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The scuffing load capacity of involute spur gear systems based on dynamic loads and transient thermal elastohydrodynamic lubrication

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ABSTRACT

A method for predicting scuffing failure in spur gear pairs by means of a transient thermal solver is proposed in this paper. The quasi-static and non-linear dynamic models of spur gear systems are established based on time-varying stiffness. The dynamic loads in meshing cycle are calculated. The transient effect is most pronounced at exchange point between single and double teeth region as well as greatly influenced by damping ratio. The temperature rise formula is proposed based on transient heat flux. The transient thermal elastohydrodynamic lubrication (TEHL) model which takes the dynamic loads into account is proposed and the distributions of pressure, film thickness and temperature are obtained. The flash temperature and minimum film thickness in meshing cycle are obtained and applied to check the scuffing load capacity. A comprehensively comparison between TEHL theory and Blok theory is implemented and the results turn out that the TEHL theory is more close to the actual conditions in theory. The distributions of flash temperature and minimum thickness with TEHL method are consistent with that of Blok flash temperature and thickness of Dowson, respectively. The scuffing load capacity is greatly influenced by dynamic effect.

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1. Introduction

The tribological processes of the tooth surface contact greatly influence the operational reliability and efficiency of involute spur gear systems. High contact temperature of lubricant and tooth surfaces at the instantaneous contact region may cause a breakdown of the lubricant film at the contact interface. The scuffing failure is unpredictable and severe for the gears systems and the scuffing load capacity is of great significance in the process of preliminary design and strength check of gear systems, especially for heavy-load and high-speed gear systems. Two methods are available in ISO standard [1,2] to evaluate the scuffing load capacity of the gear systems. Nevertheless the simplified form of load distribution was used and the flash temperatures were calculated based on Blok flash temperature equation [3], with which we cannot get the accurate flash temperature so that we need to attach a large safety factor to amend it.

Gear transmission regime is a non-linear dynamic system. In contrast to the pitting and fatigue breakage, a short time of transient overloads can result in scuffing failure; therefore the dynamic loads should be taken in account in the study of scuffing load capacity. Dynamic behavior of gear pairs is crucial to reduce vibration and noise as well as to prolong the lifetime of gear systems. Some papers have studied the dynamic characteristic of involute spur gears [4–15]. In the dynamic model, gears are represented by rigid disks that are connected to each other along the line of action through the time-varying meshing stiffness and viscous damper. The dynamic system of gear pair has six degrees of freedom (DOF) [5,8]. It can be studied by means of one-DOF model in terms of transmission error [9–15] under the condition that the compliance of bearings and shafts was neglected as well as the effect of manufacturing errors and misalignments, without loss of generality. The time-varying meshing stiffness, which can be calculated with finite element method (FEM) [5] and analytical method [5,8], is the main cause of the dynamic load. The time-varying stiffness was represented by multi-order harmonic series through Fourier expansion with the incremental harmonic balance method [4,9], or the backlash was represented by truncated series expansion [10,11], to study the dynamic load and vibration characteristics. This method is suitable to study the frequency response [4,9] and amplitude frequency diagram [5,9], which was used to predict the root crack failure [5] and to optimize the gear system [9]. The dynamic load response in terms of transmission error was calculated by Karpat et al. [6]. The excitation of gear system based on transmission errors and the maximum meshing loads varying with pinion speed were calculated by Velex and Ajmi [16].

