Numerical Investigation of the Heat-fluid Characteristic inside High-speed Angular Contact Ball Bearing Lubricated with Grease

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1. INTRODUCTION

Rolling bearing is a critical component of many modern engines and machinery, it can reduce the friction and ensure rotary accuracy [1, 2]. Lubrication, as an effective way to reduce friction, is one of the important issues in rolling bearing. With an increase in bearing speed and load, lubrication performance has become more and more demanding. According to available findings, grease lubrication has been widely used in high-speed rolling bearing because of its good sealing performance, wide application and convenient operation [3-5]. However, it is difficult for a grease-lubricated rolling bearing to dissipate heat by friction because of the...
limited flow ability of grease, high temperature rise often results in premature failure of rolling bearing [6]. Hence, it is very important to investigate the thermal behavior and fluid characteristics of greases in operating rolling bearings.

Considerable efforts have been devoted to investigate the work mechanism [7-9] and rheological behavior [10-12] of grease lubrication in high-speed rolling bearing. Thermal characteristics of rolling bearing has also attracted much attention due to the industrial demands for high speed and long working life. Ma Fangbo et al. [13] provided a calculation model for heat generation rate in a grease-lubricated spherical roller bearing (SRB) by adopting the local heat source analysis approach, the heat generation rates of roller-raceway contact, cage and inner ring land contact, roller and cage pocket contact, and power loss of roller churning were calculated. Wurzbach et al. [14] investigated the thermal behavior, flow characteristics of the non-Newtonian greases, and changes in grease properties in operating rolling-element bearings. Neurouth et al. [15] analyzed the heat transfer inside a grease lubricated thrust ball bearing, two models were developed using the thermal network method. Based on generalized Ohm's law, Siyuan et al. [16] developed a thermal network model for double-row tapered roller bearing lubricated with grease, bearing temperature at different speeds, grease filling ratios, and roller large end radius was investigated. Fangbo et al. [17] developed a transient thermal model for a grease-lubricated spherical roller bearings-shaft-bearing housing system using a thermal network method, the effect of rotating speed, radial load and grease filling ratio on bearing temperature rise was analyzed. Xue et al. [18] conducted the temperature performance tests of high-speed sealed angular-contact ball bearings using different grease filling rate, the temperature performance of bearing with different grease filling rate was compared and analyzed under different working conditions.

The studies mentioned above investigated the bearing thermal characteristics and fluid properties of grease. However, fluid flow and heat transfer are not two independent processes [19, 20], the flow of fluid contributes to heat transfer, the temperature change of fluid will affect its viscosity, and then affects its flow in turn; so, the flow and heat transfer of fluid interact with each other [21, 22]. But few studies have been done on the coupling relationship between them and didn’t consider the heat transfer between grease and bearing components, the exact behavior of greases in operating rolling bearing needs to be well understood. Therefore, in this paper a simulation model for angular contact ball bearing was established with CFD software Fluent, based on this model, the distribution and flow of grease, heat transfer, and temperature field inside the bearing chamber were analyzed using the heat-fluid-solid coupling method. A bearing test rig was designed and manufactured, bearing temperature measurement was conducted to verify the validity and accuracy of numerical simulation.

2. TEST RIG

In order to investigate the thermal behavior of high-speed angular contact ball bearing, some measurement of bearing temperature was conducted on self-made test rig. As shown in Figure 1, the test rig is mainly composed of a test section and a motorized spindle unit. The test section is driven by the motorized spindle through a flexible coupling. The highest speed and rated power of the motorized spindle are 36000rpm and 5 kW, respectively. More detail of the tested bearings assembly is shown in Figure 2. The two tested bearings are B71909 angular contact ball bearings lubricated with grease; their major specifications are listed in Table 1. The tested bearings were press fitted onto the rotation shaft with back-to-back configuration, and the rotation shaft was supported by two pairs of ball bearings. Bearing loads were applied on tested bearing housing which installed between two tested bearings.

