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On the thermal elastohydrodynamic lubrication of tilting roller pairs

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ABSTRACT

In order to investigate thermal effect of tilting roller pairs, a numerical solution for TEHL of tilting roller pairs has been presented. Variations in the lubricating performance with tilting angle have been investigated. Comparison between thermal and isothermal solutions has been made. Effects of the end profile radius, the velocity, and the maximum Hertzian pressure have been discussed. Profile modification of the roller generatrix has been assumed. Results show that all of the highest temperature, the maximum pressure, and the minimum film thickness occur at the load-carrying end. Larger tilting angle results in more evident thermal effect.

1. Introduction

Because of the deflection of a shaft caused by applied loads, dimensional error of the shaft and housing, and mounting errors, the inner and outer rings of a roller bearing may be slightly misaligned. Therefore, in the design of cylindrical roller bearings, the permissible misalignment varies depending on the bearing type and operating conditions, but usually it is just a small angle less than 0.0012 rad (4'). Similarly, any loading is likely to produce deflections of the shaft and hence roller tilt.

Harris et al. [1] investigated the cause and effect of roller skewing in cylindrical roller bearings, and pointed out that the roller thrust loading causes roller tilting and friction moments, and the friction moment may give rise to roller skewing which will affect bearing friction heat generation and fatigue endurance.

For both aligned and misaligned rollers in dry contacts, Heydari and Gohar [2] and Johns and Gohar [3] discussed the influence of the roller axial profile on the pressure distributions. However, as it was known that, machine elements are usually working under lubricating condition.

Mostofi and Gohar [4], and Park and Kim [5] obtained complete numerical solutions for isothermal EHL finite line contacts. Liu and Yang [6] obtained the complete solution under different oil-supply conditions for isothermal finite line contacts without considering the tilt effect. Kushwaha et al. [7] provided aligned and misaligned contacts of roller to races in elastohydrodynamic finite line contact under isothermal condition, and presented the film shape and pressure distribution at the extremities of a finite line contact.

More recently, Liu et al. [8] analyzed the lubricating mechanism for tilting rollers in rolling bearings, provided the film thickness and pressure with different tilting angles and compared solutions between the starved and fully flooded lubrication in tilting roller contacts. However, thermal effect was not considered in their analysis.

Liu and Yang [9] proposed a complete thermal elastohydrodynamic lubrication (TEHL) solution for a finite line contact, and compared solutions between the thermal finite, isothermal finite and thermal infinitely long line contacts. However, their analysis was based on that the roller is parallel to the infinite plane.

As it was known, thermal effect is very important in mechanical elements such as gears, cams, and high speed bearings. However, it appears that TEHL analysis in tilting roller contacts has never been reported yet. This paper attempts to give TEHL solution in tilting roller contacts, and explains both the tilting effect and thermal effect on the lubricating performance of roller pairs.

2. Mathematical model

The tilting roller contact pairs can be viewed as a finite line contact formed between a cylindrical roller "b" with dub-off end profile modification, and an infinite plane "a", as shown in Fig. 1. When the roller is tilted, its axis is no longer parallel to the plane "a", and a tilting angle, θ, is produced. The x-axis is perpendicular to the sheet of the paper and points to readers. The coordinate systems for solids "a" and "b" are similar to that for the film.

Considering the variations in the viscosity and density of the lubricant across the film, the generalized Reynolds equation [10]

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Nomenclature

b_H	Hertzian contact radius, $\sqrt{8wR_x/(\pi L E')}$, m
c, $c_{a,b}$	specific heat of lubricant and solids, J/kg K
E'	reduced elastic modulus, Pa
H	dimensionless film thickness, hR_x/b_H^2
H_{00}	dimensionless rigid central film thickness, $h_{00}R_x/b_H^2$
h	film thickness, m
h_{00}	rigid central film thickness, m
h_{min}	minimum film thickness, m
k, $k_{a,b}$	thermal conductivities of lubricant and solids, W/m K
L	length of roller, m
l	length of the straight part of roller, m
P	dimensionless film pressure, p/p_H
p	film pressure, Pa
p_H	maximum Hertzian contact pressure, $2w/(\pi b_H L)$
R_x	radius of cylindrical roller, m
R_y	end profile radius of the roller, m
$\bar{T}$	dimensionless temperature, T/T_0
T	temperature, K
T_0	ambient temperature, K
U, V	dimensionless film velocities, u/u_e, v/u_e
u, v	film velocities, m/s
$u_{a,b}$	velocities of surfaces "a" and "b", m/s
U_e	dimensionless velocity parameter, $u_e\eta_0/(E'R_x)$
u_e	entrainment velocity, $(u_a+u_b)/2$, m/s
w	applied load, N
X, Y	dimensionless coordinates, x/b_H, y/b_H
x, y	coordinates, m
x_{in}, x_{out}	domain boundaries, m
y_{in}, y_{out}	domain boundaries, m
Z	dimensionless coordinate, z/h
z	coordinate across the film, m
Z_a, Z_b	dimensionless coordinates in solids "a" and "b", z_a/b_H, z_b/b_H
z_a, z_b	coordinates in solids "a" and "b", m
α	Barus viscosity–pressure coefficient, Pa^{-1}
β	Reynolds viscosity–temperature coefficient, K^{-1}
δ_A	amplitude of the profile modification function, m
$\bar{\eta}$	dimensionless viscosity of lubricant, η/η_0
η	viscosity of lubricant, Pa s
η_0	ambient viscosity of lubricant, Pa s
μ	frictional coefficient
θ	tilting angle of rollers, rad
$\bar{\rho}$	dimensionless density of lubricant, ρ/ρ_0
ρ, $\rho_{a,b}$	densities of lubricant and solids, kg/m^3
ρ_0	ambient density of lubricant, kg/m^3
ξ	slide-roll ratio, $(u_a-u_b)/u_e$

