A method to calculate the life of sliding bearings based on fatigue and wear failure considering fluid lubrication

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Abstract. In this paper, a sliding bearing is taken as the research object to study the life simulation and calculation under conditions of a certain eccentricity, a radius clearance, inlet and outlet pressures, a radial load, and taking into account fatigue and wear failure modes. For a calculation case when the eccentricity is 0.6, the average radius gap is 0.05 mm, the inlet pressure is 2 MPa, the outlet pressure is 0.1 MPa, and the radial load is 5000 N. The calculated fatigue life is infinite and the wear life is $7.1 \times 10^4$ revolutions. This paper focuses on providing a method for quantitatively estimating the life of sliding bearings.

1. Introduction

Sliding bearing is one of the key parts of large shafting because of its high bearing capacity, impact resistance and reliable operation. It supports and reduces load friction, and its performance is directly related to the overall performance of the equipment [1]. Under ideal conditions, the bearing works in a completely oil-film state with very low wear. However, due to manufacturing and installation errors, mixed film lubrication and boundary lubrication of sliding bearings can occur when starting, stopping, extreme conditions, low speed standby, etc. As a result, the bearing wear increased and the failure rate increased, which had a certain impact on the reliable operation and working life of the sliding bearing. Therefore, it is necessary to study the life evaluation method of sliding bearing.

At present, the research on sliding bearings is mainly focused on lubrication characteristics, contact stress, health monitoring, etc. There is little research on life assessment. Among them, the research on lubrication characteristics is mainly about the solution method [2-3] and boundary conditions [4]. Studies on contact stress mainly focus on dynamic analysis [5-7], deformation and friction [8]. Researches on health monitoring mainly focus on wavelet analysis method [9], empirical mode decomposition method (EMD) [10], and support vector machine (SVM) [11], etc.

In this paper, a sliding bearing is taken as the research object to study the life simulation and calculation under conditions with a certain eccentricity, a radius clearance, inlet and outlet pressures, a radial load and so on, taking into account fatigue and wear failure modes.

2. Geometric model of sliding bearing

A simplified geometric model of sliding bearing and shaft is built using the 3d modeling software Inventor, as shown in Figure 1, including the shaft, the bearing bush, and the oil film between the shaft and the bearing bush, on which there is an oil inlet and an oil outlet.
In Figure 1, the inner diameter and outer diameter of the bearing bush are 90mm and 80mm, respectively, and the axial length is 80mm. Both the oil inlet and the oil outlet are through holes with a diameter of 2mm. The diameter of the shaft is 79.9 mm, which is not coaxial with the bearing bush. The eccentricity is 0.6 (the eccentricity is 0.03 mm/average radius clearance is 0.05 mm), and the axial length is also 80 mm. As shown in Figure 2, the narrow gap between the bearing bush and the shaft is oil film. Ideally, when the rotor rotates at high speed, the oil film in the sliding bearing is in the state of full-film lubrication due to the hydrodynamic pressure effect. The oil film bears the radial load transferred from the bearing bush and separates the bearing bush from the shaft to avoid friction and wear. Therefore, fluid lubrication modeling and analysis of oil film is an important basis for studying the life of sliding bearing.

**Figure 1.** Geometric model of sliding bearing and shaft.  
**Figure 2.** Schematic diagram of oil film position and shape (exaggerated display of oil film size).

3. Fluid lubrication model

3.1. Overview of solution methods

The Reynolds equation is usually used to describe the lubrication characteristics of fluids in a narrow space, such as the sliding bearing oil film. Reynolds equation is a special and simplified form of Navier-Stokes equation, which is famous in fluid mechanics. Although compared with Navier-Stokes equation, the computational amount can be simplified, it is still difficult to obtain the analytical solution for most cases in practice, and numerical solution is required. It is also very difficult for commercial computational fluid dynamics (CFD) software to solve the problem of sliding bearing lubrication. The size of oil film thickness is usually more than 3 orders of magnitude smaller than the size of the other two dimensions, resulting in the need for ultra-fine mesh, which brings great challenges to the calculation.

