Effects of Profile Corrections on Peak-to-Peak Transmission Error

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Management Summary

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 that could reduce the calculation time.

This approach uses an algorithm that 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. This new method was tested by comparing its results with Finite Element Method (FEM) and *LVR* software.

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 AGMA925 and DIN 3990.

Introduction

Profile correction (PC) of gears is 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: 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 an 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 be even further 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 for an extended study of the effect of different profile corrections.

The aim of this 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—to minimize corner contact (tooth interference) and to reduce dynamic excitation (transmission error).

In literature, few if any conclusions about the effect of different profile corrections (or profile "modifications," as called by ISO) can be found. Some information exists in the American literature, mainly from or in connection with research done at the Ohio State University (Ref. 1). And in the German literature, where many publications about gear and gearbox design exist, scant information is found. For example, in the classic Niemann book about cylindrical gear design (Ref. 3), only a few words are devoted to the effect produced by profile corrections. Little more is explained in the book by Linke (Ref. 2), from Dresden University. And in the U.K., some specific literature is available (Ref. 4).

A simple variant of a profile correction is a tip relief. When defining a tip relief, two major parameters are important—the tip relief *Ca* (Fig. 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 perhaps some part of manufacturing errors (pitch deviation)—are compensated. There is also agreement that the profile modification strongly impacts 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 designs (Fig. 2).

As for the optimum length L_{Ca} of the profile correction and thus the best result in reduction of the TE, opinions differ (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. It is also astonishing that in the literature few or no 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 both

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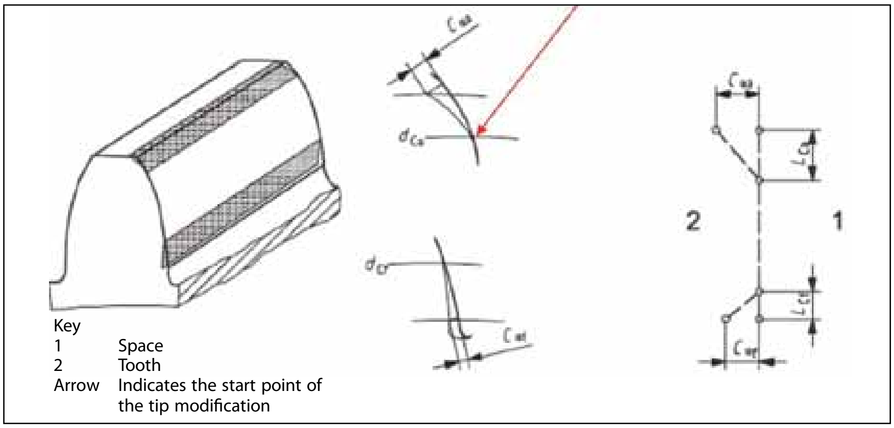

Figure 1—Linear tip and root relief as defined by ISO 21771:2007.

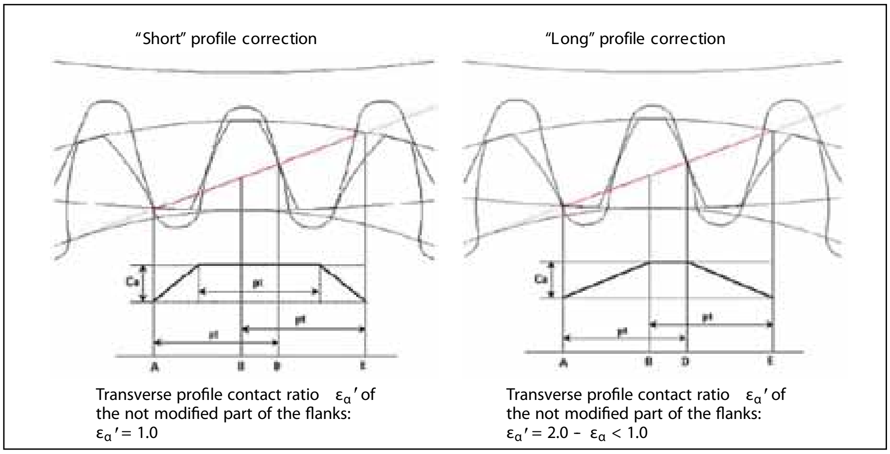

Figure 2—Definition of short and long profile correction (Refs. 1 and 4).

Table 1—Effects of Short or Long Profile Modification in Literature.

Author	Short profile correction	Long profile correction
Niemann (Ref. 3) p.112	Avoid corner contact No effect on TE	Avoids corner contact Reduces TE considerably
Linke (Ref. 2) p.465	Avoids corner contact Reduces TE	Avoids corner contact Reduces TE, but is worse for low load
Houser (Ref.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 (Ref. 4) p.58	Reduces TE for low load	Reduces TE for high load

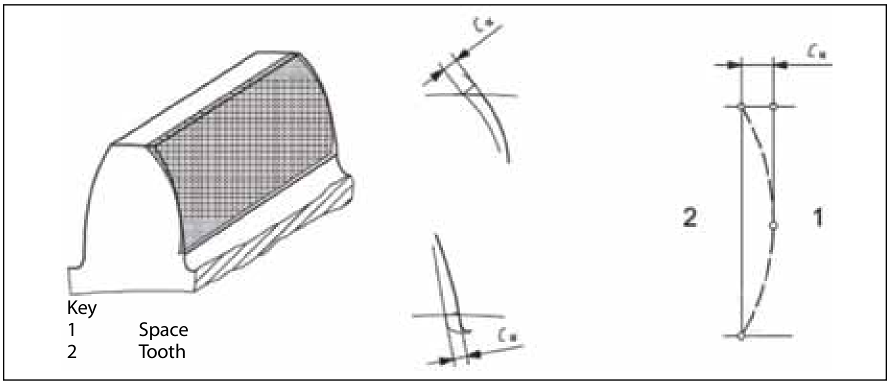

Figure 3—Profile crowning (barreling) as defined by ISO 21771:2007.

gears) (Fig. 1). Crowned profiles, for example, are often used in automotive gearboxes (Fig. 3). Another variant is a parabolic or arc-like tip (and/or root) relief (Fig. 4). Compared to the linear relief, these types of corrections have an advantage in that the pressure angle of the profile does not have an instant change at the start point of the modification (Fig. 4). Munro and Houser (Ref. 1) are using a parabolic correction, Smith (Ref. 4) is using profile crowning and Niemann (Ref. 3), most probably, a linear correction. But it is not discussed whether this type of curve has a major influence on transmission error.

