Lubricating Properties Research on Automobile Rolling Based on Numerical Calculation

Xiaohui Xia¹, Rui Yu², Weiqin Shi²

¹Changzhou College of Information Technology, Changzhou 213164, China
²Changzhou Vocational Institute of Engineering, Changzhou 213164, China

Abstract

The rolling bearing as one of the important parts of automobile transmission system, its lubricating property and fatigue life directly decides the service life of the automobile, in order to improve the service life of automobile transmission system, the automobile rolling bearing was regarded as the research object, the mathematical model of the automobile rolling bearing was established in the process of movement, the different external load, speed and other parameters on the influence of the rolling bearing lubrication characteristics were studied by the use of the numerical method. The research results show that with the automobile external load increase, the rolling bearing oil film pressure increased, while thickness decreases, it is not all rolling bearing oil film lubrication; with the increase of vehicle speed, the thickness of the rolling bearing with increased, while the oil film pressure decreases, the lubrication condition can reduce rolling bearing wear and invalidation such as agglutination; but as the automobile speed increasing, the rolling bearing working temperature increases sharply, its working condition is worsening, unfavorable to automobile rolling bearing life; which provide important reference for selection of rolling bearing load, speed to realize its good lubrication condition, and prolonging the service life of automobile rolling bearing.

Keywords: automobile rolling bearing, external load, speed, numerical calculation, lubrication characteristics

1. INTRODUCTION

The rolling bearing is widely used in the transmission mechanism of aviation, aerospace, automobile, machinery and so on due to its small start-up torque, simple structure and other advantages(Giovanetti et al., 2016). However, with the development of power machinery towards high speed and large scale, the capacity of rolling bearing is facing higher requirements (Cheng et al.,2012). There are many ways to improve the service life of rolling bearings, and one of the most simple and effective methods is reasonable lubrication (Ninno et al,2016). The oil film with a certain thickness will be formed between the rolling element and roller path in their operation process (Cardinale et al.,2016). When the full oil film lubrication is formed, the service life of the rolling bearing is about twice the specified value(Rafiee and Sadeghiazad,2016). Many researchers have made more in-depth study on the mechanics and dynamics of the rolling bearing, but there are relatively small researchers on its lubrication characteristics, and the lubrication characteristics of rolling bearings directly decide its agglutination and failure(Chairez, 2013). Because of large external load, high-speed rotation and other harsh working conditions of rolling bearings, its lubrication characteristics directly decide the life of the automobile (Elshafey et al., 2013). Therefore, the automobile rolling bearing is selected as the research object, and the more accurate multi-grid algorithm in the numerical calculation is used to calculate and analyze the influence of different external load, speed and other parameters on its lubrication characteristics, in order to provide theoretical basis for the lubrication and failure of automobile rolling bearings; and provide important reference for the improvement of service life of automobile rolling bearings at the same time.

2. NUMERICAL CALCULATION MODEL

The thickness of oil film between the rolling element and outer roller path is larger than that between the rolling element and inner roller path during the movement of the rolling bearing(Fan,2015). In this paper, the contact area between the rolling element and inner roller path in the automobile rolling bear is taken as the research object to study the lubrication characteristics of rolling bearing. The research model is shown in Figure 1(a). As can be seen in the literature, the elastic modulus is $E_1$ and $E_2$, and the Poisson’s ratio is $\mu_1$ and $\mu_2$ respectively.
The contact between equivalent elastic cylinders can be simplified to a contact between the elastic cylinder of the equivalent elastic modulus and rigid plane, and then the analytical model in figure 1(a) can be simplified to figure 1(b).

The dimensionless equation is characterized by the advantages of simplification, small number of variables contained, and the strong commonality without being limited by the dimensions, and the following equations are given in dimensionless form (Jian et al., 2015).

