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Thermal elastohydrodynamic simulation of involute spur gears incorporating mixed friction

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ABSTRACT

A model for calculating transient, three-dimensional, thermal elastohydrodynamic tooth flank contacts in spur gears with involute gearing is presented. The calculation model is based on the combined numerical solutions of the generalized Reynolds, energy and Fourier heat equations. Mass conserving cavitation, non-Newtonian flow and the real involute characteristics of the tooth flanks are incorporated. States of mixed friction and microhydrodynamic effects are ascertained integrally based on real-measured surface topographies. Straight spur gear pairs serve as an example to present the results of calculating the influence of surface roughness asperities and gearing geometry.

1. Introduction

Spur gears are implemented in a transmission to transmit and gear torques and rotary motions between parallel shafts (see Fig. 1). The involute has been established to be a common tooth profile. The rolling contact between the tooth flanks has a curvature that varies with the direction of the addendum. A combined sliding and rolling motion always exists on the tooth flanks, with the exception of the pitch point. The amounts of sliding motion reach their maximums in the regions of the addendum and root. The lubricated tooth flank contact constitutes a thermal elastohydrodynamic line contact that expands finitely in the direction of the width of the tooth flanks. Because velocity, curvature and load change as the contact point moves along the tooth flanks, the operating conditions are transient, regardless of the external operating parameters.

The tribological processes at the tooth surface contact greatly influence the operational reliability and efficiency of a spur gear. Significant variables of these processes are the pressures and shear stresses occurring in the contact and the resultant temperatures. High pressures and shear stresses can cause fatigue in the tooth flanks; high temperatures can cause warm scuffing. Calculating the thermal elastohydrodynamic tooth surface contact requires simultaneously analyzing the hydrodynamics in the lubricating gap, the elastic deformations of the contact partners and the thermal conditions in the lubricating gap and solid bodies. In addition, the lubrican's rheological properties and the affect of mixed friction and microhydrodynamics must be included in these equations. Furthermore, a sound calculation requires an exact description of the real kinematic and geometric relationships of the tooth surface contact and load sharing among the pairs of teeth with multiple meshing.

The development of models that calculate lubricated rolling contact has been a priority of tribology research for many years. The Reynolds equation is applied to calculate hydrodynamic pressure distribution and the theory of the elastic half-space can be applied to incorporate elastic deformation. Normally, the energy equation is applied to ascertain the thermal conditions of the lubricant, and the Fourier heat equation is used to determine the same conditions for the solid bodies. The entire problem and the individual sub-problems can only be completely solved numerically. Dowson and Higginson presented initial solutions for the isothermal elastohydrodynamic contact [1]. The contact was simplified to a line contact that expanded infinitely in the direction of width. This finding reduces the problem to a planar, two-dimensional contact. Because the contact of the two tooth flanks is normally longer in the direction of width than in the direction of the addendum, this simplification has been adopted for most models that calculate spur gears.

In 1981, Wang and Cheng presented an early, comprehensive study on a numerical simulation of the contact conditions of straight spur gear pairs [2,3]. This group calculated the transient, elastohydrodynamic tooth surface contact along the length of action, which was simplified two-dimensionally on the basis of the numerical approximation functions from Vichard [4]. Models have been introduced that determine the dynamic load sharing of