At present, with the improvement of computer performance, the time cost of using commercial CFD software to solve the fluid lubrication model of sliding bearing becomes acceptable. In this paper, Fluent software is used to solve the fluid lubrication model.

3.2. Mathematical model

Considering that the oil film liquid can be regarded as an incompressible fluid, the mathematical model describing the steady state flow characteristics of the oil film includes the continuity equation and the momentum conservation equation as shown in Formulas (1) and (2), respectively.

\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0
\]  

\[
\begin{align*}
\rho \left[ \frac{\partial (uu)}{\partial x} + \frac{\partial (uv)}{\partial y} + \frac{\partial (uw)}{\partial z} \right] &= \frac{\partial \tau_{xy}}{\partial y} + \frac{\partial \tau_{yz}}{\partial z} + \frac{\partial \tau_{zx}}{\partial x} \\
\rho \left[ \frac{\partial (vu)}{\partial x} + \frac{\partial (vv)}{\partial y} + \frac{\partial (vw)}{\partial z} \right] &= \frac{\partial \tau_{yx}}{\partial x} + \frac{\partial \tau_{zy}}{\partial z} + \frac{\partial \tau_{xz}}{\partial y} \\
\rho \left[ \frac{\partial (wu)}{\partial x} + \frac{\partial (wv)}{\partial y} + \frac{\partial (ww)}{\partial z} \right] &= \frac{\partial \tau_{zx}}{\partial z} + \frac{\partial \tau_{zy}}{\partial y} + \frac{\partial \tau_{zy}}{\partial x}
\end{align*}
\]
where, \( p \), \( \rho \) and \( \mu \) are respectively the pressure, density and viscosity of the oil film. \( u \), \( v \) and \( w \) are the flow rates of the oil film in three orthogonal directions in Cartesian coordinates. \( \tau_{xx} \), \( \tau_{xy} \), \( \tau_{xz} \), \( \tau_{yx} \), \( \tau_{yy} \), \( \tau_{yz} \) and \( \tau_{zz} \) are the viscous shear stress components.

### 3.3. Simulation model and boundary conditions

The following assumptions are made in the model: 1) Oil film fluids are regarded as incompressible three-dimensional stationary flows; 2) The inertial force of the oil film fluid is ignored [12]; 3) The oil film fluid has no relative sliding on the axial contact wall; 4) Reynolds \( \text{Re} = \frac{\rho U \delta}{\mu} \), is estimated to be much less than 2300 and calculated according to laminar flow. Where, \( \rho \) is the oil film density, \( U \) is the linear velocity at the journal, \( \delta \) is the average radius clearance, and \( \mu \) is the dynamic viscosity of the oil film.

As shown in Figure 3, the outer circumferential surface of the established fluid lubricating oil film model is in contact with the bearing bush, which is the static wall surface. The inner circumferential surface of the model is in contact with the axis. The position of two oil inlet holes is the inlet, while the axial two end faces are the outlet. As shown in Figure 4, there are about 43,500 nodes and 126,000 cells in total.

![Figure 3. Boundary condition diagram of oil film in fluid lubrication model.](image1)

![Figure 4. Mesh generation of oil film in fluid lubrication model.](image2)

Dynamic viscosity of oil film is 0.02 Pa·s, and the oil film density is 891 kg/m³. The solver adopts Fluent, coupled algorithm is applied, the pressure adopts second-order difference scheme, the momentum adopts second-order fractional algorithm, the relaxation factor is 0.25, and the number of iterations is 50.

### 4. Static model based on fluid-solid coupling

#### 4.1. Overview of solution methods

In life analysis, it is very important to obtain the stress data of sliding bearing, which is an important basis for fatigue life analysis and wear life analysis. As the oil film pressure is relatively high, it has a significant impact on the contact state of the bearing bush and the shaft and the stress in the contact area. Therefore, it is necessary to consider the oil film pressure data calculated by the fluid lubrication model and establish a static model considering the fluid-solid coupling.

In the static model, it is necessary to establish the contact pair between the bearing bush and the shaft to simulate the contact relation and calculate the contact stress.

