ISSUE FOCUS

BEVEL GEARS / PLASTIC GEARS

INDEPENDENT PINION AND GEAR PROFILE SHIFT FOR BEVEL GEARS

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The independent pinion and gear profile shift, as it is introduced in this article, allows for the first time in bevel gear technology to use positive profile shift amounts in both gear members; the result is a larger working profile that provides a higher contact ratio and quieter operation accompanied with a higher power density.

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Bevel and hypoid gears as well as straight bevel gears require, similar to cylindrical gears, a positive profile shift for the pinion in case of low number of teeth. If the number of pinion teeth is below 18, or even below 10, the pinion develops a severe undercut, which eliminates a large portion of the tooth profile in particular at the toe area (small diameter area). This effect leads to a weakening of the tooth root and also a reduction of the flank contact area. As result, the root bending stress and the surface stress are high, and the bevel gear pair has only a fraction of the load carrying capacity compared to a bevel gearset without undercut.

Profile shift has, according to international standards, the symbol "X." In order to determine the amount of profile shift, X is multiplied with the normal module mn of the gear, resulting in the amount of shift [1, 2, 3].]

The effect of profile shift is shown in Figure 1. The left profile in Figure 1 has no profile shift. The profile in the center of Figure 1 has a positive profile shift coefficient of X = 0.3. In practice, this means the tooth is shifted toward the manufacturing tool by 0.3 • m_n, and the manufacturing tool is shifted away from the work axis by the same amount. That way, the tooth depth stays the same, but the tooth is now at a larger diameter. As a result, the tooth becomes thicker in the root and thinner at the tip. The right-side graphic in Figure 1 shows a negative profile shift coefficient of X = -0.3. The tooth is shifted away from the manufacturing tool by 0.3 • m_n, and the manufacturing tool is shifted toward the work axis by the same amount. That way, the tooth depth stays the same, but the tooth is now at a smaller diameter. This makes the tooth thinner in the root and thicker at the tip. The tooth thickness at the reference circle (pitch circle) is kept constant in all profile shifted cases, which happens automatically in cylindrical gear hobbing when the same hob is used to cut gears with different profile shifts. The negative profile shift weakens the root of the tooth and also causes undercut, while the positive profile shift reduces or eliminates undercut [4, 5].

In bevel and hypoid gears, as well as straight bevel gears, a so-called V0 profile shift is always applied. V0 profile shift means the pinion profile shift coefficient X1 and the gear profile shift coefficient X2 have the same absolute amount but opposite signs (X1 = -X2 or X1 + X2 = 0) [6]. In cylindrical gears, the V0 profile shift prevents a change of the center distance between pinion and gear. In bevel gears, this is analogous to preventing a shaft angle change. This means that a positive profile shift in both members of a bevel gearset would change the shaft angle Σ by (Equation 1):

$$\Delta\Sigma = (X1 + X2) \cdot m_n / \text{Mean Cone Distance}$$

Equation 1

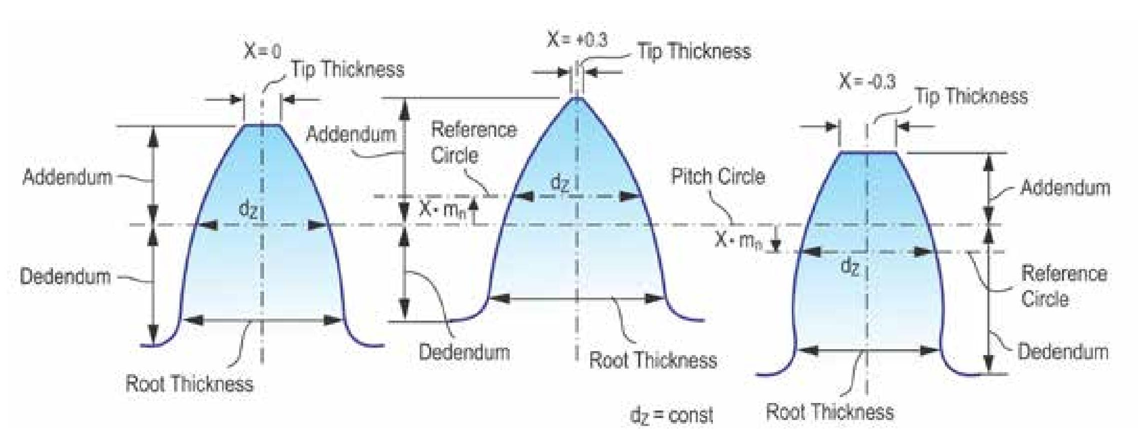

Figure 1: The effect of profile shift.

If a bevel gearset is designed with a shaft angle of 90°, any profile shift that is not a V0 shift would result in a shaft angle being not equal to 90°, which is not permissible. This is the reason why all state-of-the-art bevel gear design calculations and manufacturing systems strictly apply the V0 profile shift.

The side effect of the V0 profile shift is the fact that, although a "healthier" pinion profile can be achieved (if X1 is positive), the profile of the mating gear teeth takes an adverse effect (except with non-generated gear members, where the trapezoidal profile is just shifted). Even a gear with three times the number of pinion teeth will develop undercut from a certain amount of negative profile shift. This means, in turn, the amount of V0 profile shift is limited to the point where the lost root area on the gear teeth above the root fillet (or above the undercut) increases visibly and diminishes the improvement in the pinion teeth profiles.

