FATIGUE LIFE ANALSIS OF THRUST BALL BEARING A REVIEW

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ABSTRACT

This review paper presents various researches on the analysis of fatigue life of thrust ball bearing through 3 dimensional software Pro-e wildfire-5.0. An analylitical method is good, less expensive and gives the best results. Analytical results give good agreement with the experimental data. The thrust ball bearings are subjected to various, thrust & dynamic loads, which simulated easily through pro-e software & analysis because experimentally calculation is very complicated. In this study the researchers used various analysis codes which are based on very high cycle fatigue Theory (VHCF) and got a good result through these codes.

Keywords -: Very high cycle fatigue, fatigue limit stress.

INTRODUCTION-

Thrust bearing - A type of bearing designed to reduce friction by carrying thrust or axial loads. Thrust bearings can be either plain or anti-friction bearings. The type of component supported determines the type of thrust bearing used. A thrust bearing is a particular type of rotary bearing. Like other bearings they permit rotation between parts, but they are designed to support a high axial load while doing this. Single direction thrust ball bearings consist of a shaft washer, a housing washer and a ball and cage thrust assembly. The bearings are separable so that mounting is simple as the washers and the ball and cage assembly can be mounted separately. Thrust bearings come in several varieties.

- **Thrust ball bearings-** It composed of ball bearings supported in a ring, can be used in low thrust applications where there is little axial load.
- **Cylindrical thrust roller bearings-** It consist of small cylindrical rollers arranged flat with their axes pointing to the axis of the bearing. They give very good carrying capacity and are cheap, but tend to wear due to the differences in radial speed and friction which is higher than with ball bearings.
- **Tapered roller thrust bearings-** It consist of small tapered rollers arranged so that their axes all converge at a point on the axis of the bearing. The length of the roller and the diameter of the wide and the narrow ends and the angle of rollers need to be carefully calculated to provide the correct taper so that each end of the

roller rolls smoothly on the bearing face without skidding. These are the type most commonly used in automotive applications (to support the wheels of a motor car for example), where they are used in pairs to accommodate axial thrust in either direction, as well as radial loads. They can support rather larger thrust loads than the ball type due to the larger contact area, but are more expensive to manufacture.

- **Spherical roller thrust bearings-** It use symmetrical rollers of barrel shape, rolling inside an outer race with a spherical inner surface. They can accommodate combined radial and axial loads and also accommodate misalignment of the shafts. They are often used together with radial spherical roller bearings.
- **Fluid bearings-** Where the axial thrust is supported on a thin layer of pressurized liquid—these give low drag.
- **Magnetic bearings-** Where the axial thrust is supported on a magnetic field. This is used where very high speeds or very low drag is needed, for example the Zippe-type centrifuge.

USE OF THRUST BEARINGS-They are commonly used in automotive, marine, and aerospace applications. Thrust bearings are used in cars because the forward gears in modern car gearboxes use helical gears which, while aiding in smoothness and noise reduction, cause axial forces that need to be dealt with. The double helical or herringbone gear balances the thrust caused by normal helical gears. One specific thrust bearing in an automobile is the clutch" throw out" bearing, sometimes called the clutch release bearing. Thrust ball bearings consist of two precision chrome steel washers (ring) and a ball complement spaced by bronze retainer. They can be supplied with or without radius ball grooves in the rings. Thrust bearings are used under purely axial loads.

Bearing Race

The rolling-elements of a rolling-element bearing ride on races. The large race that goes into a bore is called the outer race, and the small race that the shaft rides in is called the inner race.

Design of thrust ball bearing

In the case of ball bearings, the bearing has inner and outer races and a set of balls. Each race is a ring with a groove where the balls rest. The groove is usually shaped so the ball is a slightly loose fit in the groove. Thus, in principle, the ball contacts each race at a single point. However, a load on an infinitely small point would cause infinitely high contact pressure. In practice, the ball deforms (flattens) slightly where it contacts each race, much as a tire flattens where it touches the road. The race also dents slightly where each ball presses on it. Thus, the contact between ball and race is of finite size and has finite pressure. Note also that the deformed ball and race do not roll entirely smoothly because different parts of the ball are moving at different speeds as it rolls. Thus, there are opposing forces and sliding motions at each ball/race contact. Overall, these cause bearing drag.

