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Dependency of the Peak-to-Peak-Transm ission-Error on the Type of Profile Correction and the Transverse Contact Ratio of the Gear Pair

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[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Profile corrections on gears are a commonly used method to reduce transmission error, contact shock, and scoring risk. There are different types of profile corrections. It is a known fact, that the type of profile correction used will have a strong influence on the resulting transmission error. The degree of this influence may be determined by calculating tooth loading during mesh. The current method for this calculation is very complicated and time consuming; however, a new approach has been developed which could reduce the calculation time.

This approach uses an algorithm which includes the conventional method for calculating tooth stiffness in regards to bending and shearing deformation, flattening due to Hertzian pressure, and tilting of the tooth in the gear body. The new method was tested by comparing its results with FEM and LVR.

This paper illustrates and discusses the results of this study. Furthermore the maximum local power losses are compared with the scoring safety calculated following the flash temperature criteria of AGMA 925 and DIN 3990.

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Introduction

Profile corrections (PC) on gears are a commonly used method to reduce the transmission error (TE) of a gear pair, the contact shock (corner contact) and the scoring risk. There are different types of profile corrections as short or long linear corrections, short or long corrections in arc form, fully crowned profile and others.

The calculation of the meshing of a gear pair under load is very complicated and therefore time consuming. Over the gear meshing cycle, from the start of contact in the pinion root area to the end of contact on the tip, a sufficient number of steps must be calculated. Using a FEM program, this requires many hours. More specialized programs as LDP or LVR perform this process in much shorter time, but even then the evaluation of different variants needs much time.

Based on a new approach for the calculation of the meshing under load, the calculation time could furthermore be reduced. An algorithm using a conventional method for the calculation of the tooth stiffness considering bending and shearing deformation, flattening due to Hertzian pressure and tilting of the tooth in the gear body, is used. With this approach the calculation of a gear mesh is carried out within seconds. In combination with an efficient user interface, this allowed to perform an extended study of the effect of different profile corrections.

The aim of the study is to analyze the effect produced by short linear, long linear, short arc-like, long arc-like and fully crowned profile corrections on gears with different transverse contact ratio (ε_α), as standard reference profile gears and high tooth profile gears may show very different characteristics when using profile corrections.

Effect of profile corrections on transmission error and noise in literature

Tip relief is applied for two reasons, the first being to

minimize corner contact (tooth interference) and the second being to reduce dynamic excitation (transmission error).

In literature few comments about the effect of different profile corrections (or profile modifications as called by ISO) types can be found. In the American literature some information can be found, mainly from or in connection with research done at the Ohio state university [1]. In the German literature, where many publications about gear and gearbox design exist, only very limited information can be found. For example in the classic Niemann book about cylindrical gear design [3], only few words are reserved on the effect produced by profile corrections. Little bit more is explained in the book of Linke [2] from the Dresden University. Also in UK literature some few specific literature is available [4].

A simple variant of a profile correction is a tip relief. When defining a tip relief, two mayor parameters are important, the tip relief Ca (figure 1) and the relief length L_{Ca} . In literature, everybody agrees, that the tip relief Ca has to be dimensioned in such a way, that the tooth bending and may be a part of manufacturing errors (pitch deviation) is compensated. There is also agreement, that the profile modification affects strongly the Peak-to-Peak-Transmission-Error (PPTE). Furthermore, it is evident that the PPTE is quite directly related to the noise level produced by a gear pair.

There are basically two options for the length L_{Ca} of the profile correction, the so called "short" and "long" relief design (figure 2).

About the optimum length L_{Ca} of the profile correction for best result in reduction of the Transmission Error (TE), the opinions are diverging (table 1).

The indications in literature are partially contradictive. The reason is possibly that the effect of long or short profile correction depends also on the transverse contact ratio of the gear pair.

