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Performance Behaviors of Rolling Bearings and New Developments

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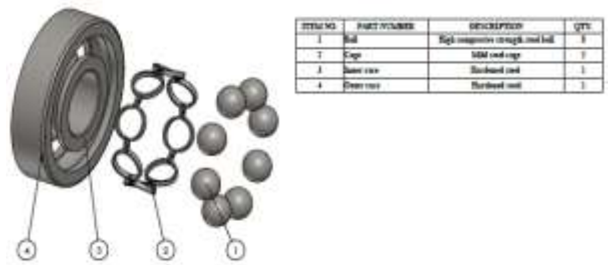
***Abstract-** Rolling bearings are broadly used in many industrial applications. The fatigue life failure of one of the elements of rolling bearing leads to loss of productivity in industries. The major causes of failure of rolling bearings are a rise in operating temperature, an increase in frictional torque, increased vibrations, centrifugal force effect in thrust bearings and lubrication starvation. All these factors lead to increased wear of balls and races. The literature survey reveals that the researchers studied the performance behaviors and characteristics of rolling elements of the bearing through analytical models and experiments. Many techniques (such as Envelop Analysis, Fourier Transform Analyzer, etc.) presented for continuous health monitoring of rolling bearings with/without modification in geometry and with the use of coated surfaces of the rolling elements. This paper presents a literature review about the analytical models and experimental methods for predicting fatigue life failure and a comprehensive study about the problems that occur and recent development in the rolling bearing.*

***Key words:** Rolling Bearing, Fatigue Life, Frictional Torque, Film Thickness, Vibrations Spectra and Characteristic Frequency.*

I. INTRODUCTION

The concepts of bearing originate from the basic process of using wooden rollers piece to carry out movement of heavy rocks with less human power. This rolling motion was developed into rolling bearing made up of metals, which consisting of rolling elements (balls or rollers) between two rings with equal spacing element (called cage or retainer) (Fig.1). In modern engineering, the advancement in material, the bearing components were designed with varying composition of materials to fulfill the end users' requirements. Generally outer- inner rings were made up with hardened steel, chromium steel, etc., and balls/ rollers manufacture to have with high compressive strength, while the retainer made up with softer material than the other rolling elements such as mild steel, bronze, aluminum, brass, polytetrafluoroethylene (Teflon or PTFE), fiberglass, and plastics.



(a)

(b)

Fig. 1: Schematic of ball bearing (a) components, and (b) assembled ball bearing.

Rolling bearing has an impact on providing support to rotating components of machinery and guiding them energy efficiently. As it experienced high friction in confirming contacts than non-confirming contacts, the rolling bearing preferred over the sliding bearing for high load and high speed applications. The rolling element bearing has non-confirming contact between the elements, so it has low friction results in less loss as compare to confirming contacts. According to modern industry requirement like in the power transmission, it is required to deliver the power from driver to driven member with high efficiency. So, rolling bearing reduces power loss to a minimum level. The friction in the rolling bearing, would greatly affect the loss of power in power transmission. And, in a heavy load application a very small mechanical power loss can be equivalent to large loss of kilowatts. Also, friction cause to heat generation in elements. As the bearing used to support the rotating members of machine, it means the bearing has to suppose to bear large load. The loading of bearing may be axially or radially depends upon the application. The load on rolling elements of bearing can produce high Hertzian pressure on the races caused to high operating temperature (range from 500˚ -700˚F). The Herts theory of surface contacts and Reynold's equation was found very useful and solved for analysis of the lubricant film thickness in spalled region.

The problem of friction in rolling elements can be resolved by the introducing a lubricating system in the bearing. By supplying the lubricants in the bearing, friction got reduced to minimum level and it reduces the heat generation. In heavy load and high speed application, friction losses become high and lubrication reached in starved lubrication condition whatever the regime hydrodynamic, elastohydrodynamic, etc. in starved condition the balls/ rollers skidding started in the races. The power transmission in the bearing got affected due to generation of heat with friction in the rolling element. So, the losses in power would greatly affected by the factors such as skidding of balls/ rollers, relative motion of ball and cage, and lubricant starvation [1-3].

