MODAL ANALYSIS OF SELF-ALIGNING BALL BEARING USING FEA – A REVIEW

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Abstract-This paper presents state of art of review on self-aligning ball bearing using FEA. The moving bearing, with external ring fixed, is a multi-body mechanical framework with moving components that transmit movement and burden from the inward raceway to the external raceway. Present day pattern of Dynamic examination is helpful in the early forecast; reproduction of rotor bearing framework as the assembling of the model is tedious, expensive, and required further investigation for weariness disappointment. The dynamic examination has turned into an exceptionally useful asset for the improvement of the real execution of the framework. The approach for expectation and approval of dynamic attributes of bearing rotor framework vibration is contemplated. Solid works and ANSYS programming are the promising devices for the displaying and modular investigation of the bearing rotor framework.

## I. INTRODUCTION

ROLLER BEARINGS

The roller bearings are generally used for higher load rating than the other ball bearing assemblies. They have higher stiffness and endurance limit than the other ball bearings of comparable size [1].

Cylindrical Roller Bearings

The radial cylindrical roller bearings as in figure 1(a) are used for very high-speed operations due to their low-friction torque characteristics. The rolling elements for the radial cylindrical roller bearing are generally crowned from the edge to avoid edge stresses and crowning also supports self-aligning characteristics of the bearing. The cylindrical roller bearing supports two types of loads Radial and Thrust load as shown in figure 1(b). The thrust cylindrical roller bearings support low loads and speed ratings as compared to the radial roller bearings.

Needle Roller Bearings

The radial needle roller bearings and thrust needle roller bearings illustrated in figure 2(a) and figure 2(b) are the cylindrical roller bearings having a greater length to the diameter ratio. They are manufactured for the applications where the space availability is limited. They have a higher frictional coefficient than the cylindrical roller bearings. The load carrying capacity of the needle roller bearings is less when compared to the cylindrical roller bearings. They are used in the applications where the continuous rotation occurs or the oscillatory motion occurs but the loading is low or intermediate.
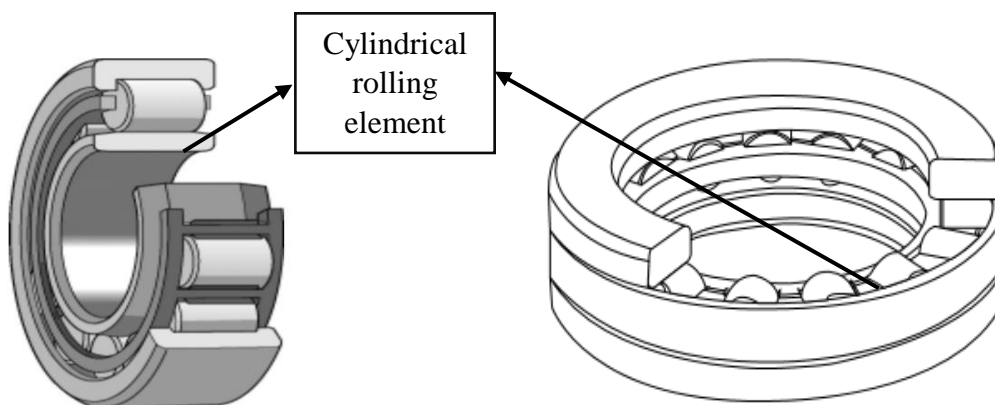
Figure 1 (a). Radial Cylindrical Roller Bearing   Figure 1 (b). Thrust Cylindrical Roller Bearing
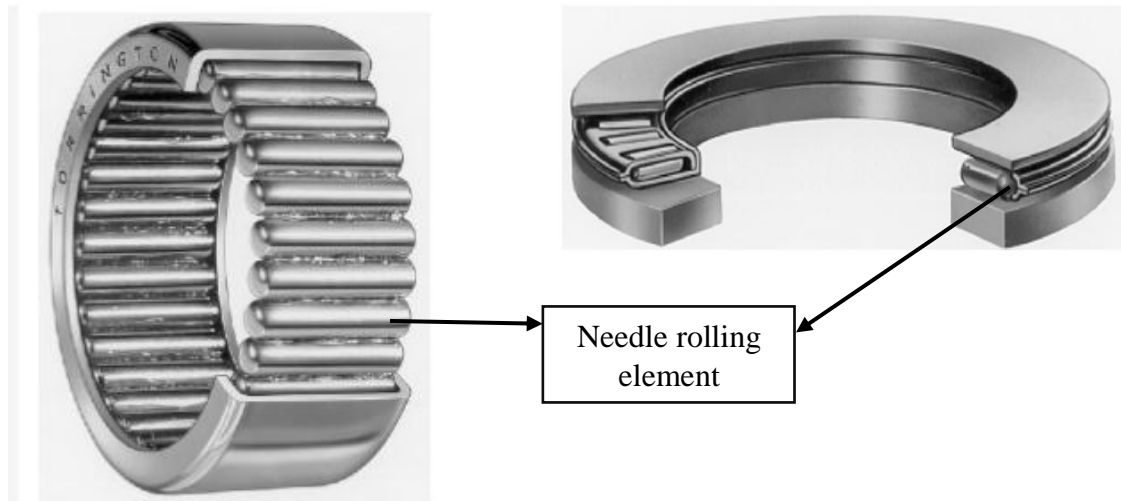
Needle rolling element

