Tribology International 87 (2015) 11–22

Contents lists available at ScienceDirect

Tribology International

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

Temperature rise of double-row tapered roller bearings analyzed with the thermal network method

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ARTICLE INFO

Article history:
Received 26 October 2014
Received in revised form
10 December 2014
Accepted 7 February 2015
Available online 17 February 2015

Keywords:
Double-row tapered roller bearing
Thermal network method
Temperature
Power loss

ABSTRACT

Based on generalized Ohm's law, the thermal network model (TNM) is developed for double-row tapered roller bearing lubricated with grease, which is commonly used in high-speed railway. The load distribution and kinematic parameters in bearing are obtained by developing a quasi-static model. The temperature of bearing at different speeds, filling grease ratios and roller large end radius are investigated. The results show that large rotating speed and filling grease ratio result in high temperature rise, especially at roller large end/flange contacts. Besides, an optimal roller large end radius is presented and its mechanism has been explored.

1. Introduction

Double-row tapered roller bearing is widely used in high-speed rail for its capacity of carrying very large radial and thrust loads. Owing to different contact angles of roller-inner and -outer ring, a component force exists between roller large-end and flange, which can cause a sliding friction. The relative sliding velocity between roller large-end and flange becomes large under high-speed conditions, which will result in high sliding friction loss and temperature rise on contact regions. The thermal network method divides bearing systems into isothermal elements connected by thermal resistances; then the temperature can be obtained through generalizing Ohm's law. Thus, the thermal network method can be applied to predict the temperature and to analyze thermal behavior of bearing.

Internal load distribution and kinematics are important to the establishment of thermal network model of rolling bearing. Harris [1] established the quasi-dynamic model of double-row tapered roller bearing and calculated the kinematic and dynamic parameters of bearing. Rydell [2] found that the point contact is appropriate for the friction characteristic at the region between roller large-end and flange. The load and velocity at different contact points have to be determined in advance, which can be obtained through the dynamic model of tapered roller bearing. Kleckner [3] calculated skew, radial and axial displacements, as well as the position of flange contact of a cylindrical roller bearing by theoretical analysis. Zhang et al. [4] obtained full numerical solution of pressure and film thickness distribution by forward iterations for elastohydrodynamic lubrication (EHL) problem of elliptical contact between rib face and roller end in tapered roller bearings. In addition, he also investigated the effects of ratios of curvature in both principal planes and position of nominal point of contact on minimum film thickness and friction.

Cretu et al. [5–8] built the quasi-static and quasi-dynamic model of tapered roller bearing through theoretical analysis. Based on that, an improved quasi-static model was established by Hu et al. [9], which can obtain the accurate kinematic and dynamic parameters such as speed and load of bearing's elements. Xia [10] employed the information poor theory to investigate the relation between the inner ring rib roughness and the vibration velocity of tapered roller bearing.

The thermal network method is commonly used to investigate the thermal behavior of bearing. Shaberth initially developed the computer program for US Army in 1974 to calculate the thermal performance of ball bearing and analyze the thermo-mechanical performance of load support systems consisting of a shaft supported by up to five rolling-elements; the program [11] was upgraded in 1981 by adding new capabilities to improve its executing performance. Parker et al. [12,13] initially verified the validity of the thermal network method in predicting bearing heat generation, inner-race and outer-race temperatures and oil-out temperatures through computer program, which agreed very well with the experimental data obtained from three different sizes of ball bearing. Following that, the lubricant volume fraction (X) in the drag force expression [14] was then adjusted based on the hypothesis that one single sphere immersed in an infinite fluid, so that the calculated global loss agrees with the experimental data. For medium rotational speeds, several models aimed at predicting

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http://dx.doi.org/10.1016/j.triboint.2015.02.011
0301-679X/