A mathematical model was established to describe two surfaces of a pair of contact pairs with the following characteristics: 1) no penetration; 2) normal compression force and tangential friction force can be transferred; 3) normally, normal tensile force is not transferred, which means free analysis and mutual movement. Among them, the most important thing was to ensure that the two surfaces in the contact pair will not permeate each other. Several commonly used contact models are presented in the Workbench, including Augmented Lagrange, Pure Penalty, MPC, and Normal Lagrange. In addition, in order to avoid the influence of oil film on establishing a contact pair between the bearing shell and the shaft, oil film was not included in the model, and oil film pressure was applied to the bearing shell and the shaft as a boundary condition.
4.2. Simulation model and boundary conditions

As shown in Figure 5, the static modelling considering the fluid-solid coupling is completed by using Workbench. A contact pair is set between the bearing shell and the shaft. The type is bonded, and the contact model is controlled automatically by the program. The fluid-solid coupling boundary is set on the inner cylinder of the bearing shell and the outer cylinder of the shaft respectively to introduce the fluid pressure of the oil film at the wall surface.

The fixed boundary is set on the end face of the shaft to constrain the radial and axial movement of the shaft. The radial force is applied to the outer cylinder of the bearing shell to simulate the radial load of the bearing. The oil film is gridded by Workbench, as shown in Figure 6, with a total of about 64,500 nodes and 21,100 cells.

The bearing bush material is tin base bearing alloy, and the shaft material is structural steel. The physical parameters are shown in Table 1.

| Bush/tin base bearing alloy | Shaft/structural steel |
|----------------------------|------------------------|
| $E$ (Pa) | $\lambda$ | $\rho$ (kg/m$^3$) | $E$ (Pa) | $\lambda$ | $\rho$ (kg/m$^3$) |
| $4.793 \times 10^{10}$ | 0.2847 | $7.545 \times 10^3$ | $2 \times 10^{11}$ | 0.3 | $7.85 \times 10^3$ |

where, $E$ is young's modulus, $\lambda$ is Poisson’s ratio, and $\rho$ is density.
5. Fatigue life analysis model
Due to the problems of asymmetrical load bearing and assembly eccentricity, the sliding bearing often bears certain alternating load in the working state, which may lead to fatigue damage. Compared with rolling bearings, sliding bearings mainly have wear failure and fatigue failure is auxiliary, so high cycle fatigue failure theory is applied to analyse. Fatigue Tool module of ANSYS Workbench was used to calculate Fatigue by means of mean stress correction model and Miner cumulative Fatigue summation rule.

In Figure 7, due to the lack of high-cycle fatigue test data of bearing shell and tin base bearing alloy, the data of structural steel brought in Workbench material warehouse were adopted for fitting. In the form of Basquin, the S-N curve equation was obtained as follows:

$$\sigma_a = 7 \times 10^9 N^{-0.34}$$

(3)

where $\sigma_a$ is stress amplitude and $N$ is fatigue life (number of cycles).

In the Fatigue Tool module, the bearing shell mainly bears compressive stress, and the asymmetric cyclic load bearing unilateral compressive stress is selected. According to the influence of average stress on fatigue strength and life, and the applicable range of different limit stress lines, Goodman average stress correction equation and s-n curve were used to calculate fatigue life.

The Goodman equation is used to convert the cyclic stress of general form into symmetric cyclic stress with equal lifetime, and the expression is:

$$\sigma_a = \sigma_{N(1)} \left(1 - \frac{\sigma_m}{\sigma_a} \right)$$

(4)

where $\sigma_a$ is the stress amplitude, $\sigma_{N(1)}$ is the fatigue strength with a life of $N$ under symmetric cycles, $\sigma_m$ is the average stress, and $\sigma_b$ is the tensile strength. For sliding bearing, the bearing bush is subjected to an alternating load for each rotation, so the fatigue life $N$ is revolution.