In experiments, motorized spindle rotates the inner ring of test bearings through the main shaft. The main shaft is connected with motorized spindle by a coupling. Special attention should be given to the coupling; design of flexible coupling was aimed to isolate test bearings.
### TABLE 1. The parameters of tested bearing B71909

| Parameter                              | Value      |
|----------------------------------------|------------|
| Bore diameter                          | 45 mm      |
| Outer diameter                         | 68 mm      |
| Width                                  | 12 mm      |
| Initial contact angular                | 15 deg     |
| Number of rolling elements             | 20         |
| Ring material                          | Steel      |
| Density of rings [g/cm³]               | 7.8        |
| Heat capacity of rings [J/kg·K]        | 460        |
| Thermal conductivity of rings [W/m²·K] | 30         |
| Ball material                          | Si₃N₄      |
| Density of ball [g/cm³]                | 3.2        |
| Heat capacity of ball [J/kg·K]         | 800        |
| Thermal conductivity of ball [W/m²·K]  | 20         |

from the vibration of motorized spindle as much as possible. The two parts of coupling are not directly contacted each other and connected through Nealon rope, so the torsional impact of motorized spindle cannot be transmitted to test bearings, this structure ensures the stability of test bearings.

In order to get the temperature data of test bearings, an infrared temperature sensor was fixed on a self-made bracket to monitor the temperature of the bearing inner ring, the bracket is easy to adjust sensor position according to the experimental requirements, as shown in Figure 3.

### 3. NUMERICAL SIMULATION

#### 3.1 Thermal Analysis Model

Due to the symmetrical structure of the tested bearings assembly, its half structure was modeled in order to save computing cost. Inside the chamber of tested bearings lubricated with grease, the heat transfers between the fluid and the solid is involved, however, it is difficult to be determined. So, the heat-fluid-solid coupling method is used in this numerical simulation, this method is often used to solve the complex heat transfer problem, it can convert the complex outer boundary conditions between the fluid and the solid into a relatively simple inner boundary condition, and make the simulation more approach to practical condition. As shown in Figure 4, the simulation model of tested bearings assembly includes solid domain and fluid domain.

The structure of bearing chamber is very complicated. Therefore, in order to make finite element meshing easy, the bearing cavity is divided into multiple fluid domains. The domain among the balls is very irregular and meshed using tetrahedral elements. The domain along both sides of the balls is relatively regular and meshed using hexagonal elements. These fluid domains are connected with each other through an interface. The 3D grid model of fluid domain in bearing chamber is shown in detail in Figure 5. The final number of elements is 3215677, and the mesh quality is above 0.6.

#### 3.2 Boundary Conditions

##### 3.2.1 Heat Generation

During bearing operation, bearing friction represents an energy loss and results in
temperature rise. The friction heat generation of the tested bearing can be obtained by the following equation [23]:

\[ H = 1.047 \times 10^{-4} Mn \]  

(1)

where \( H \) is the friction heat generation of tested bearing (\( w \)), \( n \) is bearing rotation speed (rpm), \( M \) is bearing frictional torque (N-mm). The frictional torque \( M \) consists of two components stated as follows:

\[ M = M_f + M_v \]

(2)

where \( M_f \) is bearing frictional torque due to load, and \( M_v \) is bearing frictional torque due to lubricant. \( M_v \) can be approximated by the following equation:

\[ M_v = f_1 \rho d_m \]

(3)

where \( f_1 \) is a factor depending upon bearing type and load, \( \rho \) is bearing load (N), \( d_m \) is bearing mean diameter (mm). \( M_v \) can be calculated by the following equation:

\[ M_v = 10^{-7} f_0 (v_n n)^{2/3} d_m^{1/3} \quad \text{if} \quad n \geq 2000 \]

(4)

\[ M_v = 160 \times 10^{-9} f_e d_m \quad \text{if} \quad n \geq 2000 \]

(5)

where \( f_0 \) is a factor depending upon bearing type and lubrication method, \( v_0 \) is the kinematic viscosity of lubricant (mm²/s).

In numerical calculation it’s assumed that the bearing friction heat generation is divided into two equal parts, one half was applied to bearing rings and the other was applied to rolling elements in the form of heat generating rate [24].

3.2.2. Heat Transfer

According to the working conditions of the tested bearings assembly, convective heat transfer occurs on the outer surface of bearing housing and main shaft. The convection form of bearing housing is natural convection and the convection heat transfer coefficients is assumed to be 9.7 W/(m²K) according to the practical experiences [25]. The convective heat transfer coefficients \( h_0 \) of the main shaft can be obtained by the following equations.

\[ N_x = 0.133 R_e^{0.8} P_r^{0.9} \]

(6)

\[ R_e = \omega d^2 / v \]

(7)

\[ h_0 = \frac{N_x \lambda}{d} \]

(8)

where \( N_x \) is the Nusselt number, \( R_e \) and \( P_r \) are the Reynolds number and the Prandlt number, respectively, \( \omega \) is inner ring speed, \( v \) is the kinematic viscosity of air, \( d \) is the diameter of main shaft.