This is especially significant when the number of pinion and gear teeth is equal (miter gears), or if the ratio is near one, like it is in the case of differential gears. Miter gears therefore never receive a profile shift, and differential gears only show very small positive pinion profile shift coefficient (for example X1 = +0.15) because the gear profile will develop undercut with the corresponding nega-tive profile shift coefficient (for example X2 = -0.15). Gearsets with larger ratios and a pinion tooth count below 18 also require profile shift and also in this case, only a limited V0 profile shift is possible. Figure 2 shows the tooth contact analysis of a straight bevel gearset with a ratio of 2.9 without profile shift, which is used as a baseline in the following chapters. The tooth contact is very small in profile direction. The number of pinion teeth is 12, which would require a profile shift to increase the active working area, which is only about 50 percent of the available profile. The lost areas at the top and the root are large due to physical pinion undercut and kinematic undercut in the gear.

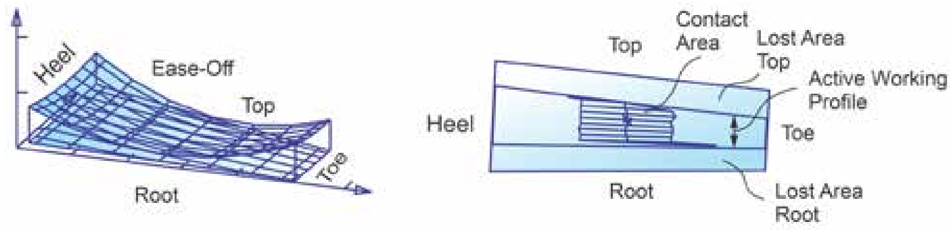

Figure 2: Ratio = 2.9, Ease-Off (left) and Tooth contact (right) for X1 = 0 and X2 = 0.

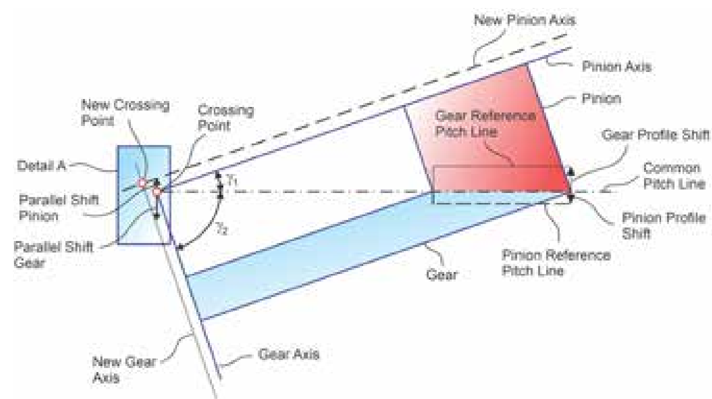

Figure 3: Geometric principle of parallel profile shift.

INDEPENDENT PARALLEL PROFILE SHIFT

Because a change of the shaft angle of a given design is not permissible, at first a profile shift was developed that is parallel to the respective pitch line. The parallel profile shift shown in Figure 3 shifts the pinion axis to the new pinion axis location and the gear axis to the new gear axis location. As a result, the crossing point between the new pinion axis and the new gear axis shifts to the new crossing point location.

The outline in a cross-sectional view of a pinion and a gear is shown in Figure 3. Pinion axis and gear axis intersect at the labeled crossing point. The common pitch line of pinion and gear is also labeled in the graphics. Before the profile shift, the pitch line is also identical to the reference pitch line, which divides the profile of both members into the addendum and dedendum. A positive pinion profile shift and a positive gear profile shift are applied parallel to their common pitch line. The positive profile shift means the cutting tool moves away from the respective member and increases their diameters. The pinion pitch line is still maintained while the new profile dividing reference pitch is shifted down, shown as a dashed line below the common pitch line. Also, the gear pitch line is maintained at the common pitch line location, while the new reference pitch line of the gear is shifted up, shown as dotted line. In order to mate and roll pinion and gear, which have now increased diameters, the reference pitch lines have to be moved back to the location of the original common pitch line. These moves can only be accomplished with a parallel shift of the pinion axis with the amount of the pinion profile shift, but in the opposite direction of the profile shift, and with a parallel shift of the gear axis with the amount of the gear profile shift but in the opposite direction of the gear profile shift. In the new axis location, the original crossing point moved to the new crossing point location.

