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Transient thermal elastohydrodynamic simulation of a DLC coated helical gear pair considering limiting shear stress behavior of the lubricant

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ABSTRACT

A transient three-dimensional thermal elastohydrodynamic calculation model is introduced to simulate the contact of tooth flanks of helical gear pairs with involute gearing. At any time, contact geometry is derived from the real involute curves of gears. In addition to the consideration of mass conserving cavitation and the non-Newtonian fluid behavior, models to capture conditions of mixed friction and microhydrodynamic effects are included. To consider the non-Newtonian fluid behavior, a shear thinning fluid model with limiting shear stress is applied. A selected helical gear pair is used as example to examine the influence DLC coated tooth flanks have on tribological behavior. It shows that friction and temperature behavior of the gearing can be significantly influenced with DLC coating.

1. Introduction

In general, cylindrical gears are used in drive trains to satisfy the requirement of an exact transmission of motion while having at the same time high power density and low friction losses. These benefits of cylindrical gears are often opposed by the impact-type loads on the gears with the consequence of inciting oscillations periodically due to the transient engagement of teeth. If the high demands are made on the load capacity and smooth running of cylindrical gear pairs, helical gears are used. On the one hand, there is a larger contact ratio due to the inclined position of the teeth and therefore, the distribution of load is more favorable. On the other hand, the gradually tapered meshing in and out leads to a significant reduction in impact load compared to a spur gear. Both in spur and helical gear pairs, the easily manufacturable circular involute has been established as the standard shape of the tooth profile. The involute gear offers the benefit to make various areas of the circular involute usable through an addendum modification at constant transmission behavior. This creates specific adaptations of the shape of the tooth profile, for example to realize a certain center distance or to improve meshing behavior. During engagement, the tooth flank contact is characterized by variable curvatures, velocities and loads. With the exception of the pitch point, tooth flanks are always subject to a superpositioned sliding and rolling motion, whereby the sliding parts of the motion achieve their largest values in the addendum and root of the tooth. Therefore, the contact of two lubricated tooth flanks must be seen as transient thermal elastohydrodynamic line contact with finite expansion in the direction of width of the tooth flanks. The tribological processes taking place in the tooth flank contact influence significantly the friction and wear behavior of the gear. In addition to a transient three-dimensional illustration of the thermal elastohydrodynamic line contact, the realistic calculation presupposes the integration of suitable rheological fluid models and the capture of mixed friction conditions and micro-hydrodynamic effects. Moreover, it requires a precise description of the real kinematic and geometric relationships of the tooth flank contact and the load distribution between tooth pairs in multiple meshing.

The first numerical calculation models to describe tribological processes in lubricated tooth flank contacts were limited to spur gear pairs and they were characterized by numerous simplifications. Compared to the present-day status, the primary reason for the simplifications was the significantly limited possibilities of computing technology. A major simplification was the reduction of the true three-dimensional tooth flank contact to a plane two-dimensional approach as line contact with infinite expansion in tooth width direction. In addition, the calculations were often isothermal using approximation equations and the Newtonian

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Nomenclature

$\overline{A'E'}$	length of action [m]
b	width [m]
c	specific heat capacity [J/kg K]
E_{red}	reduced Young's modulus [N/m^2]
f	coefficient of friction [dimensionless]
F_n	load [N]
h	film thickness [m]
h_0	nominal film thickness [m]
h_{cr}	critical film thickness [m]
h_{def}	deformed gap height [m]
$h_{\delta w}$	local deformed gap height [m]
k	thermal distribution coefficient [dimensionless]
p	pressure [N/m^2]
p_{cav}	cavitation pressure [N/m^2]
$\dot{q}$	heat flux density [W/m^2]
r	radius [m]
r_b	base radius [m]
s_{y0}	coordinate of the length of action [m]
t	time [s]
u, v	velocities in x and y direction [m/s]
x, y, z	cartesian coordinates [m]
Y_0	contact point [dimensionless]
α_n	normal pressure angle [deg]
α_p	pressure viscosity coefficient [m^2/N]
α_{yt}	transverse profile angle [deg]
β	helix angle [deg]
β_b	base helix angle [deg]
β_B	angle between contact line and flank line [deg]
β_{th}	coefficient of volume expansion [1/K]
$\dot{\gamma}$	shear rate [1/s]
δ_{xy}	gearing correction [m]
η	dynamic viscosity [Pa s]
η^*	effective dynamic viscosity [Pa s]
θ	gap fill factor [dimensionless]
ϑ	temperature [K, °C]
λ	thermal conductivity [W/m K]
ρ	density [kg/m^3]
τ	shear stress [N/m^2]
τ_c	critical shear stress [N/m^2]
τ_f	friction shear stress [N/m^2]
τ_{lim}	limiting shear stress [N/m^2]
φ_b	angle between two involute curves [deg]
Φ^p	pressure flow factor [dimensionless]
Φ^s	shear flow factor [m]
ψ_y	roll angle [deg]
ω	angular velocity [1/s]
Ω	calculation domain [m^2]