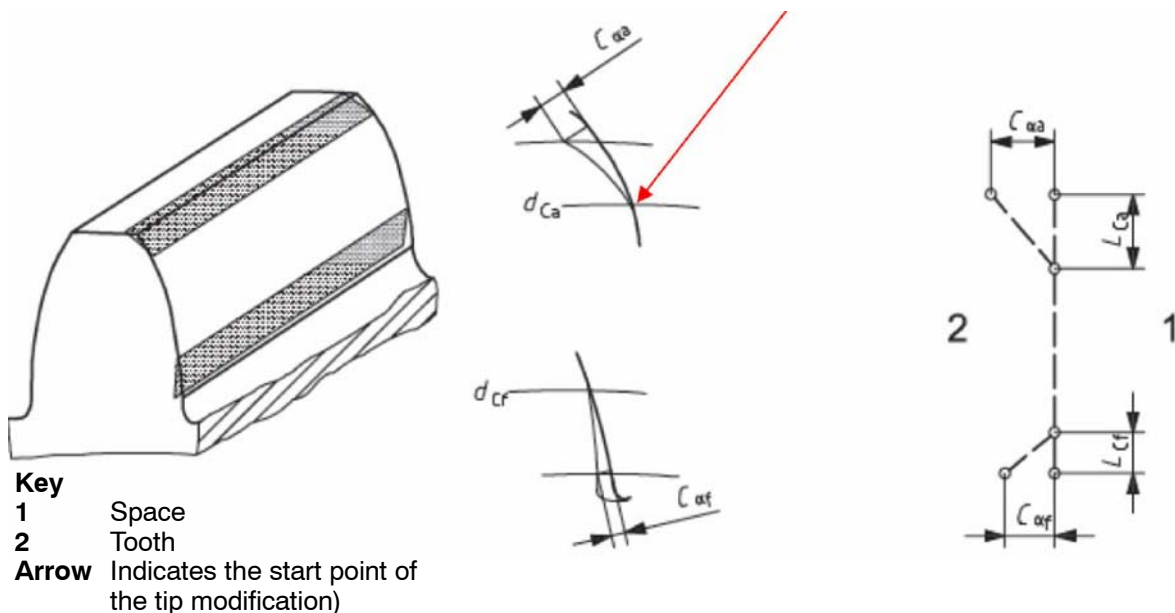


Figure 1. Linear tip and root relief as defined by ISO 21771:2007

“Short” profile correction

“Long” profile correction

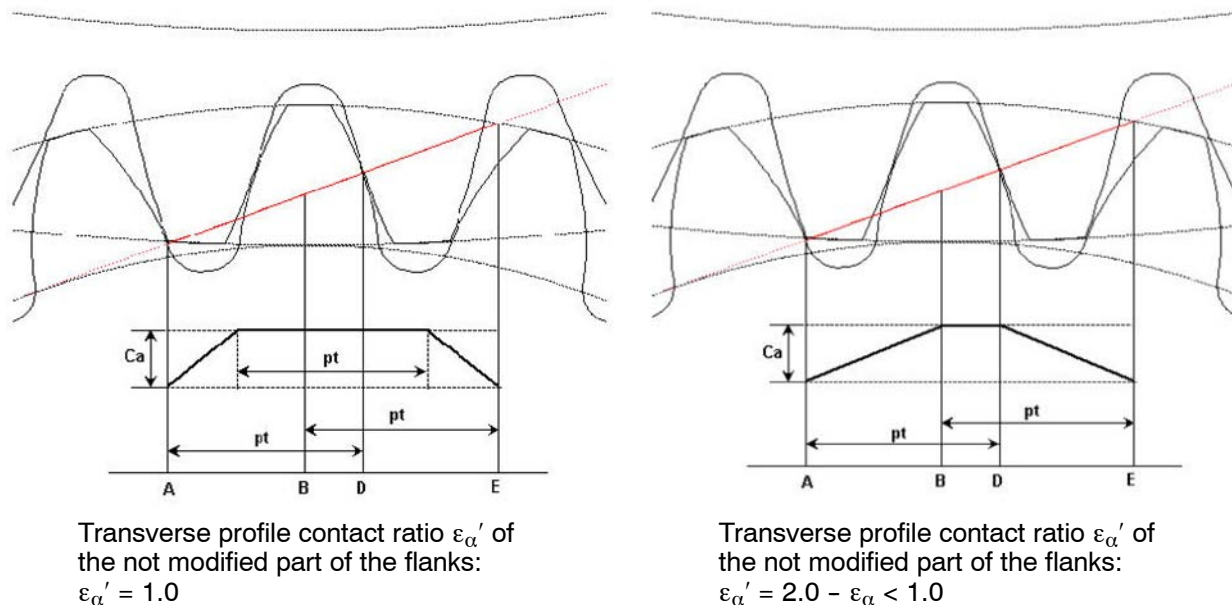


Figure 2. Definition of the short and long profile correction [1, 4]