Figure 2 (a). Radial Needle Roller Bearing   **Figure 2 (b). Thrust Needle Roller Bearing**

Tapered Roller Bearings

The tapered roller bearings as shown in figure 3(a) and figure 3(b) have the ability to carry the combination of both radial and thrust loads or to carry thrust loads only. The terminology for the tapered roller bearing is quite different from the other roller bearings, with the inner ring called cone and outer ring called a cup. The tapered roller and the raceways are crowned just like a cylindrical roller bearing to reduce the heavy stresses on the axial boundaries of the rolling contact members.
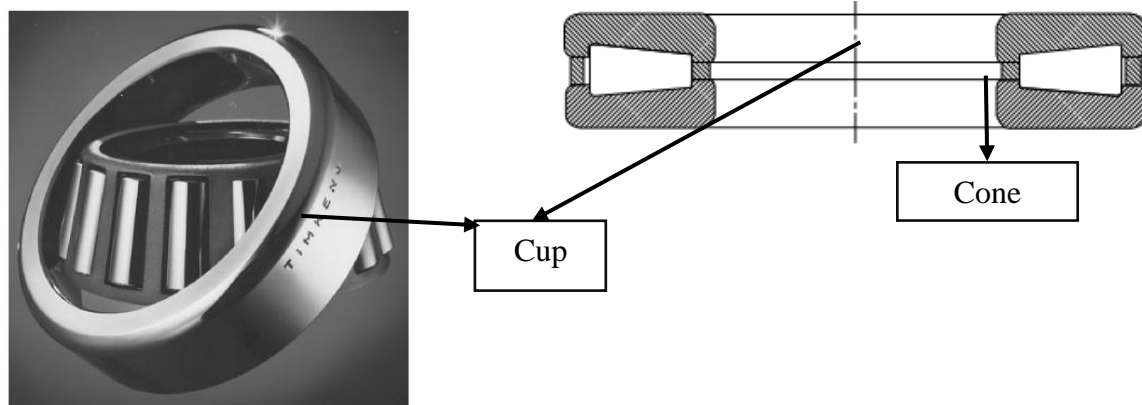

Cup

Cone

Figure 3 (a). Radial Tapered Roller Bearings   **Figure 3 (b). Thrust Tapered Roller Bearings**

BEARING SELECTION AND DESIGN

Any machine component design was based on some technical principles and an imaginative approach to performing a specific task with maximum efficiency and economy. The safety, producibility, reliability, cost-effectiveness, etc. are the desirable properties of the effective design. But there always some compromise between the required design and effective design of any component. The task of bearing selection for the specific task is very tedious for a bearing designer. Due to the good efficiency to support both axial and radial loads along with competent rigidity generally rolling element bearings are preferred over hydrodynamic and hydrostatic bearings.

The requirement of big space, less fatigue life for the variable load conditions, low damping capacity with high noise levels are some drawbacks of rolling elements over the hydrodynamic bearings. Vibrations and noises are used as indicators in the machines to identify the problem with components and to judge the need for repair and replacement.

