Response of the interface between the ball and the raceway of rolling bearing under starved lubrication

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Abstract. The interface state of the rolling bearing contact pair often directly determined the service performance of the bearing for the aviation equipment, the ball-raceway contact interface would develop from the homogeneous lubrication to starved lubrication when used for a while, the plasto-elastohydrodynamic lubrication analysis model was used to describe the ball-raceway contact interface lubrication condition. The influence of the normal pressure, the rolling speed and the service time to the ball-raceway contact interface wear behavior was investigated by the gearbox and rolling bearing on condition of starved lubrication, it was found that the centre region of the raceway generated concave residual deformation, while the convex residual deformation occurred at the lubrication region’s edge and its surrounding region, the groove and scratch marks were emerged on the contact surface of ball and raceway, which were accelerated produced by the normal load and rolling speed of rolling bearing, besides that the ball-raceway contact interface assumed adhesive wear behavior and the wear width became gradually larger over the time.

1. Introduction

Rolling bearings were important moving parts of aviation equipment, which directly affected the airplane safety and mission. Ordinarily the performance of rolling bearings was largely determined by the lubrication state and wear mechanism of the roller interaction surfaces. The rolling bearings mainly used with the grease lubrication, which could establish the uniform lubrication conditions and enhanced bearing wear resistance, but the grease affected by high temperature and high speed, the grease film on the contact interface became thinner or even vanished over times, the starved lubrication lead to local temperature increase and the sliding friction between the roller and raceway.

Therefore, the ball-raceway contact interface under starved lubrication and its wear mechanism had been emphasized. Generally, the lubrication mechanism of rolling bearings was mainly elastohydrodynamic lubrication[1], the roller was fully lubricated, when the rolling bearing worked, most of the grease was pushed to both sides of the raceway, resulted the entrance gap of the contact cannot be Fully filled, the film thickness was also smaller than that of full-film lubrication, elastohydrodynamic research under starved lubrication conditions was also constant investigated.

Wedeven[2] first analyzed the phenomenon of oil lubrication of rolling bearings through optical interference experiments, which caused people's attention. Chiu[3] analyzed and predicted the lubrication of rolling contact under spent oil conditions; Olaro and Gafitanu[4] established the
analysis model of the ball-roller contact spent oil mechanism for ball bearings; Chevalier[5] The theoretical prediction model is established for the thickness of the elastic hydrodynamic lubricant film under oil condition. Damiens[6] believed that the key factor determining the occurrence of oil shortage is the balance between lubricant loss and replenishment in the lubricant inlet area, and established a predictive model for simulating lubricant film thickness. It can be seen from the above research summary that the current research on the problem of rolling bearing lubricating oil has achieved certain results, but the existing research is mainly based on lubricants, and there are relatively few studies on lubricating grease. Coulon[7] did an experimental investigation on artificially induced indents which caused micro-pits. The time it took for a micro-pit to develop was related to the stressed volume around the pits. Golmohammadi and Sadeghi[8] coupled an elastic-plastic line model with damage mechanics to account for material degradation of pitting around a furrow. AL Mayali[9] performed experimental investigations on the evolution of surface roughness in rolling contacts. They found that plastic deformation during the run-in process contributed to the formation of micro-pits. Hooke[10] developed perturbation analysis of the Reynolds equation as a fast analysis to access the global behavior of rough EHL contacts, under the assumption of small perturbation with respect to the smooth case.

In this paper the aviation rolling bearings was researched based on the theory of interface lubrication mechanics, the interface characteristics of the interaction between the roller and the raceway determined the lubricant film thickness, the wear mechanism of the contact interface under the starved state determined the service performance of the bearing. It provided theoretical reference for the research on the failure mechanism of the aviation rolling bearing, the reasonable maintenance and use of the system, and the improvement of the reliability and using life.

2. Math method
The lubrication model between the roller and the raceway in the deep groove ball bearing is shown in Figure 1. In the figure, x, y, and z represent the circumferential direction, width direction and radial direction of the bearing. $R_x$ and $R_y$ are the radii of curvature of the roller in the circumferential direction and width direction of the bearing. $R_x$ and $R_y$ are the radii of curvature in the circumferential direction and width direction of the bearing. $R_x$ and $R_y$ are the radii of curvature in the circumferential direction and width direction of the outer raceway, $R_x = D_x / 2$, $R_y = D_y / 2$, $R_x = (D_{1x} + D_{2x} + 2D_1 + P_d) / 4$, $R_y = f_x D_x$, where $D_x$ is the diameter of the rolling element, $D_{1x}$ and $D_{2x}$ are the nominal inner diameter and nominal diameter of the bearing, $P_d$ is the radial clearance of the bearing, $f_x$ is the ratio of the radius of curvature in the width direction of the outer raceway to the diameter of the roller.

