BEARING CONDITION MONITORING AND DIAGNOSIS THROUGH VIBRATION AND ARTIFICIAL INTELLIGENCE TECHNIQUES

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ABSTRACT

The important sources of vibration generation considered here are as the non-linear stiffness/damping and localized defects of the contacting surfaces of bearing elements. It also covers non-linear dynamic mathematical modeling of cylindrical roller bearing for localized defects. The numerical integration technique Newark-has been used for solution of the system equations. The parameter of study for cylindrical roller bearings is rotor speed. Results have been analyzed by Fast Fourier Transform (FFT), phase plot and Poincare maps. focus on use of multi-parameters, synthesis of fault features and optimization of feature sets, in order to improve fault diagnostics accuracy, which would thus enhance machinery reliability, availability, safety, and reduce maintenance costs. The complex and non-stationary vibration signals with a large amount of noise make fault detection of rolling element bearings very challenging, especially at the early stage. Daubechies wavelet is popular for smoothing of signals so, it is chosen for reducing the background noise from raw vibration signal. This research investigates the possibilities of improving machinery diagnostics accuracy based on time domain features and various intelligence techniques like artificial neural network, support vector machine.

Keywords: *Cylindrical Roller Bearing, Artificial Intelligence, Condition Monitoring .*

INTRODUCTION

To understanding the behaviour of bearing during operation it is very important to develop mathematical model for real application. One such application of rolling element bearing is the high speed rotary gas compressor, which operates at very high speeds resulting in considerable increase in stress levels in bearings. The dynamics of bearing for such an application becomes difficult because of centrifugal forces acting on the rolling elements, material properties and the slipping of the rolling elements as they roll on the race. Despite such difficulty, it is very important to model the dynamic behaviour of the bearings for high performance applications. The vibration during operation. Even new and geometrically perfect bearing is also generating the vibration. This is due to the finite number of the bearing element and changing of the load zone. The other possible sources of the rolling element bearing vibrations are the defects of the rolling bearings, unbalance load and misalignments. Therefore it is very important to clear understanding of vibration signals associated with rolling element bearings. Vibration Analysis has proven to be a powerful tool in assessing the health of a machine.

With increasing use of soft technology, mechanization and automation in industries, there has been a steep rise in competition. Industries cannot afford longer break down time for the any machines.

Common Causes of Bearing Failures

There are many causes for developing defects in different parts of bearing. It is difficult to identify exact cause of bearing defects. Moreover one or more following reasons are responsible to damage the bearings. If it is taken care about all the causes it is improve the operation of bearing, reduce catastrophic failure and increases the life of bearing.

A. Foreign Matter

Foreign matter is one of most common cause to create the trouble in bearing is wear and pitting. This could be in the form of abrasive matter, dirt, dust, steel chips, etc. This type of defect may be identified by intermittent noise from the bearing during operation.

B. Bearing Fatigue

When bearing is rotated the rolling component rolls and changes its position in different loading zone. Thus the component is under the repetitive compression and tensile stress. This action is eventually converting in removal of metals from its running surfaces of the component. This will create the local defect in bearing known as Spalling or flaking. The damaged bearing would produce excessive vibration.

C. Brinelling

Permanent deformation caused by sudden impact load during operating condition or heavy loading during rotation of bearing. In this kind for deformation would be displaced or upset between the contact surfaces of bearings. Fretting corrosion of raceway surface is known as false brinelling. It is complex phenomena of mechanical and chemical action. In presence of oxygen small impact motion or vibration caused false brinelling.

D. Corrosion

Corrosion in bearing is caused by the chemical attack on the bearing metal by presence water and acids in working atmosphere. Red and brown strip on surfaces are sign of corrosion. In extreme condition of corrosion in bearing early fatigue would be developed. Pitting on surfaces is result of corrosion. It produces uneven and noisy signals.

E. High Temperature

Temperature of bearing is increases due to excessive heat generation or poor heat removal (lubrication) form the bearing. Surface cracks or rings generated in direction perpendicular to direction of motion in both the contact elements. High temperature is reduces the hardness of component of bearing leads the early failure of bearing.

