Tribology International 86 (2015) 62–71

Contents lists available at ScienceDirect

Tribology International

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

On the dynamic performance of roller bearings operating under low rotational speeds with consideration of surface roughness

Jafar Takabi, M.M. Khonsari *

Department of Mechanical and Industrial Engineering, Louisiana State University, 2058 Patrick Taylor Hall, Baton Rouge, LA 70803, USA

ARTICLE INFO

Article history:
Received 30 October 2014
Received in revised form
29 November 2014
Accepted 15 January 2015
Available online 23 January 2015

Keywords:
Surface roughness
Dynamic model of roller bearing
Traction coefficient
Heat generation

ABSTRACT

The dynamic behavior of radially-loaded roller bearings operating at high loads and low speeds with different surface roughness values is investigated. The simulations take into account different lubrication regimes (rigid solid–isoviscous, elastic solid–isoviscous, rigid solid–piezoviscous and elastic solid–piezoviscous) in the loaded and unloaded sections of the bearing. The simulation results provide a detailed understanding of the variation of the film thickness, wear rate and heat generation between the rollers and the raceways as the rollers travel in the orbital direction. Simulation results reveal that an increase in the radial load results in a proportional increase in the wear rate and an exponential increase in the heat generation, although it does not affect the film thickness noticeably.

1. Introduction

The key parameters that govern the performance of rolling element bearings are the film thickness and traction coefficient; lubricant film protects the surface and traction coefficient determines the power loss. Over the past three decades many rheological models have been proposed to estimate the lubricant film thickness and traction coefficient [1–8]. Although it is possible to calculate the film thickness in different lubrication regimes with a good accuracy, reliable prediction of the traction coefficient under severe contact pressures in rolling bearings still remains to be a challenging task.

Traction coefficient is also a highly influential parameter in dynamic modeling of rolling bearings, where elastohydrodynamic lubrication (EHL) prevails. It is a function of the temperature and pressure as well as the sliding and rolling velocities of the bodies in contact. Reported in the dynamic models of several studies [9–13] are the use of simplified traction curves that vary as a function of only the sliding velocity. A more accurate method of calculating the traction coefficient of Newtonian fluids is presented by Gupta [14]. Later, several studies utilized more advanced non-Newtonian models [8,15–24] in dynamic analysis of rolling bearings [25–27]. Extension of the results with consideration of different rheological relationships is presented in Ref. [28] that gives the state of the art in rolling bearing modeling. Recently, Takabi and Khonsari [29] employed a non-Newtonian model implemented in a dynamic model of a cylindrical roller bearing and compared the accuracy of the results with a simplified traction curve. They showed that the simplified traction curves may be used only in the case of low sliding velocities. At high sliding velocities, considerable errors can occur in the final results of dynamic models.

A further review of available models in the open literature [9,10,12–14,27,28,30–32] reveals that the influence of surface roughness is neglected in the calculation of traction coefficient in the existing dynamic models. The effect of surface roughness becomes important when the film thickness is small and does not fully separate the contact surfaces. This is typically the case at high operating temperature or low rotational speed and in heavily-loaded bearings. Accordingly, in the present paper we study the effect of surface roughness on the dynamic performance of roller bearings in low rotational speeds and under large contact loads. For this purpose, a simplified dynamic model of cylindrical roller bearings is developed which employs the results of a recent study [33] to calculate the traction coefficient with provision of surface roughness that concomitantly affects the film thickness, heat generation and wear rate.

2. Modeling and simulations

2.1. Dynamic model of roller bearings

Rolling element bearing analytical models can be broadly classified as either quasi-static or dynamic models. The widely-used quasi-static models are based on the equilibrium of static

* Corresponding author. Tel.: +1 225 578 9192; fax: +1 225 578 5924.
E-mail address: khonsari@me.lsu.edu (M.M. Khonsari).