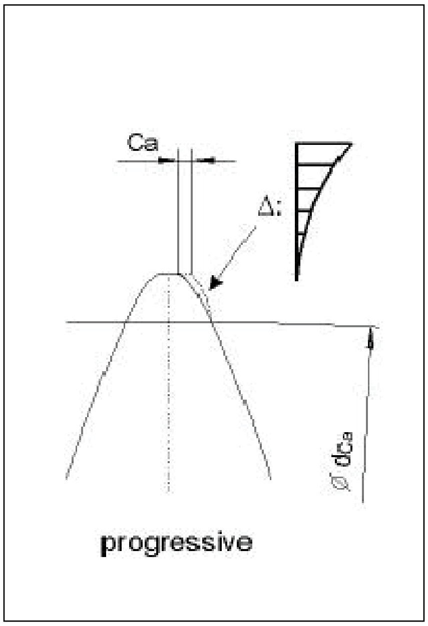

Figure 4—Arc-like or parabolic profile modification at tip.

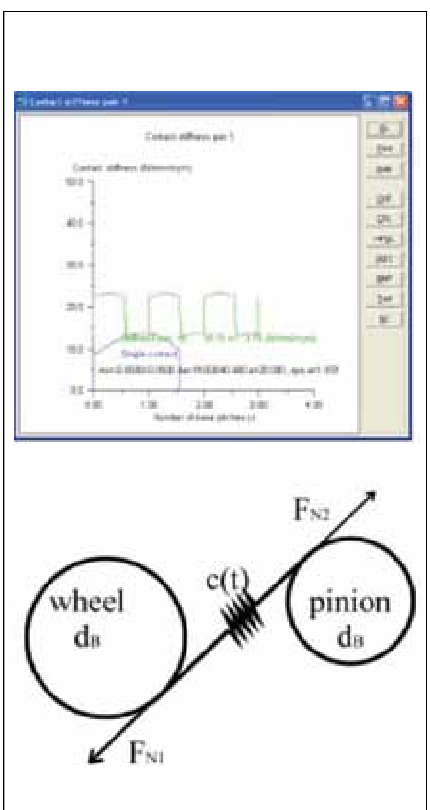

Figure 5—Typical course of the stiffness; model with spring.

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.

Calculation of Path of Contact Under Load and the 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—i.e., the number of tooth pairs in contact varies by one during the meshing and, most often, it changes from one pair to two pairs in contact. This effect causes the total stiffness in the engagement to change periodically (Fig. 5). The teeth themselves are deflected due to the torque applied, thus shifting the point of change from one to two pairs in contact and leading to premature contact.

There are often two different approaches in solving a problem in mechanical engineering—1) the very general FEM method and 2) the specific classical methods available for most of the common machine parts. The classical methods are tailored to one specific type of part—i.e., bolts, gears or bearings. The advantage of these methods lies in their fast and easy application. But in many cases, no classical method is available. Consider housings, for example, where 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 keyway on a shaft.

On the strength of a dissertation by Peterson (Ref. 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 (Ref. 6). For the calculation of the stiffness, Peterson's model covers the deflection of the teeth, the bending of the teeth in the wheel body, the Hertzian pressure and the shearing-induced deformation. 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, while increasing the delay of initial 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 of teeth comes into contact, stiffness increases sharply and 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:

- Calculation of the real path of contact under load
- Course of the normal force on the flanks
- Determining TE, stress in the root areas of the teeth, Hertzian pressure, sliding velocities, local warming up (flash temperature) and 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 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, precisely applied. The right column of diagrams in Figure 6 shows the influence of the short linear tip relief. The path of contact is almost 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).

The course of the normal force for the gears without tip relief shows a typical picture for spur gears in that in the middle of the contact path, only one pair of gears takes the full torque. Before and after that, the normal force is shared over two pairs of teeth and thus only about 50% of the maximum value in the middle. The gears with tip relief have only one pair of teeth in contact at most times, so here the normal force is nearly constant and yet on the same level as the maximum of the gears without tip relief. Nevertheless, the maximum pressure on the flank is 20% less with tip relief since the premature contact leads to a contact shock with very high pressure. Finally, the amplitude of the transmission error (PPTE) remains the same with this type of tip relief. The tip relief results in a smoother course so that the higher frequencies are reduced. This leads to less acceleration and the smaller forces induced by the transmission error.

To check this calculation method, the same gears were calculated with *ANSYS* (Fig. 7). Both methods lead to very similar results. For the Hertzian pressure, the FEM results tend to zigzag more, caused mainly by the fact that the defined stress is given for a single point on a grid. Since the real contact point is usually somewhere between two grid points, the real maximum stress on the flank is usually larger than the plotted result.

Other comparisons were made with the *LVR* program (from Dresden University); these results also show a very good correlation.

Since the calculated results are very similar, the main difference between the two methods is the disparity in effort expended to achieve them. Consider: it took two days to get the FEM model set up, calculate the stresses and extract them for presentation; with *KISSsoft* (Ref. 7), the same task was accomplished in two seconds. Moreover, each variant for the tooth form—such as a different amount of tip relief, different geometry (such as changed addendum modification) or different tooling takes only a few minutes to analyze. This demonstrates the advantages of the classical approach.