Manufacture-Centerless grinding

The outer diameter (OD) of the races are often centerless ground using the throughfeed process. Centerless grinding can achieve a very high degree of accuracy, especially when done in stages. These stages are: rough, semi-finish and finish. Each grinding stage is designed to remove enough stock material from the casing so that the next stage does not encounter any problems such as burning or surface chatter, the finish stage achieves the final dimension. Each grinding wheel at all of the aforementioned stages has a varying degree of abrasive quality (finish being the finest grade) to achieve the appropriate stock removal for the next stage and final surface finish required.

Feeding

Bearing casings are introduced to the grinding action via means of a transfer from the delivery system to a pair of infeed rollers, these infeed rollers are tapered to a certain angle so that the casings are driven forward until the regulating wheel and grinding wheel catch them and slow them to their grinding speed which can be altered by speed control of the regulating wheel. The casings are constantly rotating and are fed into the grinding area to prevent separation which can cause finish/size problems or even a "bump" that can potentially crack or destroy casings and will damage the grinding and regulating wheels.

FATIGUE LIFE OF THRUST BALL BEARING

The service or fatigue life of a bearing can be calculated more or less precisely, depending on the defined operating conditions. The required bearing life is set by the manufacturer of the equipment in which the bearing is fitted. The simplest method recommended by standard ISO 281, enables one to calculate the service life reached by 90% of bearings operating under a dynamic load.

- The nominal life L10,

L10 = (C / P)n 10^6 in revolutions

Where

- L10=rated fatigue life with a statistical reliability of 90%
- P=bearing equivalent load
- C=basic radial dynamic load rating (Get from individual bearing selection charts)

Basic Life Rating

The Basic Life Rating (L10) is defined in specification JIS B1518 "Dynamic load ratings and rating life for rolling bearings" as follows:

The Basic Life Rating is the life obtained with 90% reliability, when an individual bearing or an identical group of bearings are manufactured with common materials, common manufacturing processes and quality, and operate under the same conventional conditions. L10 Life is the accumulated rotation where 90% of survive without material flaking when they are operated under fixed conditions, of a population of bearings.
The calculation formula for the Basic Life Rating is the following.

$$L_{10} = \left(\frac{C_r}{P_r}\right)^3$$

L_{10} : Basic Life Rating in millions of revolutions
C_r : Basic Dynamic Load Rating
P_r : Equivalent Dynamic Radial Load Factor

There is a relationship between the Basic Life Rating (revolutions) and Basic Life (time).

$$L_{10} = \left(\frac{10^6}{60 \cdot n}\right) \times \left(\frac{C_r}{P_r}\right)^3 \quad (h)$$

n : Rotation Speed (min^{-1})
h : Time (hours)

Dynamic Equivalent Radial Load Factor (Pr)

The Dynamic Equivalent Radial Load Factor is defined as "the direction and magnitude to the bearing, which is able to obtain the same life under the actual load and rotation conditions". From the calculation formula and the table below, the axial and the radial loads are replaced by the Dynamic Equivalent Radial Load Factor (Pr).

$$Pr = XFr + YFa$$

X and Y are taken from the table below

Fr = Radial load (N or kgf)

Fa = Axial load (N or kgf)

Axial Load Ratio		$\frac{F_a}{F_r} \le e$		$\frac{F_a}{F_r} > e$		e
Units		X	Y	X	Y	
N	{kgf}					
$\frac{F_a}{iZD_w^2}$						
0.172	{ 0.0175 }				2.30	0.19
0.345	{ 0.0352 }				1.99	0.22
0.689	{ 0.0703 }				1.71	0.26
1.03	{ 0.105 }	1	0	0.56	1.55	0.28
1.38	{ 0.143 }				1.45	0.30
2.07	{ 0.211 }				1.31	0.34
3.45	{ 0.352 }				1.15	0.38
5.17	{ 0.527 }				1.04	0.42
6.89	{ 0.703 }				1.00	0.44

i : No. of rows

Z : No. of balls

Dw : Ball Diameter (mm)

The values for X andY that are not in the above table shall be calculated by linear interpolation.

