Rules for Optimal Basic Design of Bevel Gears

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Bevel Gear Technology

Chapter 6

Introduction

More strength and less noise are the two major demands on gears, which includes both bevel and hypoid gears. Within the last few years, the still modest request for higher efficiency has been added to the original two requirements. Traditionally, gear engineers have met the first demand by changing a gearset's basic parameters — tooth height, pressure angle, spiral angle, etc. — and to meet the second, by making flank form modifications in order to reduce crowning — also known as "Ease-Off."

With today's available bevel gear software — e.g., *CAGE* and *UNICAL* — many engineers are modifying flank topography and discovering their gearsets are both stronger and quieter. Unfortunately, this coincidence is tricking some engineers into believing that Ease-Off itself adds strength to gearsets. In fact, the flank form modifications are only allowing the sets to make greater use of the strength potential that was available in their basic designs.

Most strength optimization requires changing a gearset's basic parameters. Most minor reductions in gear noise are made by optimizing just flank topography. In some cases, though, there are gearsets that don't use all their possible strength because they need more sophisticated Ease-Offs, which are adjusted to the particular operating conditions. The challenge for gear engineers is to know when to change what — basic parameters or flank topography — to increase a gearset's strength. In order to make such a distinction it is necessary to understand both basic parameters and flank form modifications (Refs. 1 – 2).

Although the influence of the basic parameters seemed to be obvious in the past, today's calculation and analysis systems provided many new insights. These new insights occasionally seemed to contradict the general experiences and appeared physically inconclusive. It is interesting to note that the implication of the "deeper" analysis and their physical basics can often only be discovered after intensive studies. The results are significantly improved gear properties which can be materialized for the first time today.

This chapter opens with a closer look at the so called "dimension sheet." The Gleason Dimension Sheet has become a world standard, although at first view it does not appear to be very "user-friendly." In includes (in a highly condensed layout) all important basic dimensions, as well as many parameters that are not explicitly expressed in the geometrical data of a bevel gearset. The following sub-chapters elaborate each on one of the key gear parameters, with recommendations that comply with good practices in advanced bevel and hypoid gear design. The end of this chapter discusses the question, what can be achieved with more optimal dimension sheet parame-

THE GLEASON WORKS

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HYPOID GEAR DIMENSIONS NO. M SAMPLE VERSION: 1.0.4.3 1/16/2012 8:25

GLEASON

	PINION	GEAR
NUMBER OF TEETH.	12	41
PART NUMBER.		
MODULE		4.724
FACE WIDTH	37.12	30.00
PINION OFFSET. . . . . BC	38.10	
PRESSURE ANGLE - PIN CONCAVE	15D 28M	
PRESSURE ANGLE - PIN CONVEX.	26D 32M	
LIMIT PRESSURE ANGLE . . . . -	6D 21M	
SHAFT ANGLE.	90D 0M	
TRANSVERSE CONTACT RATIO . .	1.036	
FACE CONTACT RATIO	2.074	
MODIFIED CONTACT RATIO . . .	2.318	
OUTER CONE DISTANCE.	125.30	103.26
MEAN CONE DISTANCE	106.71	88.17
PITCH DIAMETER		193.68
ADDENDUM	7.83	1.91
DEDENDUM - THEORETICAL . . .	3.48	9.16
WORKING DEPTH.	10.11	9.78
WHOLE DEPTH.	11.31	11.07
OUTSIDE DIAMETER	95.12	195.01
CUTTER RADIUS.	3.750"	
SYM. RACK GEAR POINT WIDTH .		0.070"
CALC. GEAR FINISH. PT. WIDTH		0.070"
GEAR FINISHING POINT WIDTH .		0.070"
PINION ROUGHING POINT WIDTH.	0.075"	
OUTER SLOT WIDTH	0.084"	
MEAN SLOT WIDTH.	0.084"	
INNER SLOT WIDTH	0.084"	
FINISHING CUTTER BLADE POINT	0.050"	
STOCK ALLOWANCE.	0.009"	
MAX. RADIUS - CUTTER BLADES.	1.650mm	
MAX. RADIUS - MUTILATION . .	1.750mm	
MAX. RADIUS - INTERFERENCE .	1.625mm	
CUTTER EDGE RADIUS	1.500mm	1.000mm
CUTTER BLADES REQUIRED . . . STD DEPTH	STD DEPTH	
DUPLEX SUM OF DEDENDUM ANG .		
MAX. NO. OF BLADES IN CUTTER		
RATIO OF INVOLUTE/OUTER CONE		1.332
RATIO OF INVOLUTE/MEAN CONE.		1.560
GEAR ANGULAR FACE - CONCAVE.		18D 46M
GEAR ANGULAR FACE - CONVEX .		20D 52M
GEAR ANGULAR FACE - TOTAL. .		22D 18M

