

A Method to Define Profile Modification of Spur Gear and Minimize the Transmission Error

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A Method to Define Profile Modification of Spur Gear and Minimize the Transmission Error

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Introduction

In this paper a simple method to minimize the Peak to Peak Transmission Error (*PPTE*) for a spur gear set is proposed.

A parametric analysis using advanced software was performed to obtain a general understanding of the problem. Optimization variables were reduced to the Start Relief Roll Angles.

A parabolic topography was assumed for the tip relief and either Low Contact Ratio (LCR) or High Contact Ratio (HCR) spur gears have been considered. Meshing related constraints on optimization have been taken into account. On the basis of parametric analysis an effective optimization procedure has been suggested for LCR that can be extended even for HCR.

Nomenclature

P	Diametral Pitch
b	Face width
h	Tooth height
s	Coordinate along profile
TE	Transmission Error
$PPTE$	Peak to Peak TE
θ_P	Pinion start relief roll angle
θ_G	Gear start relief roll angle
v_e^{Pinion}	Pinion relief amount
v_e^{Gear}	Gear relief amount
v_e	Total relief amount
SAP	Start Active Profile roll angle
EAP	End Active Profile roll angle

Table 1. Nomenclature.

1 Transmission Error Effect on Gear Noise

In a gear set, Transmission Error (*TE*) is defined as the difference between the effective and the ideal position of the output shaft with reference to the input shaft. The ideal position represents a condition of perfect gear box, without geometrical errors and deflections. *TE* can be expressed either by an angular displacement or, more conveniently, as a linear displacement measured along a line of action at base circle according to [1]. *TE* is typically considered to be the primary cause of whining noise.

Indeed, whining noise is produced by:

- changing tooth load amplitude,
- changing load position along the tooth profile,
- changing tooth load direction.

These changes are consequences of tooth deflection, local contact deformation and body deformation quantities, which are the origin of *TE*. Several Authors [2-6] studied the correlation between *TE* and profile modification, in order to reduce the *TE*. Niemann [4] proposed *long* and *short* modifications.

This different denomination is based on the start point of tip relief modification along the profile.

According to experimental results, gears with long modification have the minimum *PPTE* and the minimum noise level at the design torque. At lower torque this optimum condition was not verified and an intermediate or short modification is suggested. In these works *short* and *long* modifications are considered to be technological limits and it is not useful to extend the relief to the pitch radius, as described in [1,3].

In the present paper, instead, the start tip relief modifications (θ_P , θ_G) have been considered as design variables ranging from Start Active Profile roll angle (*SAP*) up to the End Active Profile roll angle (*EAP*). Total relief v_e (at the tooth tip) is imposed equal to the deflection of tooth pairs under the nominal torque. The topography of the tip relief is parabolic. The conditions for obtaining the minimum *PPTE* configuration are found and discussed.

2 Methodology

The quantities v_e^{Pinion} (v_e^{Gear}) and θ_P (θ_G) are defined in figure 1.

The ranges for the two Roll Angle variables are the Start Active Profile roll angle (*SAP*) and the End Active Profile roll angle (*EAP*) for each gear (figure 2).

Meshing gears simulations have been carried out by means of a hybrid method, combining the Finite Element technique with a semi-analytical solution [7,8].

The main assumptions for the analysis are the following:

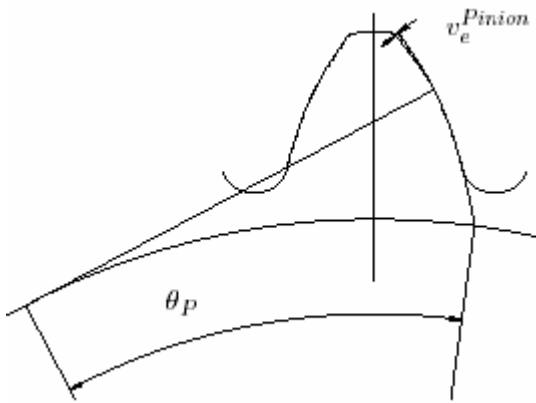


Figure 1. Definition of v_e^{Pinion} (v_e^{Gear}) and $\theta_P(\theta_G)$.

