Applied Thermal Engineering 131 (2018) 328–339

Contents lists available at ScienceDirect

Applied Thermal Engineering

journal homepage: www.elsevier.com/locate/apthermeng

Research Paper

# An optimized thermal network model to estimate thermal performances on a pair of angular contact ball bearings under oil-air lubrication

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## HIGHLIGHTS

- Ball loads with centrifugal force, gyroscopic moment and thermal expansion.
- Structure and assembly constraints effects on bearing temperature.
- An optimized nodes planning scheme for angular contact ball bearings.
- A more precise prediction model with less data overhead.

## ARTICLE INFO

*Article history:*
Received 28 August 2017
Revised 16 November 2017
Accepted 3 December 2017
Available online 5 December 2017

*Keywords:*
Angular contact ball bearing
Thermal resistance network
Temperature rise

## ABSTRACT

To estimate thermal performances of angular contact ball bearings is essential for operating precision and service life of angular contact ball bearings. However, a simple and accurate model estimating the bearing temperature is needed but absent so far. Considering the centrifugal force, gyroscopic moment and thermal expansion, ball loads equilibrium model was first established to analyze the bearing loads. The radial and axial heat transfer, especially the effect of structural constraints on bearings temperature was well characterized as well as the bearing mounting arrangement, and then an enhanced node planning scheme was proposed. With their surroundings considered, an optimized thermal grid model for a pair of front bearings mounted in the high-speed spindle was developed by using this scheme. Next Newton-Raphson method was employed to obtain the numerical solution by *Matlab*. The bearing temperature variation, moreover, was tested to validate the developed model. The results indicate that both series of results agree well. Compared with the presented models, the deviation between the calculation results and the test values is reduced to less than 9%. As a result, the bearing temperature rise can be better forecasted, which may be beneficial to improve bearing operating accuracy as well as bearing service life.



## 1. Introduction

Ball and roller bearings, together called rolling bearings, are commonly used to support various kinds of loads while permitting rotational and translatory motions of a shaft or slider. Angular-contact ball bearings designed to support combined radial and thrust loads are the most popular bearing type employed in various rotating machinery, especially high-speed devices because of a low friction coefficient, simple structure, high operation precision, low cost, etc.

Friction of any magnitude retards motion and results in energy loss. Unfortunately, it is difficult for an operating angular-contact ball bearing to dissipate friction heat because of the inherent geometric and surrounding complexity. Too much heat accumulating in the bearing will cause its temperature rising, which will have a significant impact on the overall performance of a bearing system [1].

The overall temperature level of an angular contact bearing depends on many factors, among which are: operating speeds, applied load, lubricant, bearing mounting arrangement, housing design, operational environment, etc. [2]. Conventional approaches to study bearing temperature are by experiment and the models based on the statistical characterization of the measured temperature are built. Palmgren [3] fitted an empirical formula to calculate the friction torque; Sud et al. [4] experimentally investigated the thermal performance of preloaded angular contact ball bearings lubricated with oil; Gerstenberger et al. [5] measured the temperature distribution of inner/outer ring of a grease-lubricated tapered roller bearing working in a low temperature

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https://doi.org/10.1016/j.applthermaleng.2017.12.019
1359-4311/

## Nomenclature

| | |
|---|---|
| $A$ | contact area of a contact region ($mm^2$) |
| $A_r^*$ | dimensionless real contact area |
| $a$ | semi-major width of contact area (mm) |
| $b$ | semi-minor width of contact area (mm) |
| $C_p$ | specific heat (J/kg·K) |
| $C_s$ | bearing rating static load (N) |
| $D_b$ | ball diameter (mm) |
| $D_i$ | spindle inner diameter (mm) |
| $D_s$ | spindle diameter (mm) |
| $d_{ext}$ | outer diameter (mm) |
| $d_{int}$ | inside diameter (mm) |
| $d_m$ | pitch diameter (mm) |
| $E$ | Young's modulus of elasticity (Pa) |
| $F_{cent}$ | centrifugal force of ball (N) |
| $F$ | applied load (N) |
| $F_0$ | static bearing load (N) |
| $f_o$, $f_i$ | groove curvature coefficient |
| $f_0$, $f_1$ | bearing factor |
| $G$ | heat transfer matrix |
| $g_r$ | radial clearance (mm) |
| $h_{cont}$ | contact conductance coefficient (W/($mm^2$·K)) |
| $K$ | load-deflection parameter (N /$mm^{1.5}$) |
| $k_D$ | thermal conductivity (W/(mm·K)) |
| $L$ | natural length (mm) |
| $L_g$ | thickness of the void space (mm) |
| $l$ | rows number |
| $l_{oe}$, $l_{oi}$ | initial center lengths (mm) |
| $M$ | Torque (N·m) |
| $N_u$ | Nusselt number |
| $n$ | rotation speed (r/min) |
| $P_r$ | Prandtl number |
| $P_s$ | equivalent static radial load (N) |
| $P_1$ | specified load (N) |
| $Q$ | normal contact load (N) |
| $Q_g$ | heat source matrix |
| $Q_H$ | Power loss (W) |
| $q_{oilair}$ | flow of oil-air ($mm^3$/s) |
| $q_H$ | heat dissipating capacity (W) |
| $R$ | thermal resistance (W/K) |
| $R_e$ | Reynolds number |
| $r_{ci}$ | groove radius of inner ring (mm) |
| $r_{in}$ | inside radius of rings (mm) |
| $T$ | node temperature (°C) |
| $u_{cent}$ | inner ring centrifugal expansion (mm) |
| $V_s$ | spindle speed (r/s) |
| $W$ | power (W) |
| $X$ | ratio efficient of oil-air |
| $X_0$,$Y_0$ | equivalent static load factor |
| $Z$ | balls number |