![Figure 1](image-url)

**Figure 1** Automobile rolling bearing: (a) Bearing, (b) The moving figure of Rolling bearing two surface

### 2.1 Reynolds

The one-dimensional Reynolds equation for non-Newton fluid full-film lubrication can be obtained by the force balance of any lubrication micro-element along the x axis (Costa and Baldo, 2015).

\[
\frac{\partial}{\partial x} \left[ \varepsilon \frac{\partial P}{\partial x} \right] = C_A \frac{\partial (\frac{\rho^* H}{\eta})}{\partial x}
\]

(1)

Each equivalent parameter in the equation is defined as:

\[
\varepsilon = \left( \frac{\rho}{\eta} \right)_c H^3 \quad C_A = \frac{3\pi^2 U_0}{4W^2} \quad \left( \frac{\rho}{\eta} \right)_c = \left( \frac{\rho_c}{\eta} \right) = 12 \left( \frac{\eta_c}{\rho_c} - \rho_c \right) \frac{\eta_0}{\rho_0}
\]

\[
\rho^* = \frac{\rho_0}{C_A U_0 \gamma_0} = \left[ \rho_0 \eta_0 (U_1 - U_2) + \rho_2 U_2 \right] \quad \frac{1}{\eta_c} = \frac{1}{H \eta_0} \int_0^H \frac{1}{\eta} dZ \quad \rho_c = \frac{P_0}{H} \int_0^H \rho dZ
\]

\[
\rho'_c = \frac{P_0}{H^3 \eta_0} \int_0^H \rho' \int_0^Z \frac{1}{\eta'} dZ' \quad \frac{1}{\eta_c} = \frac{1}{H^3 \eta_0} \int_0^H \frac{Z'}{\eta} dZ
\]

\[
\frac{1}{\eta} = \frac{1}{\eta_c} \left( \frac{\tau}{\tau_0} \right) \left( \frac{\tau_0}{\tau} \right)
\]

The boundary condition in equation (1) is:

\[
\begin{cases}
P(X_m, Z) = P(X_{out}, Z) = P(X, Z_{out}) = 0 \\
P(X, Z) \geq 0 (X_m < X < X_{out}, 0 < Z < Z_{out})
\end{cases}
\]

Dimensionless parameters are as follows: the coordinates along the direction of movement and the direction of the film thickness are \( x/b \), \( z/R \); the pressure rating \( sp = P/P_H \); the maximum Hertz contact stress is \( P_H = \frac{E}{4 \sqrt{\pi}} \); the dimensionless load parameter is \( W/w = E R \); where, \( E \) is the equivalent elastic modulus, and its value meets the following condition: \( \frac{1}{E} = \frac{1}{E_1} \left( \frac{1 - \mu_2}{E_2} + \frac{1 - \mu_1}{E_2} \right) \). In this equation, \( E_1, E_2, \mu_1 \) and \( \mu_2 \) are the elastic
modulus and Poisson’s ratio of the roller and inner ring material; the half width of contact is
\[ b = R \sqrt{\frac{6w}{\pi}}; \]
where \( R \) is the equivalent curvature radius, and \( R = \frac{R_1R_2}{R_1+R_2} \); in the equation, \( R_1 \) and \( R_2 \) are curvature radius of two surface contact points respectively; the dimensionless solid surface velocity parameter is
\[ U_1 = \frac{U_1}{E R}; \]
the dimensionless oil film velocity parameter is \( U_2 = \frac{U_2}{E R}; \)
the average velocity change parameter is \( C_u = \frac{u_0}{u}; \)
the dimensionless equivalent viscosity parameter is \( \eta^* = \frac{\eta}{\eta_0}; \)
\( \eta \) is the shear stress of oil film; \( \eta_0 \) is the characteristic shear stress of fluid; and \( u \) is the suction velocity, namely, the average velocity (Lua and Huang, 2015).

2.2 Film thickness equation

\[ H = H_0 + \frac{X^2}{2} - \frac{1}{2\pi} \int_{X_1}^{X_2} \bar{P}(S) \ln (X - S)^2 \, dS \]  

(2)

Where \( H \) is the parameter of dimensionless oil film thickness, \( H = \frac{hR}{b^2} \); \( H_0 \) is the dimensionless initial central film thickness.