for a Newtonian steady-state finite line contact can be written as:

$$\frac{\partial}{\partial x}\left[(\rho/\eta)_e h^3\frac{\partial p}{\partial x}\right]+\frac{\partial}{\partial y}\left[(\rho/\eta)_e h^3\frac{\partial p}{\partial y}\right]=12u_e\frac{\partial}{\partial x}(\rho^* h) \tag{1}$$

where, $(\rho/\eta)_e=12(\eta_e\rho'_e/\eta'_e-\rho''_e)$, $\rho^*=[\rho'_e\eta_e(u_b-u_a)+\rho_e u_a]/u_e$, $\rho_e=(1/h)\int_0^h\rho\,dz$, $\rho'_e=(1/h^2)\int_0^h\rho\int_0^z(1/\eta)dz'dz$, $\rho''_e=(1/h^3)\int_0^h\rho\int_0^z(z'/\eta)dz'dz$, $1/\eta_e=(1/h)\int_0^h(1/\eta)dz$, $1/\eta'_e=(1/h^2)\int_0^h(z/\eta)dz$.

In solving Eq. (1), the boundary and cavitation conditions must be satisfied, these conditions can be given by:

$$\begin{cases} p(x_{in},y)=p(x_{out},y)=p(x,y_{in})=p(x,y_{out})=0 \\ p(x,y)\geq 0 \quad (x_{in}<x<x_{out},\quad y_{in}<y<y_{out}) \end{cases} \tag{2}$$

The film thickness equation for the tilting roller contact can be written as:

$$h(x,y)=h_{00}+\frac{x^2}{2R_x}+\frac{(y\pm l/2)^2}{2R_y}f_\Delta-\delta(y)$$

$$+\frac{2}{\pi E'}\iint_\Omega\frac{p(x',y')}{\sqrt{(x-x')^2+(y-y')^2}}dx'dy'+y\tan\theta\,(-0.5L\leq y\leq 0.5L) \tag{3}$$

where Ω is the computing domain; f_Δ is a symbolic function, $f_\Delta=1$ if $y>l/2$ and $f_\Delta=0$ if $y\leq l/2$. In order to reduce the load-concentrated effect, profile modification of a parabolic function for the roller generatrix of the straight part is assumed, and expressed by:

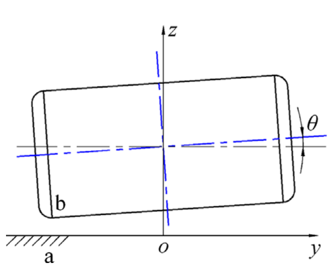

Fig. 1. Schematic of a tilting roller pair and coordinate system, x axis is perpendicular to the sheet.

$$\delta(y)=\delta_A\left[1-(2y/l)^2\right] \tag{4}$$

where, δ_A is the amplitude of the profile modification function.

It should be noted from Fig. 1 that, without the straight part of the roller, the problem will become one of a typical point contact, therefore, the half space assumption for the roller can be adopted in the present analysis.

The viscosity–pressure–temperature relationship proposed by Roelands [11] is adopted. It can be expressed in SI unit as

$$\eta=\eta_0\exp\left\{(\ln\eta_0+9.67)\times\left[-1+(1+5.1\times10^{-9}p)^{z_0}\left(\frac{T-138}{T_0-138}\right)^{-s_0}\right]\right\} \tag{5}$$

the dimensionless constants z_0 and s_0 in Eq. (5) can be calculated as follows [12]:

$$z_0=\alpha/\left[5.1\times10^{-9}(\ln\eta_0+9.67)\right]$$

$$s_0=\beta(T_0-138)/(\ln\eta_0+9.67)$$

The Dowson–Higginson's relationship [13] of density-pressure is employed with addition of a term to express the thermal expansion:

$$\rho=\rho_0\left[1+C_1p/(1+C_2p)-C_3(T-T_0)\right] \tag{6}$$

where $C_1=0.6\times10^{-9}Pa^{-1}$, $C_2=1.7\times10^{-9}Pa^{-1}$, and $C_3=0.00065K^{-1}$.

The load balance equation is given by:

$$\iint_\Omega p\,dxdy=w \tag{7}$$

The temperature distribution within the oil film can be obtained from the energy equation for the flowing film. Ignoring the heat conduction in x and y-directions, this equation can be written as:

$$c\left(\rho u\frac{\partial T}{\partial x}+\rho v\frac{\partial T}{\partial y}-q\frac{\partial T}{\partial z}\right)=k\frac{\partial^2 T}{\partial z^2}$$