F. Improper Installation

Bearing inner race and shaft is assembled in press or interference fit. Improper installation of bearing in housing and on the shaft may cause axial or radial preloading. Loose fits, tight fits, misalignment, housing shape, incorrect bearing selection and applying blows during installation are the common improper installation. Misalignment causes axial force on the bearing and generates excessive heat during operation.

G. Improper lubrication

Lack of lubrication or wrong selection of lubrication results the overheating. Inadequate lubrication between contact surfaces causes high stress zone and may get welded together. It creates deep scratches and welding rings like high temperature defect.

H. Plastic deformation

Plastic deformation occurs between contacting surfaces when excessive stationary load or impact load is applied during rest condition. This is localised defect produced indentation on raceway. This deformation would produce excessive vibration.

DYNAMICS OF BEARING

Dynamics of the bearing consists of the study of the forces acting on the different elements of the bearing during operation. The dynamic modelling of the bearing for high performance applications is complex and non-linear in nature. This non-linearity is due to the material nonlinearity, geometrical non-linearity and kinematic non-linearity. Non-linear modelling of bearing is to be more realistic under stringent conditions. During the last forty years, researchers have documented unpredictable and irregular dynamic behaviour of rigid rotor supported on rolling element bearings under external excitation. When the system is linear the output is periodic and predictable too. When the system is non-linear the output is unpredictable, it may be periodic or a periodic like sub-harmonic or even chaotic. The possible output of linear and non-linear system

OBJECTIVE OF THE STUDY

The objective of this study is establishing good condition monitoring system for cylindrical roller bearings. Many researchers used Combination of signal processing technologies and AI techniques and found good results in monitoring of bearing condition.

This work attempts to analyse the nonlinear vibration responses of a rigid horizontal rotor supported on rolling element bearings and to develop a fault diagnosis system for rotor bearing system. Finite number of rolling elements rotating with different velocities with respect to the inner race, generate a time varying stiffness component.

These finite rolling contacts between the rolling elements and the guiding races introduce a high degree of non-linearity in the bearings, which results in a nonlinear dynamic behaviour of the system.

A little work is reported in the literature on the effect of non-linear stiffness on the dynamics of behaviour of high speed cylindrical roller bearings, which is analyse in present investigation. The non-linear dynamic behaviour of a healthy, faulty and combined localized fault of cylindrical rolling element bearings is investigated.

The effects of localised and combine localized defects on the dynamics of bearings are also analysed in the present work. The inner race and outer race with localized defects are modelled with nonlinear stiffness having two degree of freedom model. Furthermore, the effect of the defective bearings on the vibration signature is also investigated by numerically and validated those vibration signatures by experimental results.

Experiments are conducted for faults diagnosis of roller bearings using various machine learning techniques and response surface method. Response surface method (RSM)

LITERATURE REVIEW

[1]. Then Gupta et al. (1977) have derived differential equation of motion for an angular thrust loaded ball bearing about its initial conditions and ball mass center[2].Furthermore, Sunnersjo (1978) was the one who has reported theoretical and experimental work on non-linear model of rolling bearings supporting a horizontally balance rotor with a constant vertical radial load. The non-linearity introduced was due to Hartzian contact stress, radial internal clearance and parametric effect owing to varying compliance[3]. To counter this effect, Rahenjatet al. (1979) have investigated the axial profile and effect of roller misalignment for taper roller bearing. Also represented numerical method for the pressure distribution for taper roller[4]. Later, Mayer et al. (1980) have considered distributed defect in rolling bearing systemand shows time-variation contact forces which exist between the rolling element and race ways of the bearing[5].

White et al. (1984) have modeled equation of motion for rolling element bearing with considering two degree of freedom and subjected to external excitations. In which outer race and inner race considered as rigidly mounted on shaft and housing. Bearing system has been modeled as spring, mass and damping system[6]. Rahenjat et al. (1988) have made model for rigid shaft and radial deep groove ball bearings. The model includes the consequences of unbalance and waviness on the bearing surface. Frequency spectra and phase plane plots had been pretended, which helps in understanding the non-linear dynamics of the system[7]. Researchers have contributed gradually increasing nonlinearity in model and degree of freedom of whole bearing systems. In these extension, the support ball bearings used in precision spindle of machine have modeled by Matsubara et al. (1988) with considering bearing as a piecewise linear springs and External concentrated load were provided on the both end of the spindle baering[8]. Aini et al. (1990) have developed simulation model for precision machine tool spindle, supported by a pair of lubricated angular contact ball bearings under moment loading at bearing support. They prepared five degree of freedom model and determined the system exciting frequencies and compared them with those obtained from the same model under dry contact conditions. It was found that the oil film contributed little to the overall frequency response[9,10].