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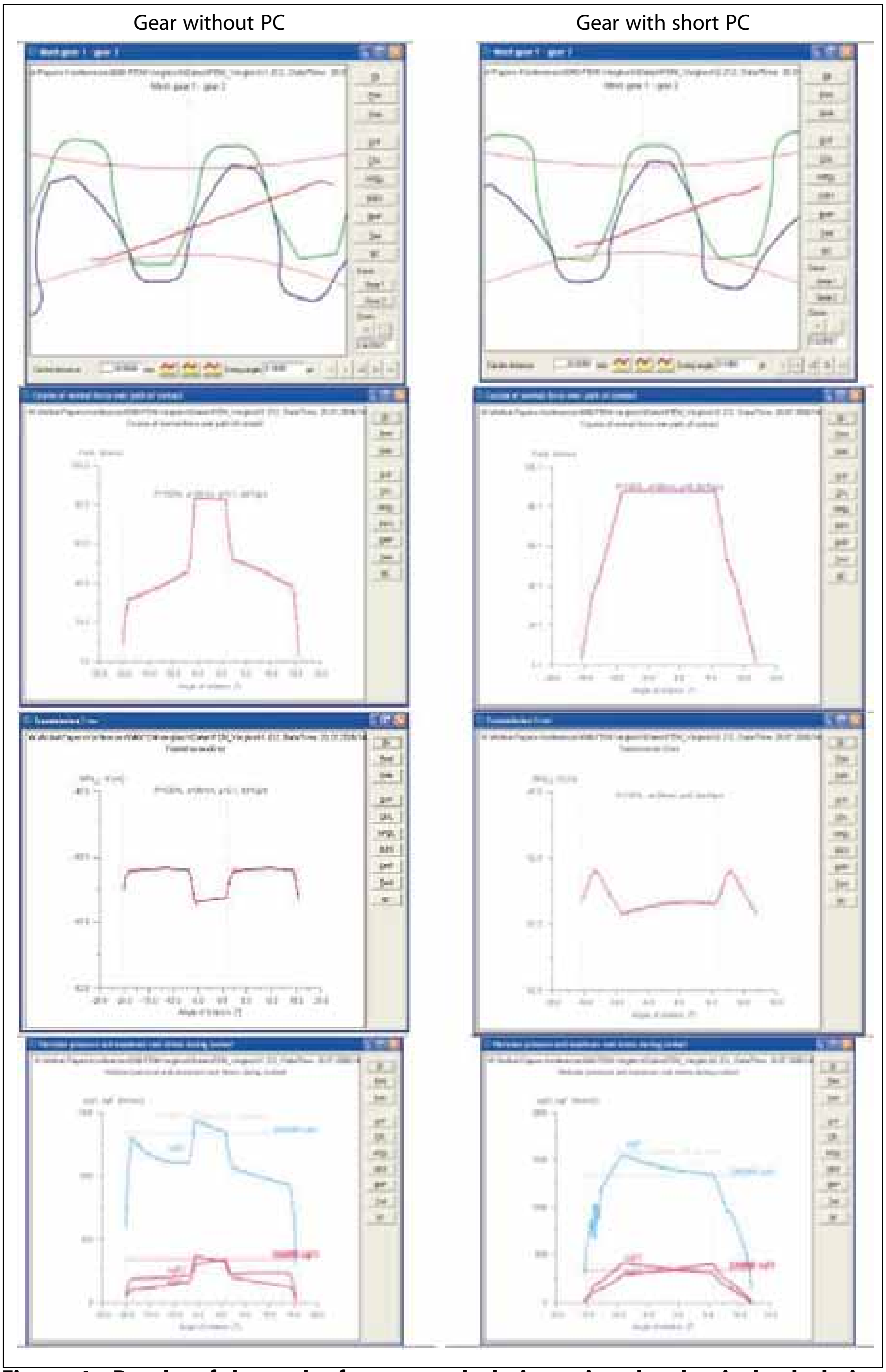

Figure 6—Results of the path of contact calculation using the classical calculation method implemented in *KISSsoft* (Ref. 7).

Study of the Influence of Different Profile Corrections

Introduction. With the discussed method, the calculation of a gear mesh is carried out within seconds. In combination with an efficient user interface, this enabled an extended study of the effect of different profile corrections. Of greatest interest was analyzing the behavior of 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.

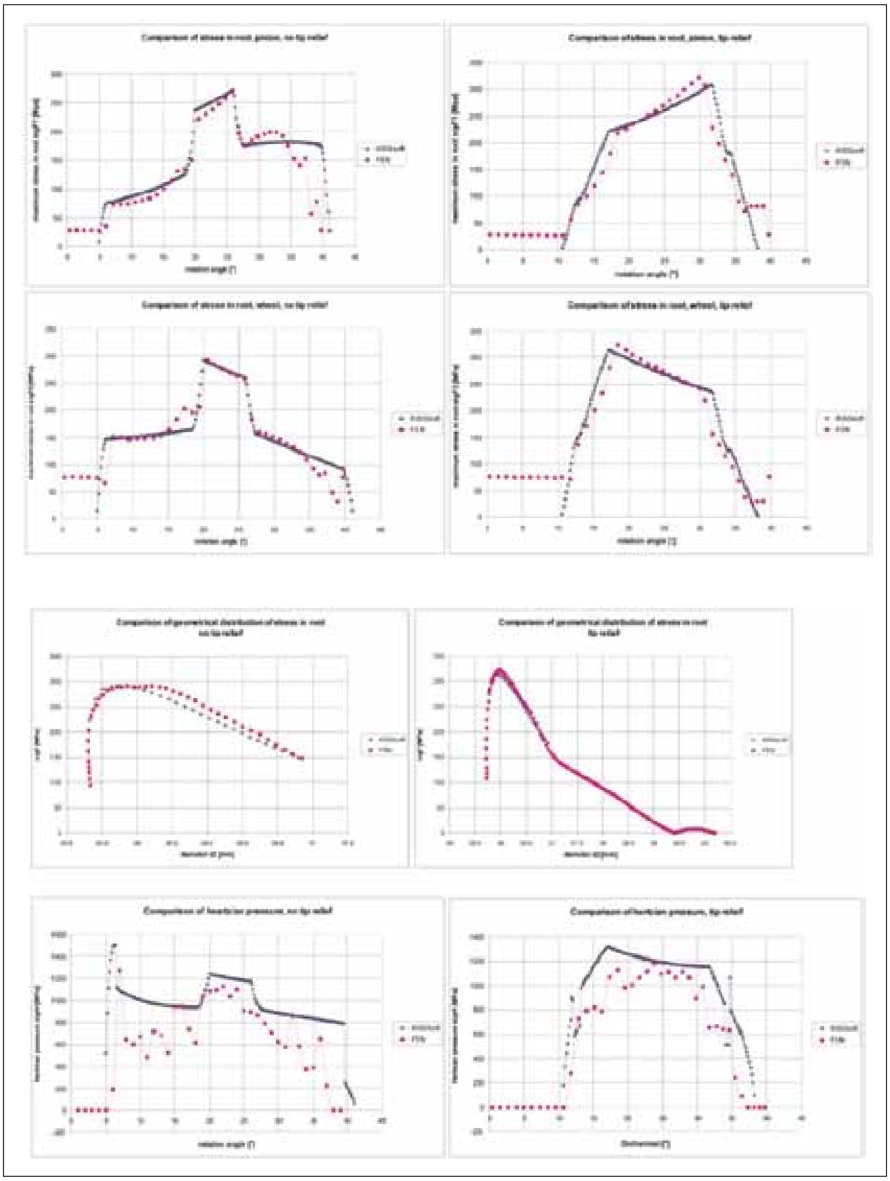

Figure 7—Comparison of the results of FEM calculation and *KISSsoft*.