Frequently used indices

c	solid contact
h	hydrodynamics
inv	involute
liq	fluid phase
mix	mixed variable for cavitation

fluid behavior without consideration of mixed friction conditions. Exemplary for these early works, are models introduced by Wang and Cheng in [1] and [2] as well as Lin and Medley in [3]. Hua and Khonsari introduced in [4] a first complete numerical solution of the two-dimensional transient elastohydrodynamic tooth flank contact. The isothermal model considers the tooth flanks as changeable radii along the line of action. However, it does neither consider non-Newtonian fluid behavior nor the impact of surface roughness. In [5] Larsson presented another numerical solution of the transient tooth flank contact. The calculation model is also limited to an isothermal approach but it considers non-Newtonian fluid behavior in the form of a simplified shear thinning fluid model with limiting shear stress. In the contact point, the involutes of the tooth flank contours are approximated through parables of second-order equations. Wang et al. introduced in [6] a calculation model for the two-dimensional thermal elastohydrodynamic tooth flank contact. But this model is limited to the Newtonian fluid behavior. Further works dealt with the consideration of mixed friction conditions in the two-dimensional tooth flank contact. Primarily works of Castro and Seabra [7] as well as of Li and Kahraman [8] can be mentioned in this context. Both cases considered directly the roughness profile of tooth flanks in plane two-dimensional tooth flank contact. Anuradha and Kumar [9] examined the influence fluid behavior of various lubricants had on the formation of lubricant film in tooth flank contact. For this purpose, a two-dimensional thermal elastohydrodynamic calculation model was applied in connection with Carreau's non-Newtonian fluid model. The authors were able to find fluid behavior influenced the formation of lubricant film in tooth contact significantly.

Further calculation models expanded the tooth flank contact from the previously viewed simplified two-dimensional tooth flank contact to a three-dimensional approach and therefore to a real line contact with finite expansion in tooth width direction. This approach is necessary to capture influences tooth trace corrections in width direction-common in practice-in form of crowning, fillets or chamfers as well as misalignment between gear wheels have on tribological processes during contact. Holmes et al. [10] examined the tooth flank contact in spur gears as a three-dimensional elliptical contact. The calculation model they introduced allows consideration of mixed friction conditions through the inclusion of real surface topographies. However, the calculations were not performed transient for the entire line of action but only statically for selected points on the line of action. The authors of this paper introduced a transient three-dimensional thermal elastohydrodynamic calculation model for spur gear pairs [11]. The tooth flank contact was calculated for a given load distribution along the entire line of action. It considers mixed friction conditions and microhydrodynamic effects just as much as the non-Newtonian fluid behavior of lubricants. In addition, the lubricating gap geometry is derived directly from the involutes of the tooth flanks. Barbieri et al. [12] linked an isothermal three-dimensional elastohydrodynamic calculation model for the elliptical point contact with a dynamic model approach for the transient calculation of the tooth flank contact of a spur gear pair. They focused on the examination of the influence additional dynamic forces had in the formation of lubricant film in tooth flank contact. Relevant changes in pressure and lubricating gap gradients could be determined in consideration of the additional dynamic forces. The influences of thermal effects, mixed friction conditions and non-Newtonian fluid behavior found no consideration in these investigations.

In the more recent past, calculation models for helical gear pairs were increasingly developed. The calculation of the tooth flank contact becomes more difficult compared to spur gear pairs because meshing is gradual, and therefore, the size of the contact