Inside bearing chamber heat transfer among grease, air and bearing components is a heat-fluid-solid coupling process and cannot be specified in advance. So, under the FLUENT platform, in numerical calculation the contacting surfaces between fluids and solid were set as default coupling-surface, and the heat transfer was calculated automatically. Finally, the second-order upside-style discrete momentum equation and turbulence equation are used. The pressure term is discretized in the PRESTO! (pressure staggering option) format, and the phase volume fraction is discretized in the geometric reconstruction format, and then the Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm is used. Solve the discrete algebraic equation and converge when the residual value drops below 10⁻³.

3.3. Solution and Parameters Set

Heat-fluid-solid coupling is an interaction process among fluid domain, solid domain and temperature field. So, in order to obtain the flow field and temperature field of bearing, it is necessary to connect them. In the heat-fluid-solid coupling model of tested bearing, heat transfers among temperature field, fluid domain and solid domain through the fluid-solid interface, the interface method was used to connect the solid domain with the fluid domain, and the standard wall function is used to deal with the flow boundary layer and the heat transfer boundary layer at the fluid-solid coupling interface, which ensures the continuity of temperature and heat flux. In order to obtain the grease distribution inside bearing chamber, the interface of grease air two-phase flow was captured by the VOF method which is specifically used to solve the flow interface position of two immiscible fluids. The VOF method is from Europe Derived from Euler method, it is based on observation points, rather than following a fluid particle for research. In the flow field of bearing grease lubrication, the boundary of the two-phase flow is judged by solving the volume fraction of the two phases at the observation point. The turbulence inside bearing chamber was simulated by the RNG k-ε model which is suitable for analyzing turbulent motion in complex regions.

Moreover, the reference pressure was set as atmospheric pressure, and the air was selected incompressible gas model, non-equilibrium wall function was selected and solved by pressure velocity coupling equation. In the numerical simulation, the control variable method was used to simulate the flow field of the bearing chamber. The specific parameters of grease are shown in Table 2 [9].

4. RESULTS AND DISCUSSION

4.1. Grease Distribution inside Bearing Chamber

In the process of bearing operation, the bearing speed has a very important influence on the distribution of grease. When grease filling ratio is 0.3, under different rotation speed the grease distribution inside bearing chamber is
shown in Figure 6. It can be seen that the grease distribution was very inhomogeneous inside bearing chamber, grease was mainly distributed over the both sides of the rolling element along outer raceway, a small amount of grease was adhered on the inner side of the cage, grease adhered on the surface of rolling elements and inner raceway was very little due to centrifugal effect. It’s also noted that the grease distribution inside bearing chamber become more inhomogeneous with the increase of bearing rotation speed, more grease is adhered on the outer raceway, the volume fraction of the grease between inner raceway and cage was accordingly decreased. It is perhaps a further finding that at high rotation speed bearing outer raceway is in full lubrication, meanwhile, bearing inner raceway is in starved lubrication.

4. 2. Flow Velocity of Grease inside Bearing Chamber The flow velocity of grease inside the bearing chamber is shown in Figure 7. It is shown that grease adhered on the outer raceway was in viscous state, its flow velocity was very low; however, the flow velocity of grease adhered on inner ring and cage was relatively high, grease between cage and rolling elements has the highest flow speed. It’s also found that, with the increase of bearing speed, the flow velocity of grease between inner raceway and cage increases obviously; however, an increase in bearing speed has very little effect on flow velocity of grease adhered on outer raceway.