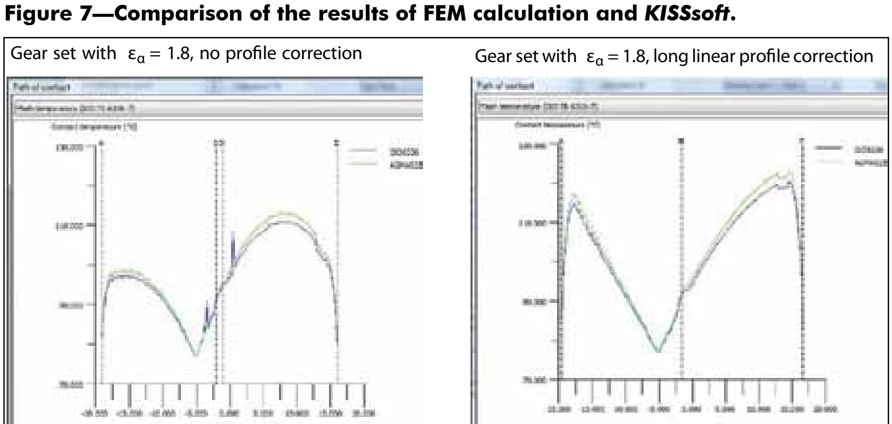

Figure 8—Flash temperature following AGMA 925-A03 (Ref. 8) and ISO 6336-7 (Ref. 9).

The aim of the study is to also analyze the effect produced by short linear, long linear, short arc-like, long arc-like and fully crowned profile corrections in the case of gear sets having a transverse contact ratio (ε_α) between 1.4 to 2.4. The profile correction was optimized for the design torque, which was defined based on a required safety factor of 1.0 for pitting and 1.4 for bending following ISO 6336. The tip relief was designed to eliminate the corner contact in the beginning and end of the contact at design torque, based on a perfect tooth form without manufacturing errors. The resulting PPTE was analyzed with different torques between 50% and 150% of the design torque. Furthermore, each variant was checked—including manufacturing errors—to evaluate the capability of the different corrections to compensate tooth form errors.

It is well known from literature that profile corrections are very important for spur gears, less so for helical gears. The reason is that helical gears and their helix angle shift the meshing contact from the left to the right side of the gear. So a gear pair with a sprung helix overlap ratio (ε_β) bigger than 1 also has, along with a badly designed profile correction, a very good PPTE. For this reason—the goal here being to analyze the effect of profile corrections—mostly spur gears were used.

Since a profile correction also has an important impact on the flash temperature and scoring risk, the highest flash temperature was calculated and compared. The calculation of the local flash temperature is calculated with two methods, i.e.—AGMA 925-A03 (Ref. 8) and ISO 6336-7. As Figure 8 shows, the flash temperature is reduced when using an optimized profile correction; the maximum temperature decreases from 120°C to 112°C and the flash temperature (difference between local temperature and gear body temperature) decreases from 43.6° to 35.7°—a significantly reduced scoring risk.

Short and long correction length and the PPTE. The profile correction, specifically the tip relief *Ca* and/or the root relief *Cf* (Fig. 1), has to be designed for a specific torque—normally for the medium or the most frequent torque. In this study, the design was done for the nominal torque (100%), but it is very important to also check the effect of a profile correction on the PPTE with different torque levels. In this study, the PPTE was calculated for 50%, 75%, 100%, 125% and 150% of the nominal torque.

Figure 9 shows the PPTE of gear pairs without profile correction. It is evident and logical that the PPTE is proportional to the torque in that the bending of the tooth increases with the load (torque) and the TE increases accordingly. It is evident from the graph that the PPTE decreases with higher transverse contact ratio. There is a significant reduction of the PPTE (about 50%) above $\varepsilon_\alpha = 1.8$. These are high-tooth gears that always have 2–3 teeth pairs in contact and, therefore, higher stiffness, normally lower stiffness variation and lower PPTE. This paper confirms these phenomena. Figure 10 shows, for example, that the PPTE of high-tooth gears ($\varepsilon_\alpha \geq 2.0$) is less than half of the PPTE of normal gears. This is valid *only* for gears without any profile correction. When applying a correction (Fig. 12), the PPTE is not proportional to the torque and the PPTE of high-tooth gears is less reduced when compared to normal gears.

The effect of a short profile correction is shown in Figure 11. As the ratio of PPTE with profile corrections to PPTE without correction is displayed, every result having a value bigger than 1 represents a situation in which the gear with correction is worse than the gear with no correction. With the exception of gear pairs with a very high ε_α (2.4), the short correction is always worse than no correction at all. For low load (75% and less), the increase of the PPTE can be

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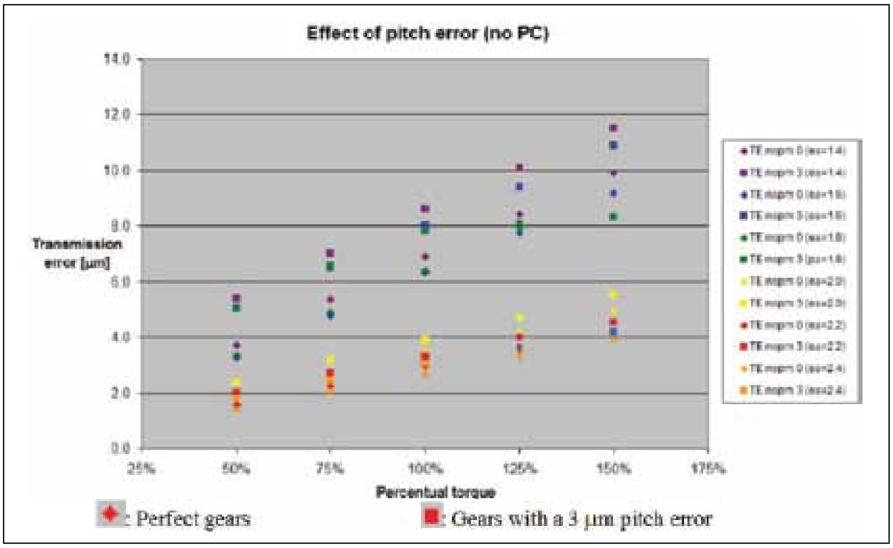

Figure 9—PPTE in µm for different gear pairs with ε_α = 1.4 to 2.4, without profile correction, depending on torque. For perfect gears and gears with pitch errors.

