Thermal Analysis of Cylindrical Roller Bearing Considering Thermo-Mechanical Coupling

Baogang Wen 1, Jiayu Wang 1, Meiling Wang 2* and Xu Zhang 1

1 School of Mechanical Engineering and Automation, Dalian Polytechnic University, Dalian, 116034, PR China
2 College of Locomotive and Rolling stock Engineering, Dalian Jiaotong University, Dalian, 116028, PR China
* Email: meilingcc@163.com

Abstract. Thermal characteristic is important performance of the bearing, which reflects the friction characteristics of bearing. Thermal characteristics of bearing will change along with internal behavior when the bearing fails. There is a very complex coupling relationship. It is an important implication for the thermal analysis cylindrical roller bearing considering thermo-mechanical coupling. A model of heat generation for cylindrical roller bearing considering thermo-mechanical coupling is established based on the method of quasi-static and thermal network of bearing to solve the complex thermo-mechanical coupling of bearing. And then, the variation of complex thermal-mechanical characteristics such as friction torque, contact force and temperature for cylindrical roller bearing with wear failure is analyzed. The results show that the heat generation characteristics of the bearing are strongly coupled with the internal contact force and clearance. And the decrease of bearing clearance and the increment of contact force and friction torque, even rotor binding with the increasing of wear degree.

1. Introduction

Thermal characteristic is important performance of the bearing, which reflects the friction characteristics of bearing. Thermal characteristics of bearing will change along with internal behaviour when the bearing fails. There is a very complex coupling relationship. It is an important implication for the thermal analysis cylindrical roller bearing considering thermo-mechanical coupling.

The basis of bearing thermal characteristics analysis is heat generation calculation. Palmgren [1] proposed empirical formulas to calculate the overall load friction torque and viscous friction torque of rolling bearings based on the experimental results. Astridge and Smith improved Palmgren's calculation method, and put forward the calculation formula of overall power loss (calorific value) of high-speed cylindrical roller bearing on the basis of test [2]. However, the overall method cannot give the specific location of the power loss necessary to predict the temperature of bearing parts. Harris [3] established a complete mechanical analysis model of rolling bearing through mathematical theoretical derivation and mathematical theoretical calculation, and proposed to separately calculate the local heat generation between each contact element by local calculation method. In reference [4-7], the local heat source of rolling bearing is calculated, and the calculation model of bearing local friction power consumption is obtained. Temperature rise and temperature distribution are important indexes for thermal characteristic analysis of rolling bearings. The thermal network method with high efficiency and accuracy is regarded as an important means of rolling bearing temperature calculation.
reference [8-12], the thermal analysis model of bearing is constructed by thermal network method, and
the power loss and bearing temperature field of rolling bearing are studied.

Although many scholars have made many research achievements in rolling bearing temperature
calculation modelling and influencing factor analysis. However, these studies are only aimed at the
temperature calculation of rolling bearing under complex working conditions [13-15] and structural
changes [16-19]. The thermo-mechanical coupling mechanism and influence relationship of bearing
fault heat generation process the temperature rise model has not been established, and the relationship
among internal clearance, which needs to be carried out in depth.

A model of heat generation for cylindrical roller bearing considering thermo-mechanical coupling
is established, based on the method of quasi-static and thermal network of bearing to solve the
complex thermo-mechanical coupling of bearing. And then, the variation of complex thermal-
mechanical characteristics such as friction torque, contact force and temperature for cylindrical roller
bearing with wear failure are analyzed. It provides a method and theory for the analysis of bearing
temperature and thermo-mechanical characteristics.

2. Model of Cylindrical Roller Bearing Temperature Rise

2.1. Model of Cylindrical Roller Bearing Heat Generation

The heat generation of bearing is related to friction surfaces of the heat generation rate which
including the sliding friction between roller and raceway, the sliding friction between roller end face
and rib, roller flow resistance and elastic hysteresis. The elastic hysteresis loss caused by rolling and
the heat generation rate of sliding friction between roller end face and rib are relatively small, so they
can be ignored. In this paper, the calculation of friction heat generation rate between bearing parts is
based on elastohydrodynamic lubrication (EHL) contact, and the friction characteristics between parts
under the conditions of lubrication failure and oil cut-off are not considered.