SIMULATION & ANALYSIS WITH 3 DIMENSIONAL SOFTWARE

Computational 3 dim simulation & analysis indicates the numerical solution of differential governing equations of thrust ball bearings, with the help of computers. This technique has a wide range of engineering applications. In the field of dynamic research this technique has become increasingly important and it is prominent for studying bearings & other materials. Computational 3 dim simulation & analysis also provides the convenience of being able to switch off specific terms of governing equations. This permits the testing of theoretical models and, inverting the connection, suggesting new paths for theoretical explorations. Computational 3 dim simulation & analysis provides five major advantages compared with various experimental, thrust & dynamic loads:

a. Lead time in design and development is significantly reduced.

b. It can simulate flow conditions not reproducible in experimental model test.

c. It provides more detailed and comprehensive information.

d. It is increasingly more cost-effective than real time testing.

e. It produces lower energy consumption.

A well developed 3 dim code allows alternative designs to be run over a range of parameter values

LITERATURE REVIEW

P. Guy*, P. Meynaud"-(1996), This paper presents an original ultrasonic technique devoted to the early detection of sub-surface fatigue cracks in lOOCr6 bearing steels. Echographic experiments were carried out during interrupted rolling contact fatigue tests. Crack initiation and crack propagation stages were both investigated. Quantitative measurements of sub-surface crack position, depth and angle with respect to the bearing surface are presented. Experimental results are analyzed and discussed in relation to rolling contact fatigue theories. Contrary to what is generally accepted the presented results show that the crack propagation stage could represent an important part of bearing life.

Arne Melander- (1996)in this paper the author has studied A finite element study was performed of driving forces of short cracks at inclusions in bearing steel exposed to rolling contact load. Five inclusion configurations were considered, namely a pore, a manganese sulphide inclusion, a through-cracked alumina inclusion, an alumina inclusion which was uncracked but which could debond from the matrix and finally a titanium nitride inclusion. Short cracks were allowed to grow from the inclusions. The inclusions were 20/xm in diameter. The cracks were allowed to grow from 2 ~m up to 8 ~m away from the inclusion interface into the matrix. The cracks were oriented 45 ° to the rolling contact plane in agreement with the experimental observations of so called butterfly cracks in bearing steels. This generated residual stresses during quenching corresponding to a hardening operation of the steel. Friction was taken into account on the crack faces and on the inclusion interface. The material was modeled as elastic.

A.A. Torrance a, J.E. Morgan b, G.T.Y. Wan-(1996)

This paper describes the results of some rolling contact fatigue tests on roller rings using a new type of fatigue machine. Two types of lubricant were used: base oil, and the same oil with a commercial sulphur-phosphorus extreme-pressure (EP) additive package. The presence of the additives gave a more polished track and a lower wear rate, but at the same time, the L," life and Weibull slope were reduced. These results confirm that such additive types can reduce the life of bearings, possibly by a mechanism of corrosion-enhanced fatigue.

Yoshida, M. Fuji –(2002),

In this work, Ni–P alloy plating and sulfur zing treatments were employed, and the surface modified steel rollers and ball bearings were fatigue-tested under a pure or free rolling contact condition. The fatigue lives of both rollers and bearings were improved by these surface modifications. The contact pressure and subsurface stresses of the surface modified rollers and bearings were analyzed. The reason why the rolling fatigue strengths of surface modified rollers and bearings were higher than those of the non-coated ones would be due to the smaller contact pressure and subsurface stresses by the smaller elasticity as well as the conformity of the plated layer.

Youngsik Choi-(2006)

A rolling contact fatigue life model encompassing crack initiation life and crack propagation life was developed in Part 1 of two companion papers. This paper concerns the experimental verification of the developed life model. The specimens were machined by using various machining conditions and at two different cutting tool conditions: new and worn tools. The experimental results indicate that tool wear, as well as the machining conditions, greatly influences the micro-hardness and residual stress distribution below the machined surface and thus the rolling contact fatigue life. Tool wear induces a highly variable change in fatigue life depending on the machining conditions. The predicted lives by the developed life Model show good agreement with the experimental lives, although the experimental lives of the specimens show significant variation.

