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Research paper

Method for calculating the tooth root stress of plastic spur gears meshing with steel gears under consideration of deflection-induced load sharing

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ABSTRACT

The operational behavior and, in particular, the bending strength of thermoplastic gears are substantially influenced by the significant increase in contact ratio under load. According to current analytic calculation standards and guidelines, this effect is neglected when tooth root stress is calculated. Using numerical methods, such as the finite element method (FEM), a consideration of deflections is possible but the usage is complex and extensive in engineering practice compared to recomputing guidelines or standards like VDI 2736 or ISO 6336. In this work, a method is presented for calculating the nominal tooth root stress based on existing analytic guidelines, taking the actual contact ratio into account. Based on the example of three test gear geometries, the results are compared to existing guidelines, as well as numerical computations, showing that the presented method correlates well with the latter.

1. Introduction

Motivation. Gear teeth are deformed under load, causing the actual contact ratio of loaded gears to be higher than the transverse contact ratio ε_α which is used to model stress conditions according to DIN 3990 [1] / ISO 6336 [2]. For steel gears with basic rack tooth profiles [3] the resulting deflections are often negligible, allowing tooth root stress to be calculated sufficiently accurately. A noteworthy increase in actual contact ratio may lead to reduced tooth root stresses compared to the rigid body-based model [4] of the standards [1,2,5]. But, as the phenomenon is connected with meshing interferences, the effect is generally critical with regard to contact load at the gear flank. To avoid preliminary flank damages when steel gears are used, it is common for the flank profile to be modified in order to compensate for load-induced increases to the contact ratio.

Thermoplastic gears tend to cause even higher increases in actual contact ratio because of the considerably higher yield strain of the materials compared to steel. Additionally the deflection-induced increase of flank loading due to meshing interferences is not as critical as for steel gears because of the distinct expansibility of thermoplastics. Experiments with

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Nomenclature

a	mm	center distance
a_{ACR}	-	parameter for calculating modified contact ratio factor $Y_{\varepsilon,\,ACR}$
α_n	°	normal pressure angle [17]
α_{wt}	°	working pressure angle [17]
b	mm	face width
c_{abort}	-	constant: abortion criteria
c'	$\frac{N}{mm \cdot \mu m}$	single stiffness [1]
c_{BS}	-	basic rack profile factor [1]
c'_C	$\frac{N}{mm \cdot \mu m}$	single stiffness according to DIN 3990 [1] Method C
c_γ	$\frac{N}{mm \cdot \mu m}$	mesh stiffness [1]
$c_{\gamma,\,w}$	$\frac{N}{mm \cdot \mu m}$	modified mesh stiffness [9]
$c_{\gamma,\,ACR}$	$\frac{N}{mm \cdot \mu m}$	modified mesh stiffness according to ACORA
d_a	mm	tip diameter [17]
E	N/mm^2	(short term) Young's modulus
ε_α	-	transverse contact ratio [17]
ε_β	-	overlap ratio [17]
$\Delta\varepsilon_{\alpha 1}$	-	increase in contact ratio due to posterior meshing[4]
$\Delta\varepsilon_{\alpha 2}$	-	increase in contact ratio due to pre meshing[4]
$\Delta\varepsilon_{\alpha 1+2}$	-	increase in contact ratio due to pre and posterior meshing
$\varepsilon_{\alpha,\,w}$	-	actual contact ratio [10,4,9]
f_{Th}	mm	adequate deflection in meshing direction[4]
$f_{\varepsilon\beta}$	-	correction factor overlap ratio [10]
F_{tb}	N	tangential load at base circle [1]
F_t	N	nominal tangential force [1,5]
h	mm	tooth depth [17]
h^*_{aP0}	-	addendum factor of tool
h^*_{fP0}	-	dedendum factor of tool
m_n	mm	normal module [1]
r_a	mm	tip radius ($= 0.5 \cdot d_a$) [17]
r_b	mm	base circle radius [17]
r_{Nf}	mm	$1/2 \cdot$ effective root diameter (=radius)
ρ^*_{aP0}	-	tip radius coefficient of tool
σ_{F0-C}	$\frac{N}{mm^2}$	nominal root stress (Method C) [1,5]
σ_{F0-ACR}	$\frac{N}{mm^2}$	modified nominal root stress
$\sigma_{F,\,FEM}$	$\frac{N}{mm^2}$	maximum root stress according to static FEM
p_{et}	mm	normal base pitch [1]
x	-	addendum modification coefficient [17]
ξ	-	material factor [1]
Y_{Fa}	-	form factor DIN 3990 [1] Method C, [5]
Y_{Sa}	-	stress correction factor DIN 3990 [1] Method C, [5]
Y_β	-	helix angle factor [1]
$Y_{\varepsilon,\,DIN}$	-	contact ratio factor [1,5]
$Y_{\varepsilon,\,DIN,\,w}$	-	contact ratio factor according to [1,5] as a function of $\varepsilon_{\alpha,\,w}$
$Y_{\varepsilon,\,old}$	-	contact ratio factor [12]
$Y_{\varepsilon,\,old,\,w}$	-	contact ratio factor [12] as a function of $\varepsilon_{\alpha,\,w}$
$Y_{\varepsilon,\,Fue}$	-	contact ratio factor [10]
$Y_{\varepsilon,\,ACR}$	-	modified contact ratio factor
z	-	number of teeth
$\zeta, \gamma, \tau, \varphi, \alpha_{...}$	rad	auxiliary angles according to Thoma [4]

Indices

1	pinion
2	wheel

plastic gears prove that, compared to (nominal) transverse contact ratio, significantly higher actual contact ratios during test runs cause tooth root breakage without critical flank damage [6,7].