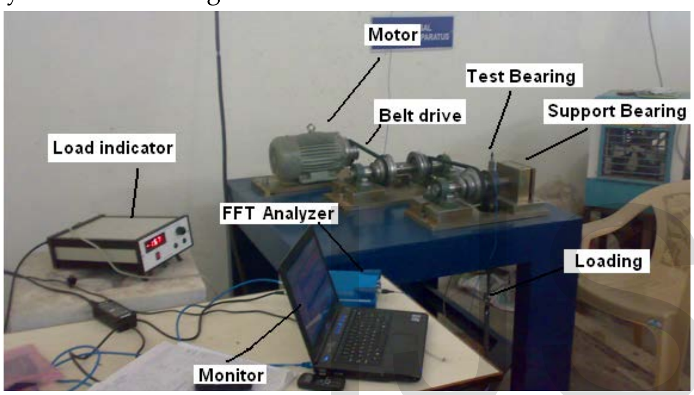

Fig 3: Schematic diagram of Experimental Test Rig

4. Results and Discussion

The numerical and experimental results at different speeds, loads and defect sizes are plotted and comparison has been made. It is observed that, numerical results obtained by ANSYS and both are validated successfully by experimental results.

4.1 Numerical results

The model has been developed in the ANSYS and the results are obtained by considering the same operating conditions. The same trend is observed as that of theoretical results. The time domain plot for 1mm defect on outer ring at 5000rpm is as shown in Fig 4. The impulses are obtained at reciprocal of defect frequency (1/fo) which is 211Hz. But in case of theoretical results, the same frequency is 208Hz and therefore the average error of frequency 3Hz is noted. The reason may be because of limitations of meshing for complex bearing elements. Fig 5a to d depicts the frequency domain plot for 1mm defect on outer ring at 1000rpm, 2000rpm, 3000rpm and 5000rpm respectively. It is observed that amplitudes of vibration increases with increase in speed. In this case the amplitudes of vibration are 29.4mm/s², 124 mm/s², 289 mm/s² and 905 mm/s² at 1000rpm, 2000rpm, 3000rpm and 5000rpm respectively. The time domain plot for 1mm defect on inner ring at 5000rpm is as shown in Fig 6. In this case, the impulses are obtained at reciprocal of inner ring defect frequency 380Hz.
The overall vibration level is less as compared to outer ring defected bearings for the same speed and defect size. Fig 7a to d depicts the frequency domain plot for 1mm defect on inner ring at 1000rpm, 2000rpm, 3000rpm and 5000rpm respectively The corresponding levels of amplitudes are 21.5 mm/s², 107 mm/s², 199 mm/s² and 693 mm/s² respectively.

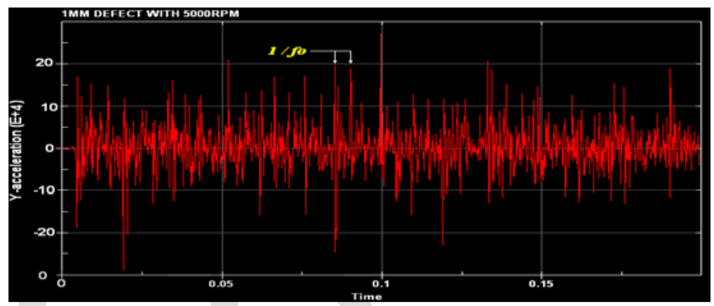

Fig 4: Time domain plot for bearing with 1mm defect on outer ring at 5000rpm (Using ANSYS).

4.2 Experimental Results

Experimental amplitudes of vibrations are measured at various speeds, loads and for different defect sizes on outer and inner ring of bearings. The peak amplitudes are obtained at outer ring defect frequency (fo) i.e. at 42Hz, 84Hz, 126Hz and 208Hz at speed 1000rpm, 2000rpm, 3000rpm and 5000rpm respectively which are almost same as that of theoretical results. The sample experimental result for amplitudes of vibration at 5000rpm and 5kg load for test bearing with 1mm defect width on outer ring is as shown in Fig 8 a) Time domain and b) frequency domain. The impulses are obtained after every 0.0048seconds which is corresponding to reciprocal of outer ring defect frequency 208Hz at 5000rpm. The strong peak at 208Hz (fo) is clearly shown in Fig 8b. Some harmonic peaks are also observed at fo+fs, fo+2fs, fo-fs, 2fo, 3fo etc. where fs is shaft rotation frequency 83Hz.
The peak amplitudes obtained at inner ring defect frequencies are 75Hz, 150Hz, 226Hz and 376Hz at 1000rpm, 2000rpm, 3000rpm and 5000rpm respectively which are again close to theoretical results. The sample experimental result for amplitudes of vibration at 5000rpm and 5kg load for test bearing with 1mm defect width on inner ring is as shown in Fig 9a) Time domain and b) frequency domain.