Robert Kunc- (2007)

For the purpose of the numerical analysis of the actual carrying capacity of rolling contacts in thrust ball bearings with surface hardened raceways, an elasto-plastic constitutive model was used which links the mechanics of material damage with the isotropic and kinematic hardening or softening. A damage material model, implemented into a commercial finite element program, allows us to monitor the elastic strain, plastic strain increase, stress changes and material damage growth, which are closely related to the number of load cycles. In this way, the location and the time of occurrence of bearing raceway damage can be determined along with the growth of damage up to the point when a micro crack is formed. In other words, low cycle life of rotational rolling connection can be assessed. The paper presents the material model, numerical analysis of the actual carrying capacity of the rolling contact in single-row ball bearings and the verification of the numerical material model with experimental results of low cycle carrying capacity.

Jesse Bockstedt –(2007)

Magnetic treatment has been shown to be capable of altering mechanical properties of materials. Since the behavior of tribological systems depends on the mechanical characteristics of the interacting bodies, applying a magnetic field to an operating tribological system is a potential means for controlling system behavior. As a measure of the effects of applied magnetic fields on tribological processes the micro hardness of ferromagnetic steel thrust bearing washers was measured for washers before and after being run under no-field and applied magnetic field conditions. Analyses of the data showed statistically significant effects. For the specific test conditions used, micro-hardness increased by approximately 5% for bearings run under no-field conditions. For bearings run under pulsed magnetic fields washer hardness decreased by approximately 9%.

T.N. Sexton-(2009)

Down-hole tools used for drilling oil and gas wells are subjected to harsh environments where abrasive fluids, high loads and speeds, and high temperatures can cause tool components, including thrust bearings,
to quickly fail. This paper discusses the advantages that polycrystalline diamond can provide when used as a bearing material in down-hole tools. Laboratory test results are presented that can be used to help predict the performance of polycrystalline diamond bearings in these down-hole applications. Results from two types of tests are presented including tests that measure bearing capacity and those that evaluate diamond bearing wear rates. Bearing failures, generated during laboratory testing, are discussed

Franz-Josef Ebert-(2010)

This article presents an overview of fundamentals of rolling element bearing designs and technologies. The article shows the different types of rolling bearing stressing and the analysis of the stress distribution (principal stresses and equivalent stresses) in the material under the rolling contact area. The article reveals the importance of the cleanliness of bearing steels as well as different types of inclusions and their effect on rolling contact fatigue. Additionally the article describes how to optimize the material properties (strength, toughness and residual stress) by the heat treatment processes. The outcome of these investigations is that endurance life of a rolling element bearing can be achieved if specific operating conditions, an adequate lubrication, good system cleanliness and specific bearing stressing are met. The article provides a guideline for bearing engineers on how designs and Technologies can be applied to optimizing a bearing for a particular industry or aerospace application.

Y.B. Liu a,*, Y.D. Li a, S.X. Li a, Z.G. Yang –(2010)

Methods to predict the S–N curves of high-strength steels in the very high cycle fatigue (VHCF) regime are reviewed. At the same time, a new prediction in form of Basquin's equation is proposed based on the prediction of fatigue strengths in both high cycle fatigue regime and very high cycle fatigue regime. The new prediction is compared to the prior predictions and experiments results obtained by our recent experiment and others' published papers.

Erick S. Alley, Richard W. Neu (-2010)

We have laid the groundwork necessary to determine the influence of microstructure in RCF and related very high cycle fatigue problems. Crystal plasticity material models provide more realistic accumulations of localized plastic strains with cycling compare to homogenized J2 plasticity. With J2 plasticity, the bearing must be overloaded to capture significant plasticity near inclusions; with crystal plasticity, realistic bench test loads can be applied with plastic strain accumulation observed near inclusions in cases where RCF failure is anticipated

Tiago Cousseau a,n, BeatrizGrac-(2011)