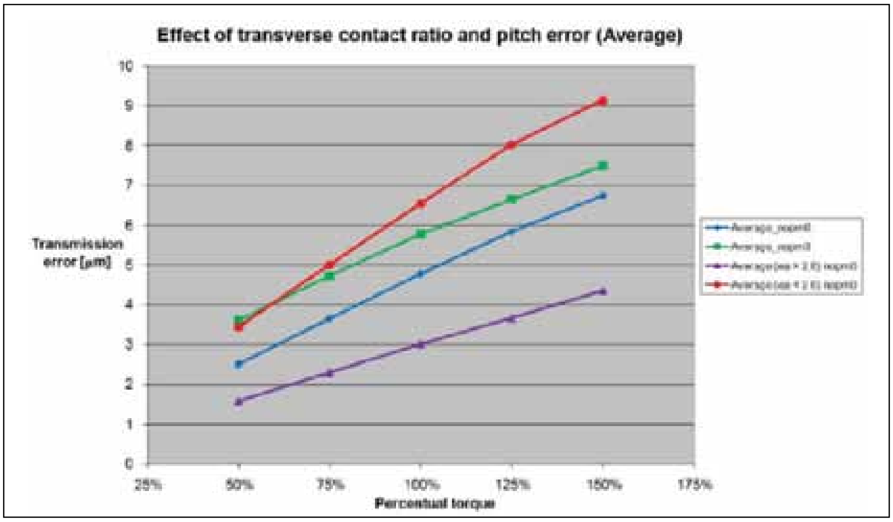

Figure 10—PPTE (same data as in Fig. 9) depending on torque, without PC. Curve "Average (ε_α < 2.0) nopm0" for gears having $\varepsilon_\alpha \geq$ 2.0; Curve "Average (ε_α > 2.0) nopm0" for gears having $\varepsilon_\alpha \geq$ 2.0; Curve "Average_nopm0" for all gears; "Curve "Average_nopm3" for all gears with pitch error 3 µm.

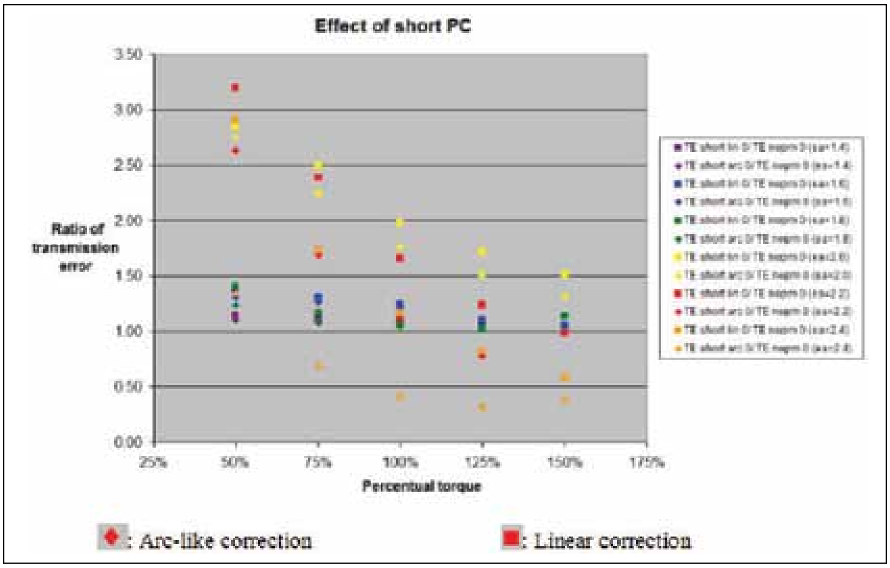

Figure 11—Ratio of PPTEwithPC to PPTEnoPC for different gear pairs, with ε_α = 1.4 to 2.4, with short profile correction, depending on torque.

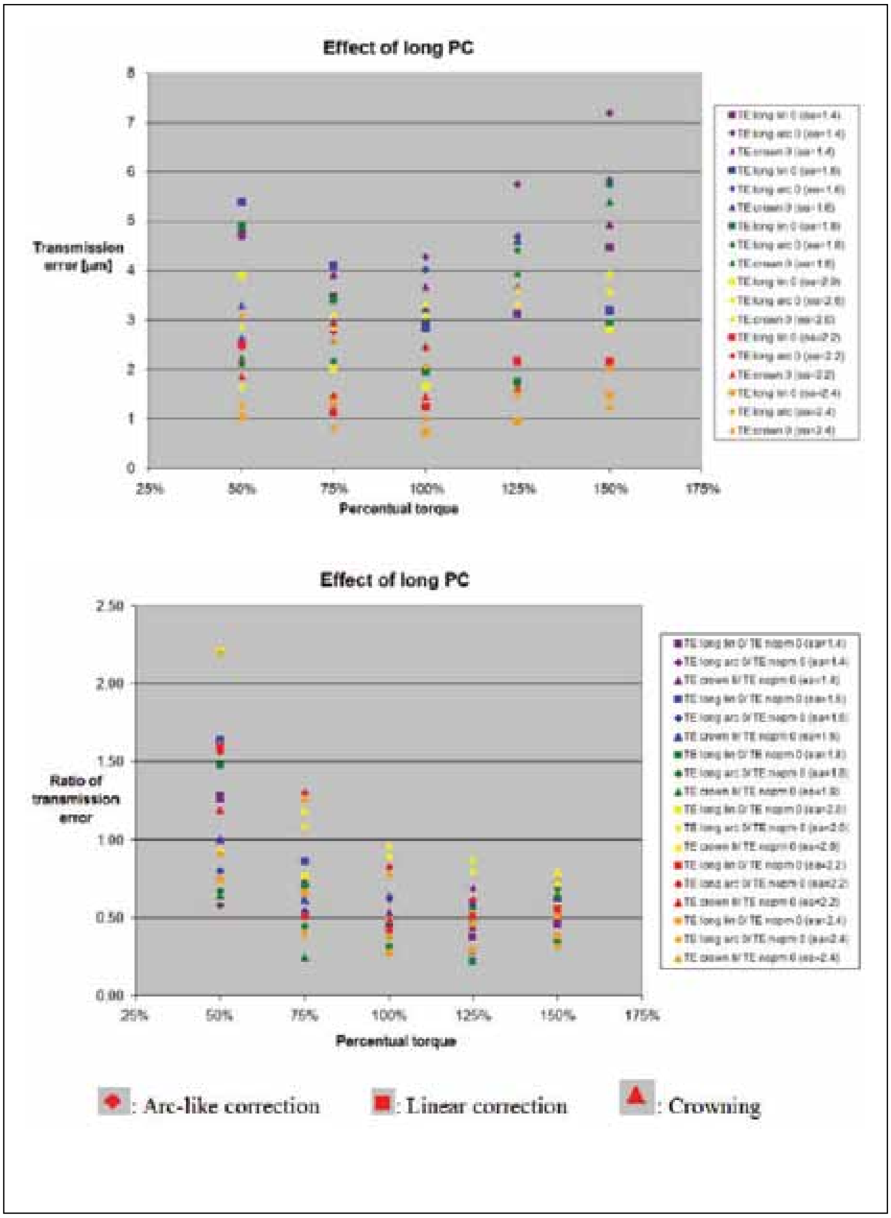

Figure 12—PPTE (above) and Ratio (below) of PPTEwithPC to PPTEnoPC for different gear pairs, with ε_α = 1.4 to 2.4, with long profile correction, depending on torque.