ALL DIMENSIONS ARE METRIC UNLESS OTHERWISE SPECIFIED

RELEASED BY - AJN

	PINION	GEAR
PITCH APEX BEYOND CROSS PT .	25.40	-2.97
FACE APEX BEYOND CROSS PT. .	15.99	-4.72
ROOT APEX BEYOND CROSS PT. .	28.32	-1.25
CROWN TO CROSSING POINT. . .	90.80	37.03
FACE ANG JUNCT TO CROSS PT .		
FRONT CROWN TO CROSS. POINT.	56.90	
MEAN NORMAL TOPLAND.	1.76	2.73
PITCH ANGLE.	18D 33M	69D 41M
FACE ANGLE OF BLANK.	24D 0M	71D 40M
INNER FACE ANGLE OF BLANK. .		
ROOT ANGLE	16D 42M	63D 44M
OUTER SPIRAL ANGLE	51D 34M	30D 54M
MEAN SPIRAL ANGLE.	50D 17M	25D 26M
INNER SPIRAL ANGLE	50D 49M	20D 9M
HAND OF SPIRAL	LH	RH
DRIVING MEMBER	PIN	
DIRECTION OF ROTATION-DRIVER	REV	
BACKLASH . . . . . MIN	0.13 MAX	0.18
GEAR TYPE.		NON-GENERATED
DEPTHWISE TOOTH TAPER. . . .	DPLX	
FACE WIDTH IN PCT CONE DIST.		29.051
DEPTH FACTOR - K		4.200
ADDENDUM FACTOR - C1		0.180
OFFSET ANGLE	8D 26M	23D 37M
GEOMETRY FACTOR-STRENGTH-J .	0.2833	0.2922
STRENGTH FACTOR - Q.	7.64202	2.68355
EDGE RADIUS USED IN STRENGTH	1.500mm	1.000mm
CUTTER RADIUS FACTOR - KX. .	1.000	
FACTOR . . . . . MN	0.7945	
STRENGTH BALANCE DESIRED . .	GIVN	
STRENGTH BALANCE OBTAINED. . GIVN		0.148
GEOMETRY FACTOR-DURABILITY-I	0.2325	
DURABILITY FACTOR - Z. . . .	3420.22	1850.35
GEOMETRY FACTOR-SCORING -G .		
SCORING FACTOR - X		
EFFICIENCY AT 30000 PSI. . .		93.703
PROFILE SLIDING FACTOR . . .	0.00359	0.00336
LENGTHWISE SLIDING FACTOR. .	0.00689	0.01735
RESULTANT SLIDING FACTOR . .	0.00777	0.01767
AXIAL FACTOR - DRIVER CW . . OUT	0.947 OUT	0.020
AXIAL FACTOR - DRIVER CCW. . IN	0.581 OUT	0.217
SEPARATING FACTOR-DRIVER CW. ATT	0.070 SEP	0.173
SEPARATING FACTOR-DRIVER CCW SEP	0.894 ATT	0.056
INPUT DATA		
SPIRAL ANGLE	49.60D	
INPUT DATA . . . . .KTT/CUTM	T2DS	
CLEARANCE FACTOR	1.150	
CALCULATED GEAR PITCH ANGLE.	69.685D	
NUMBER OF BLADE GROUPS . . .		
EFFECTIVE CUTTER RADIUS. . .		
SLOT WIDTH PCT FOR BLADE PT.		