4. 3. Heat transfer Coefficient on inner Surface of Bearing Chamber In the working process of the bearing, heat is an important factor affecting the performance of the bearing. Heat will dilute the grease and reduce the lubrication efficiency of the bearing. Figure 8 shows heat transfer coefficient on the interface between the fluid domain and the solid domain inside bearing chamber at different speeds. It’s shown that, heat transfer coefficient was high on the area of outer raceway surface where more grease was adhered on; heat transfer coefficient was also high on rolling elements surface which frequently contact with raceway, cage and grease; heat transfer coefficient was the lowest on inner raceway where little grease was adhered while its velocity was high, which was due to the fact that air convective heat transfer inside bearing chamber was

### TABLE 2. The specific parameters of the grease

| Grease          | Lithium base grease |
|-----------------|---------------------|
| The consistency of grease | NLGI 3              |
| The rheological model of grease | Herschel-Bulkley flow model |
| Viscosity coefficient $k$ | 20.5                |
| Rheological index $n$ | 0.71                |
| Yield stress $\tau_0$ | 1076                |
| Density [kg/m$^3$] | 872                 |
| grease filling ratio | 0.3                |
| Heat capacity [J/kg·K] | 2000                |
| Thermal conductivity[W/m$^2$·K] | 0.14               |

![Figure 6](image1.png)  
(a) $n=6000$r/min  
(b) $n=12000$r/min  
(c) $n=20000$r/min  

**Figure 6.** Grease distribution inside bearing chamber at different rotation speed

![Figure 7](image2.png)  
(a) $n=6000$r/min  
(b) $n=12000$r/min  
(c) $n=20000$r/min  

**Figure 7.** The flow velocity distribution of grease inside bearing chamber at different speeds
very small; heat transfer coefficient was high on the area of inner and outer raceway surface where rolling elements pass. It also can be seen that with the increase of bearing speed heat transfer coefficient was accordingly increased on surface of outer raceway and rolling elements, which was due to the temperature rise of bearing components.

It can be concluded that inside bearing chamber conduction heat transfer is dominant in which grease plays a key role, the convective heat transfer of air and grease is insignificant.

4.4 Temperature Rise of Bearing

The temperature distribution of tested bearing is shown in Figure 9. It can be seen that bearing temperature field was nonuniform. The temperature of rolling elements was the highest, its heat mainly transfers to outer raceway through grease between rolling elements and outer raceway; the heat transfer condition of inner ring was poor, the temperature of inner ring was also high and slightly lower than that of rolling elements; outer ring has an effective heat transfer way, its heat can conduct to bearing housing and then transfer to ambient air, so the temperature of outer ring was the lowest. With an increase in rotational speed bearing temperature was increased, the temperature of grease adhered on outer raceway was also increased, which indicated that more grease involves in conduction heat transfer.

4.5 Verification

In order to verify the validity and accuracy of numerical simulation, bearing temperature rise was measured on a self-made test rig. In the experiment, bearing axial load is 200N, and radial load is 100N, the tested bearing was started to run from zero speed to 9000 rpm, after running for 300 seconds at 9000 rpm, it was turned off. The measured data and simulation data of bearing temperature are shown in Figure 10. It’s indicated that the simulation results were in good agreement with the experimental results, the residual error was below 8%, and the maximum residual error happened after bearing stop rotating. When bearing stops rotating, the natural convection of bearing housing plays a dominant role in bearing temperature, and the natural convection heat transfer coefficients come from experiences, this value 9.7 W/(m²·K) is not very accurate. This is one reason of residual error. The other reason is nonlinear effects of
bearing temperature rise. Regardless of all these effects, numerical simulation showed good validity and accuracy.

5. CONCLUSION

The thermal-fluid characteristics of angular contact ball bearing lubricated grease was analyzed using thermal-fluid-solid coupling method. It was concluded that, grease distribution inside bearing chamber is very inhomogeneous, most of grease was distributed over the both sides of the rolling element along outer raceway, grease adhered on the surface of rolling elements, cage, and inner ring is very little, which becomes more inhomogeneous with the increase of bearing speed; grease adhered on the outer ring was in viscous state; however, it plays a dominant role in bearing heat transfer, convective heat transfer of air and grease was insignificant; affected by heat transfer condition the temperature rise of bearing components was obviously different, the temperature rise of rolling elements was the highest, the temperature of inner ring was slightly lower than that of rolling elements; outer ring has the lowest temperature.

The results provide some reference for lubrication design and thermal analysis of high-speed angular contact ball bearing. The method in the paper is also applicable to the analysis of heat-fluid behaviors of oil-air or oil lubricated rolling bearings. In this study bearing temperature experiment was conducted on self-made test rig and verified the validity and accuracy of numerical simulation, grease flow experiment can be considered in the future work.

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