The frictional heat generation in thrust loaded ball bearing has considerable influences on the operating condition and thermal contact resistance. To maintain low operating temperature, it is necessary to develop better lubrication. Like use of synthetic paraffinic oil exhibited approximate double of the fatigue life with 10% Fluorocarbon and Polyphenyl Ether. However near 700˚F, the rise in temperature caused to decrease the bearing torque, while beyond the temperature 700˚F the bearing torque increase gradually. And with the use of some lubricants shows poor lubricity and faces lubrication scarcity, which cause to early pitting, wear and sludge formation. Lubricants got degrade as chemical changes within the oil flow of fully developed elastohydrodynamic (EHD) contact. Also the lubricant starvation in EHD lubrication caused to chemical change in lubricants. It proved that lubricant degradation rate can be controlled by the chemical poisoning of metal of rolling bearing. The contaminated lubricants make detriment the surface of rolling elements of a rolling bearing, which caused to initiate the spall and reduce the life of bearing[23-25]. As the sliding/skidding of ball occur at the outer race, this lead to surface damage with contaminated lubricant, therefore it was recommended that the ultraclean lubricant supply in the recirculating system. Also, during operation, lubricant got out of the race or got thick in semisolid form and starved condition occurs at the races. However, the conventional grease proved to better lubricants over wide application of machinery[3]. The sealed bearing solves the problem of grease move out problem from the raceways [26- 29].

The analysis of ball motion with respect to cage motion play extensive role in the geometry consideration of rolling element bearing. Prediction was made about the ball motion using classical theory of EHL, the elastic force by the lubricants on the race surface in various types of bearing such as thrust bearing/ deep groove ball bearing. In order to avoid ball-cage collision, the proper alignment of bearing must be requiring for stability of cage ball motion. Skidding problem faced in elements of angular contact ball bearing could analyze with classical equation, which provide the prediction about minimum preload requirement. To avoid the cage instability, cage was removed from the bearing element for low operating bearing. And it was found the load bearing capacity of this kind of bearing was very less near to negligible, a group of balls was formed leaving the large gap among the balls. Race widening method adopted for pure spinning of balls by designing the widen tracks, during test no damage was detected on the races. Many computer models were developed and validate with experimental data to study the ball cage motion[4-12].

The elastohydrodynamic lubrication (EHL) theory was developed for the analysis of minimum film thickness within the non-conformal contacts of the rolling element bearing. The stress produced between the contact zone, distribute over an elliptical region under EHL regime. This theory was very useful for calculation of film thickness, pressure distribution, and shear stress and temperature characteristics with non-Newtonian fluid in gears [13-15]. It analyzed experimentally that the EHL regime is the best suit for gears or for all elastohydrodynamic (EHD) contacts. However the lubrication condition has great effect in rolling contacts like in rolling element bearing. The insufficient quantity of lubricant is term as starvation in the contact zone with small fluid film thickness. Jet lubrication with single or dual orifice was used to supply sufficient quantity of lubrication to maintain low operating temperature in the elements [16-20].

High speed up to 2-3 million DN (product of diameter and rpm) aircraft engine bearings has a reduced fatigue life significantly due to high centrifugal force occur at the outer race by balls of bearing. The parameters such as outer-inner race cooling, lubricant flow, and inlet temperature of lubricating oil affects the bearing operating temperature as well endurance life at 3million DN. The fatigue life was significantly increase with the use of spherically hollow or cylindrically drilled balls but the balls were face early flexure failure due to the high stress developed at the inner surface of balls. The centrifugal force problem in high speed bearing can be solved with the use of a ball bearing and fluid film bearing connecting in parallel and more effectively in series to share the rotational speed of the system. Further the two balls race contact points at arched outer race share the centrifugal load and improving the life of bearing[21- 22].

This literature review present recent development to improve the performance of rolling bearing with continuous health monitoring. Researchers developed the numerical model to analyze the performance of bearing and come with solution with improved parameters i.e. reduction in friction, vibration decrement, wear reduction, and increase in fatigue life.

2. Frictional torque measurement: The classical adhesion theory of friction provides the base to friction analysis of bearing. The interlocking of surface asperities is cause to friction force (F), which is directly proportional to normal load (W) [F ∝W]. When two rolling surface makes contact, the pressure distribution around the contact point caused to reacting force, which produce a reacting friction torque or frictional torque. The frictional torque plays very important role in a rolling bearing with a wide variation of speed and load, depending on the types and form of lubricant present in the bearing elements. The frictional heat generation in the rolling bearing has considerable influences on the operating condition and thermal contact resistance. Heat generation in the hybrid ceramic ball bearing vary with speed and load. For more viscous synthetic paraffinic oil, cage drag was found to account for approximately 2 to 10 percent of total experimental bearing torque and lubricant supply has an impact on frictional torque also [13-14].