Figure 1 The most important gear parameter in the dimension sheet.

ters and should not be subjected to tooth contact optimization and vice versa. The basic dimensions are "locked in" after a gearset is in service and certain optimizations which are traditionally subject of improved basic dimensions can be achieved with changes in the contact geometry. The summary chapter gives recommendations of what can be done in such a conflict.

Influence of the most important parameters. A gearset's basic parameters establish the potential of its properties which include strength and noise as well as efficiency. There are many major basic parameters, and each of them has a variety of effects upon the operating performance of a gearset. Gear engineers who practice gear design and optimization, and who must valuate the physical properties of bevel and hypoid gearsets, need to understand these effects — especially when optimizing gearsets. Figure 1 shows a Gleason Dimension Sheet of a typical automotive hypoid gearset. The parameters with the most influence are highlighted in yellow (Ref. 3).

The individually marked parameters are discussed in the order with which they are listed in the dimension sheet.

Module. A fine pitch gear pair results in a larger transverse contact ratio than in that of a comparable coarse pitch system. The reason is the increasing number of teeth in case of a smaller module and same pitch diameter, which will for an infinite number of teeth eventually deliver the relationships of a generating rack. The theoretical contact ratio of two engaged racks is infinite. Smaller modules deliver proportionally shorter and thinner teeth than larger modules in case of same pitch diameters. Fine pitch gearsets generate less vibration and noise, but also have a lower power density. The analogy for better comprehension of this phenomenon is the following: If a cantilever beam is split from its profile height into two halves, and the two thinner beams are packaged above each other like the leaves of a leaf spring, then the load carrying capacity of the beam package (same outside dimensions) is only 25% of the original beam. If now the length of the beam package is shortened to half the length, then the load carrying capacity is 50% of the original. Figure 2 summarizes the major influence of a reduction in module.

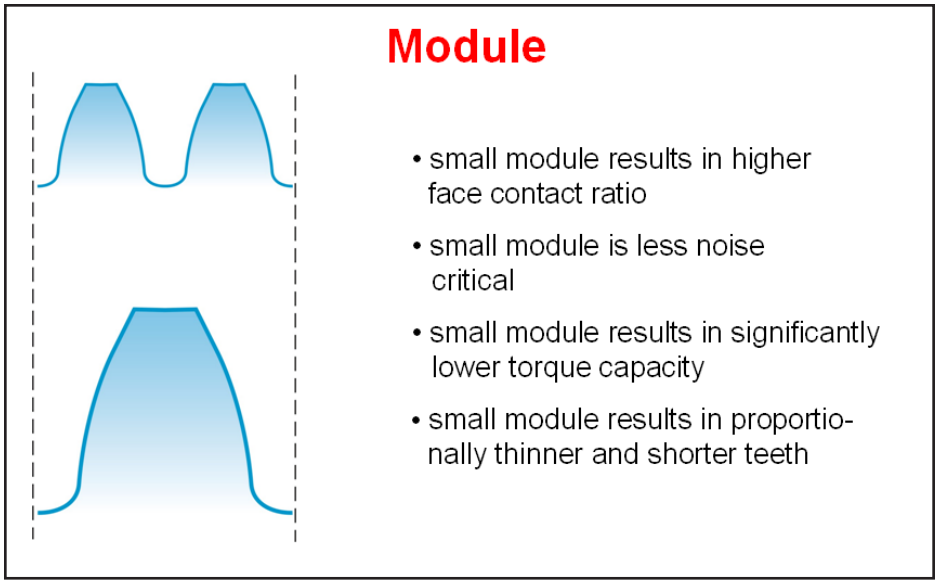

Figure 2 Influence of module.