6. Wear life analysis model
Wear failure is the main failure mode of sliding bearing. When the wear reaches a certain level, the noise and vibration of sliding bearing increase significantly, and the bearing is considered to be invalid. The factors influencing the wear are very complex, involving materials, working conditions, environmental stress, etc. Therefore, the wear problem is not perfect in theory and practice in the field of tribology. In recent decades, hundreds of formulas for calculating various kinds of wear have been published, involving at least more than 100 parameters related to materials, mechanics, thermophysics and chemistry in the wear process. However, the applicability of these formulas is greatly limited, and it is difficult to establish a unified quantitative relationship for complex and changeable wear behaviours.

For sliding bearings, the classical Archard wear model can be used to describe the wear behaviour of sliding bearings on the premise that only load, speed and material are considered. The wear volume generated by unit displacement is expressed as:

$$\frac{dV}{dS} = K_s \frac{W}{3\sigma_y}$$

(5)

where, $V$ is the wear volume, $S$ is the displacement, $K_s$ is the wear constant, $W$ is the normal load, and $\sigma_y$ is the compressive yield strength of the material. $K_s$ is not a completely constant value, which has certain correlation with load, sliding speed, temperature, etc., and generally needs to be determined by friction and wear test. According to Literature [13], the wear constants of cemented carbide and hardened steel, which are close to the sliding bearing materials, are $5 \times 10^{-5}$ at dry friction, load of 4000 N and sliding speed of 1.8m/s. For sliding bearings, the normal load $W$ can be considered as the normal contact force between the bush and the shaft.

Under the condition of constant speed and load, the wear volume can be expressed as:

$$V = K_s \frac{W}{3\sigma_y} S$$

(6)
For the maximum allowable wear volume of given failure criterion, the maximum displacement can be obtained as follows:

$$S_{\text{max}} = \frac{3\sigma_s V_{\text{max}}}{K_w}$$  \hspace{0.5cm} (7)

Furthermore, it can be deduced that wear life \(N\) (revolution) is:

$$N = \frac{S_{\text{max}}}{\pi d} = \frac{3\sigma_s V_{\text{max}}}{\pi d K_w}$$  \hspace{0.5cm} (8)

where \(d\) is the inner diameter of the bearing shell. When the maximum allowable wear volume of the failure criterion is further changed to the maximum allowable wear thickness, Equation (8) can be expressed as:

$$N = \frac{3\sigma_s}{\pi d K_w} \pi d H_{\text{max}} = \frac{3\sigma_s H_{\text{max}}}{K_w}$$  \hspace{0.5cm} (9)

where, \(l\) is the axial length of the sliding bearing.

Figure 8. Oil film pressure cloud diagram (36000 rpm, inlet pressure 2 MPa, and outlet pressure 0.1 MPa).

7. Results and discussion

7.1. Fluid lubrication result
Under the conditions of 36,000 rpm, inlet pressure of 2 MPa and outlet pressure of 0.1 MPa, the typical oil film pressure cloud diagram is shown in Figure 8. It can be seen that the maximum pressure reaches 17.07 MPa, and the minimum pressure is negative pressure, reaching 6.17 MPa. In the process of shaft rotation, the oil film in a certain area will experience the alternating process of positive pressure and negative pressure repeatedly. This pressure acts on the bearing bush, which may cause the fatigue damage of the bearing bush material.

7.2. Results of static analysis
Under the conditions of 36,000 RPM, inlet pressure of 2 MPa, outlet pressure of 0.1 MPa and radial load of 5000 N, and considering the fluid-solid coupling effect caused by the oil film pressure, the typical deformation cloud diagram, stress cloud diagram and the contact between the bearing shell and the shaft are calculated, as shown in Figure 9, Figure 10 and Figure 11, respectively. It can be seen that the maximum stress of the bearing bush is 4.4 MPa, which is caused by the oil film pressure and the contact pressure between the bearing bush and the shaft. The maximum contact stress between the bearing bush and the shaft is 17.9 MPa, while the normal contact force from the shaft borne by the
bearing bush is 1682 N, which is significantly less than the radial load of 5000 N. It can be seen that a large proportion of the radial load is borne by the oil film, and the anti-wear effect of the oil film can also be seen.