![Figure 1. roller lubrication model.](image)

Between the roller and the outer raceway, it is lubricated by grease with a suction speed of $U$. When the grease flows from point A1 to point A2, the lubrication gap first decreases and then
increases. When the lubrication gap is reduced, the grease between the roller and the raceway generates dynamic pressure effect, forming grease film. The grease film separates the rolling body from the outer raceway, and thus plays the role of supporting, lubricating and reducing friction\cite{11}.

2.1. Plasto-elastohydrodyanamic model

The Oswald constitutive equation is selected to establish the grease-lubricated Reynolds equation, the relationship is expressed as:

$$\tau = \phi \gamma^n$$ \hspace{1cm} (1)

Where the $\tau$, $\phi$, $\gamma$ and $n$ were the shear force, plastic viscosity, shear strain rate, and rheological index of the grease, the constitutive relationship takes into account the influence of the rheological index $n$ on the shear stress of the grease, which was suitable for yielding Shear stress had little effect on the lubricating performance of grease\cite{12}, the corresponding grease lubrication Reynolds equation is:

$$\frac{n}{2n+1} \left( \frac{1}{2} \right)^n \left[ \frac{\partial}{\partial x} \left( \rho h^n \left( \frac{1}{\phi} \frac{\partial P}{\partial x} \right) \right) + \frac{\partial}{\partial y} \left( \rho h^n \left( \frac{1}{\phi} \frac{\partial P}{\partial y} \right) \right) \right] = \frac{\partial (\rho U h)}{\partial x}$$ \hspace{1cm} (2)

Among them, $P$ and $h$ represent the grease film pressure and thickness, $n$, $\rho$, $\phi$ and $U$ are the grease rheological index, density, plastic viscosity and suction speed. The boundary conditions can be defined as:

$$P(x, y) = P_{(x,y)} = \frac{\partial P(x, y)}{\partial x}$$ \hspace{1cm} (3)

$$P(x, y_e) = P(x, y_s) = 0$$ \hspace{1cm} (4)

$x_s$ and $x_e$ are the starting and ending coordinates of the lubrication calculation domain along the bearing circumferential direction (x direction), $y_s$ and $y_e$ represent the starting and ending coordinates of the lubrication calculation domain along the bearing width direction (y direction). In this paper, the acceleration between the rolling body and the raceway is pure rolling, and $\omega_i$, $\omega_r$, $\omega_c$, and $\omega_o$ are defined as the inner ring rotation angular velocity, rolling body rotation angular velocity, rolling body revolution angular velocity, and outer ring rotation angular velocity, where $\omega_i > \omega_o$, the grease suction speed $U$ between the largest rolling element and the outer race is expressed as:

$$U = \frac{1}{4} d_e \left[ \left( 1 + \frac{D_e}{d_e} \right) (\omega_i - \omega_o) + \frac{D_o}{d_o} \omega_o \right]$$ \hspace{1cm} (5)

The rotation angular velocity of the outer ring $\omega_o = \pi n_o / 30$, where $n_o$ is the rotation speed of the outer ring of the bearing. The expressions of the rolling body rotation angular velocity $\omega_r$ and the rolling body revolution angular velocity $\omega_c$ are:

$$\omega_r = \frac{(d_e + D_e)(d_e - D_e)(\omega_i - \omega_o)}{2d_e D_e}$$ \hspace{1cm} (6)

$$\omega_c = \frac{(d_e + D_e)\omega_o + (d_e + D_o)\omega_o}{2d_e D_e}$$ \hspace{1cm} (7)
The rotation angular velocity of the bearing inner ring $\omega_i = \pi n_i / 30$, where $n_i$ is the rotation speed of the bearing inner ring.

2.2. Grease thickness
The film thickness equation between the roller and the raceway in the rolling bearing is described as follows:\(^{(13)}\):

$$h_{(x,y)} = h_0 + \frac{x^2}{2R_x} + \frac{y^2}{2R_y} + v_{(x,y)}$$

(8)

Where $h_0$ is the central film thickness of the rigid body, $R_x$ and $R_y$ are the equivalent radius of curvature of the rolling element and the outer raceway along the bearing circumferential direction and the width direction, and $1/R_x = 1/R_{sx} + 1/R_{sw}$, $1/R_y = 1/R_{sy} + 1/R_{sw}$, $v_{(x,y)}$ is the total deformation of the roller and the raceway.