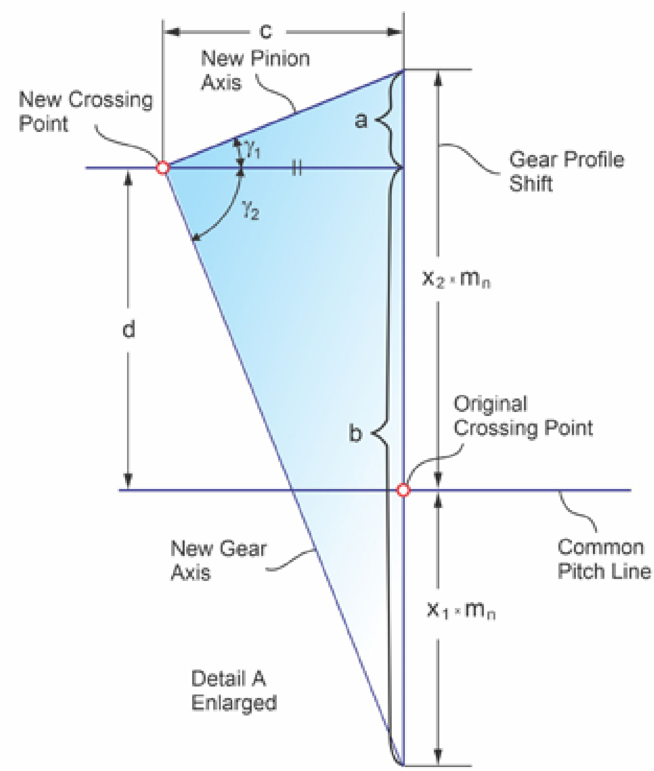

Figure 4: Enlarged detail A from Figure 3.

Enlarged detail A is shown in Figure 4. The original crossing point moved to the location of the new crossing point. The offset coordinates between the original and the new crossing point are d in vertical direction and c in horizontal direction. Because the crossing points

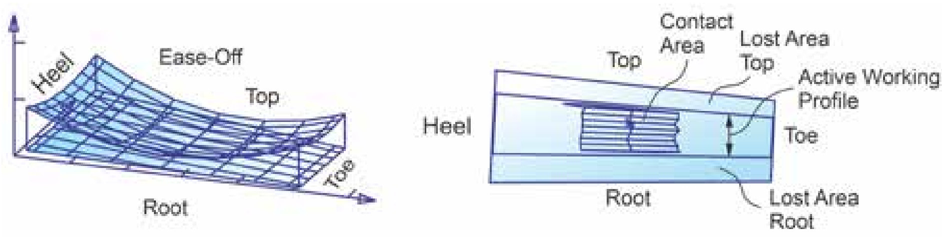

Figure 5: Ratio = 2.9, Ease-Off (left) and Tooth contact (right) for X1 = 0.15 and X2 = 0.0.

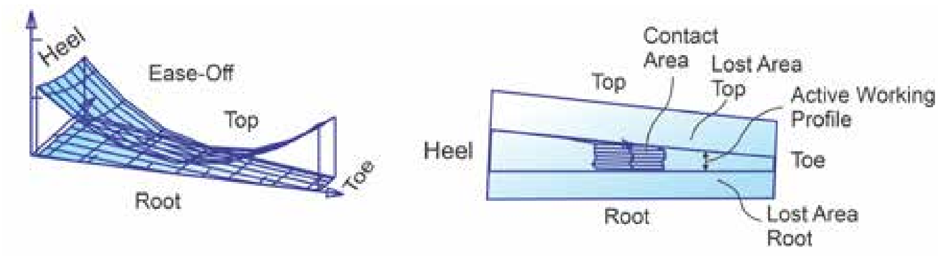

Figure 6: Ratio = 2.9, Ease-Off (left) and Tooth contact (right) for X1 = 0.5 and X2 = 0.5.

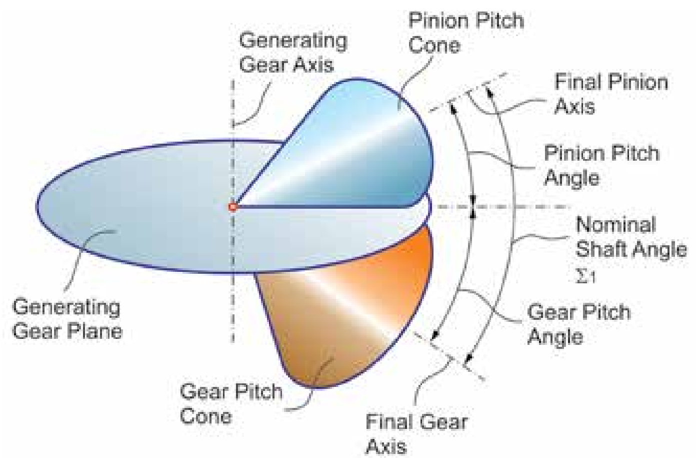

Figure 7: Initial pitch cones with original shaft angle.

are the origin of pinion and gear and determine their location and the location of their teeth, the offset coordinates have to be calculated in order to position pinion and gear correctly relative to each other in order to mesh and roll both members after the profile shift:

$$(x_1 + x_2)m_n = a + b$$ **Equation 2**

$$C = \frac{a}{\tan\gamma_1} = \frac{b}{\tan\gamma_2}$$ **Equation 3**

where:

c: horizontal offset.

a: vertical shift of pinion member axis.

b: vertical shift of gear member axis.

rearranging (2):

$$b = (x_1 + x_2)m_n - a$$ **Equation 4**

plug (4) into (3):

$$\frac{a}{\tan\gamma_1} = \frac{(x_1 + x_2)m_n - a}{\tan\gamma_2}$$ **Equation 5**

(5) rearranged:

$$a\left(\frac{1}{tan\gamma_1} + \frac{1}{tan\gamma_2}\right) = \frac{(x_1 + x_2)m_n}{tan\gamma_2}$$ **Equation 6**

(6) solving for a:

$$a = \frac{(x_1 + x_2)m_n}{\left(\frac{tan\gamma_2}{tan\gamma_1} + 1\right)}$$ **Equation 7**

from (3):

$$C = \frac{a}{\tan\gamma_1}$$

$$d = x_2 * m_n - a$$ **Equation 8**

where:

γ_1: pinion pitch angle.