![Figure 9](image)

**Figure 9.** Bearing bush and shaft deformation cloud diagram (36000 rpm, inlet pressure 2MPa, and outlet pressure 0.1 MPa, 5000 N).

![Figure 10](image)

(a) Bearing bush  
(b) Shaft

**Figure 10.** Bearing bush and shaft stress cloud diagram.
According to the results of statics analysis, the Fatigue Tool module of ANSYS Workbench was used to calculate the Fatigue life by applying the average stress correction model and Miner cumulative Fatigue summation rule based on the high-cycle Fatigue failure theory. The solution setting of Fatigue Tool module is shown in Figure 12, and the typical Fatigue life analysis results are shown in Figure 13. It should be pointed out that the fatigue life is $1 \times 10^6$ because the s-n input is only up to $1 \times 10^6$. Due to the lack of Babbitt data, considering that the compressive yield strength of structural steel is 250 MPa, the maximum stress of bearing shell and shaft is 4.4 MPa and 14.7 MPa, which is far less than the compressive yield strength. Therefore, it can be considered that the sliding bearing has infinite life.

**Figure 11.** Calculation results of contact between bearing pad and shaft.

7.3. Fatigue life analysis results

According to the results of statics analysis, the Fatigue Tool module of ANSYS Workbench was used to calculate the Fatigue life by applying the average stress correction model and Miner cumulative Fatigue summation rule based on the high-cycle Fatigue failure theory. The solution setting of Fatigue Tool module is shown in Figure 12, and the typical Fatigue life analysis results are shown in Figure 13. It should be pointed out that the fatigue life is $1 \times 10^6$ because the s-n input is only up to $1 \times 10^6$. Due to the lack of Babbitt data, considering that the compressive yield strength of structural steel is 250 MPa, the maximum stress of bearing shell and shaft is 4.4 MPa and 14.7 MPa, which is far less than the compressive yield strength. Therefore, it can be considered that the sliding bearing has infinite life.
when only fatigue damage is considered. And according to the existing experience, fatigue failure is not the main failure mode of sliding bearing.

7.4. Wear life analysis results

Under the conditions of 36,000 rpm, inlet pressure of 2 MPa, outlet pressure of 0.1 MPa and radial load of 5000 N, the normal contact force between the bearing bush and the shaft is calculated as $W=1682$ N, and the wear constant $K_s$ is $5 \times 10^{-5}$. Due to the lacking of Babbitt metal material data, the compressive yield strength $\sigma_y$ is 250 MPa according to the structural steel, the axial length $l$ of the sliding bearing is 0.08 m, and the maximum allowable wear thickness is assumed to be 0.0001 m. According to Equation (9), the wear life (revolutions) can be estimated:

$$N = \frac{3 \times 2.5 \times 10^9 \times 0.08 \times 0.0001}{5 \times 10^{-5} \times 1682} = 7.1 \times 10^4$$

(10)

It should be pointed out that this is a kind of wear life calculation result under high speed, heavy load and large eccentricity, and the wear life of conventional sliding bearing should be much longer than this result.

![Fatigue Tool module setup](image1)

**Figure 12.** Solution setup of Fatigue Tool module.

![Fatigue life analysis results](image2)

**Figure 13.** Fatigue life analysis results.
8. Conclusions
In this paper, a sliding bearing is taken as the research object, and the sliding bearing is studied under the conditions of eccentricity of 0.6, average radius gap of 0.05 mm, inlet pressure of 2 MPa, outlet pressure of 0.1 MPa, and radial load of 5000 N. Considering the two failure modes of fatigue and wear under oil film lubrication, the calculation results of fatigue life and wear life are given by means of simulation. The fatigue life is infinite and the wear life is $7.1 \times 10^4$ revolutions in the case study. This paper focuses on providing a method for quantitatively estimating the life of sliding bearings. The sliding bearing geometric model, lubrication state, working conditions, materials, environmental stress and other aspects are different from the actual situation, leading to some errors in the calculation results. The further study will conduct test to validate the model.

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