Tiwari and Vyas (1995) have estimated the non-linear stiffness parameter of rolling element bearings in a rotor bearing system. The theoretical results wereequated with experimental findings[11]. Tiwari and Vyas (1997) have further extended their work on a balance rotor to estimate bearing parameters in a non-liner rotor bearing system undergoing small residual imbalance forces along with random forces[12]. Akturk et al. (1997) have investigated the effect of preload on bearing and number of ball on the shaft vibration. Amplitude of vibrations was considerably reduced if bearing is preloaded and numbers of balls are correctly selected it means both parameters greatly affected dynamic behavior of bearings[13].

Another class of bearing defects may be categorized as point or local defects. This includes cracks, pits and spalls in the running surfaces, as well as practical contamination of the bearing lubricant. These kinds of defects manifest themselves in the bearing vibration signal as vibratory transients, which result from discontinuities in the contact forces during rotating of bearing.

NEWMARK-β METHOD

Several techniques are available for solving nonlinear equations iteratively. For finding out transient response through nonlinear differential equation of motion, Newmark-β method is used iteratively at every time increment is obtained. The Newmark –β method based on acceleration varies linearly between two instants of time. The velocity and displacement are calculated from acceleration based on two parameter α and β. The

SPIRAL BEVEL GEAR

A spiral bevel gear is a bevel gear with helical teeth. The main application of this is in a vehicle differential, where the direction of drive from the drive shaft must be turned 90 degrees to drive the wheels. The helical design produces less vibration and noise than conventional straight-cut or spur-cut gear with straight teeth. A spiral bevel gear set should always be replaced in pairs i.e. both the left hand and right hand gears should be replaced together since the gears are manufactured and lapped in pairs.

Bevel gears classification

Bevel gears are classified in different types according to geometry:

- **Straight bevel gears** have conical pitch surface and teeth are straight and tapering towards apex.
- **Spiral bevel gears** have curved teeth at an angle allowing tooth contact to be gradual and smooth.
- **Zerol bevel gears** are very similar to a bevel gear only exception is the teeth are curved: the ends of each tooth are coplanar with the axis, but the middle of each tooth is swept circumferentially around the gear. Zerol bevel gears can be thought of as spiral bevel gears, which also have curved teeth, but with a spiral angle of zero, so the ends of the teeth align with the axis.
- **Hypoid bevel gears** are similar to spiral bevel but the pitch surfaces are hyperbolic and not conical. Pinion can be offset above, or below, the gear centre, thus allowing larger pinion diameter, and longer life and smoother mesh, with additional ratios e.g., 6:1, 8:1, 10:1. In a limiting case of making the "bevel" surface parallel with the axis of rotation; this configuration resembles a worm drive. Hypoid gears are widely used in automobile rear axles.

Spiral bevel gears are used to transmit power between shafts that are typically at a 90-degree orientation to each other. The teeth on spiral bevel gears are curved and have one concave and one convex side. They also have a spiral angle. The spiral angle of a spiral bevel gear is defined as the angle between the tooth trace and an element of the pitch cone, similar to the helix angle found in helical gear teeth. In general, the spiral angle of a spiral bevel gear is defined as the mean spiral angle.

Because spiral bevel gears do not have the offset, they have less sliding between the teeth and are

more efficient than spiral and produce less heat during operation. Also, one of the main advantages of spiral bevel gears is the relatively large amount of tooth surface that is in mesh during their rotation. For this reason, spiral bevel gears are an ideal option for high speed, high torque applications.

The American Gear Manufacturing Association (AGMA) has developed standards for the design, analysis, and manufacture of bevel gears.