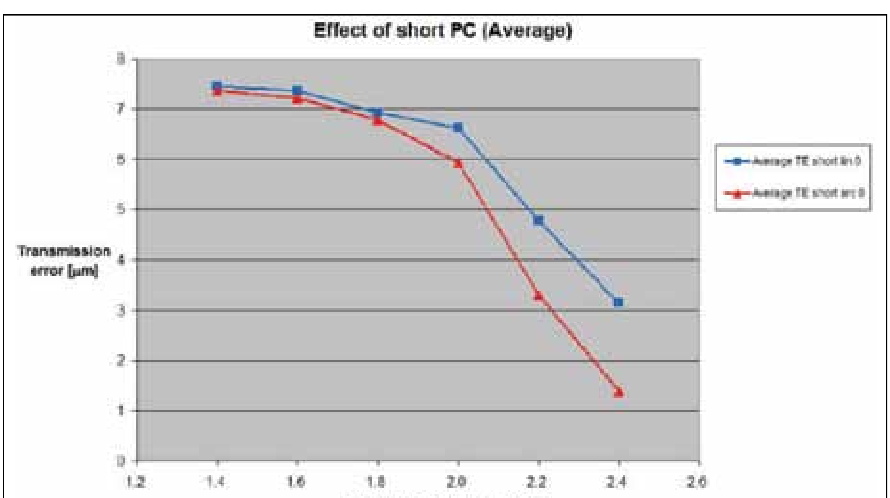

Figure 13—Mean PPTE (average over torque from 50 to 150%) depending on ε_α with short profile correction, for linear and arc-like correction curve.

300% and more.

The result is completely different when using a long profile correction (Fig. 12). All gear sets above 80% of nominal torque have a significantly reduced PPTE (30%–70%). Only for low load (60% and less of nominal torque) will the PPTE increase as compared to the gear set with no correction. But the increase is smaller than it is for short corrections.

To document such a significant difference between the short and the long correction—mainly for any transverse contact ratio ε_α—was certainly a surprise. The result is in agreement with some well-known authors—e.g., Houser (Ref. 1) and Niemann (Ref. 3). But the fact that—when using the long corrections—the transverse contact ratio of the unmodified part of the flanks is far lower than 1 (Fig. 2) served to reduce our expectancy of such a good result for the long correction.

Influence of curvature of the profile correction. It may be interesting to analyze the influence of different curvatures of the profile correction on the PPTE. For the short profile correction, a linear (Fig. 1) and an arc-like (Fig. 4) curve were used. The same was used for the long correction and profile crowning (Fig. 3).

When using the short profile correction (Fig. 13) for normal gears (ε_α < 2.0), the form of the curve has no significant influence. But for high-tooth gears, the arc-like curve is much preferred.

With a long profile or crowned correction, there is really no significant difference between the effects of different curve types when comparing the effect over the full ε_α scale. Rather, it appears that the linear correction is a bit better than the arc-like version (Fig. 14). It is particularly interesting when this effect is shown to be torque-dependent (Fig. 15). The linear correction is very effective for design torque, yet worse than the other corrections for lower torque.

Influence of manufacturing errors on the PPTE. The calculations presented here were repeated with a manufacturing error in order to evaluate the capability of the different corrections to compensate for tooth form errors. In this case a pitch error of 3 μm was applied—i.e., half of the maximal admitted error for Q-6 (ISO1328 or AGMA 2015) with gears of this size. The PPTE of gears with the manufacturing error is clearly increased, but the mean increase of the PPTE is much smaller than the pitch error (Table 2). Figure 10 shows that the 3 μm pitch error increases the PPTE on gears with no modification by only 1 μm or less.

If specific profile corrections are in fact best-suited to absorb pitch errors, it is not apparent—as the evaluation of the increase of PPTE due to a 3 μm pitch error in Table 2 shows. Arc-like (short and long) and short linear corrections yield the best results, but it is also assumed that further checks are performed to ensure that this result is indeed significant. In addition, profile errors and other manufacturing errors should be checked to ensure the clearest picture possible.

Supplementary study of the length of the profile correction. Thus far, the effect of short or long profile correction has been considered. The results showed clearly that the long correction reduces the PPTE. Houser recommends use of a "medium" profile correction, as "Long and short reliefs represent useful design limits for spur gears and generally some intermediate type of relief gives the best compromise, depending upon the range of operating loads that the gear meet (Ref. 1)." It is thus clearly possible that an intermediate length of the correction might provide even better results, and so is well worth checking.

The PPTE calculated on a specific gear with a different length of profile correction is shown in Figure 16. Here the PPTE typically increases slightly—from zero to the short profile correction. With increasing correction length, the PPTE decreases significantly and is quicker with the linear correction than with the arc-like correction. The PPTE reaches a minimum around the long profile correction. With the linear correction, the minimum is reached shortly before the long profile correction, the arc-like correction following

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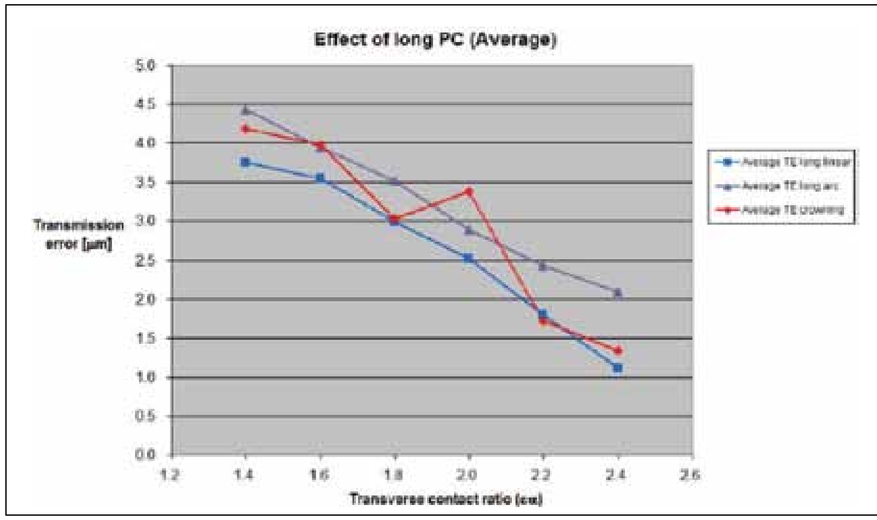

Figure 14—Mean PPTE (average over torque from 50 to 150%) depending on ε_α with long profile correction, for linear, arc-like and crowned correction curve.

