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Research paper

An adhesive wear prediction method for double helical gears based on enhanced coordinate transformation and generalized sliding distance model

Changjiang Zhou[a,b,*], Hongbing Wang[a]

[a] State Key Laboratory of Advanced Design and Manufacturing for Vehicle Body, Hunan University, Changsha 410082, PR China
[b] State Key Laboratory of High Performance Complex Manufacturing, Central South University, Changsha 410083, PR China

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ABSTRACT

An adhesive wear prediction method for double helical gears is proposed according to enhanced coordinate transformation and generalized sliding distance model in conjunction with Archard's wear equation. To describe transient contact ellipse and identify the contact point pairs conveniently, a transform coordinate plane is set in coincidence with the plane of action and a coordinate axe parallels to the contact line. The contact pressure distribution is determined by contact line length, contact width and normal force, and a modified sliding distance model is proposed by generalized moving distance replacement of Hertz contact width. As the wear coefficient, contact pressure and sliding distance are given, the tooth wear depths are predicted by a developed numerical procedure. Effects of major geometrical and working parameters on the wear depth are investigated. The results show that the wear depth becomes smaller, which is mainly determined by the contact force per unit length, equivalent curvature radius and sliding distance as normal module, normal pressure angle, helix angle, tooth width or transmission ratio increases. However, the wear depth becomes larger when input torque is improved. It is indicated that rational parameters match in gear design and uniform wear distribution are beneficial for wear resistance.

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1. Introduction

Owing to their smooth transmission, high load capacity and small axial force, double helical gears are widely applied in the main reducers of heavy helicopters and warships, and in the increasers of energy equipment [1]. For the double helical gears with heavy haul and larger width diameter ratio, welding (slight scratch, scuffing or severe adhesive wear) tends to be developed by high contact pressure and sliding speed. Adhesive wear, which is closely related to the contact pressure, oil film fracture and material plasticizing on the tooth surface, is a main failure mode according to the experiments [2]. While the tooth surface and meshing stiffness are modified by adhesive wear, the transmission precision and dynamic performance are degraded [3,4]. Therefore, a prediction model for adhesive wear of double helical gears and a practicable design solution to wear reduction are necessarily investigated.

* Corresponding author at: Hunan University, Changsha, Yuelu District 410082, PR China.
E-mail address: yangtsezhou@hnu.edu.cn (C. Zhou).

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Nomenclature

B tooth width (mm)
B_w groove width (mm)
B_i centre of contact ellipse
b half-width of Hertzian contact (mm)
C centre of the cutting fillet
E' effective Young's modulus (Pa)
F normal force on a single tooth pair (N)
f length of the plane of action (mm)
F_z total normal force of the mating tooth pairs (N)
h wear depth (μm)
h_{ij} wear depth of a contact point ij on the tooth surface of the driving pinion (μm)
k wear coefficient (m^2/N)
$L(i, t)$ the length of the i_{th} contact line at the time of t (mm)
$L_a(t)$ the contact line length of a single tooth pair at the time of t (mm)
L_z total length of contact line (mm)
M arbitrary point on the machined tooth profile in the static coordinate system
M' arbitrary point on the hob profile in the dynamic coordinate system
M_1' an arbitrary point on the fillet
M_2' an arbitrary point on the straight cutting edge
m module (mm)
m_n normal module (mm)
N relative instantaneous center of two conjugate points
N_1 intersection of the normal line $M_1'N_1$ and the coordinate axis O_0Y_0
N_2 intersection of the normal line $M_2'N_2$ and the coordinate axis O_0Y_0
n total number of the contact lines
P contact pressure (Pa)
P_{ij} contact pressure of a contact point ij on the tooth surface of the driving pinion (Pa)
P_{bt} base pitch of transverse face (mm)
P_0 intersection of the coordinate axis OX and the reference circle of the machined tooth
Q equality divided part number of rotation angle of the driving pinion in a mesh cycle
R_o cutter fillet radius (mm)
r radius of the reference circle of gear blank (mm)
r_{b1} radius of the base circle of the driving pinion (mm)
S sliding distance (mm)
S_a move distance (mm)
S_{ij} sliding distance of a contact point ij on the tooth surface of the driving pinion (mm)
T_{in} input torque (N·m)
t engagement time (s)
V_b speed of the contact line along the transverse plane of helical gear (mm/s)
V_y moving speed of the contact line along tooth width (mm/s)
x_m modification coefficient
Z_p tooth number of driving pinion
Z_g tooth number of driven gear
α_n normal pressure angle (°)
α' working pressure angle (°)
β helical angle (°)
β_b helix angle of base circle (°)
θ_p rotation angle of the driving gear in a mesh cycle (°)
$\Delta\theta_p$ angle increment of the driving pinion (°)
$\Delta\theta_g$ angle increment of the driven gear (°)
Δz movement distance of the ellipse along the negative direction of Z axis (mm)
φ roll angle of the hob (°)
ω_1 angular velocity of the driving pinion (rad/s)
$(\mathbf{X}_{ij})^p{}_q$ position vector of the driving pinion tooth surface at q
$(\mathbf{X}_{ij})^p{}_{q=0}$ initial position vector of the driving pinion tooth surface

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