The driving and driven gears are the most important components of the Gear box of any automotive. Modelling allows the design engineer to let the characteristic parameters of a product drive the design of that product. During the gear design, the main parameters that would describe the designed gear such as module, pressure angle, root radius, tooth thickness and number of teeth could be used as the parameters to define the gear.

LITERATURE REVIEW

Benedict and Kelley [1] performed experiments with cylindrical rollers to investigate the gear tooth friction. They presented their results as an empirical formula to predict the instantaneous friction coefficient. However, a very limited range of variables within which the experiments were run around the validity of this equation.

Diab et. al. [2] derived a semi-empirical traction formula based on experiments on a disk test rig at low rotational speeds. Xu et al. used an EHL model along with a multiple regression analysis to obtain a new friction coefficient formula which they used in predicting mechanical efficiency of parallel axis gear pairs and used mechanical efficiency model together with a gear design optimization model to show that measures to maximize the mechanical gear efficiency often impacts the other noise and durability.The final design must be a compromise that delivers reasonable efficiency levels with reasonably low vibration excitations and contact and bending stresses.

Boness [3] performed experiments on a disc and a gear operating partially submerged in lubricant to measure drag torque and estimate churning losses. Based on these experiments, he obtained empirical relations for churning losses within the ranges of the experiments and also conducted experiments on individual and meshed spur gears to measure the churning losses. Their results were used to show certain discrepancies formulae.

Changenet and Velex [4] also predicted churning losses in a single and a pair of gears. Their study was based on results from a dimensional analysis andcompared well with the experiments they conducted for validation and proposed a physics-based model oto predict spin losses of a spur gear pair including drag and pocketing loss components. They investigated the impact of static oil level, speed, module and face width on the load independent losses.

Dawson [5] performed experiments on large spur and helical gears to measure the windage losses and quantify the effects of speed, gear size and geometry as well as the shape ofthe enclosure. Research on power losses of cross-axis gears goes all the way back to Buckingham [6-7] who proposed an approximation of hypoid gear efficiency by assuming that a conjugate action between the gear teeth that was taken to equivalent to that of spiral bevel gears and the sliding action of the pitch surfaces is equivalent to that of worm gears. He then approximated the power loss of a hypoid gear as the sum of power losses of a spiral bevel and a worm gear.

Handschuh and Kicher [8] developed a method to analyze the thermal behavior of spiral bevel gears. They assumed an elliptical Hertzian contact and used a simplified expression for friction coefficient as a function of slide-to-roll ratio and rolling velocity. Then they employed a finite element model to determine the heat generated as a result of the relative sliding of the tooth surfaces. In terms of hypoid gear power losses, Simon [9] used an EHL lubrication formulation along with a hypoid gear a finite-element load distribution model to predict mechanical power losses.

Jia et al [10] used a multilevel-mutligrid technique to solve the implement the same EHL equations with accelerated convergence. Taking these preliminary studies on cross-axis gears one step further, Xu and Kahraman developed a model to predict load dependent power losses in hypoid gear pairs.

They utilized a finite element based hypoid analysis software package [11] for contact analysis and a deterministic EHL model proposed by Cioc et al. [20] to predict the friction distributions. They performed experiments and validated the proposed EHL based model. Later, Kolivand et al [12] extended this study by utilizing the contact model developed by Kolivand and Kahraman, in the process removing the dependency of the load distribution computation on a FE package. Investigating the components of the rear axle including cross axis gearing, experimental studies were performed measuring overall axle power losses.

Johnson et al. [13] conducted experiments to investigate the effects on windage power loss of a single spiral bevel gear and showed that optimally placed shrouds could reduce windage as much as 70%. Then, Johnson et al. [14] extended this study to a shrouded spiral bevel gear pair and presented their results concluding that gear windage becomes a significant contributor to spin losses at high speeds.

Gabiccini et al. [15] presented an automatic procedure to optimize the loaded tooth contact pattern of face-milled hypoid gears with misalignments varying within prescribed ranges. Through the formulation of an appropriate nonlinear optimization problem,

Artoni et al. [16] proposed a novel methodology to systematically define optimal ease-off topography to simultaneously minimize the loaded transmission error and tooth contact pressures, while concurrently confining the loaded contact patterns within a prescribed allowable region on the tooth surface to avoid any edge- or corner-contact conditions.