2.3 Load equation

Under the condition of full oil film lubrication, the bearing load is completely borne by the oil film; the oil film pressure force of the entire contact area must be balanced with the external load \( w \), and the following equation can be obtained after the nondimensionalization (Ninno et al., 2016):

\[ \int_{X_1}^{X_2} \bar{P}(X) dX = \frac{\pi}{2} \]  

(3)

2.4 Viscosity equation

The viscosity equation can be expressed as the following through nondimensionalization:

\[ \bar{\eta} = \exp \left\{ \ln \eta_0 + 9.67 \right\} \left[ 1 + 5.1 \times 10^{-5} p u P \right]^{\varphi} \left( \frac{t \bar{T} - 138}{t_0 \bar{T} - 138} \right)^{\psi} - 1 \} \]  

(4)

In the equation, \( \bar{\eta} \) is the dimensionless viscosity, \( \bar{\eta} = \frac{\eta}{\eta_0}; \) \( \bar{T} \) is the dimensionless temperature, \( \bar{T} = t/t_0; \) \( t \) is the working temperature (K); \( t_0 \) is the ambient temperature (K); \( z_0 = \frac{a}{5.1 \times 10^{-5} P u (\ln \eta_0 + 9.67)}; \) \( s_0 = \frac{\beta (t_0 - 138)}{t_0 (\ln \eta_0 + 9.67)}; \)
where \( a \) and \( \beta \) are the viscosity-pressure and viscosity-temperature coefficient.

2.5 Density equation

\[ \bar{\rho} = 1 + 0.6 \times 10^{-9} p u P \frac{1}{1 + 1.7 \times 10^{-7} p u P} - 0.0007 (\bar{T} - 1) \]  

(5)

Where, \( \bar{\rho} \) is the dimensionless density of the lubricating oil, \( \bar{\rho} = \rho/ho_0; \) \( \rho_0 \) is the environmental density of lubricating oil (kg/m³).

2.6 Dimensionless friction equation
\[ F = \int_{X_1}^{X_2} \left( \frac{p_H b^2 \partial P}{R \partial X} + p_H \tau_0 \sinh c \right) dX \]  

(6)

Where, \( F \) is the dimensionless friction;

\[ c = \frac{(u_2 - u_1) F_x - \sqrt{(u_2 - u_1)^2 + F_1^2 - F_2^2} \cdot F_x}{F_1^2 - F_2^2} \]

\[ F_x = \frac{\tau_0 p_H b^2}{R \eta_0} \int_{0}^{\eta} \frac{1}{\eta} \cosh \left( \frac{b}{R \tau_0} \frac{\partial P}{\partial X} \right) dZ \]

\[ F_2 = \frac{\tau_0 p_H b^2}{R \eta_0} \int_{0}^{\eta} \frac{1}{\eta} \sinh \left( \frac{b}{R \tau_0} Z \cdot \frac{\partial P}{\partial X} \right) dZ \]

2.7 Dimensionless energy equation

\[ C_1 \frac{\partial T}{\partial Z} + C_2 \rho U \frac{\partial T}{\partial X} + C_3 U \frac{\partial P}{\partial T} \frac{\partial P}{\partial Z} + C_4 \tau = 0 \]

(7)

where, \( C_1 = -K \frac{T_R}{b^2} \); \( C_2 = \rho c T \); \( C_3 = \frac{ER \rho \mu}{\eta b} \); \( C_4 = -p_H \frac{ER^2}{\eta \mu} \).

2.8 Dimensionless thermal interface equation

For thermal-elastohydrodynamic lubrication, people pay attention to the temperature of the contact surface of the solid and the lubricating film rather than the temperature inside the solid (Cannistraro and Lorenzini, 2016). Therefore, in the calculation of thermal elastohydrodynamic lubrication, the thermal interface equation is usually used to directly solve the temperature distribution of the contact surface of the solid and oil film. Since the contact body is in the moving state, the temperature of the contact surface can be reduced to the heat conduction problem of semi-infinite body with a moving heat source, and its dimensionless thermal interface equation is:

\[ \left\{ \begin{array}{l}
T(X, 0) = \left[ \frac{R}{\sqrt{\pi \rho_c k_1 b}} \right]^{\frac{1}{2}} \int_{-\infty}^{S} \left. \frac{\partial T}{\partial Z} \right|_{X=S} ds + 1
\end{array} \right. \]

\[ \left\{ \begin{array}{l}
T(X, H) = \left[ \frac{R}{\sqrt{\pi \rho_c k_2 b}} \right]^{\frac{1}{2}} \int_{-\infty}^{S} \left. \frac{\partial T}{\partial Z} \right|_{X=S} ds + 1
\end{array} \right. \]