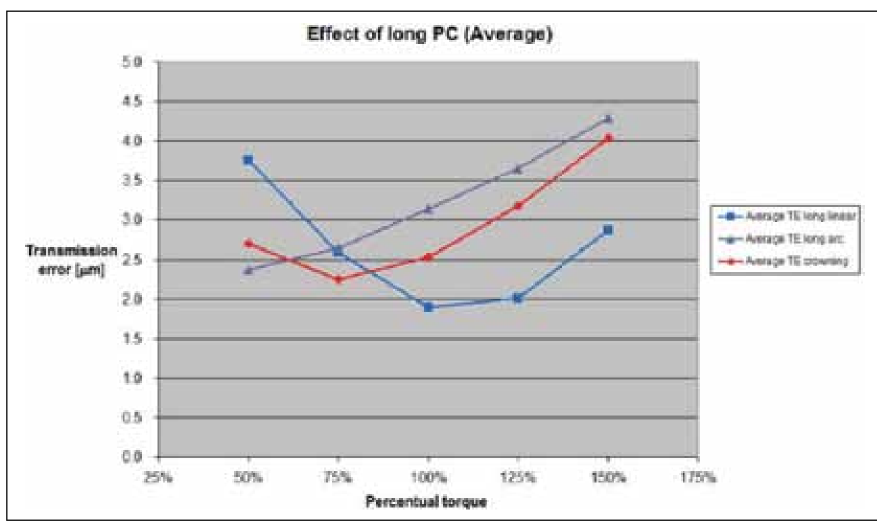

Figure 15—Mean PPTE (average over ε_α from 1.4 to 2.4) depending on torque with long profile correction, for linear, arc-like and crowned correction curve.

Table 2— Mean increase of PPTE due to a pitch error of 3 μm for the nominal torque (100%).

Profile correction	Mean PPTE (μm) with no pitch error	Mean PPTE (μm) with pitch error of 3μm	Mean increase of PPTE (μm)	Increase of PPTE in % of the pitch error
No correction	4.8	5.8	0.99	33.0
Short, linear	6.2	6.9	0.69	23.2
Short, arc-like	5.4	6.1	0.71	23.8
Long,linear	1.9	2.9	1.05	35.0
Long, arc-like	3.1	3.9	0.78	26.1
Crowning	2.5	3.7	1.18	39.2

shortly after. Clearly, the curves may change with torque and gear geometry, but the tendency is repeatable.

Different Profile Corrections and Flash Temperature

Normally the local flash temperature is highest at both the beginning and end of the contact between two gears. As the tooth thickness reduction produced by a profile correction is reducing the Hertzian pressure at these exact points, the result of the reduced load is that the local temperature at beginning and end of the contact will decrease. It is therefore logical that any profile correction will be helpful for reducing the risk of scoring.

Figure 17 shows the relative reduction of the flash temperature when using a short PC; Figure 18 shows the same with a long PC. The results are very similar, with no significant difference between the short and long correction. But with few exceptions, the reduction of the flash temperature when using any PC is very relevant.

It is revealing that the reduction of flash temperature (always compared to the gear without PC) is smallest with ε_α = 1.4; then the temperature decreases significantly with higher ε_α and the optimum ε_α = 2.0 (a temperature decrease of 60%); and finally—with even higher ε_α—the reduction is again less significant. Indeed, there is no significant influence in different curvatures of the profile correction on the flash temperature.

Dimensioning the Tip Relief *Ca*

It is perhaps important to discuss the layout of the optimum tip relief *Ca* (Fig. 1). The tip relief was designed in order to eliminate corner contact at the beginning and end of the contact at design torque without reducing the length of the contact between the gears. Figure 19 illustrates how the effect of the tip relief must be checked.

To be clear, tip relief was not varied in this study. Without any profile correction, the PPTE is quite proportional to the torque (Fig. 9). But upon applying a profile correction, the PPTE is lowest at 75% of design torque—and rises with smaller torque. So decreasing the amount of *Ca* by 5% does not provide the same result as would a change in torque of +5%. Although it might work, somebody could suggest, if torque and PPTE are proportional.

Regardless, the amount of *Ca* is an additional parameter to investigate.

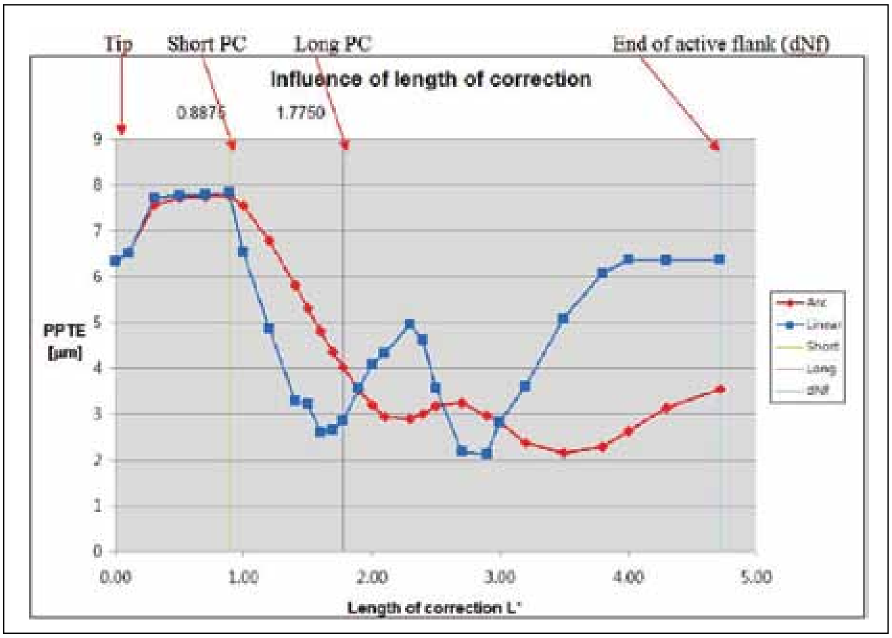

Figure 16—PPTE at design torque depending on the length of profile correction (for a gear set with ε_α = 1.6), when using linear or arc-like tip correction.