2.1.1. Friction Heat Generation Rate of Cylindrical Roller Bearing

The total friction heating rate of cylindrical roller bearing is calculated as follows:

\[ H = H_1 + H_2 + H_3 + H_4 \]

Where \( H_1, H_2, H_3 \) and \( H_4 \) are friction heating rate which are sliding friction between roller and
raceway, oil mixing friction between roller, sliding friction between cage and guide surface of inner
ring and sliding friction between roller and cage pocket.

The calculation formula of each parameter is as follows:

\[ H_1 = \mu Q \Delta V \] (2)

\[ H_2 = 0.5F_d d_m \omega_w \] (3)

\[ H_3 = 0.5d_m F_{CL} (\omega_e - \omega_m) \] (4)

\[ H_4 = \sum_{j=1}^{Z} 0.5 \omega_v F_m \mu \] (5)

Where \( \mu \) is the friction coefficient between roller and raceway, \( Q \) is the contact load between roller
and raceway, \( \Delta V \) is the relative sliding speed of roller and raceway, \( F_d = 1/8C_d \rho D_w l_c (d_m \omega_m) \) is
the viscous resistance of lubricating oil to roller, in which \( C_d \) is the viscous resistance coefficient
of lubricating oil, \( \rho \) is the density of the lubricant in the bearing chamber, \( l_c \) is the effective length
of the roller, \( d_m \) is the pitch diameter of the bearing, and \( \omega_m \) is the angular velocity of roller revolution,
\[ F_{CL} = \frac{\eta_0 \pi w_{cR} d_{cR} (\omega_i - \omega_m)}{1 - d_{cR}/d_{iR}} \]
is the lubricant viscous friction between the cage and the guide surface of the inner ring, in which \( \eta_0 \) is the viscosity of the lubricating oil under atmospheric pressure at 20 °C, \( w_{cR} \) is the total width of the guide surface of the cage, \( d_{cR} \) is the diameter of the guide surface of the cage and \( d_{iR} \) is the diameter of the guide surface of the inner ring, \( Z \) is the number of rollers, \( \omega_{bj} \) is the roller rotation angular velocity and \( F_{mj} \) is the force between the roller and the cage.

2.1.2. Friction Heat Generation Rate of Cylindrical Roller Bearing

Bearing wear can lead to increased surface roughness, and the wear rate considering temperature is calculated as follows [20]:

\[ W = 1.9877 \times P^{1.503} \times e^{-332/T} \] (6)

Where \( P \) and \( T \) are the pressure and temperature of the contact area, respectively. Bearing wear will deteriorate the lubrication state, cause the oil film to be unstable, and cause the change of the surface friction state. The friction state is related to the surface roughness when the oil film thickness is the same. Oil film lubrication parameters are as follows:

\[ \lambda = \frac{h_{\text{min}}}{\sqrt{\sigma_1^2 + \sigma_2^2}} \] (7)

Where \( \sigma_1 \) and \( \sigma_2 \) are the root mean square deviation of two surface roughness respectively, the average roughness of the surface is usually calculated by arithmetic mean deviation. The calculation formula is expressed as \( \sigma = 1.25Ra \), \( h_{\text{min}} \) is the minimum oil film thickness [21].

When \( \lambda \geq 3 \) becomes full elastohydrodynamic lubrication, the oil film completely separates the surface. When \( \lambda \) is close to 1 or less than 1, it cannot be lubricated normally, and the friction and wear are serious. and the bearing cannot be lubricated normally.

![Figure 1. Variation of friction coefficient with time.](image)

The change of the friction coefficient \( \mu \) is caused by the change of the lubrication state. It changes between 0.002 and 0.02 under full elastohydrodynamic lubrication, and between 0.1 and 0.2 under dry friction state.
2.1.3. Calculation of Bearing Load Distribution Considering the Variation of Thermal Characteristic Clearance

The movement and force relationship between a single roller and the inner and outer rings of the bearing is shown in figure 2. Assuming that the center of the roller is fixed and the inner ring and the outer ring rotate in opposite directions for a rotating bearing.