(8)

Where, \( \rho_1, \rho_2, c_1, c_2, k_1 \) and \( k_2 \) are the density, specific heat capacity and thermal conduction coefficient of the bearing inner race and roller materials.

3. NUMERICAL CALCULATION AND SOLUTION

According to the previous experience and combined with the numerical calculation model established above, \( X_{in} = 5 \) and \( X_{out} = 2 \) are taken as the starting point and ending point of the calculation area of the computational example in the numerical calculation model respectively to carry out the numerical analysis for above model. The numerical calculation model takes into account the temperature and other main influence factors (Mocanu et al., 2016). The oil film pressure in the model is numerically solved by the multi-grid method, and the oil thickness is solved by the multi-grid integration method, so as to obtain the complete numerical solution through the continuous iteration (Song and Chen, 2015). The grid is divided into five layers in the calculation process, with 145×1025 as the highest number of grid nodes. During the calculation, the judgment basis of convergence is set as that the both the relative errors of the pressure and load are less than \( 10^{-5} \), and the relative error of temperature is less than \( 10^{-6} \); the calculation and solution process is shown in Figure 2.
The lubricating oil used in automobile rolling bearings is HIV 1605, and its relevant lubricating parameters are: the dynamic viscosity is $\eta_0=0.068\text{Pa}\cdot\text{s}$; thermal conduction coefficient is $K=0.14\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$; the environmental density is $\rho_0=878\text{kg}\cdot\text{m}^{-3}$; the specific heat capacity is $c=2035\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$; the viscosity-pressure coefficient is $\alpha=2.24\times10^{-8}\text{Pa}^{-1}$; and the viscosity-temperature coefficient is $\beta=0.047\text{K}^{-1}$.

The relevant parameters of numerical calculation are: the bearing $d\timesD\timesB=220\times400\times108\text{mm}$; the effective length of roller $L=72\text{mm}$; the roller radius $R_2=22.5\text{mm}$; the contact radius of bearing inner race and roller $R_1=132.5\text{mm}$; the radial load of bearing $F_r=18675\text{N}$; the speed of bearing inner race $n=450\text{r/min}$; the temperature of contact area $T_0=328\text{K}$; the thermal conductivity of bearing inner race and roller material $K_1,2=65\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$; the density of bearing inner race and roller material $\rho_1,2=7800\text{kg/m}^3$; the specific heat capacity of bearing inner race and roller material $c_1,2=425\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$; and the elasticity modulus of bearing inner race and roller material $E_1,2=2.05\times10^11\text{Pa}$.

### 4. Analysis of Calculation Results

In this paper, the influence of different external loads, speeds and other parameters on automobile rolling bearing is studied, and the oil film thickness and contact pressure of its contact area are solved according to the multi-grid method in the numerical calculation; the following analysis results are obtained from the procedure calculation, and the ordinate and abscissa in the figure are all expressed in dimensionless numerical values in order to facilitate the result analysis.

Figure 3 and 4 are the distribution curves of the oil film thickness and oil film pressure of the contact area solved by the calculation of basic parameters. It can be seen from Figure 3, the automobile rolling bearing pressure distribution curve is almost the same with Hertz distribution curve, and the secondary peak of the oil film pressure of automobile rolling bearing under certain load and speed is not very obvious; Figure 4 shows that with the increase in pressure, the oil film thickness of the contact area of automobile rolling bearing is drastically reduced.
The pressure distribution and oil film shape of the contact area of the automobile rolling bearing obtained under the same working conditions as above are shown in Figure 5. It can be seen from Figure 5 that the oil film pressure of the automobile rolling bearing increases first and then decreases, the middle oil film pressure is the largest, and the pressure distribution result is consistent with the result of the two-dimensional analysis; the oil film shape is similar to the figure symmetric with the x-axis, similar to the results calculated previously.