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Nomenclature

b	gear face width, mm
w	load per unit of length, N m^{-1}
m	module, mm
t	temperature, K
τ	time, s
$q(\tau)$	transient heat flux, $W(m\ s)^{-1}$
$k(\tau)$	stiffness at time τ, N m^{-1}
z	coordinate across lubricant film, m
h	film thickness, m
χ	thickness ratio
f	friction coefficient
u_1, u_2	sliding velocity, m s^{-1}
n_1	rotational speed of pinion, r min^{-1}
P	input power, kW
p	pressure, Pa
$\bar{\lambda}$	shear stress, N m^{-2}
η	viscosity of lubricant, N s m^{-3}
r_{b1}, r_{b2}	base radius of pinion and gear, m
b_0	half width of the Hertzian line contact, m
P_H	maximum Hertzian pressure, Pa
α	Braus viscosity–pressure coefficient, m^2 N
β	Reynolds viscosity–temperature coefficient, K^{-1}
R_1,R_2	radii of curvature of pinion and gear, m
R	curvature sum, $R = R_1R_2/(R_1+R_2)$, m
$\lambda, \lambda_1, \lambda_2$	thermal conductivities of lubricant and solids, $W(m\ K)^{-1}$
ρ, ρ_1, ρ_2	densities of lubricant and solids, kg m^{-3}
c, c_1, c_2	specific heat of lubricant and solids, J kg K^{-1}
x	contact coordinate in rolling direction ,m
$\Delta\tau$	half Hertzian time width, s
ζ	damping ratio
T	the meshing cycle, s
κ	heat flux distribution ratio
R_{a1}, R_{a2}	roughness of pinion and gear
$x(\tau)$	transmission error, m
t_M	bulk temperature, °C
t_s	scuffing temperature, °C
ω_n	natural frequency
m_e	equivalent inertia mass of gear pair
S_B	safety factor of scuffing load capacity
E	equivalent Young's modulus, $E = 2/((1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2)$, GPa
$k_1(\tau), k_2(\tau)$	stiffness of the first and second pair of gear system, N m^{-1}
z_1, z_2	teeth number of pinion and gear
J_1,J_2	polar mass moments of inertia of pinion and gear
t_f	flash temperature, °C
$c(\tau)$	damping of gear system
θ_1, θ_2	angular displacement of active pinion and gear
d_0	inner diameter
$\overline{d}$	average value of the dedendum and addendum circle diameter

The elastohydrodynamic lubrication (EHL) supplied a method to study the tribological behavior of contact region of gear systems. A multitude of studies have been carried out on the spur gear systems [17–25]. A considerable number of literatures are focused on the equilibrium EHL [17,18] without considering transient effect. However, for gear systems, the conditions are by no means steady due to the load, the relative curvature and the relative velocity changing during gear meshing process. The transient EHL was carried out in many literatures [19–25]. In 1997, Larsson [19] presented a solution for the lubricated line contact problem under a square-wave load, to calculate the film thickness, pressure, and friction in the contact area of two gear teeth. Later on, Li and Kahraman [24] proposed a transient, non-Newtonian, mixed EHL model for involute spur gear based on the load distribution from ISO standard [1]. The similar method was used in the research of Wang [21] and Bobach [22] as well as in some other literatures [23,24]. The transient EHL based on the load deduced under condition of equilibrium state cannot totally reflect the characteristics of transient EHL.

The combination of transient EHL and dynamic analysis was developed to describe the lubrication problems [26–30]. In 1981, Wang and Cheng made a comprehensive study on numerical simulation of the contact conditions of straight spur gear pairs [26,27]. In their study, the gear dynamics, lubricant film thickness and flash temperature were comprehensively analyzed to research the lubrication performance of gear systems. Li and Kahraman [28,29] proposed an EHL model of a gear pair which was used in conjunction with a dynamic model to predict the lubrication behavior under various dynamic speed conditions; in this study, the predicted instantaneous tooth load was fed into the gear EHL model to simulate the lubrication behavior of the spur gear contact under the dynamic loading condition. Furthermore, the instantaneous pressure and film thickness were calculated with this method. De la Cruz et al. [30] proposed a full transmission model, comprising system dynamics, lubricated contacts, asperity interactions and thermal balance, to analyze the transmission efficiency and operational refinement.

The lubrication properties of contact region are closely related with scuffing failure. The high contact temperature or thin lubricant film may lead to break-down of the lubricant film. So the scuffing load capacity is always an important indicator in the process of designing and checking of gear systems [31–36]. Seabra et al. [31–34] evaluated the influence of the operating conditions, gear geometry and base oil viscosity on gear scuffing as well as studied the scuffing criteria based on the FZG gears. Tuszynski et al. [36] presented a new test method to test scuffing resistance of gears. Nuruzzaman et al. [37] investigated the effects of contact pressure, rolling speed and slip ratio on the minimum oil film thickness.

The dynamic load and transient TEHL model were combined to study the scuffing load capacity of spur gear systems in this paper. More details on thermal analysis of contact process were carried out. The distribution of temperature varying over time in contact region was calculated. A TEHL model which is more in line with the actual conditions was put forward and the hydrodynamic pressure, film thickness and contact temperature as well as the flash temperature under dynamic condition were calculated with numerical method. The scuffing load capacity safety factor was calculated based on TEHL theory. A comprehensive analysis and comparison between the THEL and Blok theory were implemented.

2. The dynamic load of the gear system

2.1. The quasi-static model of gear system

In the quasi-static model, the inertia mass and damping effect of gear systems were ignored at any meshing moment and we also assumed that the distribution of load only depended on the

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