3.1 Self-Aligning Double-Row Ball Bearing

In this study, the SKF 1205 EKTN9 shown in figure 4 bearing has been used to study and simulate the vibration characteristics. The material properties used for the simulation are standard steel for the bearing rings, 100CR6 contains

approximate 1% carbon and 1.5% chromium and silicon nitride (Si_3N_4) for the rolling elements. The basic geometry and design for the bearing explained are follows [2].
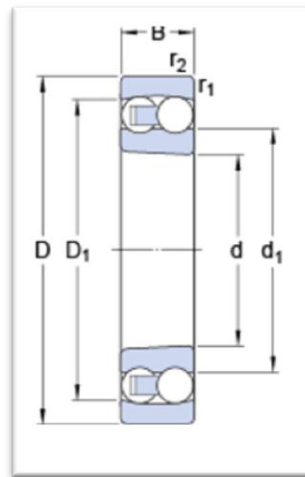
Figure 4 Dimensions for the SKF 1205 EKTN9 Bearing

The dimensions for the geometric model has given below in table 1.

Table 1 Principal Dimensions for the SKF 1205 EKTN9 Bearing

S.NO.	Dimensions	Value(mm)
1	d	25
2	D	52
3	B	15
4	d_1	33.3
5	D_1	44.6

FAILURES OF BALL BEARINGS

A bearing can fail by one or more of following ways: -

i) **Excessive loads:** A bearing can fail due to the excessive load condition**.** It can be caused by one or more reasons viz. tight fits, brinelling, and improper preloading. It usually cause premature fatigue as shown in figure 5.
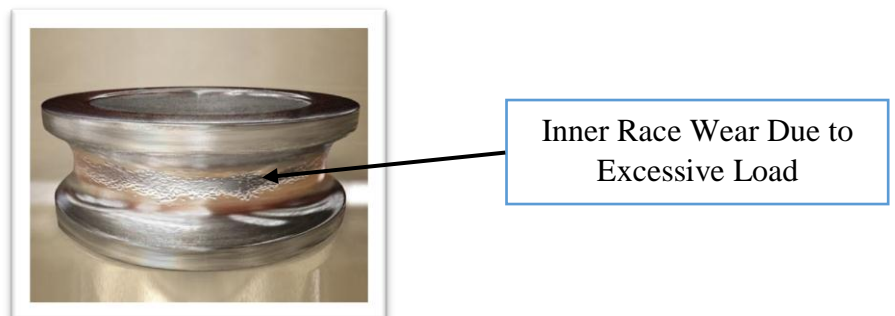

Inner Race Wear Due to Excessive Load

Figure 5. Defect Due to Excessive Load.

ii) **Overheating:** Temperatures more than 400°F can anneal the ring and ball materials. The resulting loss in hardness reduces the bearing capacity causing early failure. e. In extreme cases, balls and rings will deform. The temperature rise can also degrade or destroy lubricant as shown in figure 6.
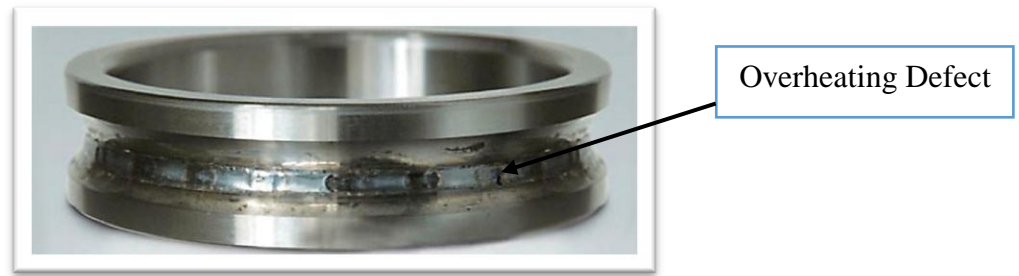
Figure 6. Wear of Inner Race Due to Over Heating.

iii) **False brinelling:** False brinelling-elliptical wear marks in an axial direction at each ball position with a bright finish and sharp demarcation, often surrounded by a ring of brown debris-indicates excessive external vibration. A small relative motion between balls and raceway occurs in non-rotating ball bearings that are subject to external vibration. When the bearing isn't turning, an oil film cannot be formed to prevent raceway wear. Wear debris oxidizes and accelerates the wear process as shown in figure 7.
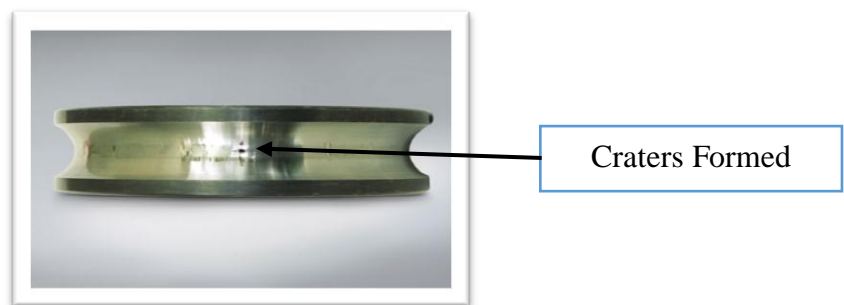