An analytical model was developed to analyze the spinning torque, cage drag, rolling resistance and compared with experimental results, which indicates that high viscous fluid has less cage drag as compared to low viscous fluid. The spin motions of balls in the raceway create the gyroscopic effect. Spinning torque and rolling resistance are the function of relative spin velocities of inner and outer races. A dynamic model predicted the friction torque and noise variation with time. Spin torque and load of contact equation estimates the gyroscopic effect in heavy loaded bearing. Frictional torque relation with contact load given to model the drift analysis (eq.1) [15-16].

$$\tau = CfP^{4/3} \tag{1}$$

'Dalh hysteresis model' was developed for the analysis for solid lubricants with MoS2 coated surface, silver nitride balls or steel balls (eq.2). A relationship established between bearing friction torque (T) and study rolling friction torque with angular displacement (xa) as;

$$\frac{T}{T_s} = 1 - \left[\frac{x_a}{x_m}\right]^{-2} \quad (2)$$

xm is modeling parameter depends upon study friction torque and rest slop. A numerical method presented for ball bearing and tapered roller bearing, the bearing torque found similar and ball bearing compete, the roller bearing life with larger size balls. A rib torque has been observed in tapper roller bearing [14].

A running torque(M) and suggested a running torque formula for deep grove ball bearing with/without filled polymer coated cage cavity [18].

$$M = M_p + M_e + M_d + M_s + M_{EHL} + M_b \quad (3)$$

$$M = M_p + M_e + M_d + M_s + M_{EHL} + M_c \quad (4)$$

Mp, Me, Md, Ms, MEHL, Mb and Mc are running torque caused by the shearing resistance of the polymer lubricant, elastic hysteresis, differential slip, spinning friction of the balls, EHL viscous rolling resistance, the friction between the balls and the polymer lubricant, and friction between the balls and the cage respectively. The above basic equation was very helpful in the development of a tribodynamic model of ball piston motion to study the friction force and lubricants behavior. In the friction and whirl analysis using FFT (Fast Fourier Transformation) of a float ring turbo charger and ball bearing turbocharger, it was found that the ball bearing turbocharger has more stability at all the speeds and oil pressure. Effect of behavior ball-race contact and kinematics of ball was studied to attain the correct ball equilibrium, the ball race contact angle and normal load affect the external torque on the balls [26-28].

Experiments were performed to resolve the frictional torque issue in the bearings. By conducting various experiments for study of lubrication system and effects of properties of lubricant on the frictional torque in the bearing i.e. using electrical conductivity method and found that lubricant viscosity varies with lubricant condition for identical lubricants and torque can be monitored with viscosity for various lubricants. The conventional lubricant can't withstand with extreme environmental conditions such as very low temperature in space craft application, use of solid lubricant proved very effective to beat such extreme condition [17- 18].

MoS2 coated surface act as self-lubricated and has variation in frictional torque with normal load and coating thickness. The additional experiments revealed that the friction torque in MoS2 coated surfaces remained relatively constant with velocity[15]. In contrast, the friction torque characteristics of oil-lubricated (SAE 20 and DTE 24) steel and silicon nitride balls tend to undergo a change with velocity as a bearing crossed from boundary, to mixed, to elastohydrodynamic lubrications (EHL) regimes. It was found an increment in frictional torque in conventional and textured races with respect to high load. But it was decreased by 27%, 4% and 10% at the speed of 1.8, 3 and 4 m/s in textured race bearing with MoS2 blended grease[23-24].

The textured (spherical dimples over the surface area of 120.42 mm2 of stationary race) ball bearing outer race showed lesser frictional torque as compared to conventional race. A low order numerical model was developed for friction analysis and design prediction of linear ball bearing with less computation cost and accuracy of 7% approximately. A turbocharger test rig with proximity sensors was developed for performance analysis of ball bearing of a turbo charger. It correlate the frictional measurement with the shaft whirl and conclude that shaft whirl caused to skidding and results in increased frictional torque [29-31].