http://dx.doi.org/10.1016/j.triboint.2015.01.011
0301-679X/

Nomenclature

a_x	diameter of area associated with an adsorb molecule (m)
c_f	frequency of the elastic contact (Hz)
E	elastic modulus (Pa)
E_a	heat adsorption of the lubricant on surface (kJ/mol)
E'	effective modulus of elasticity $[0.5[(1-\nu_1^2)/E_1+(1-\nu_2^2)/E_2]]^{-1}$
F_s	traction force (N)
f	traction coefficient
f_c	asperity friction coefficient
g_E	dimensionless elasticity parameter $(w_l^2/\mu_0 E' Ru)^{1/2}$
g_v	dimensionless viscosity parameter $\left(\alpha^2 w_l^3/\mu_0 R^2 u\right)^{1/2}$
G	dimensionless material number $(\alpha E')$
h	film thickness (m)
h_{min}	minimum film thickness (m)
h'	dimensionless film thickness $(h_{min} w_l/\mu_0 Ru)$
H	Vickers hardness (Pa)
H_c	dimensionless central film thickness (h_c/R)
H_{min}	dimensionless minimum film thickness (h_{min}/R)
k	thermal conductivity of lubricant (W/m K)
k_w	original Archard wear coefficient
K	dimensionless thermal number $(E'R/\sqrt{k\mu_0 T_0})$
l	contact length (m)
L_a	asperity load ratio (percentage)
m	mass (kg)
n	load–deflection exponent, line-contact, 1.11
p_h	hydrodynamic pressure (Pa)
Q	contact load (N)
$\dot{Q}_{gen}$	rate of heat generation (W)
R	equivalent contact radius, $(1/R_1 \pm 1/R_2)^{-1}$ (m)
R_g	gas constant (J/mol K)
S	slide-to-roll ratio (u_s/u)
t_0	fundamental time of vibration of the molecule in the adsorb state (s)
T_o	inlet temperature (K)
T_s	absolute temperature of the surface (K)
u	rolling velocity (m/s)
u_s	sliding velocity (m/s)
U	dimensionless velocity number $(\mu_0 u/E'R)$
V	dimensionless hardness number (H/E')
w_l	load per contact length (N/m)
W	dimensionless load number $(w_l/E'R)$
$\dot{W}_r$	wear rate (m^3/s)
z_1	$\alpha/5.1\times10^{-9}(\ln\mu_0+9.67)$
α	pressure–viscosity coefficient (m^2/N)
φ	fractional film defect coefficient
δ	load–deflection (m)
ν	Poisson ratio
μ	lubricant viscosity (Pa s)
μ_0	inlet viscosity (Pa s)
σ	standard deviation of the surface heights (m)
$\bar{\sigma}$	dimensionless surface roughness number (σ/R)
τ	shear stress (Pa)
Λ	limiting shear stress coefficient

forces and moments acting on each rolling element, whereas in dynamic models the integration of the differential equations of motions with respect to time is considered [30,31,34–39]. Unlike the quasi-static models, dynamic models can provide a real-time simulation of the motion of the bearing elements under any operating condition.

The first fully dynamic model of the rolling element bearings was presented by Gupta [40–43] where six degrees of freedom (DOF) were considered for all the bearing elements including rollers, cage and race-ways. Based on the Gupta's model the translational and rotational motions of each bearing element are predicted by solving the Newton and Euler equations of motions, respectively. Presented in the literature are also several simplified dynamic models [9,10,12,13,32,44] to reduce the required computational time and to make parametric design studies. Ref. [28] presents a thorough review and the current state-of-the-art of dynamic models of rolling element bearings.

Developed for the present study is a dynamic model of cylindrical roller bearings similar to Gupta's model. The basic concept and details of this dynamic model is presented in Refs. [14,29]. The next section presents the simplified assumptions made for the dynamic model of the present study and their effects on the final results. Then, the approach developed in the current work to calculate the traction forces in different regimes of lubrication is presented in Section 2.2.

2.1.1. Simplified dynamic model

In the general form of a fully dynamic model, rollers can freely move between the races, and this free motion of the rollers between the races leads to a high frequency response in the results. The frequency (c_f) of this vibratory motion of the rollers can be calculated by employing the Hertzian load-deflection formula for the elastic contact between the rollers and the raceways [14]:

$$c_f = \frac{1}{2\pi}\sqrt{\frac{nQ}{\delta m}} \tag{1}$$

where δ and Q are the normal displacement and the normal contact force between the roller and the raceway, respectively. m is the mass of the roller, and n is equal to 1.11 for the case of line contacts. This high frequency motion can be filtered out by an equilibrium constraint for the special cases, such as the current study, where the position of the inner raceway and the outer raceway are fixed with respect to one another [14]. Thus, the radial and axial position of the loaded rollers between the races is fixed for any orbital position and can be calculated from the quasi-static solution associated with any orbital position of the rollers; see Fig. 1. Considering the fact that the radial load and the rotational

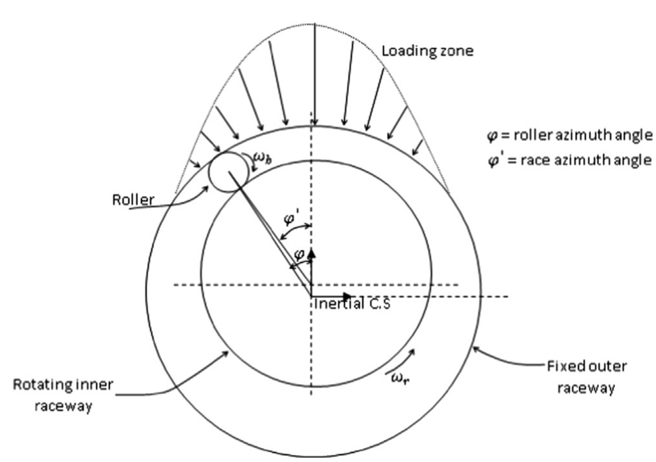

Fig. 1. Positions of rollers and races in a bearing under a constant unidirectional radial load.