2.3. Viscosity and density
The viscosity uses the Roelands viscosity model\(^{(14)}\), expressed as:

$$\phi = \phi_0 \exp \left[ \ln \phi_0 + 9.67 \left( 1 + 5.1 \times 10^{-5} p \right)^z - 1 \right]$$

(9)

In this investigation, $\phi_0$ is the initial plastic viscosity of the grease, $Z = \alpha \left[ 5.1 \times 10^{-5} \left( \ln \phi_0 + 9.67 \right) \right]$, and $\alpha$ is the viscosity-pressure coefficient of the grease. The density in this project was expressed as:

$$\rho = \rho_0 \left[ 1 + \frac{0.6 \times 10^{-9} p}{1 + 1.7 \times 10^{-9} p} \right]$$

(10)

where the $\rho_0$ is the grease original density.

2.4. Pressure
The pressure distribution of the grease film between the roller and the raceway of the deep groove ball bearing should satisfy the load balance condition\(^{(15)}\), the load balance can be expressed as:

$$F = \int_{\Omega} P_{(x,y)} dx dy$$

(11)

Where, $F$ is the normal load between the largest roller (when the roller at azimuth angle $\theta = 0^\circ$) and the outer raceway, $\Omega$ represents the lubrication area, $dx$ and $dy$ indicate the grid spacing along the bearing circumferential direction (x direction) and width direction (y direction) in the lubrication calculation domain.

3. Experimental details
The deep groove ball bearing (SKF 16064MA) was used in the experiment, the basic parameters of the bearing are shown in Table 1. The deep groove ball was installed at the shoulder of the large gear shaft in the gearbox, the double row deep groove bearing was selected for the pinion shaft, the detailed experimental setup was shown in Figure 2. The lithium grease is used as the bearing grease. The performance parameters of the grease are shown in Table 2.

| Table 1. Basic performance index of test bearing. |
|-----------------------------------------------|
| performance index                             |
Test bearing type 16064MA
Limit speed 2000 r/min
Basic dynamic load rating 281kN
diameter of Rolling 12.3mm
Contact angle 90°
Number of Rolling 17

**Table 2.** grease properties.

| Type                   | 3# Kunlun   |
|------------------------|-------------|
| Base oil               | Lithium mineral oil |
| Base oil viscosity (η, 25°C) | 14.9 |
| Cone penetration (25°C, 0.1mm) | 253 |
| Dropping point         | 221         |
| Temperature range      | -30~180     |

In the test rig, two test bearings and two assistant bearings were carried in a group, which were worn together on a test shaft, totally 12 sets test bearings were used for the experiment. Normal load was used in the experiment to reduce the wear on the sample surface. In the paper, the normal load, shaft speed and working time were analyzed. The temperature of the lubricating oil was maintained at 40°C. In order to reduce the effect of surface wear, each balling bearing was used only once during the experiment. The test was carried out at room temperature (25°C), the samples were cleaned with ultrasonic waves and cleaning agents, then dried with an electric hair dryer, and then lubricated. The grease was Kunlun 3# aluminum-based grease. The shaft speed is set at 300rpm/min, and the load is 50N, 100N, 150N and 200N, the rolling bearing wear test completed at 500min and 1000min, the surface morphology of the raceway was tested after the experiment. The experimental condition was shown in Table 3.
| Type                        | Value          |
|-----------------------------|----------------|
| Nomal load F(N)             | 50, 100, 150, 200 |
| Shaft speed n(rpm/min)      | 300            |
| Testing time t (min)        | 500, 1000      |

4. Results and discussion

4.1. Micromorphology of rolling bearing raceway

The Contour GT three-dimensional optical interference profilometer was used to detect the surface morphology of the bearing raceway. The two-dimensional and three-dimensional surface topography of the specimen was observed in the experiment, the depth of the worn-out under different working conditions was obtained by computer image. It was found that the micro-topography would greatly change the grease thickness between the contact pairs, and further affected the internal stress distribution and surface wear behavior of the raceway.

Figure 3 was the real-time picture of the surface morphology of the test bearing before and after wear using a profilometer. It can be seen from the Figure 3 and Figure 4, the grease thickness was obvious deteriorated due to the load cycles, the nomal load greatly affected the grease thickness, the morphology changed faster and worsen as the nomal load increase, while the shaft speed accelerated the film generate, as the speed increases, the lubrication situation was improved.