γ_2: gear pitch angle.

m_n: normal module.

d: vertical offset.

Test calculations showed the parallel profile shift is limited to small profile shift coefficients. Fig-ure 5 shows the analysis results of a parallel profile shift of X1 = 0.15 and X2 = 0.0 (compared to the baseline without profile shift in Figure 2). The tooth contact looks similar to the contact of the baseline in Figure 2, but the active working profile is larger than the baseline contact. The lost top area is only about 50 percent of the baseline, and the lost root area is about the same size as the lost root area of the baseline. In summary, the parallel pinion profile shift achieved a noticeable improvement by increasing the active working profile.

Large profile shift coefficients reverse the positive effect of increasing the active working profile, which is attributed to the fact the pinion pitch line does not pass through the crossing point of the pinion and the gear axes. Also, the gear pitch line does not pass through the crossing point. The reason is the departure from the kinematic coupling condition between pinion and gear, which is part of the Law of Gearing.

An example for X1 = 0.5 and X2 = 0.5 is shown in Figure 6, which shows a significant degradation of the tooth contact and an increase of the lost area. The tooth contact in Figure 6 is signifi-cantly smaller than for the baseline, and the active working profile is less than half of the active working profile of the baseline. The lost root area is slightly smaller than the baseline, but the lost top area has increased.

GLOBAL EXPLANATION OF THE INDEPENDENT ANGULAR PROFILE SHIFT

The limitations seen in the parallel profile shift and the fact that a profile shift should be proportion-al to the profile depth along the face width of tapered depth teeth lead to the development of an angular profile shift. Because the angular profile shift changes the shaft angle, a unique method was developed, which applies a pre-correction to the shaft angle in such a way that the resulting shaft angle is equal to the nominal and required shaft angle, after the angular profile shift is applied.

The following sequence of pitch cone representations gives a global explanation of the two-step approach for the proportional (angular) profile shift. In Figure 7, the original pitch cones with the nominal shaft angle are represented as the starting point of the proportional

$$Z1/Z2 = \sin\gamma_1 I / \sin\gamma_2 I$$ **Equation 9**

profile shift transformations. The pitch angles are calculated for cones that roll on an enveloping line without slippage:

With:

$$\gamma_2 I = \Sigma_2 - \gamma_1 I$$ **Equation 10**

(9) plugged in (10) and solved for γ_1:

$$\gamma_1 I = \arctan(\sin(\Sigma) / (Z2 / Z1 + \cos(\Sigma))) \quad \text{Equation 11}$$

whereas:

γ_1:pinion pitch angle.
γ_2: gear pitch angle.
Z1: pinion number of teeth.
Z2: gear number of teeth.

The first transformation step is the pre-correction of the pitch angles of pinion and gear. If a posi-tive pinion profile shift $X1 \bullet m_n$ and a positive pinion profile shift $X2 \bullet m_n$ should be applied to the gearset, then the shaft angle has to be reduced in a first step by $x\varphi_1 + x\varphi_2$ (calculation of $x\varphi_1$ and $x\varphi_2$ will be explained later) as a pre-correction, preparing for the following profile shift. With the reduction of the shaft angle, also the pitch angles γ_1 and γ_2, will be reduced vs. $\gamma_1|$ and $\gamma_2|$, and therefore have to be re-calculated applying Equations 8 to 10, resulting in the graphic of Figure 8.

The proportional profile is applied after the pre-correction shown in Figure 8. Pinion and gear profile shift X1 and X2 will result in reversing the cone and shaft angle reduction by increasing the shaft angle by the amounts of $x\varphi_1$ and $x\varphi_2$, resulting in the original (nominal) shaft angle as shown in Figure 9. Although the cones in Figure 9 have the same angles as the original pitch angles $\gamma_1|$ and $\gamma_2|$, they are merely reference cones and no pitch cones. The correct pitch angles after the proportional profile shift are still γ_1 and γ_2, of the pre-corrected system. This effect might surprise gear engineers when studying a bevel gear dimension sheet after a proportional profile shift has been applied. Applications like the Gleason ConiflexPro design and optimization software treat a V0 profile shift like an independent, proportional pinion and gear profile shift and, therefore, modifies the pitch angles as part of the profile shift. For experienced bevel gear designers, this will require some learning experience; however, the proportional angular pinion and gear profile shift is the geometrically correct principle that provides significant advantages.

INDEPENDENT ANGULAR PROFILE SHIFT

The angular profile shift is achieved in two steps: In step 1, the shaft angle is pre-corrected before the profile shift is applied. This principle is explained in Figure 10.