Nomenclature

A	Area
a	Semi-major axis of ellipse in rolling direction at contact
b	Semi-minor axis of ellipse in the direction transverse to the rolling at contact
c	$0.5d_m \tan\theta + 0.5l \sin(\alpha_f + \frac{\alpha_i+\alpha_e}{2}) sec\theta$
C_m	Dimensionless torque
d_{ext}	External diameter
d_{int}	Internal diameter
d_m	The pitch diameter of bearing
d_n	The distance between two columns
D	Shear ratio
D_h	Outside diameter of the tube
$D_{\omega 1}$	Diameter of roller small-end
$D_{\omega 2}$	Diameter of roller large-end
E	Young modulus
F_0	Preload
F_a	Radial force
F_c	Centrifugal force
F_r	Thrust force
h_0	Central lubricant thickness
H_d	Heat flow by heat conduction
H_g	Heat generation
H_v	Heat flow by heat convection
i	The sequence number of columns
j	The sequence number of rollers at each column
k	Thermal conductivity
k_e	Ellipticity
K	Load-displacement coefficient
l	Length of roller
m	Mass
M	Tilting moment
n	Flow index
Nu	Nusselt number
p	Pressure
Pe	Peclet number
$\dot{q}$	Heat flow
Q	Load
Q_e	Contact force of roller-outer ring
Q_f	Contact force of roller large-end-flange
Q_i	Contact force of roller-inner ring
$\dot{Q}$	Sliding friction loss
$\dot{Q}_c$	Viscous drag loss
R	Thermal resistance
Re	Reynolds number
R_m	Pitch radius of bearing
R_s	Radius of sphere
R_x	Equivalent radius of contact
S	Conductance matrix
S_m	Immersed surface of a roller
T	Temperature
T_0	Atmospheric temperature
T_1,T_2,T_3	Coordinates of the center of the sphere of roller large-end
Ta	Taylor number
U	Relative sliding velocity
U_1	Sliding velocity of P_i on the roller large-end
U_2	Sliding velocity of P_i on the flange of inner ring
X	Volume fraction of grease
x,y,z	Coordinates
α	Pressure-viscosity coefficient
α_e	Contact angle of roller-outer ring
α_f	Contact angle of roller large-end-flange
α_i	Contact angle of roller-inner ring
β	Angle
γ	Temperature-viscosity coefficient
δ_0	Axial displacement caused by preload
δ_r	Axial displacement
δ_r	Radial displacement
δ_θ	Angular displacement
ε	Relative error
ε_R	Radial clearance
Δt	Temperature difference
Δv	Relative sliding velocity
η	Viscosity of lubricant at pressure p and temperature T
η_0	Viscosity of lubricant at atmospheric pressure and temperature T_0
η_{air}	Viscosity of air
η_{eff}	Viscosity of grease-air mixture
η_{grease}	Viscosity of grease
θ	$0.5\pi - \alpha_f$
μ	Dynamic viscosity of lubricant
μ_s	Velocity of forced flow of air
ν	Kinematic viscosity of air
ρ_{eff}	Density of grease-air mixture
τ	Shear stress
τ_y	Yield stress
ω_c	Orbital angular velocity of roller
ω_i	Angular velocity of inner ring
ω_R	Spinning angular velocity of roller

global losses have been proposed [15], but they gave no information on the locations of the sources of power loss. Harris [16] detailedly formulated the sliding friction force, viscous drag force and sliding force between cage and bearing rings to calculate the temperature of bearing network nodes under the condition of oil-bath lubrication. Pouly et al. [17,18] presented a thermal network model of ball bearing and clarified thermal resistances between thermal elements; the results show that bearing temperature distribution is very sensitive to the localization of the heat sources. Apart from geometry, the drag coefficient and the fraction of air in the lubricant are the main parameters to control their intensity; their results show a good agreement with the experimental results carried out by NASA [19] on a jet-lubricated high-speed ball bearing subjected to a pure axial load.

As the tapered roller bearing in railway is lubricated with grease, the grease lubrication has to be considered during establishment of the thermal network model. Kauzlarich et al. [20] published the first theoretical analysis of EHL with grease in 1972; they formulated the Reynolds equation with the Herschel–Bulkley model and examined the validity of this model. Cann [21] researched the mechanism of grease lubrication and the replenishment problem, and studied the non-Newtonian rheology of grease, surface chemistry and capillary flow. Jin-Gyoo Yoo et al. [22] conducted an analysis of grease thermal elastohydrodynamic lubrication problems based on the Herschel-Bulkley model.

It can be found that the studies focused on thermal behaviors of double-row tapered roller bearing are very limited; however, double-row tapered roller bearing is now frequently used in high-speed railway and its thermal behaviors may greatly influence the bearing performance. In this paper, using a quasi-static model and the Herschel-Bulkley model of grease, a thermal network model of double-row tapered roller bearing is established