3. Effect of film thickness on wear: The influence of lubricant film thickness on wear was studied using EHL theory. Researchers developed the model for Newtonian and non-Newtonian fluid film thickness with elastohydrodynamic lubrication. To study the wear behavior of rolling bearing was difficult, researchers had made extreme efforts to analysis the wear using many arrangements and simulation model such as ball on rolling disc arrangements, strain gauge method [23-25]. Heavy load and lubrication starvation cause to race wear,

also preload in angular contact bearing has certain effect on wear due to heat generation. [26-28].
The equation for dimensionless film thickness (H) for non-Newtonian and Newtonian fluids developed presented as follows:

$$H_{min} = H_{min,N} f(U^*, \gamma, U, W, G) \quad (5)$$

$$H_{min,N} = 3.07 U^{0.71} G^{0.5} W^{-0.11} \quad (6)$$

Where W, U, G, U* are the dimensionless parameters for load, speed, material and sliding velocity respectively. The sliding velocity measures the limiting shear stress constant (γ) at the contact zone. Above equations were used to develop more advance model for estimating the lubrication starvation.

It was observed that minimum and central film thickness (H0,F) varied with the sliding velocity. A lubricated steel ball over a glass race shows Hertzian pressure distribution and analysis was made to study the effects of the load over the Hertzian contact zone, which found to stick with the EHL theory. Also, the lubrication conditions in the contact area were determined as fully flooded and starved. Researchers developed bearing test rigs based upon EHD theory, which analyze the traction force in hydrodynamic bearing [29-31].

An analytical model was developed to differentiate the dimensional distance of fully flooded and starvation condition (d*) by.

$$d^* = 1 + 3.06 \left[\frac{H_{0,F}}{B^2}\right]^{0.58}$$, where B- semi minor axis of elliptic contact … (7)

Ultrasonic method to monitor the film layer of lubricant in the rolling element was used. A Lubricant film thickness (h) relation with respect to reflection coefficient (R) and acoustic factor (Fa) based upon ultrasonic theory; provide the idea about failure mechanism of bearing elements.

$$R = \sqrt{\frac{F_a^{\,2}}{1 + F_a^{\,2}}};$$

Where $F_a = \frac{\pi h f z_i}{B}$ (8)

This acoustic factor depends on the measured acoustic impedance (zi), frequency of ultrasound (f) and bulk modulus of lubricants (B). The variation in lubricant thickness with reflection coefficient helps in the monitoring of failure of bearing elements. The porous polyimide cage and cotton phenolic cage manufactured for better oil retention in the race. Water absorption and oil exchange in the porous polyimide cage fully immersed in the oil [32-34].

The surface wear of rolling elements of a rolling bearing caused to early failure of the bearing, occurrence of wear due to many reasons such as lubricant impurities, geometrical imperfections, ball waviness and fatigue spall initiation etc. Lubricant breakdown rate increased with the exposure of surface to some chemicals such as alkylglyceryl ether sulfonates with in the EHD contacts and spall initiation occur with use of contaminated lubricant, therefore the use of non-contaminated and fine/absolute filtered lubricant has more life and less wear. However, the presence of micro pitting on the surface of bearing operated with 49-micron absolute filtration of contaminated lubricant [35-37].The wear of textured outer race was more than the conventional race in dry condition (without lubrication). SEM image shows some wear debris also entrapped in these dimples which serves as micro reservoirs for lubrication.

To examine flow regimes of flooded ball bearing, computation fluid dynamic (CFD), numerical and experimental analysis was carried out by varying the flow parameters such as flow rate, pressure and velocity, it was found that a large velocity gradient exists at the wall of moving race. Also, the existence of reverse flow region in the gap between cage, balls and rings of bearing regions with generation of source-sink flow was observed[38].

The chaotic component in starved lubricating conditions occurs during the running of grease lubrication for long duration and small effect of load on film thickness. The real film thickness was decreased with the speed increment. EHL film thickness of a grease lubricated deep groove ball bearing was measured using the electrical capacity method and the speed effect on film thickness for all rolling component was studied by fixing the load and changing the speed[39].Later on, EHL film behavior was studied, the lubricant film thickness got reduced to minimum on sudden drop in speed to rest of bearing. In perfluoropolyether (PFPE) and MoS2 film coated bearing, the generation of dust increase with number of cycles and form wear of the surface [40-41].