*Greek symbols*

| | |
|---|---|
| $\alpha$ | contact angle (°) |
| $\alpha_b$ | thermal expand constant |
| $\alpha_d$ | diffusion coefficient |
| $\omega$ | angular velocity (rad/s) |
| $\varepsilon$ | thermal expansion (mm) |
| $\delta$ | contact deformation (mm) |
| $\zeta$ | relative displacement (mm) |
| $\theta$ | relative angular displacement (°) |
| $\varphi_s$ | constant of load rating |
| $\rho$ | density (kg/$mm^3$) |
| $\mu$ | Poisson ratio |
| $\mu_s$ | friction coefficient |
| $\Delta T_b$ | uniform temperature of the ball (°C) |
| $\sum$ | elliptic integrals of the second kind |
| $\lambda$ | gyroscopic moment coefficient |
| $\eta$ | dynamic viscosity (Pa·s) |

*Subscripts*

| | |
|---|---|
| $a$ | axial direction |
| $b$ | ball |
| $c$ | cage |
| $i$ | inner ring |
| $j$ | the *j*th ball |
| $o$ | outer ring |
| $r$ | radial direction |

environment (−25 to 20 °C); Jiang et al. [6] measured the temperature rise of hybrid ceramic ball bearings and steel bearings lubricated under oil-air lubrication, respectively. Some parameters influencing bearing's temperature variation were also investigated by Jiang et al. [6], such as lubricating oil type and viscosity, lubrication interval, bearing's preload and shaft rotating speed; on the basis of Palmgren's research, Harris et al. [7] calculated the friction heating power between balls and rolling contact, the friction heat caused by lubricating oil viscosity, the friction heat between the cage and the ring, and the friction heat between the roller end and the inner ring side.

At the same time, there are many presented reports on the mechanism of thermal performance for bearings. Ma et al. [8] discussed the thermally induced deformations of high-speed spindle system, and built a three-dimensional (3D) finite element analysis model to conduct transient thermal-structure interactive analysis; Lin et al. [9] studied the heat expansion of trap and rolling body affecting bearing temperature; Kim et al. [10] studied the effect of the contact thermal resistance between rollers and rings on heat transfer; Yang et al. [11] investigated the automatic adjustment of preload with bearing material thermal expansion; Brecher et al. [12] estimated the cage-induced frictional losses under minimal lubrication; Yan et al. [13] studied the thermal- deformation coupling of spindle-bearing system; Liu et al. [14] integrated the thermal response and preload into a new thermal-mechanical coupling model of angular contact ball bearing to analyze the friction loss of shaft-bearing; Li et al. [15] analyzed the effect of improper assembly on thermal characteristics of rolling bearings and spindle.

Various factors impacting the temperature of angular contact ball bearings, as mentioned above, have been investigated by pioneers successively. Until now heat convection effect on the thermal behaviors, however, is not thorough. The housing convecting with atmosphere was considered by few scholars as well as the coolant and lubricant exchanging heat with the substructures of bearings. Additionally, the structures outside bearings influencing heat convection have been not discussed in depth.

In modeling the bearings to estimate the temperature fields, many contributions were conducted based on finite element method (FEM), such as Huang et al. [16] and Pouly et al. [17]. Due to the limitation in meshing and boundary condition treatment, the computation of FEM is complicated, however. By contrast, the advantage of thermal network method in complex system analysis is embodied and therefore is popular in analyzing the heat transfer of complex systems.

Brown et al. [18] studied the heat dissipation of lubricant and the temperature of outer raceway; Parker et al. [19] introduced the oil scaling factor to discuss the thermodynamic performance of bearing and verify the thermal mesh model of angular contact