![Kinematic and Stress Relationships](image)

**Figure 2.** Relationship between motion and force of single roller.

Where \( \omega_i, \omega_{bj}, \) and \( \omega_m \) are the angular speeds of inner race, roller rotation and holder considering slip respectively, \( \mu \) is the friction coefficient between the contact surfaces, \( Q_{ij} \) and \( Q_{ej} \) are the contact load between the roller and the inner and outer race raceways, \( P_{ij} \) and \( P_{ej} \) are the tangential dynamic pressure of lubricating oil acting on the roller, \( T_{ij} \) and \( T_{ej} \) are the tangential friction force exerted by the oil film on the roller.

A set of force balance equations between rollers, inner and outer rings and holder when the bearing is running at a constant speed, which can be expressed as follows:

\[
\begin{align*}
P_{ij} + T_{ij} - P_{ej} - T_{ej} &\pm F_{mj} = 0 \\
Q_{ij} + F_c - Q_{ej} &\pm \mu F_{mj} = 0 \\
T_{ij} + T_{ej} - \mu F_{mj} & = 0 \\
\sum_{j=1}^{Z} F_{mj} & = 0 \\
F_r - \sum_{j=1}^{Z} Q_j \cos \psi_j & = 0
\end{align*}
\]

Where \( F_r \) is the radial load of the bearing, \( F_c = 0.5m_b\omega_m^2 \) is centrifugal force, \( m_b \) is the mass of a single roller, \( P_{ij} \) and \( P_{ej} \) are the tangential dynamic pressure on the roller when the lubricating oil between the inner and outer rings is squeezed.

The calculation formula of contact load between each roller and inner and outer rings is as follows:

\[
\begin{align*}
Q_{ij} &= K_g (\delta_y + 0.13h_y)^{10/9} \\
Q_{ej} &= K_g (\delta_y + 0.13h_y)^{10/9}
\end{align*}
\]
Where $K_{ij}$ and $K_{ej}$ are the load deformation coefficients of rollers and inner and outer rings respectively, $\delta_{ij}$ and $\delta_{ej}$ are the contact deformation between the roller and the inner and outer rings respectively, $h_{ij}$ and $h_{ej}$ are the minimum oil film thickness between the roller and the inner and outer race raceways respectively.

The total deformation of the contact between the raceway and the roller at the angular position of the load zone $\psi_j$ when considering the influence of the minimum oil film thickness on the contact deformation, the formula is calculated as follows:

$$\delta_j = \delta_i \cos(\psi_j) - \frac{P_d}{2} + h_i + h_j = (\delta_{\text{max}} + \frac{P_d}{2} - h_i - h_e) \times \cos(\psi_j) - \frac{P_d}{2} + h_i + h_j$$

(10)

Where $\delta_j = \delta_i + \delta_e$, $\delta_{\text{max}} = \delta_{i1} + \delta_{e1}$, $\delta_r$ is the total radial displacement of the bearing.

The deformation coordination equation of the bearing zone is obtained by combining equations (13) and (14).

$$[(\frac{Q_{ii}}{K_n})^{0.9} + (\frac{Q_{ei}}{K_n})^{0.9} + \frac{1}{2} P_d - 1.13(h_i + h_e)] = [(\frac{Q_{i1}}{K_n})^{0.9} + (\frac{Q_{e1}}{K_n})^{0.9} + \frac{1}{2} P_d - 1.13(h_i + h_e)] \cos(\psi_j)$$

(11)

Where $h_i$ and $h_e$ are the minimum oil film thickness between the roller and the inner and outer raceways [20], considering the shear thinning non-Newtonian fluid and thermal effect in the elastohydrodynamic lubrication oil film thickness.