Figure 10 shows the outline in a cross-sectional view of a pinion and a gear (drawn with solid lines). The pinion and gear axis intersect at the crossing point and include the nominal shaft angle Σ_1. In a first step, before the profile shift is applied, a pre-corrected shaft angle Σ_2 is calculated based on the desired profile shifts of pinion and gear. The desired pinion profile shift is applied at the outer cone distance ROUT to calculate an angle xj1:

$$x\gamma_1 = \operatorname{atan}(X1 \cdot m_n / ROUT) \quad \text{Equation 12}$$

The pinion axis is then rotated around the crossing point about the angle xj1 to the pre-corrected pinion axis position, drawn with dashed lines. The original pinion pitch line moves to the new pinion pitch line location. The desired gear profile shift is also used at the outer cone distance ROUT to calculate an angle $x\gamma_2$:

$$x\gamma_2 = \operatorname{atan}(X2 \cdot m_n / ROUT) \quad \text{Equation 13}$$

The gear axis is then rotated around the crossing point about angle $x\varphi_2$ to the pre-corrected pinion axis position, drawn with dotted lines. The original gear pitch line moves to the new gear pitch line location. The pre-corrected shaft angle between pinion and gear is:

$$\Sigma_2 = \Sigma_1 - x\gamma_1 - x\gamma_2 \quad \text{Equation 14}$$

whereas:

Σ_1: initial and final required shaft angle.
Σ_2: temporary shaft angle for pre-correction.

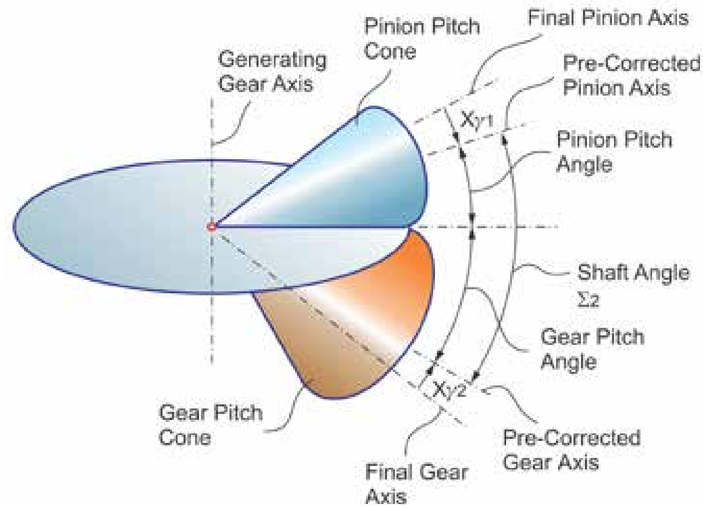

Figure 8: Pre-corrected pitch cones (by $x\gamma_1$ and $x\gamma_2$) with reduced shaft angle.

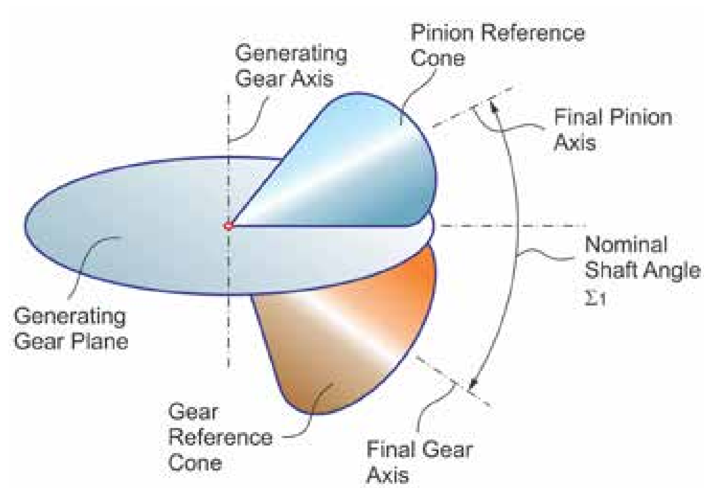

Figure 9: Final reference cones with nominal (original) shaft angle.

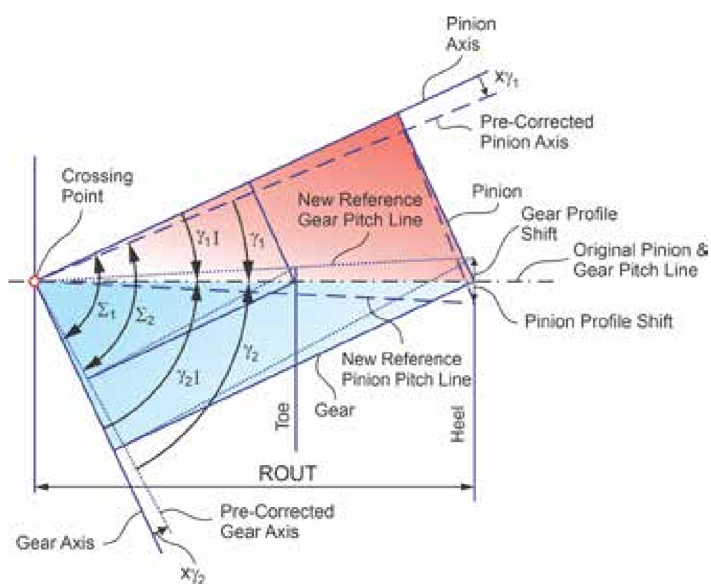

Figure 10: Principle of proportional (angular) profile shift with pre-corrected pitch angles $x\gamma_1$ and $x\gamma_2$ in order to achieve the correct shaft angle Σ_1 after profile shift application.

