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The influence of the lubricant viscosity on the rolling friction torque

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ABSTRACT

Authors propose a theoretical model and an experimental methodology for defining the friction torque in a modified thrust ball bearing, operating in mixed and full film lubrication conditions.

The friction torque was measured at low loads and large Λ parameter range using a spin-down method.

A comprehensive analytical bearing torque model is described using elastic rolling resistance, curvature effects, inertia forces, disc-air resistance and ball-races hydrodynamic rolling forces, the latter explaining 98% of the final bearing torque. Several sets of hydrodynamic rolling force relationships respecting the transition from IVR to EHL lubrication regime were tested. Final numerical results are shown to be very close to the experimental ones in both full film and mixed lubrication conditions.

1. Introduction

The rolling resistant torque of a ball on the raceway in a bearing is due to miscellaneous concepts, including elastic hysteresis losses due to micro slip in the contact and curvature effects, hydrodynamic lubricant resistance, roughness and form deviations effects on torque.

The resistant moment around the contact center to rolling due to elastic hysteresis losses in the rolling process and contact micro slips can be approximated by the following relationship [1,2]:

$$MER = 7.48\ 10^{-7}\left(\frac{d}{2}\right)^{0.33} Q^{1.33}\left\{1-3.519\ 10^{-3}\ (k-1)^{0.8063}\right\}\frac{\mu_s}{0.11} \quad [\mathrm{N\ m}] \tag{1}$$

Where Q is the normal load, d is the ball diameter, k represents the ratio R_y/R_x, in which R_x and R_y are the reduced radii of curvature in the rolling direction and the transverse direction, respectively.

The sliding friction coefficient μ_s has a maximum value of the order of 0.11 in dry contact conditions, while in the presence of a lubricant film this value decreases [1].

At the contact between a ball and the rolling track in a thrust ball bearing, in the absence of additional spin as a result of zero ball-race contact angle and in the absence of the gyroscopic motions, only two symmetrical lines of pure rolling exist in the contact ellipse [2]. Elsewhere, positive and negative sliding speed can be defined because the contact ellipse is curved in space.

As a result of this raceway curvature, an additional moment MC around the contact center can be defined using the following relationship [1,3]:

$$MC = 0.08\mu_s\frac{Q{a_c}^2}{R_d} \quad [\mathrm{Nm}] \tag{2}$$

where μ_s is the average friction coefficient on the contact ellipse, defined as a function of the lubrication regime, a_c is the major semi-axis of the contact ellipse, R_d is the radius of the deformed contact surface $\left(R_d = \frac{2dR_c}{2R_c+d}\right)$, in which R_c is the transversal curvature radius of the rolling track.

For a contact between a ball and the raceway in a ball bearing operating in dry conditions the total rolling moment around the contact center can be obtained by summing Eqs. (1) and (2).

Dry friction, is however not common in bearings and it is essential to also account for lubricant effects on the final bearing torque.

In order to estimate the global friction losses in ball bearings and tapered roller bearings, Houpert [1,2,4] developed several analytical models in which the lubrication regime affects both the friction coefficient μ_s and a hydrodynamic rolling force FR.

The accurate estimation of the friction coefficient μ_s has been the subject of several theoretical and experimental studies using non-linear visco-elastic and thermal models. Roughness effects on friction are also introduced via the parameter Λ (defined as film thickness divided by the composite RMS surface roughness), [1,11,12,16].