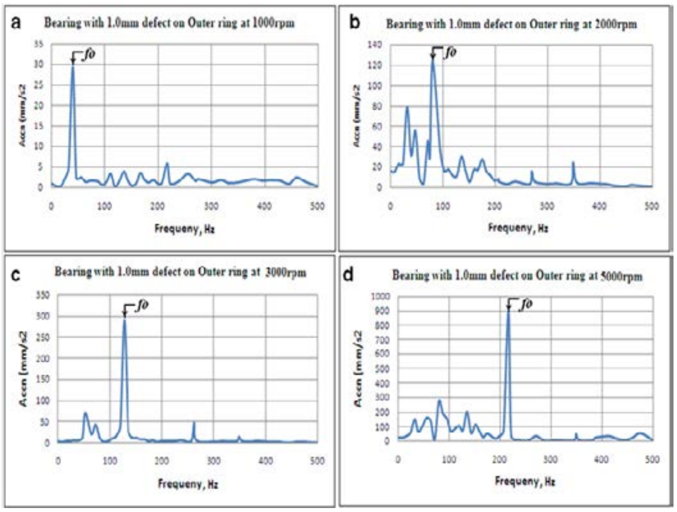

Fig 5:Frequency domain plot for bearing with 1mm defect on outer ring at RPM a)1000 b) 2000 c) 3000 d) 5000

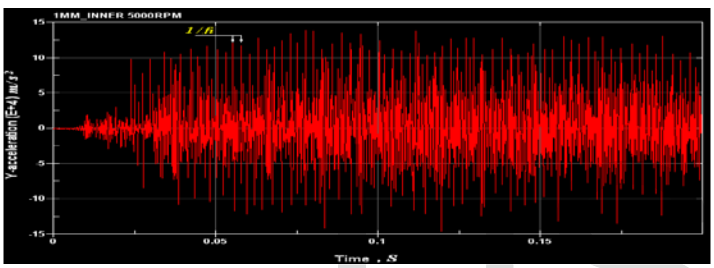

Fig 6: Time domain plot for bearing with 1mm defect on inner ring at 5000rpm (Using ANSYS).

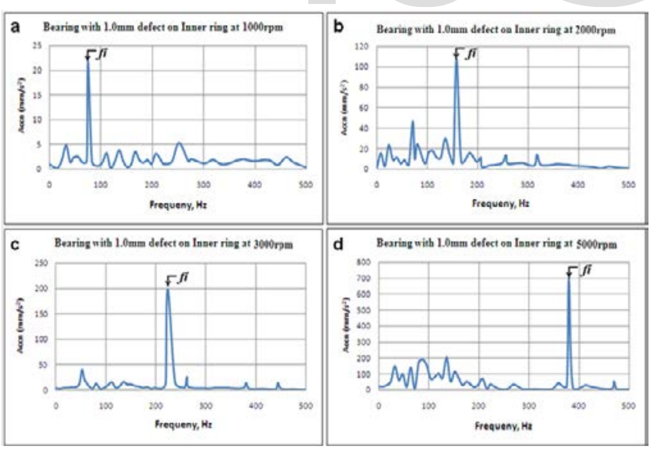

Fig 7:Frequency domain plot for bearing with 1mm defect on inner ring at RPM a)1000 b) 2000 c) 3000 d) 5000

Here the impulses are obtained at every 1/fi period which is 0.00267seconds. A strong peak at inner ring defect frequency, fi is clearly observed as shown in Fig 9b. It also shows some harmonic peaks corresponding to fi+fs, fi-fs, 2fs etc. For any speed and defect size, the outer ring defected bearing gives better response in terms of peaks and characterized defect