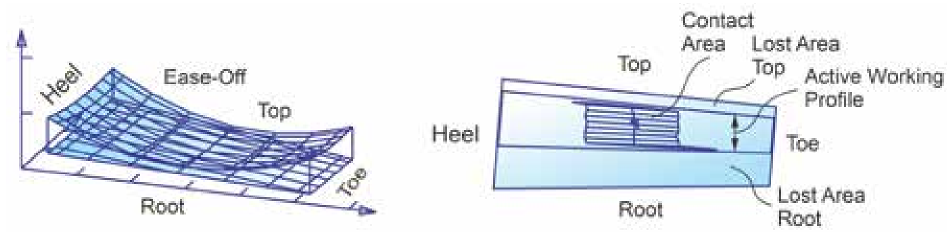

Figure 11: Ratio = 2.9, Ease-Off (left) and Tooth contact (right) for X1 = 0.7 and X2 = -0.7.

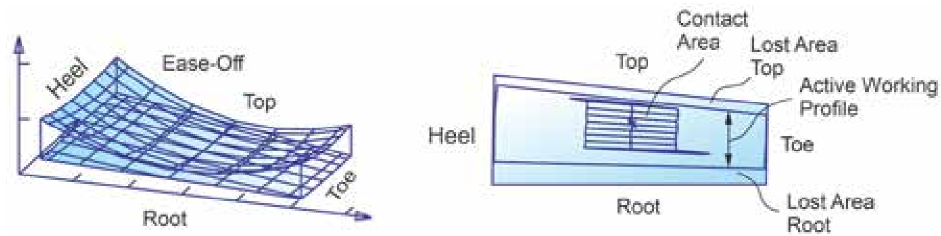

Figure 12: Ratio = 2.9, Ease-Off (left) and Tooth contact (right) for X1 = 0.7 and X2 = 0.7.

After the pre-correction of the shaft angle, the initial pitch angles $\gamma_1|$ and $\gamma_2|$ are now meaningless. New pitch angles have to be calculated according to the kinematic rule of cones rolling without slippage:

$$Z1/Z2 = \sin\gamma_1 / \sin\gamma_2 \quad \textbf{Equation 15}$$

With:

(15) plugged in (14) and solved for γ_1:

$$\gamma_2 = \Sigma_2 - \gamma_1 \quad \textbf{Equation 16}$$

$$\gamma_1 = \arctan(\sin(\Sigma_2) / (Z2 / Z1 + \cos(\Sigma_2))) \quad \textbf{Equation 17}$$

whereas:

Z1: pinion number of teeth.

Z2: gear number of teeth.

At this point, a gearset with the correct number of teeth and a pre-corrected shaft angle and with pitch angles, adjusted to the pre-corrected shaft angle is the result.

After the pre-correction of the shaft angle and the pitch angles, the pinion and gear profile shifts are applied in a second step that moves the pinion reference pitch line away from the original common pitch line (see Figure 10). The pinion profile shift $X1 \cdot m_n$ will increase the pinion diameter and create the pinion reference pitch. In order to achieve congruent pinion and gear reference pitch lines, the pinion axis is now rotated counterclockwise by the angle $x\varphi_1$ to the original location, which rotates the pinion reference pitch line to the location of the original pitch line.

The gear profile shift $X2 \cdot m_n$ will increase the gear diameter and create the gear reference pitch line. The gear axis is now rotated clockwise by the angle $x\varphi_2$ to the original location, which rotates the gear reference pitch line to the location of the original pitch line.

As a result, the nominal and required shaft angle Σ_1 is re-established in combination with the de-sired profile shift. The pitch lines of pinion and gear are not congruent anymore, but the reference pitch lines of pinion and gear are congruent and match the original pitch line. With this two-step approach, an angular profile shift, which is proportional along the face width (direction ROUT) with the distance from the crossing point, is the result. A proportional profile shift is adjusted to the changing tooth depth between toe and heel. The profile shifts of pinion and gear can be chosen individually and independently. There are no negative side effects that would limit the amount of proportional pinion and gear profile shifts.

Applying a global logic would deem that reducing the shaft angle by $x\gamma_1 + x\gamma_2$, then calculating the new pitch angles and after that adding $x\gamma_1 + x\gamma_2$ to the reduced shaft angle would arrive at the same tooth proportions as if the angular profile shift was never applied. In spite of this global logic, the pre-corrected pitch angles sustain when the angular profile shift is added. The reason is the generating ratio (number of generating gear teeth divided by the number of work gear teeth) remains when the profile shift is added, and the nominal shaft angle is established. The pre-correction of the shaft angle in the first step (if X1+X2 >0) reduces the mean diameter of pinion and gear, which also reduces the module. The profiles of pinion and gear will sustain their characteristic because the diameter as well as the module have been reduced by the same factor that will maintain the root transitions and just result in a size reduction of the profiles. The angular pro-file shift in the second step will then change the tooth profile according to Figure 1.

In order to prevent a center distance change of cylindrical gears, Max Maag [7] developed a method where the pitch diameter was first reduced and, after a positive profile shift on both mem-bers was applied, the working pitch diameter matched the original pitch diameter, and the original center distance was re-established.