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Nomenclature

b	width (m)
c	specific heat capacity (J/kgK)
E_{red}	reduced Young's modulus (N/m^2)
f	coefficient of friction (dimensionless)
F_n	normal force (N)
g_α	length of action (m)
h	film thickness (m)
h_0	nominal film thickness (m)
h_{cr}	critical film thickness (m)
h_{def}	deformed gap height (m)
h_{min}	minimum film thickness (m)
$h_{\delta w}$	local deformed gap height (m)
k	coefficient of thermal distribution (dimensionless)
O	center point (dimensionless)
p	pressure (N/m^2)
p_{cav}	cavitation pressure (N/m^2)
p_e	base pitch (m)
$p_{c,lim}$	plastic flow pressure (N/m^2)
$\dot{q}$	heat flux density (W/m^2)
r	radius (m)
s_{y0}	coordinate of the length of action (m)
t	time (s)
u, v, w	velocities in x, y and z direction (m/s)
u_d	relative velocity (m/s)
w	deformation (m)
x, y, z	Cartesian coordinates (m)
Y_0	contact point (dimensionless)
α	pressure angle (Grad)
α_w	working pressure angle (Grad)
α_y	profile angle (Grad)
β_{th}	coefficient of volume expansion (1/K)
$\dot{\gamma}$	shear rate (1/s)
δ_y	tooth trace variation in the direction of width (m)
η	dynamic viscosity (Pas)
η^*	effective dynamic viscosity (Pas)
θ	gap fill factor (dimensionless)
ϑ	temperature (K)
λ	thermal conductivity (W/mK)
ρ	density (kg/m^3)
τ	shear stress (N/m^2)
τ_0	boundary shear stress, Eyring (N/m^2)
τ_f	friction shear stress (N/m^2)
Φ^p	pressure flow factor (dimensionless)
Φ^s	shear flow factor (m)
ψ_y	roll angle (Grad)
ω	angular velocity (1/s)
Ω	calculation domain (m^2)

Frequently used indices

c	solid body contact
el	elastic
gas/vap	gas–vapor phase
h	hydrodynamics
inv	involute
liq	fluid phase
mix	mixed variable for cavitation
pl	plastic
solid	solid bodies

multiple meshing and calculate the bulk and flash temperatures. Simple finite element models, already used to an extent, were applied. As is common in elastohydrodynamic calculations, the parabolas of each contact point were used to approximate the contact of the tooth flanks. Lin and Medley presented a model to calculate straight spur gear pairs, which was also based on the approximation functions of Vichard [5]. Unlike Wang and Cheng, this group regarded the contact as isothermal and disregarded dynamic load sharing. However, their calculation model includes a method that ascertains the change of the tooth flanks' radii of curvature, which is strongly variable in the root region. Oster presented a calculation model that was comparable to the aforementioned models [6]. However, the tooth surface contact, which is variable as a function of the length of action, was simplified to a sequence of independent and stationary, two-dimensional elastohydrodynamic contacts. The

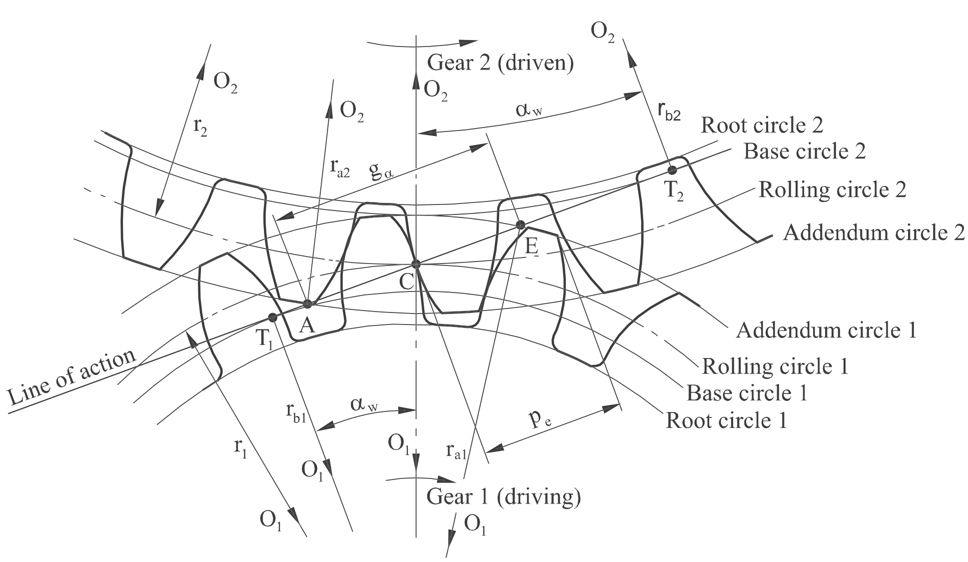

Fig. 1. Geometric conditions of involute gearing.