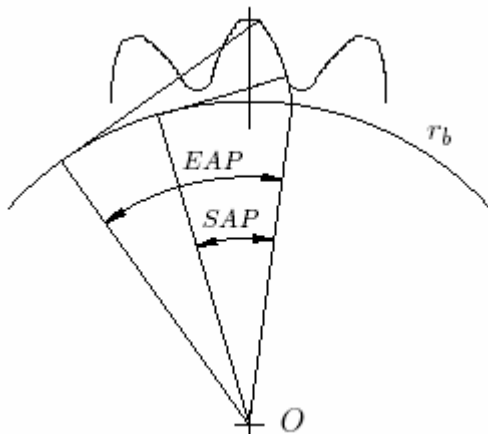


Figure 2. Symbols SAP and EAP .

Plain strain conditions: suggested by the spur gear geometry (high ratio b/h and so not affecting bending deformation behavior). 2D plane strain analysis is adequate for this kind of tooth. Moreover the bi-dimensional version of the software requires little time both for model definition and simulations, with a very high precision of the results.

Static analysis: static TE was determined neglecting rotational speed and inertia forces.

Friction neglected: in order to get rid of the uncertainty about Coulomb frictional coefficient. The friction force is assumed not affecting the TE shape function substantially.

Spacing error and pitch error not considered: the nominal TE was calculated, and no statistical consideration have been performed in the analysis. Otherwise required if geometrical errors considered.

Two kinds of spur gear set have been analyzed, in which the TE function is rather different:

- Low Contact Ratio gears (LCR),
- High Contact Ratio gears (HCR).

For the LCR gear contact ratio is between 1 and 2 and when the contact is at the pitch position, only

one tooth pair is in contact, while for HCR , 3 tooth pairs are in contact at the pitch position.

At the beginning each spur gear set was analyzed by assuming no modification. The maximum deflection of the gear pair tooth, TE_{max} (evaluated along the pinion base circle) was then calculated and imposed equal to the total relief amount.

$$v_e = TE_{max}$$

The amount of the tip relief was equally shared between pinion and gear assuming *long* modification suggested in [4,5].

$$v_e^{Pinion} = v_e^{Gear} = \frac{v_e}{2}$$

Successively, contact pressure was calculated along the meshing zone in order to detect possible corner contacts.

As shown in figure 3 the corner contact can be observed at both sides. v_e^{Gear} has to be increased in order to avoid corner contact at the start of meshing, likewise v_e^{Pinion} has to be increased to avoid the corner contact at the end of meshing.

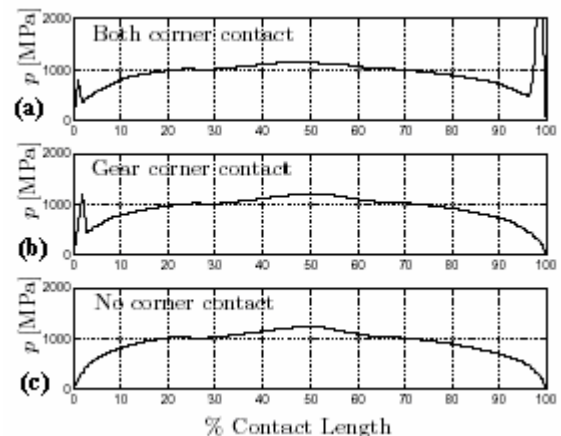


Figure 3. Adapting v_e to avoid Corner Contacts: (a) Corner Contact both at Start and End meshing, (b) Corner Contact only at Start meshing, (c) No Corner Contact detected.

When corner contact was eliminated, pinion relief (v_e^{Pinion}) and gear relief (v_e^{Gear}) were held fixed through the subsequent calculations.

To perform the minimization, an object function has to be defined along the variables and the ranges in which the optimum is searched.

In a similar analysis [9] the object function was related to the Fourier expansion of the TE and the first three harmonics considered. However as shown in figures 4 and 5, in this analysis the first three harmonics could not be the main part of the signal reproducing the TE function.