EXAMPLE WITH RATIO 2.9

Figure 11 shows the analysis results of a straight bevel gearset with a ratio of 2.9 with a V0 pro-file shift. The pinion profile shift coefficient is X1 = +0.7, and the gear profile shift coefficient is X2 = -0.7. The tooth contact is the same size as the baseline in Figure 2. The active working profile increased slightly compared to the baseline, and the lost top area is significantly reduced. The lost area at the root increased by the same amount; the lost top area reduced. It can be observed in the example of Figure 2 and Figure 11 that the effect of the V0 profile shift improves the profiles of one member and deteriorates the profile of the mating member.

Figure 12 shows the analysis results of a straight bevel gearset with a ratio of 2.9 with a proportional pinion profile shift coefficient of X1 = +0.7 and a proportional gear profile shift coefficient of X2 = +0.7. The positive profile shift in both members increased the tooth contact in profile direction and accordingly doubled the active working profile compared to the baseline. The lost areas at root and top are reduced vs.

the baseline. A gearset according to Figure 12 will have a re-duced root bending stress and a reduced flank surface stress compared to the gearset versions in Figures 2 and 11.

EXAMPLE WITH MITER RATIO 1

Figure 13 shows the analysis results of a straight bevel miter gearset with no profile shift (X1 = 0 and X2 = 0). The Ease-Off surface to the left represents the consolidated length and profile crowning of a pair of meshing teeth. To the right in Figure 13, the tooth contact area and the active working profile are shown. The lost root area consists partially of the root fillet radius and partially of kinematic undercut of the gear tooth. The lost top area represents a kinematic undercut of the pinion tooth. Figure 13 is the baseline for the comparisons in the following Figures 14 and 15.

Figure 14 shows the analysis results of a straight bevel miter gearset with a V0 profile shift (X1 = 0.7 and X2 = -0.7). To the right in Figure 14, the tooth contact area and the active working profile are shown. The lost root area has doubled compared to the baseline in Figure 13 and also the lost top area increased. In the case of this example miter gearset, the V0 profile shift had a negative influence on the tooth profiles of pinion and gear.

Figure 15 shows the result of a proportional pinion profile shift coefficient of X1 = +0.7 and a pro-portional gear profile shift coefficient of X2 = +0.7 applied to the gearset in Figure 13. The tooth contact is larger in profile than the baseline in Figure 13 and the active working profile has significantly increased versus the baseline. The lost top area is nearly zero and the lost root area has reduced by 30 percent compared to the baseline. The larger active flank area reduces the surface stress proportionally.

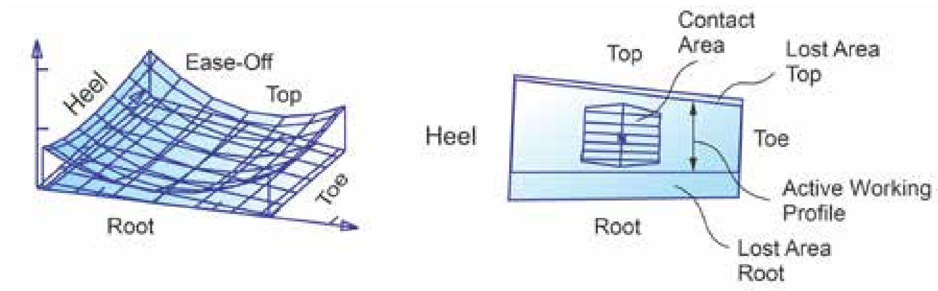

Figure 13: Ratio = 1, Ease-Off (left) and Tooth contact (right) for X1 = 0 and X2 = 0.

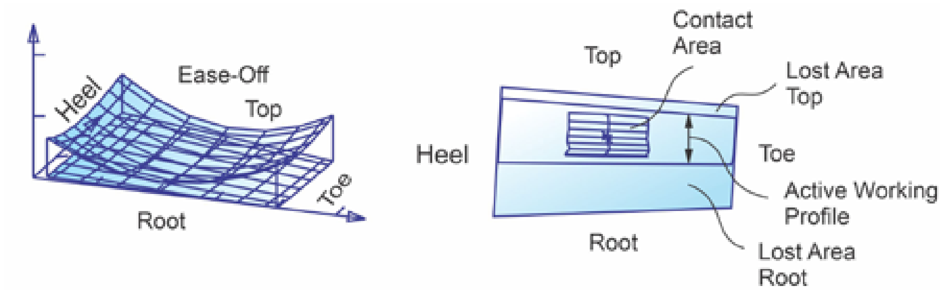

Figure 14: Ratio = 1, Ease-Off (left) and Tooth contact (right) for X1 = 0.7 and X2 = -0.7.

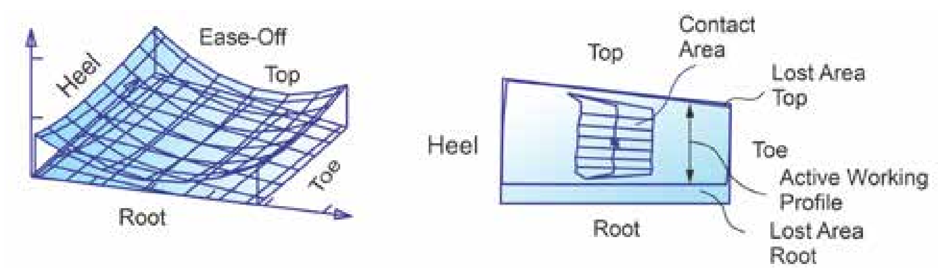

Figure 15: Ratio = 1, Ease-Off (left) and Tooth contact (right) for X1 = 0.7 and X2 = 0.7.