Bearing clearance $P_d$ affected by thermal expansion can be expressed as follows:

$$P_d = u^0_r - \Delta d_i^T - \Delta d_e^T - 2\Delta D_w + \Delta_r$$

(12)

Where $u^0_r$ is the original radial clearance, $\Delta D_w$ is the increase of roller diameter caused by temperature change, $\Delta r$ is the increase of clearance caused by ferrule expansion due to temperature change, $\Delta d_i^T$ and $\Delta d_e^T$ are the diameter changes of inner and outer race respectively caused by interference fit when considering temperature change.

The increase of inner raceway diameter is calculated as follows:

$$\Delta d_i^T = \frac{2(l_i + \Delta l_i)(\frac{d_i}{d})}{[(\frac{d_i}{d})^2 - 1] \left[ \frac{d_i}{d} \left( \frac{d_i}{d} \right)^2 + v_i \right] + \frac{E_i}{E_2} \left( \frac{d_i}{d} \right)^2 - 1}$$

(13)

The reduction of outer raceway diameter is calculated as follows:

$$\Delta d_e^T = \frac{2(l_e + \Delta l_e)(\frac{D}{d_e})}{[(\frac{D}{d_e})^2 - 1] \left[ \frac{D}{d_e} \left( \frac{D}{d_e} \right)^2 + v_k \right] + \frac{E_i}{E_4} \left( \frac{D}{d_e} \right)^2 - 1}$$

(14)
Interference amount $I_i$ is changed when the shaft and inner ring are thermally expanded, its formula is calculated as follows:

$$\Delta I_i = \alpha d(T_i - T_a) - \alpha d(T_i - T_s)$$  \hspace{1cm} (15)

Similarly, the increment of interference $I_e$ between bearing pedestal and outer ring after thermal expansion is calculated as follows:

$$\Delta I_e = \alpha D(T_e - T_a) - \alpha D(T_e - T_h)$$  \hspace{1cm} (16)

Where $I_i$ and $I_e$ are the interference of inner ring / shaft and outer ring / bearing seat respectively in the initial state, $d$ is the inner diameter of hollow shaft in the initial state, $E_1 \ldots E_4$ and $v_1 \ldots v_4$ are the elastic modulus and Poisson's ratio of inner ring, shaft, outer ring and bearing seat respectively, $T_a$, $T_i$, $T_s$ and $T_h$ are the reference temperature, the temperature of inner ring, shaft and bearing seat respectively and $\alpha$ is the linear expansion coefficient.

The increase of clearance caused by ferrule expansion due to temperature change is calculated as follows:

$$\Delta_r = \alpha d_e(T_e - T_a) - \alpha d_i(T_i - T_s)$$  \hspace{1cm} (17)

The increase in roller diameter due to thermal expansion is calculated as follows:

$$\Delta D_w = \alpha D_w(T_w - T_u)$$  \hspace{1cm} (18)

Simultaneous equations (13) - (18) can calculate the clearance considering the bearing installation tightness and thermal effect.

2.2. Model of Transient Temperature of Bearing Base on Thermal Network
The heat generated by bearing friction is transferred through heat conduction, heat convection and heat radiation. Generally, the effect of thermal radiation is very small and can be ignored when the temperature is lower than 200\degree C. Therefore, the analysis of heat exchange mainly includes heat conduction and heat convection. The heat dissipation network model of bearing system is established, as shown in figure 3.

The heat of bearing friction is transferred through heat conduction, heat convection and heat radiation. Thermal radiation has little effect on heat transfer, which can be ignored when the temperature is lower than 200\degree C. Therefore, heat conduction and heat convection are the two main parts of heat exchange analysis. The thermal network model of bearing system is established, as shown in figure 3.
Figure 3. Thermal resistance transmission diagram of various components of bearing system.

The bearing system is divided into 17 thermal nodes, ignoring the sealing and edge chauffeuring structures, as shown in figure 4. The bearing body has 8 nodes, 3 nodes at different shaft sections, and 4 nodes, including inlet lubricating oil, bearing cavity lubricating oil, outlet lubricating oil and external environment. The detailed description of node division is shown in table 1.