Artoni et al. [17] presented a novel methodology to restore the designed functional properties of hypoid gear sets whose teeth deviate from their theoretical models due to inevitable imperfections in the machining process.

Ozel et al. [18]. For the spiral bevel gear machining by using CNC milling machine, people don't need to invest huge money on specific bevel gear machine since the general CNC milling machine are used, but the accuracy is difficult to achieve, and takes longer time compared to the specified cradle machining method.

Suh et al. [19] proposed a virtual gear model. In this model, the sampling points were measured by using CMM, and fitted by NURBS surface. Compare the virtual gear model to the theoretic model, the geometric error was evaluated. Weimin developed accurate way to measuring the spiral gear tooth by optimize the measuring process parameters.

EXPERIMENTAL SETUP

For experimentation work, the experimental test rig has been developed in vibration laboratory at UVPCE-GNU as shown in Fig. 1. Roller NJ 305 bearing have been taken for the experimental work, which is same as taken in the theoretical simulation. The horizontal shaft of weight 1.5 kg is used and rig is connected to a data acquisition system through proper instrumentation as shown

Data Acquisition System

Two channel vibration analyzer VIBEX-II (Pruftechniqe) used for signal acquisition. VIBEX consists of omnitrend software, which is designed for quick data acquisition, review and storage. A detail of VIBEX-II is given in Appendix C. Data acquisition hardware consists of a remote optical sensor with a visible red LED light source is used to measure rotor speed and two Piezoelectric accelerometers are used for picking up the vibration signals from various stations on the rig. For getting optimum signals, accelerometers are fitted in vertical and horizontal direction on the bearing housing.

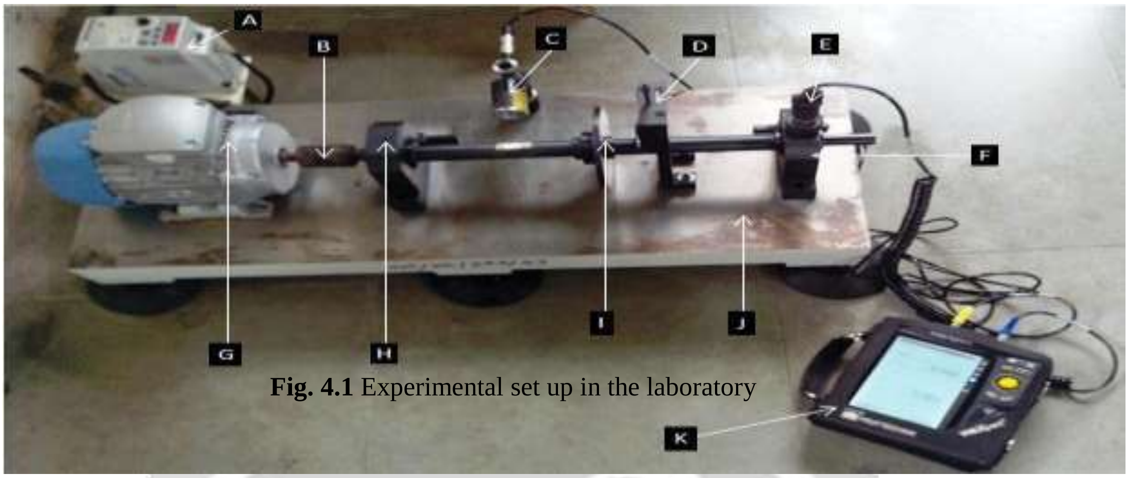


Fig. 4.1 Experimental set up in the laboratory

(A)Variable Speed Control, (B) Flexible Coupling, (C) Laser Beam Prob., (D) Load Adjustment Screw, (E) Accelerometer, (F) Tested Bearing with Housing, (G) Motor, (H) Healthy Bearing with Housing, (I) Load Disc, (J) Base with Anti vibration Pad, (K) Vibration Analyzer

Experimental Procedure

The research on bearing faults has been carried out, considering single factor affecting of bearing faults such as fault in either of inner race, outer race or ball etc. Data generated using traditional method of research, using single factor effect is valuable, but fails to indicate the effects of interactions between various test parameters.