Figure 7. Shows False Brinelling.

iv) **True brinelling**: Brinelling occurs when loads exceed the elastic limit of the ring material. Brinell marks show as indentations in the raceways which increase bearing vibration (noise). Severe Brinell marks can cause premature fatigue failure. Any static overload or severe impact can cause brinelling. Examples include: using hammers to remove or install bearings, dropping or striking assembled equipment, and a bearing onto a shaft by applying force to the outer ring. Install bearings by applying force only to the ring being press fitted, i.e., do not push the outer ring to force the inner ring onto a shaft as shown in figure 8.

Figure 8. Show True Brinelling.
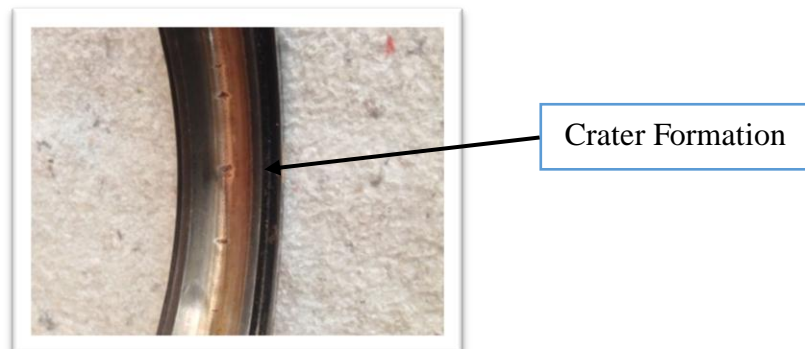
v) **Contamination:** Contamination symptoms are denting of the bearing raceways and balls resulting in high vibration and wear. Contaminants include airborne dust, dirt or any abrasive substance that finds its way into the bearing. Principal sources are dirty tools, contaminated work areas, dirty hands and foreign matter in lubricants or cleaning solutions as shown in figure 9.
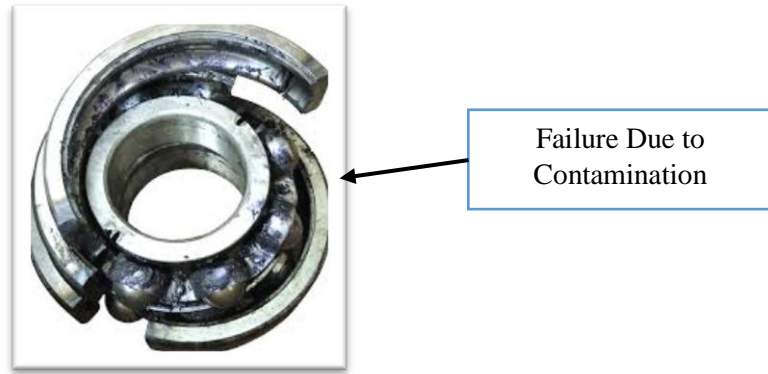
Failure Due to Contamination

Figure 9. Contamination Failure.

ROLE OF VIBRATION

Vibration can cause catastrophic failures and can reduce machine efficiency and utilization. The running and repairing cost substantially increases due to the impacts caused by vibrations. As for the pumps, compressors, and turbines, etc. mostly run above their resonant frequency range the study of their dynamic characteristics becomes important. The classical vibration and various rotor dynamic techniques were used for the analysis and the new emerging methods like simulations, F.E.A is also used to study the vibration characteristics of mechanical systems. More complicated applications arise day after day that requires more reliability, higher speed limits, and greater accuracy. Therefore, the bearing manufacturers improve the bearing designs and their qualities with respect to the vibrations and noise **[3]**.