Figure 5 Contact area: (a)Pressure distribution, (b) shape of oil film

Figure 6 and figure 7 are the result of oil film pressure and thickness distribution of bearing contact area under different loads on the premise of unchanged other conditions respectively. It can be seen from Figure 6 that with the increase of the external load of the automobile rolling bearing, the oil film pressure of the contact area of the rolling bearing obviously starts to appear secondary peak; at the same time, the secondary peak will increase along with the load, and then the maximum pressure will be followed by the sharp decrease; it also can be seen from Figure 6 that with the increase of external load, the secondary peak of the contact area of rolling bearing is gradually shifted toward the outlet direction. It can be seen from Figure 7 that the oil film thickness of the contact area of automobile rolling bearing is gradually thinned with the increase of external load; and the necking part at end gradually narrows with the increase of external load, that is, the proportion of the necking part in the whole contact area gradually decreases with the increase of external load; as a result, the increase of external load of the automobile will hinder the rolling bearing to achieve the full oil film lubrication, and in
order to improve the life of the automobile rolling bearing, the external load of the automobile shall be reduced appropriately to achieve full oil film lubrication.

Figure 6 Pressure distribution at different loads

Figure 7 Film thickness distribution at different loads

Figure 8 and 9 are the result of oil film pressure and thickness distribution of the contact area solved by changing the rotational speed of automobile rolling bearing when other conditions remain unchanged. As can be seen from Figure 8, with the increase of the automobile rotational speed, the variation of the pressure distribution curve of the automobile rolling bearing is more obvious, and its distribution gradually deviates from the Hertz distribution. When the rotational speed is small, the secondary peak is more obvious. The secondary peak disappear with the increase in the rotational speed, and the maximum oil film pressure is gradually moved toward the inlet area. It can be seen from Figure 9, with the increase in the automobile rotational speed, the oil film thickness of the contact area of automobile rolling bearing gradually increases, resulting in the formation of better lubrication conditions to prevent the wear, failure and other adverse effects of automobile rolling bearing due to the lack of lubrication; but because of the increase in automobile rotational speed, the operating temperature of automobile rolling bearing increases sharply, resulting in deterioration of the working conditions of automobile rolling bearings to speed up the deformation and damage of the rolling; Figure 9 also shows that as the automobile rotational speed increases, the proportion of the necking part of the automobile rolling bearing in the whole contact area gradually increases, that is, the necking part gradually become widened.

Figure 8 Pressure distribution at different speed
CONCLUSION

In this paper, the automobile bearing is used as the research object, and the model of automobile rolling bearing in the process of movement is established to analyze the influence of different automobile external loads, speed parameters on the thermal-elastohydrodynamic lubrication of automobile rolling bearing. The following conclusions are obtained by combining the analysis results:

(1) the increase of external load of automobile rolling bearing makes the oil film thickness smaller and the oil film pressure increases, and the reasonable reduction of bearing external load is the key to realize the full oil film lubrication of bearing and improve the service life of the bearing; with the increase of the rotational speed, the oil film pressure decreases and the oil film pressure decreases, which can prevent the wear and other failures caused by too small oil film thickness. However, the bearing temperature becomes larger with the increase in rotational speed, and further accelerates the deformation and damage of bearing;

(2) This study provides an important reference for the reasonable selection of the rotational speed of automobile rolling bearing, and improve the service life of bearing and the lubrication characteristics of automobile rolling bearing. However, due to the limited experimental conditions, the experimental analysis and verification of the automobile rolling bearing needs to be carried out under the mature conditions in the future, and the more in-depth study will be implemented for the influence of movement, dynamics and more factors of automobile rolling bearing on the lubrication characteristics.

ACKNOWLEDGMENTS

This work is supported by Top-notch Academic Programs Project of Jiangsu Higher Education Institutions(PPZY2015C235).

REFERENCES