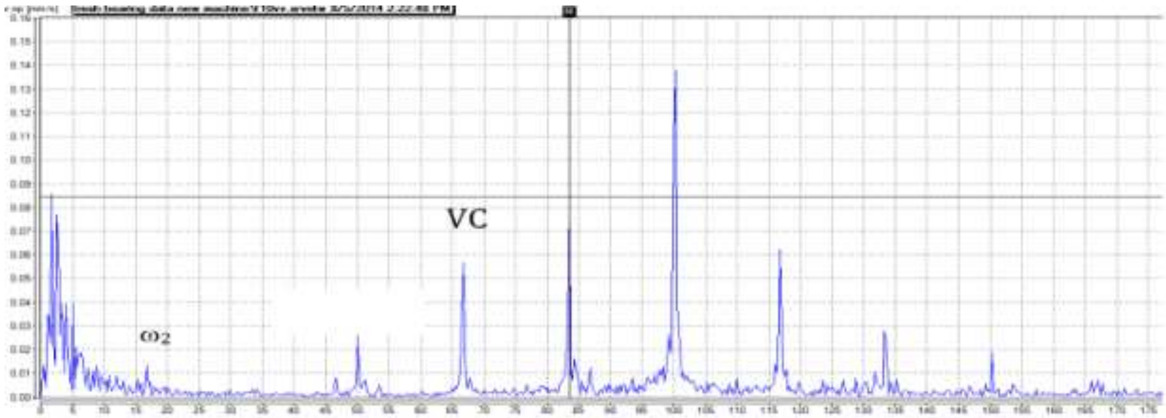
Therefore, for an efficient experimentation, a systematic scientific approach is necessary to design and carry out the experimentation properly. A properly planned experimentation is of utmost importance for deriving clear and accurate information from the experimental observations. The present work aims to identify effect of various faulty bearing components on the vibration response of a rotor bearing system. Faults on bearing components are considered as defects on outer race, inner race and ball. Different experiments are being carried out for healthy bearings and faulty bearings.

Results and Discussion

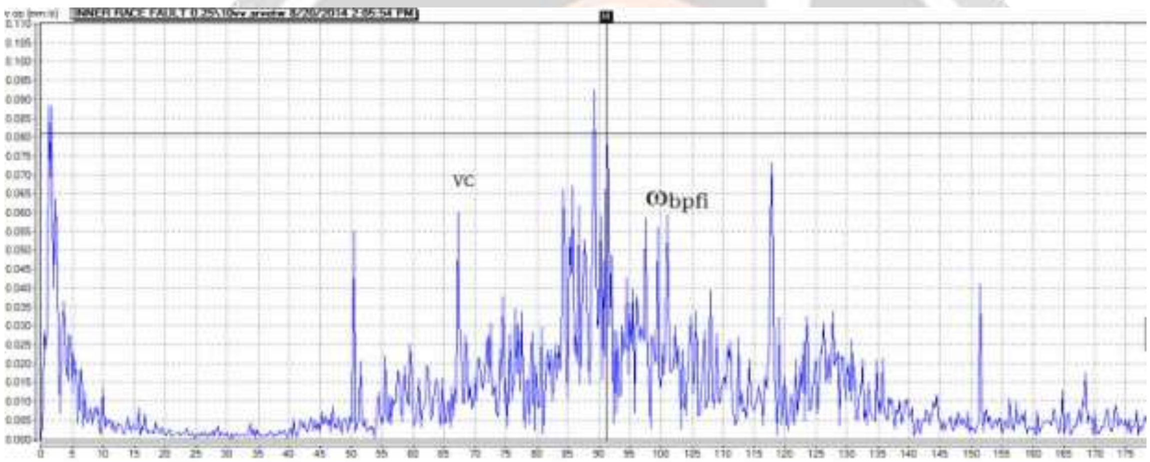
The vibration signal samples are collected as 32768 in 2000 ms for displacement, 2560 in 5000 ms for velocity and 4063 in 62 ms for acceleration.