Grease lubricant behavior testing was performed for conducting and resisting electrical current. It was observed in railway wagon bearing that the damage of bearing due to high resistivity of grease cause to starvation. During the other experiment for three greases, it was found that the grease film thickness decreased with rise in temperature, which results in reduction of base oil viscosity and starvation. With increasing speed, the normalized grease film thickness decrease after churning phase, results in increment in starvation level but it become stable at high speed. The film thickness for grease decreases with increase in load for long running time. Also a small effect observed in the normalized film thickness for Lithium (Li/M) thickened grease (hg/hff= 0.3 at high speed), while higher than for the Polyurea thickened grease, the normalized film thickness observed that of the lithium greases with the lower base oil viscosity[42].

A failure study was reported using fractography, SEM, EDS and Raman spectroscopy for a Zirconium Dioxide rings and Silicon Nitride balls made ball bearing and concluded that Y-TZP zirconia is not a suitable combination with silicon nitride ball bearing. This was damaged during service due to the premature wear. During the observation of various components of ball bearing, many deep grooves, cracks on surface and 50 µm pits approximately were found. It was cleared from the analysis that material abrasion and wear debris due to carbon contamination on the surface of inner race was reported. From the Raman Spectroscopy the monoclinic and tetragonal spectrum were found prevalent at the surface of inner race as phase transformation. SEM analysis of silicon nitride balls gave the evidence of a 60 um diameter pit approximately on the ball surface and a white color layer contains about 40% of zirconia by mass recognized too [43].

Relative investigations of dimensional abilities of estimating wear using non-contact methodology of confocal microscopy. Wear measurement test of balls by gravimetric method didn't clear the uncertainty with worn mass. When dimension metrology used for wear evaluation of balls, optical confocal microscopy provide a better understanding associated with the uncertainty of volume and density of balls. When the first worn surface was generally settled, a joined strategy for mass loss with nearby geometry of volume loss adjustment may be a helpful technique. The portrayal of the perplexing systems of wear through the surface dimensional metrology can't be connected effectively with the profile unpleasantness parameters of the well-used surface. In any case, this fundamental investigation demonstrates some normal variety patterns of areal parameters with the coefficient of friction. Some coating on bearing elements has great impact on wear like Ag and OTS-SAM were found protecting coating for micro-ball bearing and reducing friction coefficient. Specifically OTS-SAM was more effective wear characteristics as compared to Ag thin film [44-45].

Entrapped MoS2 particles of lubricant in the dimples provide a cushion to the rolling element caused to less wear. SEM image cleared that disappearance of scratches from the convention/textured races and balls with MoS2 grease. Also, better stability and long life of the soap fibers of grease due to the presence of dimpled race. Mechanical deformation and friction reduced significantly with SiC coating, The SiC coating vanish faster than TiN and volumetric adhesive wear rates are observed between 4x10-4 and 4x10-5 mm3/mN.rev during the study. The clearance effect on wear observed and found a slight change in clearance severe wears shown on the rolling elements as well on the race. Experimentally it was clear that the wear increased with decreasing cage ball pocket clearance in the study of dynamic behavior of ball cage motion [46-49].

Vibrational monitoring: The performance of rolling bearing is strongly correlated with the vibration of rolling elements. Rolling bearing face the problem of vibration due to the defects occurs at the races during operation. Sometimes the dirt/ foreign particle contamination in the lubricant cause to spall at races of bearing. Fourier decomposition of the signal received by an accelerometer fixed up on stationary outer race associated with uncertainties of more than one frequency. These characteristic frequencies were termed as 'elastic contact frequency' and 'bearing kinetic frequency' and varies as 1/6th and 1/2nd power of ball race contact. Ball, race and cage stability model was developed for various material combinations and solved with ordinary differential equation system. The prediction was made for ball velocity and acceleration with respect to load versus stress and ball position. Dynamic model presented, describe the ball and ball retainer instability effect, viscous drag produced between ball retainer and races by lubrication. Planar Analysis of a Dynamic Retainer (PADRE) simulation predicts the instability relationship with ball- retainer friction and highly unstable characteristic frequency. A rigid body model defines the cage instability effectively than the earlier resonance model. For highly imbalance ball retainer motion, the whirl motion of retainer shown as peak point frequency in the signal analyzer. To solve the problem, the finite element rotor dynamic analysis the ball bearing showed the gyroscopic effect results in high natural frequency, while the spiral grooved bearing operate with less natural frequencies and higher the damping due to lower shaft rigidity [50-54].