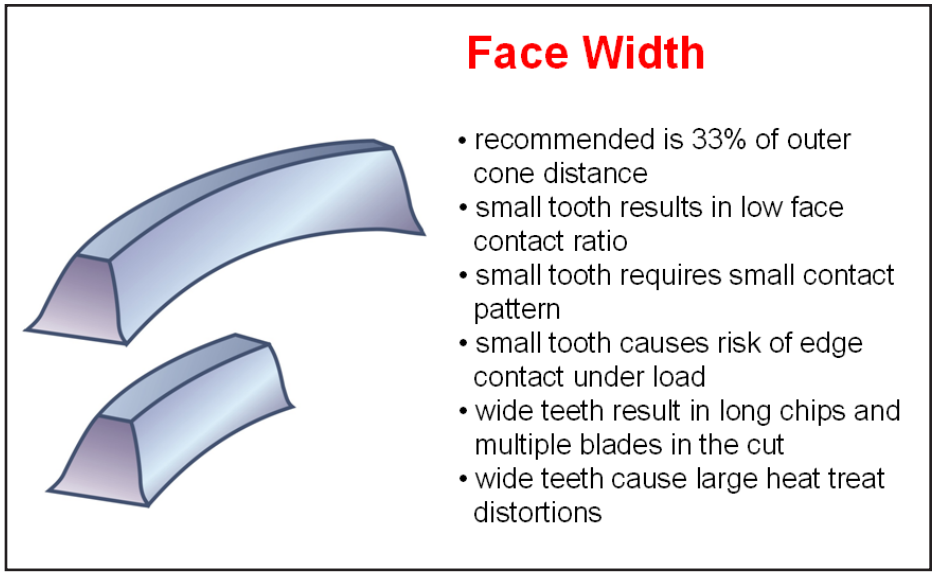

Figure 3 Influence of face width.

Face width. A smaller face width delivers a smaller face contact ratio and requires a small tooth contact. There is a constant risk of edge contact at heel and toe, even with small load-affected deformations. In the case of an oversize face width, problems in manufacturing might occur, as the chips in the soft cutting process are too long and heat treat distortion of the long spiral shaped teeth are significant. In addition, a hard finishing by grinding is problematic because of the long contacting zone between teeth and grinding wheel, which in turn leads to higher risk of burning (structural damage and soft spots along the face width and in the root). The expected strength increase due to a wider tooth is not achieved because a uniform load distribution is less likely to be realized if the face width is increased above the recommended numbers. The optimal face width for bevel and hypoid gearsets is 33% of the mean cone distance; Figure 3 summarizes the most important influences of the bevel gear design parameter face width.

Point radius of cutting blade or grinding wheel. The maximal radii that can be ground on the tip of a cutting blade are limited by the point width of the cutter and the top width of the blade (Figure 4). Blades with a small clearance side edge radius, e.g. — a sharp clearance side

corner — can cause mutilations in the root fillet side, which is not "officially" machined by the observed blade (Fig. 6).

Hypoid offset. A hypoid offset is generally used for rear wheel driven passenger cars in order to lower the center of gravity and to avoid a bigger propeller shaft tunnel. The hypoid offset causes a relative length sliding between the meshing flank surfaces and enlarges the diameter and spiral angle of the pinion. This leads to better hydrodynamic lubrication, additional dampening in the tooth mesh, increased contact ratio, and improvement in pinion strength.

The gear engineer should design bevel gears with an offset whenever possible — especially small offsets that achieve the desired advantages without the potential disadvantages that the hypoid offset can produce. Even a very small offset helps to avoid pitting population along the pitch line, as they are known in spiral bevel gears. The possible disadvantages of hypoid offsets, above 5% of the ring gear diameter, are the required use of hypoid oils with additives and the potential for scoring during the break-in period of the gearsets. Hypoid gears with offsets above 20% of the ring gear diameter show a low efficiency and an increased operating temperature. The optimal hypoid offset lies between 10% and 15% of the ring gear diameter. Hypoid gears with optimal offsets are superior to spiral bevel gears with regard to strength, quiet operation, and efficiency. Hypoid gears also have the lowest operating temperatures, compared to other angular transmissions. Figure 7 contains a summary of the most important influences of the hypoid offset.