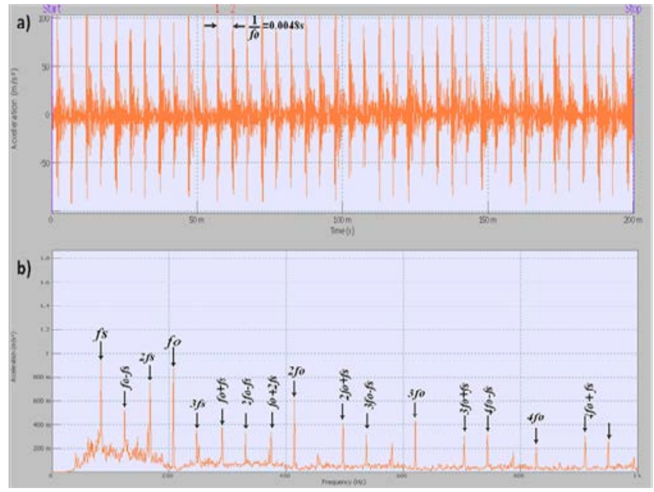

Fig 8: Vibration amplitudes of test bearing at 5000 rpm with 5kg load for 1mm defect size on outer ring a) Time domain b) Frequency domain

frequencies in comparison with inner ring defected bearings which is as shown in Fig 10a and Fig 10b. The reason is that in prior case, the defect unaltered its position with the rotation and remained in the load zone at maximum position and hence no variations in amplitude of vibration is noted where as in second case, defect is rotating with the shaft from maximum to minimum position and vice versa in the load zone and therefore the variation in amplitudes of vibration is noticed.

The vibration amplitudes measured in the spectra of healthy test bearing are 25 mm/s^2 at frequency fo (208Hz) and 15 mm/s^2 at frequency fi (376Hz) at 5000rpm. For outer ring defected bearings with defect size of 0.25mm, 0.5mm, 1.0mm and 2.0mm, the amplitudes of vibrations are 250mm/s^2 , 505 mm/s^2, 955 mm/s^2 and 1844 mm/s^2 respectively at maximum 5000rpm as shown in Fig 10a. Whereas, for inner ring defected bearings with respective defect sizes, the amplitudes of vibrations are 235 mm/s^2 , 454 mm/s^2, 739 mm/s^2 and 1093 mm/s^2 at maximum 5000rpm as shown in Fig 10b.

The efforts have also been carried out in measurement of amplitudes of vibration of outer ring defected bearing in which defect is manually shifted to new position in the load zone. In this situation, it is observed that, the position of defect has significant effect on the peaks. During the experimentation, some non zero amplitudes even outside the load zone are observed which may be due to influence of other source of vibrations in the set up at higher speed.

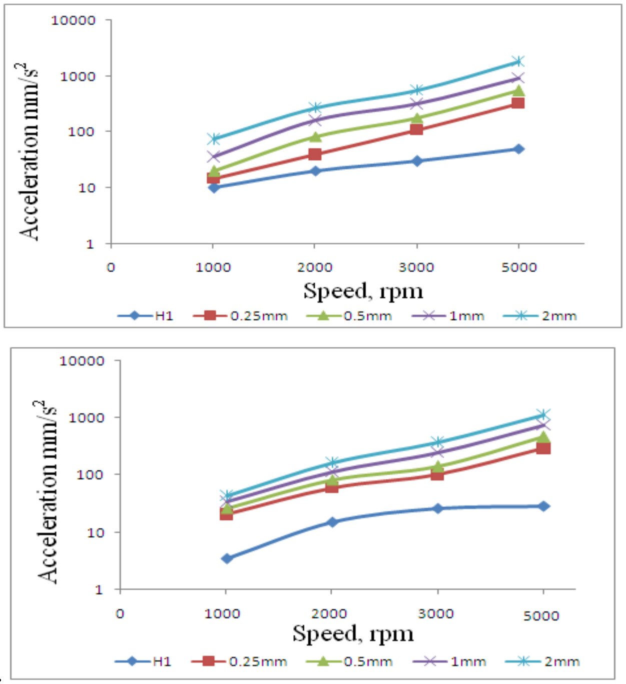

Fig 10:- Vibration amplitudes of test bearings at different speeds with 5kg load for various defect sizes on a) Outer ring of bearing b) Inner ring of bearing

4.3 Comparison of Results:

Comparison of numerical and experimental results at various speeds and loads with various defect sizes on outer ring and inner ring of the bearing has been made. Fig11a and b depicts the numerical and experimental amplitudes of vibration with respect to defect sizes on outer ring and on inner ring bearing at 5000 rpm. It is obvious that, for the defected outer ring bearing, the vibration of amplitude is in the range of 224 to 1840 where as for inner defected bearing it is in the range of 196 to 1093 mm/s².

In all the three approaches, a good agreement is observed for both outer ring and inner ring defected bearings.