PRACTICAL RESULTS

The key advantage of positive profile shift in pinion and gear is the increased working profile in particular at the lower dedendum of the gear teeth. The first practical results are shown in Figure 16. The contact pattern after roll testing with marking compound shows in Figure 16 to the left the real version of the contact analysis from Figure 11 (X1 = 0.7, X2 = -0.7). A large lost area can be observed at the lower dedendum. The improved design in Figure 12 with X1 = +0.7 and X2 = +0.7 was also cut and shows after rolling with marking compound a full working profile with a smooth transition to the root fillet (right photo in Figure 16).

The roll testing results of the gearset with a double positive profile shift proved the theoretical findings of a larger active working profile, which also has a smooth root transition, due to the in-creased contact ratio in profile direction.

SUMMARY AND CONCLUSION

The V0 profile shift was the only way profile shift was applied to bevel and hypoid gears in the past. Pinions will generally benefit from a positive profile shift, which means that gears had to be designed with the same amount of negative profile shift. In the case of non-generated gear members, there was no disadvantage of using a negative profile

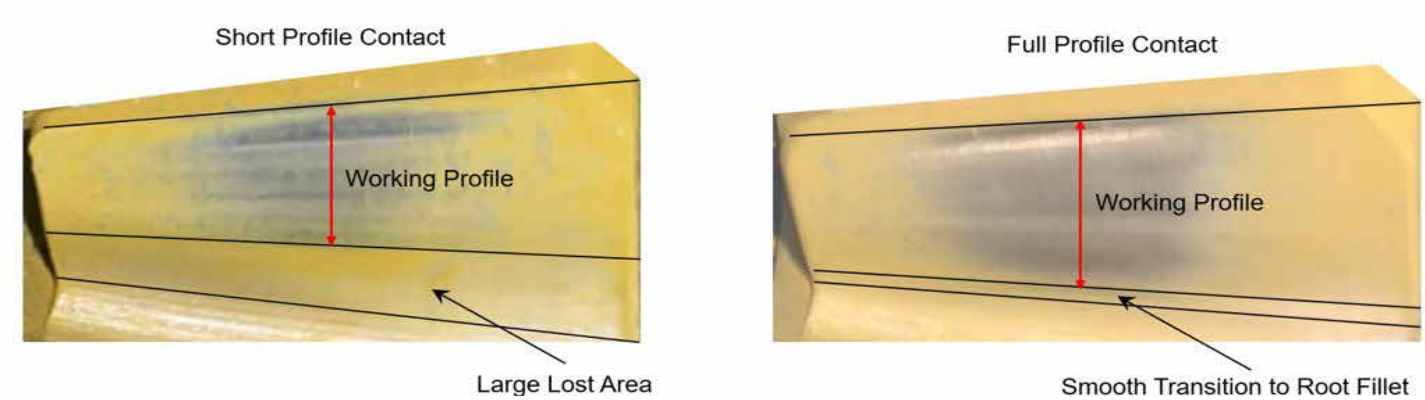

Figure 16: Roll test results V0 profile shift (left) and positive pinion and gear profile shift (right).

shift in the gear. The straight profile of the ring gear will not develop any undercut if a negative profile shift is applied. However, in the case of generated gears, kinematic undercut and physical undercut is recognized if the ratio is lower than 3. This is more severe for straight bevel gears than for spiral bevel and hypoid gears.

The independent pinion and gear profile shift, as it is introduced in this article, allows rather large positive profile shift amounts. The profile shift limiting factor is the width of the topland, which reduces with increasing positive profile shift. It is recommended to use an iterative process to increase the active working profile of pinion and gear with profile shift factors, starting at $X = 0.2$ and increasing in steps of 0.1, while checking the mean normal topland after each step. If the same amounts of profile shift are applied to the pinion and gear, then the operating pitch line will be where the original pitch line was and the sliding conditions along the profile remain (with no profile sliding at the center of the profile). To move the working pitch line of the pinion toward the pinion root, the pinion profile shift needs to be larger than the gear profile shift. A balanced profile shift between the pinion and gear has the advantage that the tip of the one member fits the root of the mating member without collision in the opposite member's root transition area. As visible in Figure 1, the topland of the tooth with a profile shift of $X = -0.3$ shows potential not to fit in the re-duced size root fillet of the tooth with a profile shift of $X = +0.3$.

It has to be considered that the pre-corrected pitch angles of the independent profile shift method will also change the pitch angles. In case of the common V0 profile shift, the original pitch angles (without profile shift) will remain. With the independent profile shift, the pitch angles are changing with the pre-correction of the shaft angle. After the profile shifts are applied to pinion and gear, the shaft angle will return to its original value, but the pre-corrected pitch angles will stay and will not reflect the relationship $\gamma_1 + \gamma_2 = \Sigma$. However, the working pitch angles replace the original pitch angles and add up to the shaft angle of the gearset.

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