Pressure angle. A reduction of the pressure angle increases the topland and width of the root fillet. A pressure angle reduction is a welcome freedom if an increase of the topland or the root fillet is required in the course of a gearset optimization. This often results in a natural combination of depth and low pressure angle, as it is used in well proportioned "high tooth designs." Suitable standard pressure angles are 20°. In the past, bevel and hypoid gears often used 22.5° in order to achieve higher gearset strength. Today's tools like flank optimization with

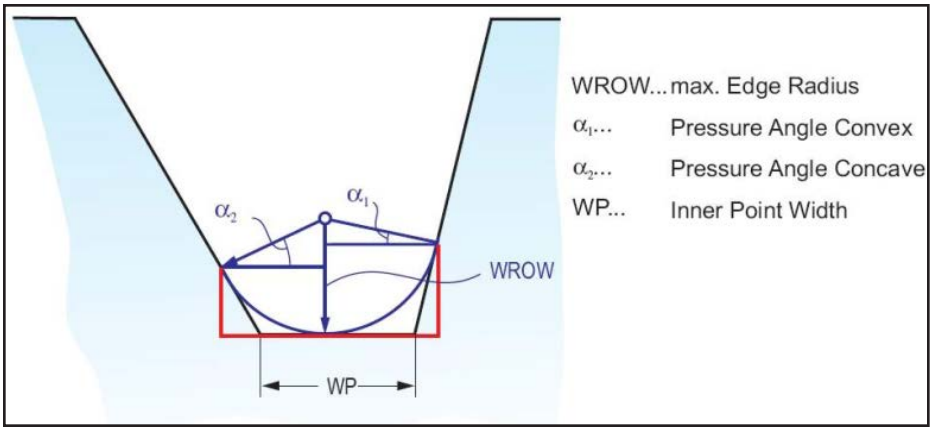

Figure 4 Maximal geometrically possible blade edge radius.

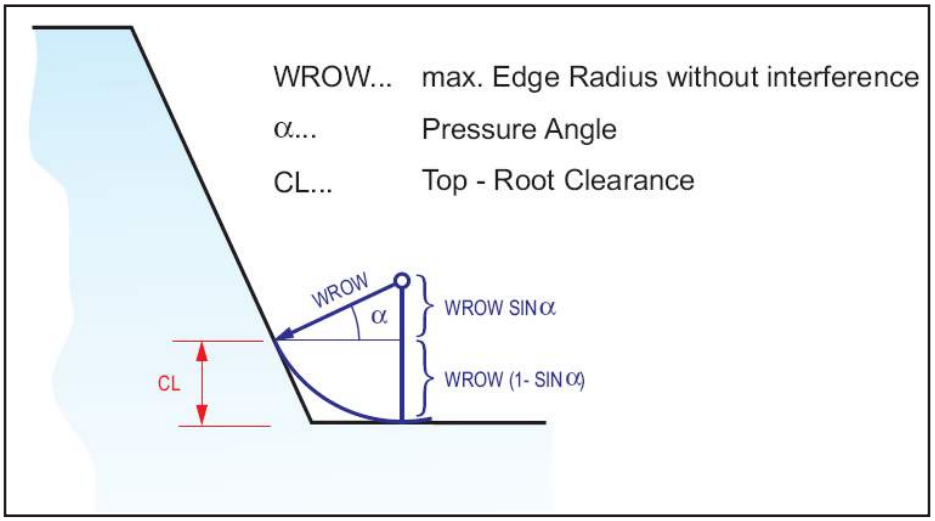

Figure 5 Interference limit for maximal blade edge radius.