5. Conclusions

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

1. At constant speed and constant load with different defect sizes on outer ring, amplitudes of vibration varies with increase in defect size. Thus at 1000, 2000, 3000 and 5000 rpm , the amplitudes of vibration are in the range of 10-60 mm/s² , 40-250 mm/s² ,100-600 mm/s² and 224 to 1840 mm/s² respectively for defect size in the range of 0.25 to 2.0 mm .

Similarly at constant speed and constant load with different defect sizes on inner ring, amplitude of vibration varies with increase in defect size.

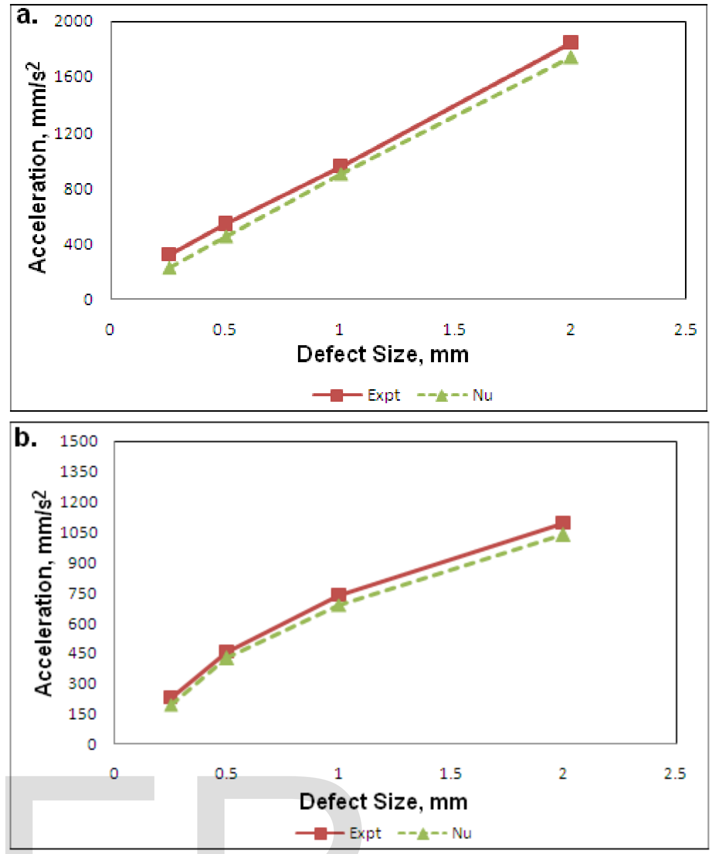

Fig 11:-Comparison of Vibration amplitudes at 5000rpm for various defect sizes on a) Outer ring b) Inner ring

Thus at 1000, 2000, 3000 and 5000 rpm, the amplitudes of vibration are in the range of 7-40 mm/s² , 25-150 mm/s² ,50-325 mm/s² and 196 to 1093 mm/s² respectively for the said range of defect size.

Hence it is concluded that, the outer ring defected bearings has higher amplitudes of vibration in comparison with inner ring defected bearings for the same speed.

2. 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.

3. It is noticed that at constant defect size and constant speed with different radial load, the amplitudes of vibration has negligible variation with increase in load.

Hence it is concluded that the defect on outer ring is more serious than the defect on inner ring. The predicted Model successfully gives the results based on the characteristics defect frequencies. Maximum values of amplitudes are observed at corresponding characteristics defect frequencies (fo in case of outer ring defect and fi in case of inner ring defect). Theoretically obtained defect frequencies showing significant effect in the experimental spectra with minor error of 1Hz to 2Hz. Whereas the error of 3Hz to 4Hz is observed in experimental and numerical defect frequencies. This is due to complexity of meshing for bearing elements. For the same defect size, the amplitudes of vibration also are large for outer ring defected bearings than the inner ring defected bearings.

Experimental and numerical results give a good agreement for the defected outer and inner ring bearings. It is also concluded that, for the outer ring the position of defect has significant effect on the peaks. During the experimentation, some non zero amplitudes even outside the load zone are observed which may be due to influence of other source of vibrations in the set up at higher speed. It is also observed that when both the healthy support bearings are replaced with defected bearings, the misalignment in the spectrum is noted. Further it is concluded that the theoretical, numerical and experimental results for 1mm defect size at any speed and load are very close.

The scope of this work is limited to measurement of amplitudes of vibration at the outer and inner ring defects. Focus is not on the defects on the balls in the bearing as vibrations of balls are controlled by misalignment, lubrication and ball slip. However, by controlling these parameters, a model can be developed for defected ball to which the excitation forces may be produced by impact of balls both on inner ring and outer ring.

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