A rolling bearing friction torque model was used and the coefficient of friction in full film lubrication was determined for each grease, depending on the operating conditions. The experimental results obtained showed that grease formulation had a very significant influence on friction torque and operating temperature. The friction torque depends on the viscosity of the grease base oil, on its nature (mineral, ester, PAO, etc.), on the coefficient of friction in full film conditions, but also on the interaction between grease thickener and base oil, which affected contact replenishment and contact starvation, and thus influenced the friction torque

Junbiao Lai a,⇑, Thore Lund-(2012)

In the present paper (Part I) a deterministic defect model is developed to describe the fatigue crack growth from de-bonded hard inclusions. The model is shown to provide a unified prediction of fatigue behavior in different regimes, i.e. low cycle fatigue regime dictated by the tensile strength, high cycle fatigue regime obeying Basquin's law and the very high cycle fatigue regime featured by the fisheye and ODA (optically dark area) surrounding an interior fatigue-initiating inclusion on the fracture surface.
The model predictions agree well with experiments. A combination of the deterministic model with a stochastic model that describes the inclusion size distribution allows prediction of fatigue strength and fatigue limit associated with certain reliability of a steel component

V. Martsinkovskya, V. Yurko- (2012)

The problem of increasing the bearing capacity of the thrust bearings is a subject of numerous scientific and technological researches, the results of which are used by a variety of firms, but the proposed measures and recommendations for solving the problem are insufficient to fully compensate for increasing stresses of loads acting on the thrust bearings and to ensure reliable and efficient operation of the units in the entire range of operating conditions.

Antonio Gabelli a, Junbiao Lai , Thore Lund-(2012)

The dynamic load ratings and life rating standard for rolling bearings, ISO 281:2007, makes use of a fatigue limit stress of hardened bearing steels as a parameter in the estimation of the fatigue life of rolling bearings. Part I of this paper series presented an analytical-probabilistic model that provides a unified prediction of fatigue strength corresponding to various fatigue regimes including the very high cycle fatigue (VHCF) regime. Based on the fatigue model, Monte Carlo simulation of rotating bending and push– pull tests of bearing steels indicates the existence of a horizontal asymptote in the S–N curve in excess of 1012 cycles, which corresponds to the fatigue limit. Prediction of fatigue limit associated with certain reliability can be realized using the developed model and methodology. In the present paper (Part II of the paper series), the application of a fatigue limit stress in the standard fatigue load ratings of rolling bearings is described.

R.K. Upadhyay *, L.A. Kumaraswamidhas, -(2013)

This paper addresses Rolling Contact Fatigue (RCF) occurs due to the result of cyclic stress developed during operation and mechanism that involve in fretting failure of rolling element bearing. As bearing raceways of non-rotating rolling element bearings exposed to vibration or sliding oscillation false Brinelling occurs. Bearing surface due to false Brinelling tends to damage within a short period, due to cavities created on the bearing raceway. Recommendation towards enhancement of bearing life is also suggested.

Chengqi Sun, Zhengqiang Lei-(2013)

Experimental results indicate that the fatigue life reduces by about two orders of magnitude when inclusion size doubles. Then, a model is proposed for predicting the fatigue strength of high-strength steels with fish-eye mode failure based on the experimental results for the effect of inclusion size and stress ratio. In the model, the effect of inclusion size a0 and stress ratio R on fatigue strength ra is expressed as ra / a0 m[(1 _ R)/2]a, where m and a are material parameters. The predicted results are in good agreement with our experimental results and the ones reported in literature.

CONCLUSION

The whole paper presented the experimental analysis of thrust ball bearing steels where the 3-dim modeling software is widely used for calculated the fatigue life analysis around the thrust ball bearings & steels,(crack growth Tensile strength and fatigue strength, pressure distribution, load rating factor, Basquin's law .) which is affected by changing formed carbon steel design, angle of contact, speed ratio etc. All simulation result calculation from software plays a significant role for modeling the thrust ball bearings which is working efficiently or not the present given condition. The 3-dim analysis follows the basic theory and assumptions of life rating standard for rolling bearings, ISO 281:2007, makes use of a fatigue limit stress of hardened bearing steels as a parameter in the estimation of the fatigue life of thrust ball bearings. to various fatigue regimes including the very high cycle fatigue (VHCF) regimes.

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