Table 1. Effects of short or long profile modification in literature

Author	Short profile correction	Long profile correction
Niemann (DE) [3], p.112	Avoids corner contact No effect on TE	Avoids corner contact Reduces TE considerably
Linke (DE) [2], p.465	Avoids corner contact Reduces TE	Avoids corner contact Reduces TE, but is worse for low load
Houser (USA) [1], p.25	Avoids corner contact No effect on TE	Avoids corner contact Reduces TE considerably at design load, but is worse for low load
Smith (UK) [4], p.58	Reduces TE for low load	Reduces TE for high load

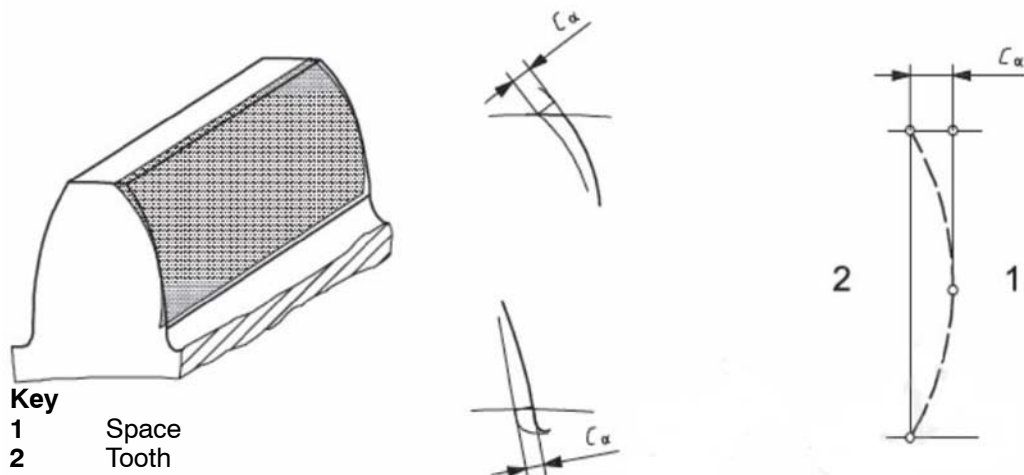


Figure 3. Profile crowning (barrelling) as defined by ISO 21771:2007

It is also astonishing that in the literature few or none indications are given for the best type of curve to use for the profile correction. There are different possibilities. The simplest is a linear tip relief on both gears (or linear tip and root relief on one or on both gears) as in figure 1. Often used, for example in automotive gearboxes, are crowned profiles (figure 3). Another variant is a parabolic or an arc-like tip (and/or root) relief (figure 4). Compared to the linear relief these types of correction have the advantage that the pressure angle of the profile does not have an instant change at the start point of the modification (figure 4). Munro/Houser [1] are using a parabolic correction, Smith [4] is using profile crowning, Niemann [3] most probably a linear correction. But it is not discussed, if the type of curve has a big influence on the transmission error.

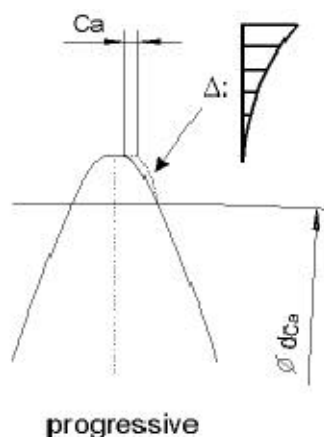


Figure 4. Arc-like or parabolic profile modification at tip

A perfect involute gear pair with infinite stiffness has no transmission error. For the consideration of the effect of profile corrections the bending of the teeth must be included. This is not a simple calculation task.