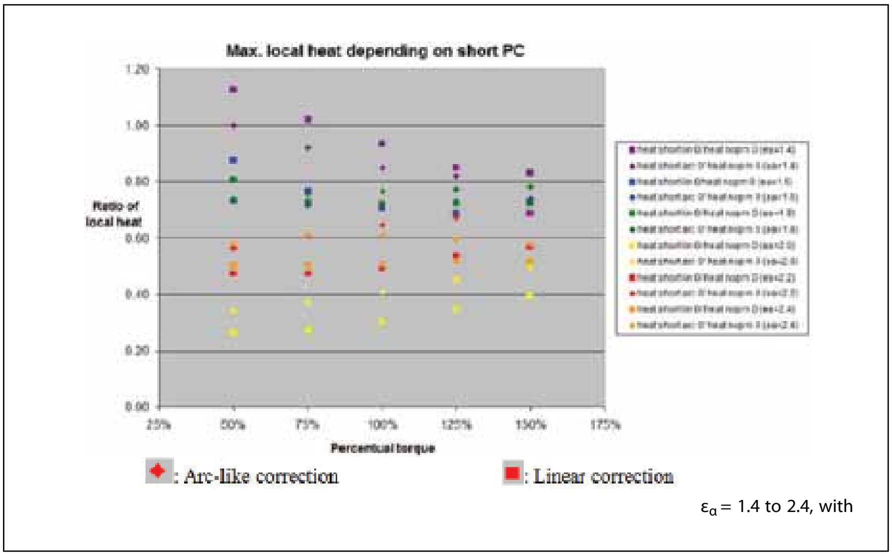

Figure 17—Ratio of HEATwithPC to HEATnoPC for different gear pairs, with ε_α = 1.4 to 2.4, with short profile correction, depending on torque.

Summary

The effect on the transmission error and the scoring risk produced by short-linear, long-linear, short-arc-like, long-arc-like and fully crowned profile corrections to gears with different transverse contact ratio (ε_α) was analyzed. There is little in literature regarding these issues. Important parameters were systematically varied and hundreds of PPTE calculations performed.

High-tooth gear sets (with transverse contact ratio $\varepsilon_\alpha \geq 2.0$) have in most cases only about half of the PPTE, in comparison with normal gear sets with no profile correction (PC) applied. With PC, high-tooth gears also have lower PPTE, but the difference to a normal gear is smaller.

A PC, short or long, effectively reduces the scoring risk and the contact shock (corner contact). For the PPTE, the difference between the short and the long correction is that for any transverse contact ratio, ε_α is very significant. Where short PC shows—even at design torque—a small increase of the PPTE compared with gears without PC, long PC shows a reduction of 30–70%—a result that has been supported by a number of authors. But the fact remains that when using long correction, arriving at such an overall good result is surprising. It is nevertheless important to know that any PC increases the PPTE with small torque (50% or less of design torque), compared to gears without PC.

The influence of different curvatures of the profile correction on the PPTE is less significant. For the short profile correction, a linear and arc-like curve were used. The same was also used for the long correction and profile crowning. When applying the short profile correction to normal gears ($\varepsilon_\alpha < 2.0$), the form of the curve has no significant influence. But for high-tooth gears, the arc-like curve serves best. With a long profile or crowned correction there is no significant difference between the effects of different curve types.

Differing parameters—such as the amount of tip relief *Ca* or the type of curve of the correction—have significant, but not always equal, influence on the PPTE. It is always recommended in a specific gear transmission case to calculate and optimize the transmission error when adaptive software is available.

Furthermore, in this paper only the peak-to-peak value of the transmission error was considered, which is normal practice in industry.

But we are convinced that the slope of the TE curve is also important, as a steeper slope will produce higher accelerations and vibrations. To date, these phenomena have yet to be researched accurately.

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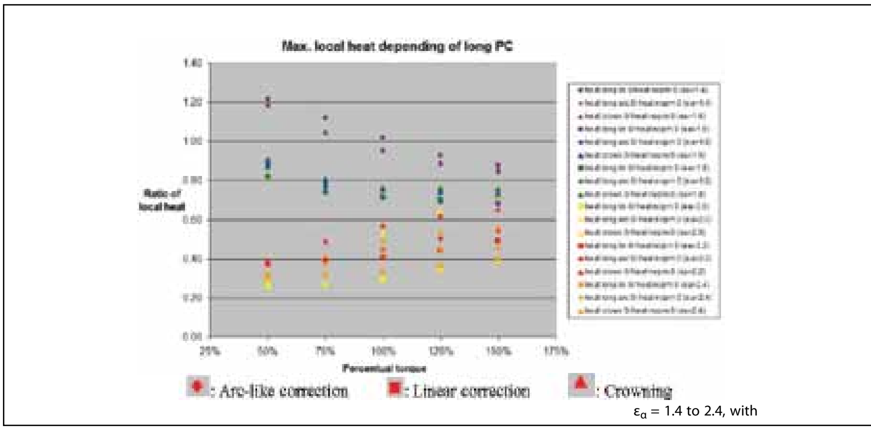
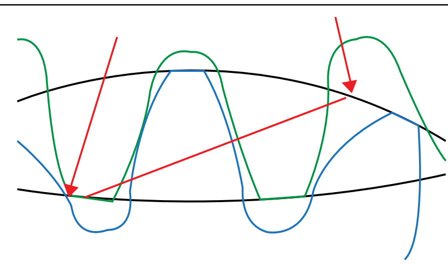
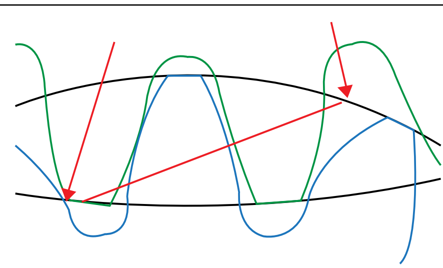


Figure 18—Ratio of HEATwithPC to HEATnoPC for different gear pairs, with ε_α = 1.4 to 2.4, with long profile correction, depending on torque.



Ca is too small: Path of contact continues on tip circle:
Indicates corner contact. Prolonged contact.



Ca is too small: Path of contact continues on tip circle:
Indicates corner contact. Prolonged contact.

Figure 19—Check of the tip relief *Ca*.

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