In the angular contact ball bearing problem of the occurrence of resonance could rectify with appropriate design by considering a suitable ball pass frequency and preload and suitable number of balls [35]. Effect of preload on damping and damping ratio was studied using Fourier Transform Analyzer (FFT) for a unifying deep groove ball bearing with its inner race with shaft and outer race with rotor. This single fitted bearing was analyzed by varying other parameters such as speed, preload, clearance and number of balls.

Hertzian load- deflection relationship Q=Kδ^(3/2) (Q is load on ball, K is stiffness of ball material and δ is deflection) used for modeling of the system and Runge-Kutta (RK) numerical method gives the solution of equation of motion and deflection of ball, which gives the optimum value of number of balls and preload in an angular contact ball bearing. This fundamental relationship was very useful in order to minimize the ball passage vibration of linear guided ball bearing arrangements as minimizing the stiffness of material. The pitching and yawing effect was analyzed by arranging the balls in different groups [55-58].

The vibration spectrum due to waviness was analyzed and found the dependency of frequency of vibration with the order of waviness and Ball Passage Frequency (BPF). The measurement and continuous monitoring of sound and vibration of linear guiding type (LGT) was done using a drive unit system and digital spectrum analyzer, results showed the increased level of sound pressure with increase of linear velocity and high peak spectrum of sound with natural vibration. A collision model presented the high acoustic emission in term of high amplitude of the vibration in the defect region [59-62].

Harris developed a mathematical model to calculate the shear stress on the spalled surface and conducted a ball & v-ring test to monitor the spall propagation. The relation between dynamic load (Qd) and spall length was very useful to estimate the value of constant (KS) in order to predict the life of ball under vibratory condition (eq.) [63].

$$Q_d = K_S \sqrt{\frac{l_S}{l_T}} \tag{9}$$

Accelerometer mounted record the vibrations, which was increased with spall progression. Vibratory velocity measured by a Measuring Amplifier and frequency analyzed by Fourier Transform (FFT) analyzer, results found lower vibratory velocity for all ceramic ball bearing than hybrid ball bearings (steel rings & ceramic balls) and conventional steel ball bearings. Analytical model was presented to analyze the stability of a parametrically excited system due to ball bearing waviness, Fourier expansion provide the idea about effect of harmonic frequency and principle frequency on the system. Frequency domain analysis used by researchers for vibration monitoring, damage location identification and Poincare map gave approximation of level and type of damage[64-66].

In the vibration monitoring system an ultrasonic piezoelectric accelerometer attached to bearing housing, send the reflected signal to the signal conditioner for storage and analysis purpose [33]. On analysis the failure mechanism detected with variation in the reflection profile. In order to measure the performance of a rolling bearing a finite element simulation was presented for flexible cage and rigid cage, the ball retainer force was reached down and had less slip between race and ball, less heat generation with lower the wear [67].

A time frequency domain used to measure the energy degradation of MoS2 coated ball of thrust bearing in three phases [68]. The Duffing oscillator gave swift result for defect detection on rolling components of ball bearing, while results provided by envelop analysis dependent on the selected central frequency and bandwidth. An envelope analysis performed using Hilbert transformation of a faulted test deep groove ball bearing with added external excitation of range 10-1000 Hz. Vibration spectra of resonance produced by the balls and defected race interaction was recorded by accelerometer and a central frequency of spectral kurtosis (sk) of filter length 32 and 64 was selected for inner race (2722 Hz & 1325 Hz for 32 and 64 filter length) as well for outer race (1578 Hz & 1306 Hz for 32 and 64 filter length) [69]. Through MATLAB analysis, it was determined that the least ratio of the highest amplitude of the frequency to the amplitude of defective frequency for all the central frequency combination. These least ratios occur due to the weak signal generation from the defected area interaction of race and balls. The changed chaotic motion of Duffing oscillator into periodic motion indicates the presence of defects in bearing.