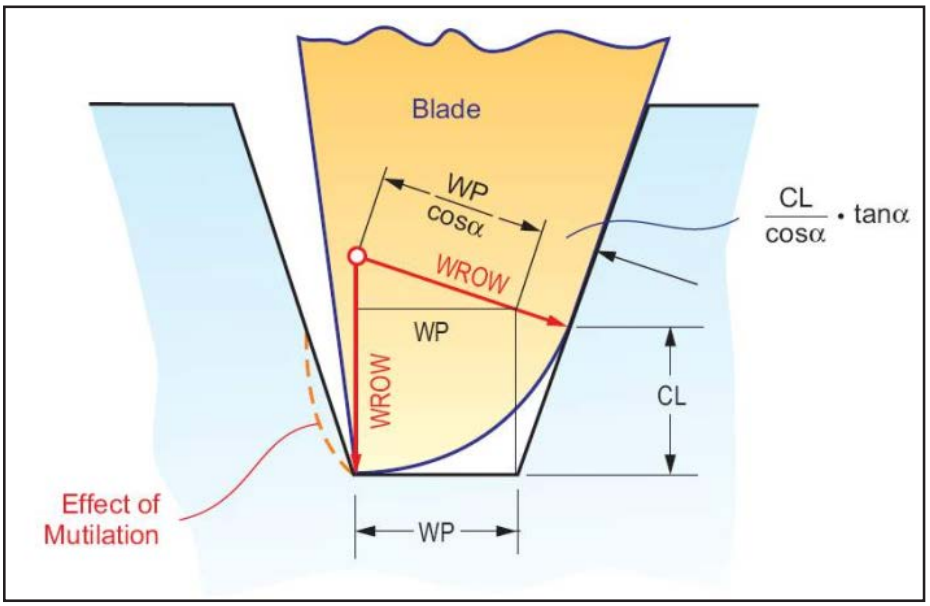

Figure 6 Mutilation limit of opposite root fillet due to edge radius.

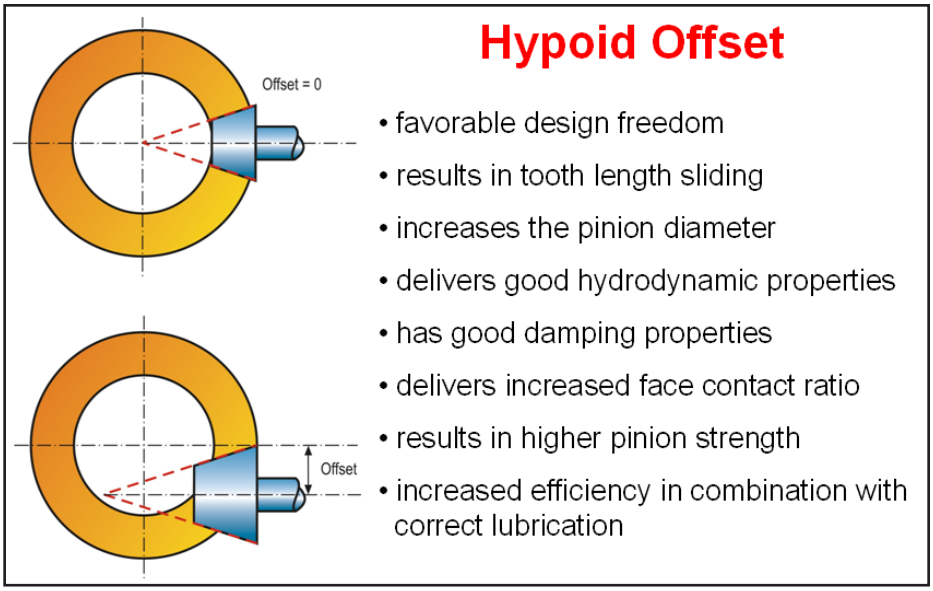

Figure 7 Influence of hypoid offset.

