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Research paper

An efficient model of load distribution for helical gears with modification and misalignment

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ABSTRACT

An efficient model is introduced for evaluating the load distribution of helical gears, considering tooth modifications and misalignment errors. Instant contact points are obtained by unloaded meshing simulation. Combined with the full numerical method for elliptical Hertzian contact, the contact line under load is determined. Then the load distribution is derived from the minimization of potential energy. The proposed model is numerically implemented by a fixed-point iteration method based computational scheme, assuring high efficiency and superior versatility for tackling modifications and misalignments. And it is verified by finite element analysis. The effects of profile crowning, lead modification, misalignment and input torque on load distribution are investigated. Results indicate that contact pattern shrinks with increasing magnitude of profile crowning and decreasing input torque, resulting in abrupt load transitions between two and three meshing tooth pairs. While lead modification can transfer load from the edge of tooth surface to the center by inclining contact pattern to longitudinal direction, providing desirable load distribution. In addition, misalignment deviates contact pattern and sharing load respectively towards the ends of tooth profile and meshing cycle.

1. Introduction

Helical gears have been widely applied in modern transmission systems due to their features regarding compact structure, high load capacity and excellent meshing performance. The evaluation of load distribution plays a crucial role in helical gear design, which paves the way for the analyses of contact stress, lubrication, dynamics and efficiency. Whereas it is characterized by arduousness because of the non-uniformity of load distribution along contact line. This is generally attributed to elastic deformations, tooth modifications, manufacturing and assembly errors. The current methods are either too simplified or too complicated and time-consuming. To address such a problem, this paper presents an efficient model to determine load distribution with consideration of modifications and misalignments.

A great deal of research effort has been devoted to the evaluation of load distribution of helical gears [1–8]. Different methods broadly fall into three categories according to their nature: analytical methods, numerical methods and experimental methods.

For the analytical methods, ISO [9] and AGMA [10] standards employed a load distribution factor in empirical formulas to account for the effect of non-uniform load distribution along contact line. Chen [11] derived load distribution on the

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Nomenclature

a_i	profile parabola coefficient ($i=1$ for the pinion, $i=2$ for the gear), mm^{-1}
$a_{pl}{}^i$, $a_{mr}{}^i$	longitudinal parabola coefficient ($i=1$ for the pinion, $i=2$ for the gear), mm^{-1}
B_i, e_{begin}, e_{end}	boundary values in program ($i=1, 2, 3, 4$)
b	tooth width, *mm*
C_s	shear potential correction factor
E	modulus of elasticity, *MPa*
E', E_{wi}, E'_{wi}	center distance ($i=1$ for the pinion, $i=2$ for the gear), *mm*
e	ellipticity
e_r	normal space width on the midline of rack cutter, *mm*
F	load, *N*
G	transverse modulus of elasticity, *MPa*
$\mathbf{M}_{ij}$	matrix of coordinate transformation from S_j to S_i
m_n	normal modules, *mm*
N_i	model parameters ($i=L, s, m$)
$\mathbf{n}_i$	normal of surface in coordinate system S_i
P	transmitted power, *kW*
p_i	screw parameter ($i=1$ for the pinion, $i=2$ for the gear), *mm/rad*
r_b	radius of base circle, *mm*
r_c	radius of contact point, *mm*
r_f	radius of tooth root circle, *mm*
$\mathbf{r}_i$	position vector of a point in coordinate system S_i
S_c	displacement of rack cutter, *mm*
T_{in}	input torque, *N·m*
U	elastic potential energy, *N·mm*
u	unitary potential, mm^2/N
$u_o{}^i$	vertex position of parabolic profile ($i=1$ for the pinion, $i=2$ for the gear), *mm*
u_i, l_i	surface parameters of rack cutter ($i=1$ for the pinion, $i=2$ for the gear), *mm*
v	inverse unitary potential, N/mm^2
y	coordinate along the tooth centreline from the gear rotation center, *mm*
z_i	number of teeth ($i=1$ for the pinion, $i=2$ for the gear)
α_c	load angle, *rad*
α_n	normal pressure angle, *rad*
β	standard helix angle, *rad*
β_b	base helix angle, *rad*
γ_c	tooth angular thickness of contact point, *rad*
γ_{wi}	crossing angle ($i=1$ for the pinion, $i=2$ for the gear), *rad*
ε, ε_i	error tolerances ($i=1, 2$)
$\boldsymbol{\eta}$	unit vector
θ	auxiliary angle, *rad*
κ	load sharing ratio
ξ	profile parameter
φ_i, φ_i'	rotation angle of gears ($i=1$ for the pinion, $i=2$ for the gear), *rad*
φ_{wi}	rotation angle of grinding worm ($i=1$ for the pinion, $i=2$ for the gear), *rad*
ω	angular velocity, *rad/s*
$\triangle E'$	center distance error, *mm*
$\triangle e$	step size in search for ellipticity
$\triangle L$	axial deviation error, *mm*
$\triangle S_{wi}$	translational motion of grinding worm ($i=1$ for the pinion, $i=2$ for the gear), *mm*
$\triangle\gamma$	shaft angle error

assumption that the load sharing between the meshing tooth pairs was proportional to the length of contact line. Jamali [12] obtained the load boundary condition for lubrication analysis based on the proportional load sharing theory.

The ever-improving computer technologies are creating new possibilities for realizing complex numerical calculations. A considerable number of research has been conducted on the numerical models of load distribution for helical gears. Conry and Seireg [13] proposed a well-known load distribution model for gears, and a modified simplex-type algorithm based procedure was employed to solve the deformation compatibility equations. Afterwards, Wink [14] presented three different solution methods for this model, and compared them in terms of accuracy and computational effort. Furthermore, Zhang