It was performed vibrational analysis of textured race (Fig.2.) and investigated that in textured reduction in amplitude of vibration and temperature rise with MoS2 Blended grease than fresh grease than conventional races ball bearing [70]. Experiments were conducted on a SKF-51308 test ball bearing of steel (EN-31) with LGMT3 fresh grease and MoS2 (by mixing 3% Molybdenum disulfide powder with general purpose grease) blended grease. Also, it was investigated the effect textured race and found a significant reduction of approximate 21% in vibration amplitude and 14% reduction in temperature due to better grease retention in the dimples of outer race (cause to the tribo-film formation).

Fig.2. Textured outer race of test bearing, [46].

The whirling motion of the cage decreased for all mass imbalance conditions as the speed increased with the increasing fluid force on the cage. The intermittent collisions reduced and disturbed the constant whirling ratio of the cage[71- 72].

5. Fatigue life prediction analysis: In the rolling bearing the existence of thermal gradient caused to affect the fatigue life adversely. Heat generation in the bearing produced failure in the form of loss of surface hardness of rolling elements and loss of lubricant properties. For estimating the fatigue life of bearing, the numerical model developed gives the idea about operating temperature and rate of heat generation of bearing theoretically. Lognormal Distributions provide a reliable estimation for low failure probabilities of that ball bearing fatigue lives than by extrapolating from Weibull Distributions [73-74]. The distribution for lognormal probability or relative frequency function presented for positive time (t) and shape parameter (σ) as (eq. 10):

$$f(t) = \frac{1}{\sigma t\sqrt{2\pi}} exp\left\{-\frac{1}{2}\left(\frac{\log_e t - \mu}{\sigma}\right)^2\right\} \quad (10)$$

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Where, μ - mean scale parameter for Lognormal distribution

This is particularly important for results which give approximate values of the Weibull shape parameter. It is very important for comparing fatigue lives of ball bearing track materials for early failure region. The investigation of fatigue life of all elements of rolling bearings has been limited by the data available for large homogeneous samples.

A comprehensive analysis of fatigue life with the severities of load find that the compressive stress and stress frequency influences the bearing fatigue life. In order to reduce the compressive stress the conformity could be increase by adding more balls/ rollers between the races. The radial stiffness would be reducing by creating undercuts on the cross-sectional geometries of bearing, which resulting the increased fatigue life. Authors investigated the fatigue life was increase for a thrust loaded arched bearing with centrifugal force effect at high speed more than 20000rpm and the size of arch has a relation with spinning of ball at outer race. This modified geometry of outer race cause to increase the outer race temperature and cage slip, some noise was found with varying load and speed. Subsequently the outer race temperature decrease with increasing the oil flow. A mathematical model was developed and solved using Newton-Raphson method to analysis the modified geometry as double arched raceway. Difficulties of conventional single arched and two contact raceway extended to three-four contact races and further to two ached raceways were remain unsolved in terms of fatigue life [75-78].

A series hybrid bearing setup was developed to increase the fatigue life of ball bearing by reducing the speed of bearing with 3.4 and 5.9 life improvement factors for 3 million and 4 million DN values. Fatigue life of ball bearing varied with load inversely and a hollow ball bearing test was performed to determine the fatigue strength of ball for various reductions (21.7%, 50% and 56.5%) in mass to reduce the centrifugal effect of balls in thrust loaded bearing, some of the hollow balls fail during flexural stress test. During five ball test for fatigue life it was found that the life (L) has a relation with Hertz stress, it means the stress-life exponent for vacuum-processed AISI52100 balls were 12 at 500˚F. The study proved that the drilled ball bearings run cooler than solid ball bearing by few degrees, while solid ball bearing has more fatigue life than drilled ball bearing at high speed up to 3 million DN [79-81].

The series hybrid bearing system reduces the speed of ball bearing near to 0.5 times the shaft speed. If the lubricant supply failure occurs due to oil flow cutoff, the inner race of ball bearing reached to the speed value of shaft and again reached back to series-hybrid system speed on restarting the oil flow. Oil lubricant parameters such as inlet temperature, flow rate has great impact on fatigue life of thrust loaded ball bearing. Researchers used PTFE, lead and lead alloy coated retainers and proved that the electroplated/ion plated lead retainers form a good thickness transfer film of lubricant on the rolling elements up to 50 micron thickness to increase the bearing life in cryogenic environment [82]83].