Factors influencing the bearing vibration generation are enlisted below:-

- **Variable compliance: -** Due to the misaligning and radial loads, the line of action of the load's changes with respect to the time and the position of the rolling elements which were supporting the load. This leads to the generation of bearing vibration in the system. Where lesser the number of rolling elements supporting the load greater will be the vibration.
- **Waviness:** - The vibration levels in the bearing generally depends on the bearing and its contact geometry. The speed and loads equally share the reason.
- **Surface Roughness:** - Due to the metal-to-metal contact the small impulses of the random vibration excites all natural modes of the bearing the system.
- **Distributed defects:** - The manufacturing process used to make bearings produce geometric imperfections like surface roughness, waviness, and misaligning races, etc. These imperfections lead to defects and then defects lead to the vibrations.
- **Localized or discrete defects:** - The defects like indentations, scratches over the rolling surface, pits, debris, etc. these are created by the operation, assembly, poor maintenance, etc. are called localized defects on the rolling surface.
- **Raceway defects, rolling element and cage defects:** - The raceway defects like pits and indentations, etc. makes a high energy pulse at the same rate of ball pass frequency relative to the inner race and vice-versa for the outer race. This higher energy pulse can be related to the system vibrations.

    As for the rolling element defects the vibrating frequency could be as high as two times of the rolling element spin frequency. The cage defects create random bursts of energy when ball rolls and the cage starts to wear or deform.

The physical imperfections can happen on bearing components, for example, internal race, external race, ball and pen. The deformities will cause a high sufficiency of vibration. Every component has variation trademark frequencies such as basic recurrence, ball turn external recurrence, (BPOF) and ball turn internal recurrence, (BPFI) can be determined by utilizing the recipe underneath:

$$BPFI = \frac{N}{2} \times F \times (1 + \frac{B}{d_m} COS\theta) \qquad \ldots\ldots 1$$

$$BPFO = \frac{N}{2} \times F \times (1 - \frac{B}{d_m} COS\theta) \qquad \ldots\ldots 2$$

$$FTF = \frac{F}{2} \times (1 - \frac{B}{d_m} COS\theta) \qquad \ldots\ldots 3$$

$$BSF = \frac{d_m}{2B} \times F \times \left(1 - \left[\frac{B}{d_m} COS\theta\right]^2\right)$$ 4

II. LITERATURE REVIEW BASED ON VIBRATION GENERATION BY THE HEALTHY AND INNER/OUTER RACE DEFECTED BALL BEARING

A. Utpat [4] have discussed the effects of defect size and of rotational speed on the vibration amplitude. The frequency response plots for defects was used in the failure analysis of the ball bearing. The model assumed to be a spring-mass system, races as masses and balls as spring and finite element analysis with a defect at the outer ring and on the inner ring was performed. The defect size ranging from 250 microns to 2000 microns. For the validations of simulation results, the experimental results were also generated.

A. Nabhan *et al.* [5] have discussed the effects of defect on the outer race of the deep groove ball bearing by experimentation and simulation (ABAQUS/CAE). The angular position for the defect on the outer race changing from 0° to 315° with 45° interval. The validation of simulation results is done by obtaining the experimental results on the experimental setup.

T. Zhaoping *et al.* [6] have discussed the contact analysis parameters such as stress, strain, penetration, sliding distance and frictional stresses between the rolling elements, an inner and outer race of the deep groove ball bearing. 3-D modal of the bearing is created in the ANSYS using APDL (ANSYS Parametric Design Language) to generate simulation results. Simulation results are consistent with the theoretical values and design of bearing can be optimized for the complicated load values.

J. Liu *et al.* [7] have discussed the effects of defect size, shaft rotational speed, the axial and radial load on the vibration spectrum of the ball bearing. They established the relation between the pulse waveform characteristics generated by defects on the bearing race. The shape and size of the localized defect induce the vibration spectrum of the impulse originated by the ball surpassing over the defect on the inner race. In the proposed work they modeled three defect types: i). Rectangular ii). Hexagonal and iii). Circular. The experimental and explicit dynamic FEA method used to explains the proposed work.

Z. Yang *et al.* [8] have discussed the faulted bearing vibration spectrum. For a different dimension of bearing the natural vibration frequency of bearing would be different. So, they took three different models one healthy bearing, and the other two with surface defects at the outer and inner races. They obtained simulation data from the ADAMS software and the vibration spectrum from the experiments. The time-frequency plot of the experimental data was obtained by using spectrum analysis method. By comparing the results of simulation and experimentation they concluded that simulation was effective for the analysis purposes.

III. LITERATURE REVIEW BASED ON MODAL ANALYSIS BY USING FINITE ELEMENT ANALYSIS

Xia *et al.* [9] have studied and concluded the influence of crowning on the dynamic characteristics of tapered roller bearings. The non-linear finite element analysis software ANSYS/LS-DYNA used to decide the dynamic performance. They concluded that the crowning value for the tapered roller affects the vibration accelerations of the tapered roller bearing.