Figure 4. Thermal network node division.

Table 1. Thermal node division description.

| Node number | Position                              |
|-------------|---------------------------------------|
| 1, 14       | Bearing pedestal                      |
| 2, 15, 16   | Outer ring                            |
| 3           | Roller                                |
| 4, 17       | Inner ring                            |
| 5, 6, 7     | Axis                                  |
| 8           | Contact between outer ring and roller |
| 9           | Contact between inner ring and roller |
| 10, 12      | Inlet oil and outlet oil              |
| 11          | Oil in bearing cavity                 |
| 13          | Air                                   |
The linear equations of the relationship between heat source, thermal resistance and temperature of the bearing are established according to literature [23]. Ambient temperature and oil supply temperature are regarded as constants, and the relevant parameters are programmed with MATLAB.

2.3. Calculation Process

The calculation flow is shown in figure 5.

![Calculation flow of bearing temperature considering thermal solid coupling effect](image)

Figure 5. Calculation flow of bearing temperature considering thermal solid coupling effect.

The transient temperature calculation model is established to calculate the transient temperature of each node of the system considering the bearing structure and load characteristics. The multi-level and coupling analysis of structural mechanics, heat generation, heat conduction and heat convection are carried out.

Taking the ambient temperature as the initial value, first set the time step, and perform a quasi-static analysis of the bearing according to the boundary conditions and working conditions to obtain the motion and mechanical relationship between the components. Then calculate the heat generation rate, conduction thermal resistance and convection thermal resistance of the system. Finally, the transient equations are solved to obtain the temperature of the bearing system. According to the change of lubricating oil temperature in the bearing cavity, the lubrication parameters are corrected, and the structural size and load distribution of the system are corrected by calculating the thermal deformation of each component and the bearing clearance. The updated data is used as the initial condition for the next time step iteration. When the deformation of the last two iterations is less than the set value, the calculation ends.

3. Thermal Characteristics Analysis of Cylindrical Roller Bearing With Roughness Deterioration Caused By Wear

The research object of this paper is N209 cylindrical roller bearing, whose inner race has double side ribs, outer race has no side rib and the holder is guided by inner race.

Sliding friction occurs between the guide surface of inner race and the holder during bearing running.
The material of bearing inner ring, outer ring and roller is GCr15, and that of cage is brass. Bearing structure size is shown in table 2, the main material parameters are shown in table 3 and operating conditions are shown in table 4.

Table 2. N209 Cylindrical Roller Bearing System Structure Dimensions.

| Structural parameters       | Symbol | Numerical value |
|----------------------------|--------|-----------------|
| Internal diameter(mm)      | d      | 45              |
| Inner raceway diameter(mm) | d_i    | 55              |
| Outer raceway diameter(mm) | d_e    | 75              |
| External diameter(mm)      | D      | 85              |
| Pitch diameter(mm)         | d_m    | 65              |
| Roller diameter(mm)        | D_w    | 10              |
| Roller length(mm)          | l_e    | 11.4            |
| Number of rollers (-)      | Z      | 17              |

Table 3. Material parameters of bearing system (20°C-120°C).

| Name                        | Parameter                                      |
|-----------------------------|-----------------------------------------------|
| Material Science            | GCr15                                         |
| Elastic modulus(MPa)        | 2.19×10^5                                     |
| Poisson's ratio(-)          | 0.3                                           |
| Coefficient of linear expansion(1/℃) | 12.03×10^{-6}                              |
| Thermal conductivity(W/m°C^-1) | 45                                         |
| Density(Kg/m^3)             | 7.8×10^3                                      |
| Specific heat(J/Kg°C^-1)    | 450                                           |

Table 4. N209 Cylindrical Roller Bearing Condition.

| Working condition          | Numerical value |
|----------------------------|-----------------|
| Axial load(N)              | 0               |
| Radial load(N)             | 1000            |
| Ambient temperature(℃)     | 20              |
| Lubricating oil temperature(℃) | 20               |
| Speed(r/min)               | 6000            |

Figure 6 shows the effect of bearing wear on the temperature of inner and outer rings. It can be seen that the temperature of the inner and outer rings of the bearing increases continuously, and the temperature of the inner ring is always higher than that of the inner ring with the increase of bearing wear; The surface roughness increases with wear, and then the lubrication state of the bearing is changed, resulting in a large amount of frictional heat inside the bearing, and ultimately increasing the temperature. The thermal resistance of the bearing inner ring is greater than that of the outer ring, so the temperature of the inner ring is always higher than that of the outer ring.
Figure 6. Bearing temperature change.