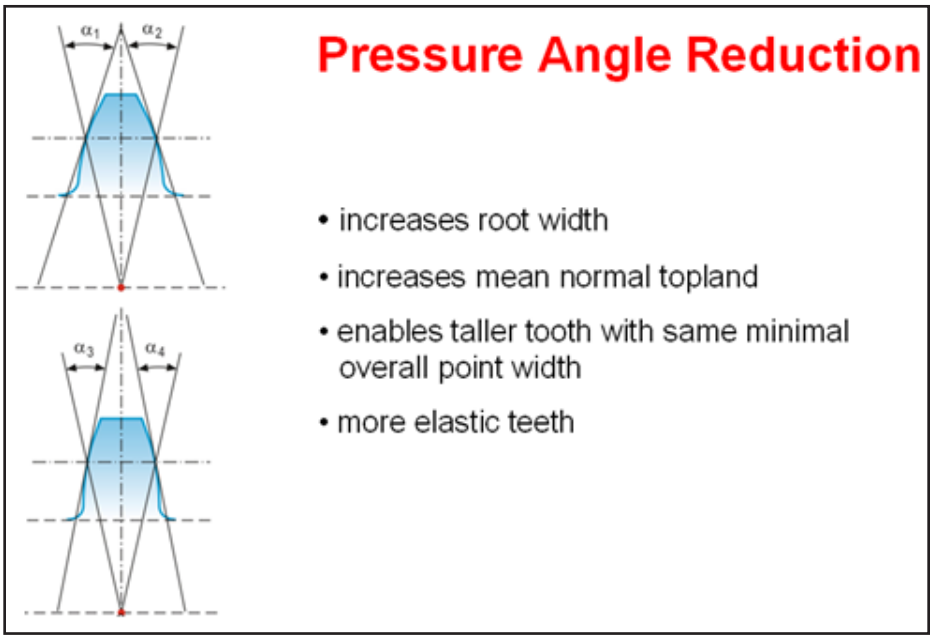

Figure 8 Influence of pressure angle.

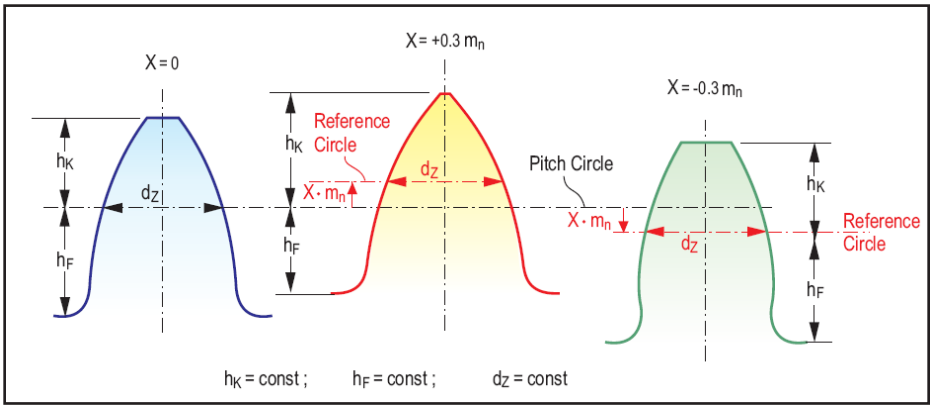

Figure 9 Influence of profile shift.

UMC (*Universal Motion Concept*; *see original text chapter 16*) and strength analysis with finite element calculation allow for a better utilization of the tooth elasticity, resulting in a better load sharing.

Subsequently, many quiet bevel and hypoid gearsets today, with high power density feature a large whole depth and low pressure angles. The recommendation for bevel and hypoid gears is to apply an included pressure angle between 36° and 40°. The most important influences of the pressure angle are summarized in Figure 8.