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Nomenclature

a_c	major semi-axis of the contact ellipse [m]
d	ball diameter [m]
E^*	equivalent Young modulus of the two elements in contact [Pa]
F_{ib}	inertial force of the ball [N]
FP	pressure force in lubricated ball race contact [N]
FR	hydrodynamic rolling force in lubricated ball race contact [N]
FS	traction force in lubricated ball race contact [N]
Ft	tangential force in lubricated ball race contact [N]
g_v	viscosity parameter [dimensionless]
g_E	elasticity parameter [dimensionless]
G	weight of the disc [N]
G	material parameter [dimensionless]
h	film thickness [m]
J	moment of inertia [kg m^2]
k	reduced radius ratio $k=R_y/R_x$ [dimensionless]
K_M	dimensionless coefficient in laminar flow [dimensionless]
K^*	constant factor used in solving of the differential Eq. (26) [N m s^{-n}]
m_b	ball mass [kg]
m_d	disc mass [kg]
M	Moes parameter [dimensionless]
M_{rol}	total rolling friction moment around the ball center [N m]
MC	resistant moment around the contact center generated by the raceway curvature [N m]
MER	resistant moment around the contact center to rolling due to elastic hysteresis losses in the rolling process and contact micro slips [N m]
M_f	friction torque between the disc and air [N m]
n	constant exponent used in solving of the differential Eq. (26) [dimensionless]
$P_{i,d}$	inertial power of the system including upper race and disc [W]
$P_{s,b}$	total power losses generated by the pivoting motion of the balls over the raceways [W]
Q	normal load on a ball [N]
r	raceway radius related to the rotational axis [m]
R	radius of the upper disc [m]
R_c	transversal curvature radius of the rolling track [m]
R_d	radius of the deformed ball-race contact surface [m]
Re	Reynolds parameter [dimensionless]
R_q	rms of the surface roughness [m]
R_x	reduced radii of curvature in the rolling direction [m]
R_y	reduced radii of curvature in the transverse direction [m]
t	time [s]
Tz	total friction torque in modified thrust ball bearing [Nm]
U	speed parameter [dimensionless]
v_m	average entrainment speed in ball-race contact [m s^{-1}]
W	load parameter [dimensionless]

Greek characters

α_p	piezo-viscous parameter of lubricant [Pa^{-1}]
η_0	oil dynamic viscosity at the operating temperature [Pa s]
Λ	lubricant parameter [dimensionless]
μ_s	sliding friction coefficient [dimensionless]
υ_f	kinematics air viscosity [m^2 s^{-1}]
ρ_f	air density [kg m^{-3}]
ϕ_2	angular position of the upper race [rad]
ω_b	angular speed of the balls in rotational motion around the bearing axis [s^{-1}]
ω_1	angular speed of the lower race [s^{-1}]
ω_2	angular speed of the upper race and disc [s^{-1}]
$\omega_{2,0}$	initial angular speed of the upper race in decelerating process [s^{-1}]

Subscripts

1	lower race
2	upper race
b	ball
exp	experimental
EHL	elastohydrodynamic lubrication regime
EHL_B	Biboulet's equations for elastohydrodinamic lubrication regime
EHL_H	Houpert's equations for elastohydrodinamic lubrication regime
IVR	isoviscous rigid lubrication regime
IVR_B	Biboulet's equations for isoviscous rigid lubrication regime
IVR_H	Houpert's equations for isoviscous rigid lubrication regime
r	race
Trans	transition between IVR to EHL lubrication regime
Trans_B	Biboulet's proposal equation for transition from IVR to EHL
Trans_H	Houpert's proposal equation for transition from IVR to EHL

Recently, Guddei and Ahmed [17] determined experimentally the influence of the surface roughness on rolling friction coefficients using low load and low surface roughness conditions.

According to Eqs. (1) and (2), the presence of the lubricant in the rolling contact leads to a decrease of the rolling moment since the friction coefficient μ_s will decrease. But the presence of lubricant in the rolling contact is responsible for a bearing torque increase due to the hydrodynamic rolling force FR calculated when integrating the Reynolds equation and accounting for the Poiseuille flow. Miscellaneous relationships have been suggested for calculating FR.

In the full film lubrication conditions between a ball and the rolling track, Houpert [14] suggested the following in 1987:

– in the IVR lubrication regime

$$FR_{\mathrm{IVR}\_H} = 1.213E^*R_x^2k^{0.358}U^{0.636}W^{0.364} \tag{3}$$

– in the EHL lubrication regime

$$FR_{\mathrm{EHL}\_H} = 2.765E^*R_x^2k^{0.35}U^{0.656}W^{0.466}G^{0.022} \tag{4}$$