The calculation of path of contact under load and the transmission error (TE)

To get the TE of a gear set during meshing, the contact path under load is calculated. This means a contact problem must be solved: The number of tooth pairs in contact varies by one during the meshing, most often it changes from one pair to two pairs in contact. This effect causes the total stiffness in the engagement to change periodically (figure 5). The teeth themselves are deflected due to the torque applied. This shifts the point of change from one to two pairs in contact a small amount and leads to a premature contact.

There are often two different approaches to solve a problem in mechanical engineering, the very general finite element method (FEM), and the specific classical methods available for most of the common machine parts. The classical methods are tailored to one specific type of part only, e.g. bolts, gears or bearings. The advantage of these methods is their fast and easy application. In many cases no classical method is available, for instance for housings. In these cases the application of FEM is the only possibility. In other cases the application of FEM would be much too expensive, like for a key and key way on a shaft.

Based on a dissertation by Peterson [5] proposing a classical method for the calculation of the tooth stiffness, it was possible to develop a quick and accurate method to solve this problem [6]. For the calculation of the stiffness, the model according to Peterson covers the deflection of the teeth, the bending of the teeth in the wheel body, the Hertzian pressure, and the deformation due to shearing. The gear is cut into several transverse sections and the stiffness is calculated for these slices. For a spur gear the stiffness is multiplied by the width which leads to the final value. For a helical gear the beginning and end of contact of the slices is dependent upon the position of the slices along the tooth width. The final course of the total stiffness is calculated by integrating the stiffness functions for the slices over the width, with increasing delay of the begin of contact. Figure 5 shows a graphical representation of the model: a spring is fixed on the path of contact, which means that it is located on the common tangent of the two base circles of the gears. This spring has a periodically changing stiffness $c(t)$. If in this model the pinion is rotating with constant speed and torque, and the output torque is constant on the gear, the spring will be deflected periodically. This deflection is the transmission error, typically quantified in micrometers.

In the simulation of the meshing the deflection of the teeth is given by the normal force applied to a single tooth divided by the stiffness. Since the point where

the force is applied varies in the height direction, the stiffness will also depend on the meshing position. Further, if the second pair teeth comes into contact the stiffness increases sharply, so the deflection of the first pair of teeth is reduced. To find the correct point of contact, an iteration must be performed.

The reward for all this effort is the calculation of the real path of contact under load, the course of the normal force on the flanks, the transmission error, the stress in the root areas of the teeth, the Hertzian pressure, sliding velocities, the local warming up (flash temperature) and even the prediction of local wear on the tooth flanks.

Figure 6 shows an example of the effect of the tip relief: in the top left diagram the path of contact for the full involute gears is shown. In the middle part the path is a straight line. In the section of contact start and end, however, the line is curved, and the contact follows the tip circle of the pinion or the gear. This is the region of the prolonged contact (corner contact). The tip relief is designed to compensate the deflection of the teeth and thus eliminate the premature contact. This, however, only works for a specific torque applied exactly. The right column of diagrams in figure 6 shows the influence of the short linear tip relief. The path of contact is almost a straight again. The only deviation is a nick in the region of the beginning of the tip relief. This marks the rapid change of the pressure angle at this point (where the linear tip relief starts).

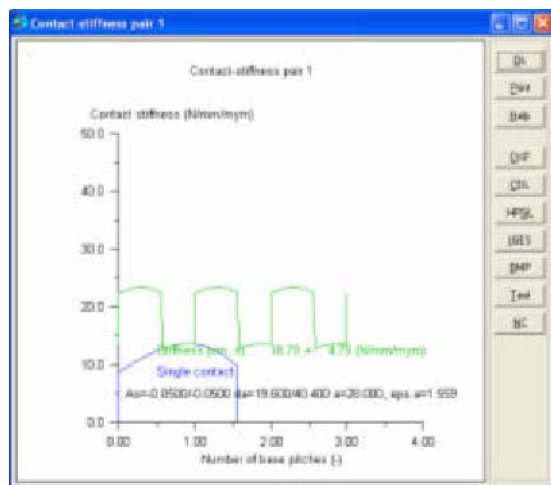


Figure 5. Typical course of the stiffness, and the model with a spring