During number of research, it found that Silicon-nitride ball bearing shows equal life to the steel ball bearing but higher than other ceramic material at same load condition. Silicon Nitride balls with steel rings bearing generate heat approximate 10-20% less than the bearing having steel balls and required less preload as in case of angular contact ball bearing, which results in low thermal expansion and less aerodynamics axial thrust imbalance of the engine and increased fatigue life of bearing. Rise in temperature for hybrid ceramic balls was found smooth and less as compare to steel balls for less preload with 1.5 meter long pipe in oil-air lubrication system. The ceramic balls were to be found as self-lubricated for high speed. However, it was difficult to determine the variation of temperature with change of various types of lubricating oil. The oil air jet system found suitable for control of temperature rise but there was increase in power loss [84-88].

In the deep-groove ball the cyclic stress- strain effect caused to plastic deformation of raceway contact, it was observed that the circumferential residual stresses were in controlled manner, while in later cyclic stress were high and caused to plastic

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deformation in the form of cracks or loss of material from raceway of bearing. Finite element analysis (FEA) was carried out of angular contact ball bearing by considering hoop stress effect in the race surface equivalent to cylindrical bearing. The performance of rolling bearing was analyzed with consideration of geometrical imperfection of races using a computer model ADORE [89-91]. The particular geometrical parameters (A̅) were predicted with the variation in amplitude (a), frequency (n) and angular position (θ) for the race out roundness and groove curvature (A) variation as (eq.11);

$$A = \bar{A} + a\cos n\theta \tag{11}$$

Fatigue life prediction of main shaft bearing in a gas turbine was made using 'Lundberg and Palmgren' basic equation of survival probability(S), failure stress(τ) and depth of stress occurrence with number of stress cycle (N), (eq. 12) and using L10 fatigue life relation of load rating (C) with the equivalent applied load (P) (eq. 13). It was found the fatigue life of bearing material follows the above equation for all conditions, speed- load and lubrication was not satisfactory. A more improved theory proved to predict more effectively using a computer program namely 'SHABERTH' . Also, ball fatigue life prediction was made using extended model based on life factors and considering the raceway friction stress. The incremental volume (V) found to be less than the shear stress prone volume. Hertzian stress proved to high stress defth (z) considered to make fatigue life lesser than the predicted life [92-95].

$$\ln\frac{1}{S} \sim \frac{N^e \tau_0^C V}{Z_0^h} \tag{12}$$

$$L_{10} = \left(\frac{C}{P}\right)^3 \tag{13}$$

Spall initiation in the bearing component cause to increase stresses on the surface. Surface stresses were developed with the progression of spall region. A mathematical model provided for fatigue life prediction (eq.14) [63].

$$w_S = 0.4177 + 8.647\sqrt{\frac{l_S}{l_T}} \quad \text{and}$$

$$l_s = 2.847\, b \tag{14}$$

The spall width (wS) and spall length (lS) relation estimate the spall progression volume as volume of material affected in the spall progression. The basic 'Lundberg and Palmgren' equation is not enough to predict the life of bearing manufactured with modern homogeneous bearing material, a modern approach used in mathematical model (eq. 15) to determine the probability of survival of rolling bearing [96].

$$S = \sum_{n=0}^{k}(-1)^n \frac{\bar{A}^n N^{ne} I^n}{n!} \tag{15}$$

Where, A is constant and I moment of inertia. This model was effectively used in fatigue life prediction for complete failure analysis of rolling bearing with various operating conditions.

II. CONCLUSIONS

This review focused on the recent work done in the area of performance monitoring relative to life of rolling bearing. Tremendous efforts have been made to improve the fatigue life with the use of modified geometry of bearing, use of various combinations of ball-race-cage material and provided protecting coatings.

- It provided better techniques for vibration analysis as envelop analysis, FFT analyzer, duffing oscillator for rolling bearing.
- Textured race and Nano particle blended lubricants were used to reduce the vibrational amplitude, frictional torque and to improve the wear performance of bearing and researchers performed a relative investigation of dimensional abilities of estimating wear.
- Hertzian contact theory based numerical models were developed for monitoring and fault prediction show very fine results while compared with the experimental data.
- More experimental data require for validation the mathematical model presented.

- More future research needed to develop the numerical model for analysis of textured races and ball contact to achieve the more accuracy level and to improve the performance parameters of rolling bearing.

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