	Operating speed of roller bearing in rpm						
Defect in Bering	**600**	**1000**	**1400**	**1800**	**2200**	**2600**	**3000**
Rotating Frequency (ω2) Hz	10.0	16.7	23.3	30.0	36.7	43.3	50.0
Inner race defect Freq. (ωbpfi) Hz	60.3	100.5	140.6	180.8	221.0	261.2	301.4
Roller Defect Freq. (ωbfs) Hz	46.6	77.7	108.8	139.8	170.9	202.0	233.1
Case Freq. (ωc) Hz	4.0	6.6	9.3	11.9	14.6	17.2	19.9
Outer race defect Freq. (ωbpfo)Hz	39.7	66.2	92.7	119.2	145.7	172.1	198.6

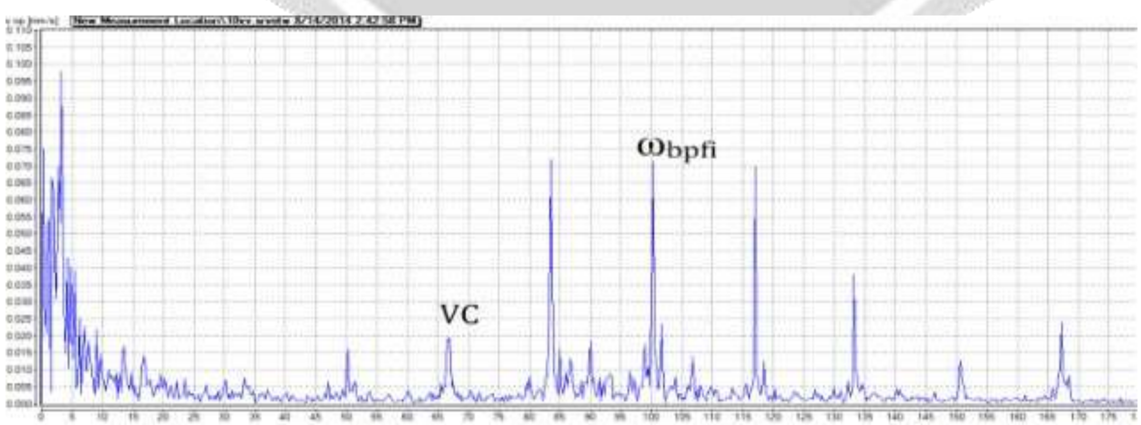
VC Freq.	Hz	39.7	66.2	92.7	119.2	145.7	172.1	198.6