Figure 7 shows the working radial clearance of $30 \mu m$, the effect of roughness on radial clearance. It can be seen that the bearing clearance decreases with the increase of bearing wear. The thermal expansion of the outer ring is less than the sum of the thermal expansion of the inner ring and the roller when the bearing temperature increases, resulting in the reduction of the clearance.

Figure 7. Variation of bearing working clearance.

The change of friction torque under bearing wear is shown in figure 8. It can be seen from the figure that the total friction torque of the bearing increases with the increase of wear, among which the friction torque of roller and ferrule increases the most, and there is no significant change in the friction torque of roller oil mixing, the sliding friction torque of cage and guide surface, and the sliding friction torque of roller and pocket.

Figure 8 shows the variation of friction torque considering bearing wear. It can be seen that the total friction torque of the bearing increases with the increase of wear degree, among which the friction torque between roller and raceway the largest increase, and there is no obvious change in roller oil mixing friction torque, sliding friction torque between cage and guide surface, and sliding friction torque between roller and pocket. This is due to the increase in roughness and the deterioration of the friction state between the roller and the raceway, resulting in an increase in the friction torque between the roller and the raceway.
Figure 8. Variation of bearing friction moment under wear failure.

Figure 9 shows the change in contact force between each rolling element of the bearing and the raceway when wear is considered. It can be seen that the contact force between the roller and the raceway increases with the increase of roughness.

Figure 9. Change of contact force between bearing roller and raceway consider wear.
With the increasing wear degree of the bearing, the working temperature, total friction torque and internal contact stress of the bearing are increasing, and the working clearance is decreasing.

In combination with the above analysis, it can be seen that, the temperature, total friction moment and internal contact stress of bearings increase continuously and the working clearance decreases continuously with the increasing wear of bearings. The wear of the bearing will cause the friction coefficient between the raceway and the roller contact surface to increase, which will increase the total friction moment. The increase of the total friction moment will increase the working temperature of the bearing. The increase of the temperature of the bearing will decrease the clearance, and the decrease of the clearance will increase the contact stress, which in turn will increase the friction moment. Thermo-mechanical coupling within the bearing eventually leads to bearing failure.

4. Conclusions

A thermal characteristic analysis model of cylindrical roller bearing consider the action of thermo-mechanical coupling is proposed to analyse the friction, clearance and contact mechanical characteristics of the bearing under the heating condition. The conclusions are as follows:

- There is a strong coupling relationship between heat, clearance and contact force inside the roller body.
- With the increase of wear degree, the working temperature gradually increases, clearance decreases or negative clearance is caused by thermal collision, and then contact force between inner rolling body and inner and outer races increases, friction moment increases, and bearing temperature shows a rising trend.

Acknowledgment

This work was financially supported by the National Natural Science Foundation of China (Grant No: 51905069) and the Department of Education of Liaoning province (No.:LJKZ0536) the Natural Science Foundation of Liaoning Province (2019-ZD-0088) and the High-level Talents of Dalian City (No.: 2018RQ18)