Mishra *et al.* [10] developed the three different numerical models in the virtual environment to generate vibration signatures. i). A planar motion block model which does not consider cage and traction dynamics developed in MATLAB Simulink. ii). A model with more complexity for cage, traction, and contact dynamics was developed in SYMBOLS software using energy domain bond graph formalism. iii). Multibody physics model with complex contact and traction mechanism developed in ADAMS software. Spectra Quest Machine Fault Simulator was used for the experimentation.

P. Kadarno & Z. Taha [11] developed a finite element model simulation for the analysis of the ball bearings. A defect at the outer race is created to generate the vibration response and then compared it to the healthy bearing vibration response. ABAQUS model shows similar characteristics between the simulated data and experimental data. The study of time signal parameters such as RMS and peak-to-peak value led to draw the effects of loads and the rotational speed of the ball bearing on the vibration response.

H. Rong Xin *et al.* [12] concluded that the nonlinear contact model of the deep groove ball bearing solved with ANSYS Workbench gives possible results when compared to the real model. With the reasonable boundary conditions, stresses and deformations at the outer and inner race of the rolling bearing were calculated in the static structural module of the ANSYS workbench. Furthermore, the simulation results show consistency with theoretical values. The Hertz elastic contact theory was used to generate the theoretical values and for the basic assumptions.

U. A. Patel & S. Rajkamal [13] have discussed the prevention of system failure by continuous monitoring of vibration signal of bearings. For that, they studied the dynamic characteristics of the ball bearing. They developed the ANSYS modal for simulation results and compared it with experimental data. FFT (Fast Fourier Transformations), used to generate the experimental data. A MATLAB program was created to plot the bearing pass frequency of outer and inner race.

P. Sulka _et al._ [14] concluded that the static structural analysis in ANSYS of the ball bearing gives almost the same results when compared to the analytic results. The Hertz theory was used to obtain the analytical solutions for contact pressure, stress and for the displacement of bearing. The Lagrange formula was used to determine the friction coefficient and normal stiffness factor between the contact points of elements.

V. Purushotham _et al._ [15] proposed a method of detections of localized bearing defects based on the wavelet transform using Hidden Markov Model. They plotted the vibration spectrum for single and multiple defects on the inner race, outer race and rolling elements. The Mel-frequency analysis approach confirms a 99.9% success rate to recognize the bearing fault in their proposed method when compared to other methods like probability distribution function, Gaussian distribution, etc. The step by step decomposition of vibration signal to the frequency-time domain using discrete wavelet transform led to detect the location of the defect on the races.

S. Tyagi _et al._ [16] discussed the effectiveness of transient analysis of ball bearing for the detection of incipient stage defects. In the study, the Rexnord ER 16K bearing was used and the dynamics of the bearing element was described in basic motions: - 1) Shaft rotational frequency. 2) The fundamental cage frequency. 3) The ball pass raceway frequency. 4) The ball rotational frequency. The general radial load was taken to simulate the bearing in ANSYS of 65.5 N for 1200 rpm (20 Hz). MATLAB software was used to process the raw experimental data into the graphical time-frequency domain.

Y. Shao _et al._ [17] studied the vibration or noise created in the bearing due to the impact generated by the defects. The finite element analysis was used to extract the characteristics of the small impacts in deep groove ball bearing 6304. The explicit dynamic module from ANSYS Workbench was selected to solve the non-linear dynamic process with contact between multiple flexible bodies. The faults were generated on the inner, outer race and on the rolling element. And it was concluded that the vibration signature of the outer race defect was much greater as compare to rolling element noise.

N. Tandon and A. Choudhury [18] presented an analytical model to predict the vibration frequencies of the rolling bearing corresponds to the localized defects on the outer, inner race and rolling element under radial and axial loads. For the characteristics defect frequencies, discrete spectrum as peak and its harmonics.

P. K. Kankar _et al._ [19] studied fault diagnosis for high speed rolling element bearings with local defects in inner and outer races and rollers by using a response surface technique. This model predicts peaks in a discrete spectrum at the characteristic frequencies. The model was validated with measurement data.

IV. CONCLUSIONS

This paper studied the review on the self-aligning ball bearing analysis using finite element analysis and it was found that the bearing are not made for high speeds and heavy loads. Generally self-aligning ball bearings are made for under low load conditions where the misalignment of the shaft takes place. The further analysis can be done by simulation of ball bearing in simulation software like ansys, comsol, which gives the accurate conditions for the analysis of ball bearings under different loads or at different speeds.

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