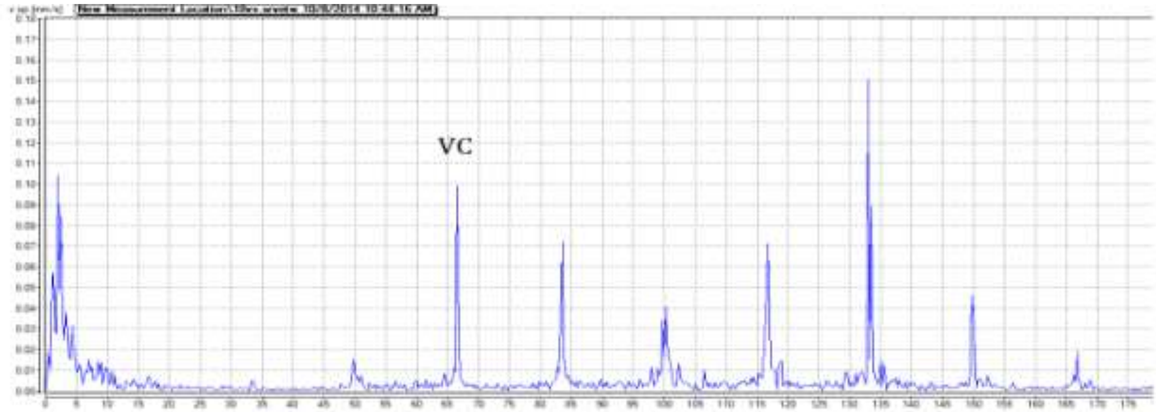
FFT plot of healthy bearing



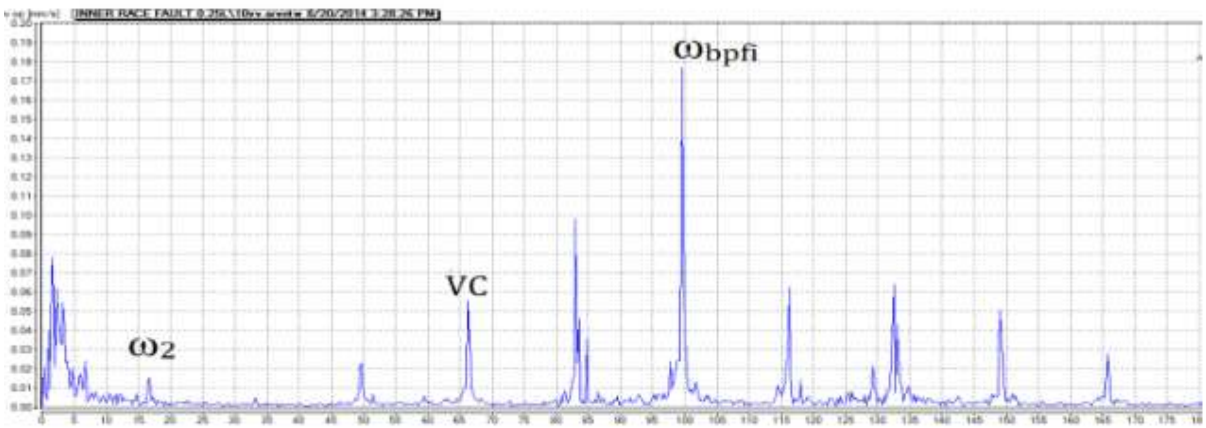
6 FFT plot of Bearing due to Inner race defect size 0.25 mm



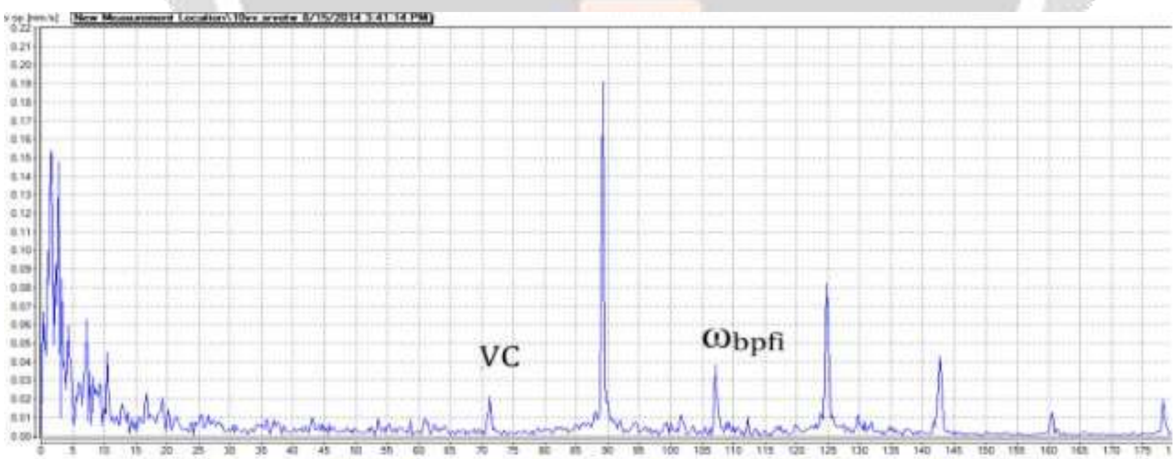
FFT plot of Bearing due to Inner race defect size 0.5 mm



FFT plot of healthy bearing with radial load W=25N



FFT plot of Bearing due to Inner race defect size 0.25 mm with radial load W=25N



FFT plot of Bearing due to Inner race defect size 0.5 mm with radial load W=25N

CONCLUSION

From the experimental work and mathematical model, investigation of a roller, due to localized defect and combined defects the succeeding findings may be predicted. With increment in defect size the amplitude of vibrations is increased and then decreases due to “self-Peening’ effect. And, with increment of rotational speed with localized defect the characteristics defect frequency also shifted on FFT diagram and the behavior of operation is change from periodic to chaotic. For individual localized defects on inner race, roller and outer race, the peaks appear at characteristics fault frequencies. The characteristic fault frequencies appear with interaction with case frequency and ball passage frequencies in combined localized faults.

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