References

[1] Palmgren A. 1959 Ball and roller bearing engineering [J] Philadelphia: SKF Industries Inc
[2] Astridge D. 1973 Smith C. Heat generation in high-speed cylindrical roller bearings [C] Proceedings of the Inst Mech Eng Elasto-hydrodynamic Lubrication 1972 Symposium, Leeds 83-94
[3] Harris T A, Mindel M H. 1973 Rolling element bearing dynamics [J] Wear 23 (3) 311-37
[4] Harris T A, Kotzalas M N. 2006 Advanced concepts of bearing technology: rolling bearing analysis [M]. CRC press
[5] Jin C, Wu B, Hu Y. 2012 Heat generation modeling of ball bearing based on internal load distribution [J] Tribology International 45 (1) 8-15
[6] Moorhy R S, Raja V P. 2014 An improved analytical model for prediction of heat generation in angular contact ball bearing [J] Arabian Journal for Science and Engineering 39 (11) 8111-9
[7] Nelias D, Bercea I, Mitu N. 2003 Analysis of double-row tapered roller bearings, Part II—results: prediction of fatigue life and heat dissipation [J] Tribology transactions 46(2) 240-7
[8] Jiang X Q, Ma J J and Zhao L C. 2000 Thermal Analysis of High Speed Precision Angular Contact Ball Bearings [J] Bearing (08) 1-4+45
[9] Wang Y Z, Huang B and Zhao K, et al. 2019 Thermal analysis of flexible bearing based on thermal network method and finite element method [J] Lubrication and Sealing 44 (01) 42-6+51
[10] Mizuta K, Inoue T, Takahashi Y, et al. 2003 Heat transfer characteristics between inner and outer rings of an angular ball bearing [J] Heat Transfer—Asian Research Co-sponsored by the Society of Chemical Engineers of Japan and the Heat Transfer Division of ASME 32 (1) 42-57
[11] Pouly F, Changenet C, Ville F, et al. 2010 Investigations on the power losses and thermal behaviour of rolling element bearings [J] Proceedings of the Institution of Mechanical Engineers, Part J: Journal of Engineering Tribology 224 (9) 925-33

[12] Zheng D, Chen W. 2017 Thermal performances on angular contact ball bearing of high-speed spindle considering structural constraints under oil-air lubrication [J] Tribology International 109 593-601

[13] Lu F, Wang T, Zhao Z Q. 2020 THERMAL CHARACTERISTICS OF BEARINGS WITH INTEGRATED EHL THEORY AND CFD METHOD [J] Aerodynamics 35 (06) 1204-11

[14] Mi W, Yan K, Wu W W, et al. 2015 Analysis of Transient Thermal Characteristics of Spindle-Bearing System Considering Thermal-Deformation Coupling [J] Journal of Xi’an Jiaotong University 49 (08) 52-7

[15] P Z, Liu Z X, Gao W J, et al. 2016 Research on Heat Generation Characteristics of Reverse Cylindrical Roller Bearing [J] Propulsion Technology 37 (12) 2329-35

[16] Hao X, Li N, Yu C X, et al. Analysis of Displacement and Stress Characteristics of Cylindrical Roller Inner and Outer Rings Considering Thermal Effect [J] Journal of Mechanical Engineering 1-10

[17] Wang L W, Yu J B. 2021 Research on thermal and mechanical characteristics of high-speed angular contact ball bearing [J] Machine design 38 (01) 1-7

[18] Yu J, Li S S, Yuan W, et al. 2018 Dynamic characteristics of spindle bearings considering thermal deformation [J] Aerodynamics 33 (02) 477-86

[19] Zhang C, Tian J X, Guo D, et al. Thermal characteristics of Grease-Lubricated high-speed angular-contact ball bearings considering thermal expansion [J] Journal of Tsinghua University (Natural Science Edition) 1-11

[20] Li B L, Liao Q M, Jiang Q Y, et al. 2004 Temperature Study on Wear Simulation of Line Contact Parts [J] Journal of Dalian Railway Institute (02) 13-6

[21] Wen B, Ren H, Dang P, et al. 2018 Measurement and calculation of oil film thickness in a ball bearing [J] Industrial Lubrication and Tribology

[22] Deng S E, Qun J Y. 2008 Design principle of rolling bearing [M] Beijing Standards Press of China

[23] Wang Y S, Liu Z, Zhu H F. 2011 Analysis of Temperature Field of Shaft Bearing [J] Journal of Mechanical Engineering 47 (17) 84-91