![Figure 3. the grease film thickness on different nomal load.](image-url)
The contact load ratio of the rolling bearing contact pair under different loads and speeds was investigated, as shown in Figure 5. The contact load ratio mainly reflected the ratio of the external load borne by the rough peak to the total load during the operation of the contact pair. The greater contact load ratio, the load-bearing effect of the lubricating film reflect the poor lubrication state. As the relative motion speed of the contact pair decreases, the contact load ratio of the roller raceway pair of the rolling bearing increased sharply, indicated that the lubricating film was too thin to isolate the interaction surface. The lubricating film numerical analysis can be used to calculate the lubricating performance of rolling bearings in the state of starved state. The results showed that under the starved lubrication, the grease film between the contact pairs was very small and thick, which caused the pressure of the dual interface to rise, the rough peaks directly worn. The contact of this metal dual surface was easy to cause stress concentration and the micro-crack initiation, diffusion and bridging in the interface, which eventually lead to early fatigue damage, resulted in the corrosion and spalling on the contact surface, which ultimately affects the service safety of rolling bearing components.

4.2 Wear analysis
The microscope is used to observe the wear progression on the raceway surface, the first symptom of wear is observed before the test, as shown in the figure 6, where depicts the presence of silver
mark on the inner race. Over a period of time, excessive wear on inner race is noticed, abrasion originated from cutting wear particles, the wear particles generated during pitting and flaking mixed with the lubricating grease and remains in contacts of operating surfaces of bearing. It induced rapid wear progression on major portion of inner race and resulted in spalling. Figure 7 shows deep scratches, pitting and spalling on raceway surfaces. There are two possible reasons concluded for these defects; first one is due to the mixing of wear particles with grease and other one is due to load cycles. The development of wear on raceway surfaces due to above mentioned reasons are negligible; however, severe wear on the surfaces of roller and raceway can produce significant increase in starved lubrication. A small portion of raceway surface is found darkened due to discoloration, this discoloration resulted due to increase in temperature at local contact surfaces over 80 ° C. It indicates that during experiments, excess temperature was generated, grease service life and viscosity is reduced significantly, maybe about half of the actual, the lubricant fails to provide effective film thickness. This leads to direct metal to metal contact on contact surfaces thereby increase in temperature which results in wear propagation.

The figure 8 and figure 9 show the raceway surface of worn representatives micromorphology after 750 min and 1500 min experiment, the cross-section profiles were laser measured and presented in the direction Y. As expected, there are strip-shaped wear scars along the sliding direction distributed in the groove which is the grinding of the raceway surface by the abrasive debris between the friction pairs The generated furrows indicate that the wear mechanism is abrasive wear on the raceway on condition of starved lubrication. Meanwhile, it can also be seen from the groove that shear deformation occurs on the contact surface of the raceway and the roller. The roller can observe scaly wear surface, which infers that the ball under the test conditions- Adhesive wear also occurs when the bearing slides relative to the lack of oil. In summary, the form of the sliding oil wear of the raceway is mainly abrasive wear and adhesive wear. Moreover, comparing the three-dimensional morphology, it can be seen that the degree of wear at 1500 minutes is significantly more serious than the case of 750 minutes.
Figure 8. cross-section of the raceway on 750 min

Figure 9. cross-section of the raceway on 1500 min

Figure 10. morphology on the cross section of raceway

Figure 10 shows the morphology of the raceway under different loads. It indicates that the wear width and depth of the inner raceway increase with the load under the condition of starved oil. However, from the local wear morphology profiles the wear is slight and the wear morphology is relatively uniform. When the load increases, some furrows appear in the morphology. In the case of dry contact the heavy load becomes more severe, the abrasive particles aggravate the process, resulting in more irregular morphology. In the raceway bottom region, microscopic images of damage seem to prove the thesis of maximum tensions being caused by bearing work located at some distance from the interaction surface. In the regions of accumulated tension, within areas of macro-fractures, there is initiation and development of further micro-fractures. The main condition for increased durability of the raceway is the need for homogeneity of its lubrication.

5. Conclusions

In the present study, experiments have been conducted to access the interface of between the roller and raceway under starved lubrication, the experimental test-rig has been developed for the investigation, deep groove ball bearing has been used for the experiment and tested over a period of 1500 min. Experimental studies include observations of the micromorphology of raceway contact surface and the wear mechanism. Following conclusions are drawn from the experimental studies:

- The grease film thickness reduced, the central and minimum film deteriorated due to the load cycles, the nominal load greatly affected the grease thickness.
- The thickener film is gradually removed from the contact, grease starvation occurs, and the film thickness is subsequently reduced after several hundreds of working cycles, the central and minimum film thicknesses fall, regardless of the variation of the entraining velocity.
- The centre region of the raceway generated concave residual deformation, while the convex residual deformation occurred at the lubrication region’s edge and its surrounding region.
- With the same load and maximum entraining speed, the shorter the stroke length, the longer the actual life of the grease lubrication.
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