Profile shift. The profile shift (or addendum modification) is used in order to improve the roll conditions (increase of active profile) and to avoid undercut in the pinion root. The resulting pitch line for spiral bevel gears with a ratio above 1x2 lies towards the pinion root. The pinion root shift towards the pitch line is even more significant for hypoid gears, often with the goal to keep the vector summation of profile and length sliding low. The center graphic in Figure 9 shows the reduction of topland and root width for this case. The reduction of topland and root fillet could be reversed by reducing the pressure angle; but this in turn would also reverse the improvement in roll conditions and increase the undercut again. In the case of form-generated sets it is possible by means of a tooth thickness balance to increase the pinion topland and reduce the gear topland by nearly identical amounts since the Formate gear will not change its profile due to a profile shift. This however will reduce the root width of the pinion, which presents another limitation. In many cases a compromise between small pressure angle reductions and minimal tooth thickness balance can be struck in realizing the desired profile shift.

Whole depth. A taller tooth has more elasticity than a tooth with the standard depth of about 2.2 * mn. The larger whole depth is achieved by an involute extension at the top and a deeper cutting at the root. The root tooth thickness will change little or not at all. Advantages of taller teeth are the increased transverse contact ratio, the lower intensity of the meshing impact and better load sharing

between adjacent teeth due to the higher elasticity. Within certain limits this means that more tooth bending leads to reduced bending stresses in the root fillet area, which can be verified with finite element calculation (load sharing). In summary, there are strength and noise advantages without disadvantages regarding the function of a gearset. The most important influences of an increased whole depth are summarized in Figure 10. Usual limits for bevel and hypoid gears are a minimal whole depth of 0.8* module and a maximal whole depth of 1.2* module.

Cutter radius. A small cutter radius increases the contact ratio of a bevel and hypoid gearset. The reason is the spiral angle, which increases faster towards the heel in the case of a small cutter radii. Load-affected deformations cause tooth contact movement towards the heel, that causes an increase of the contact ratio while the load rises. The larger spiral angle also reduces the tooth contact movement and the contact spread in heel direction under load that presents a "natural" protection effect; this effect is amplified with smaller cutter radii and reduced if larger cutter radii are used.

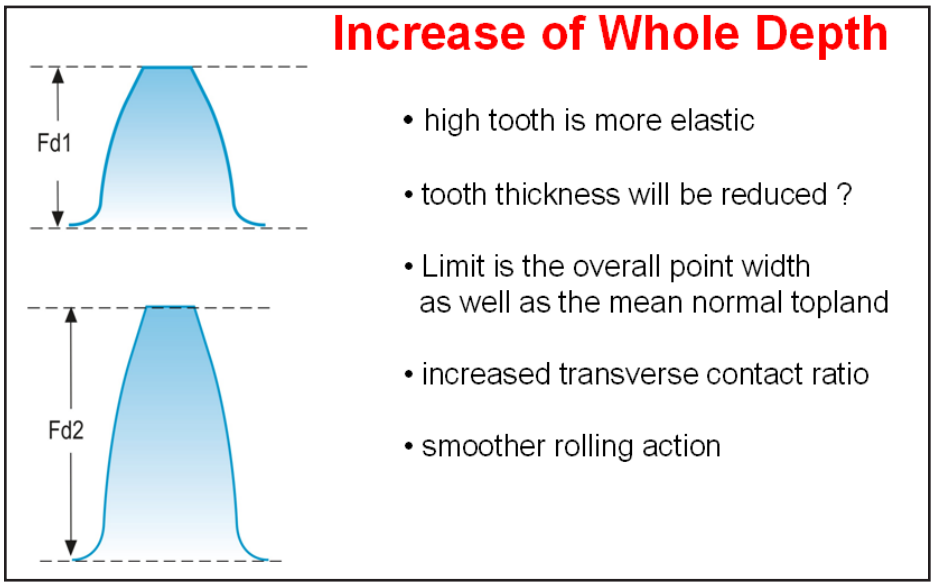

Figure 10 Influence of whole-depth.

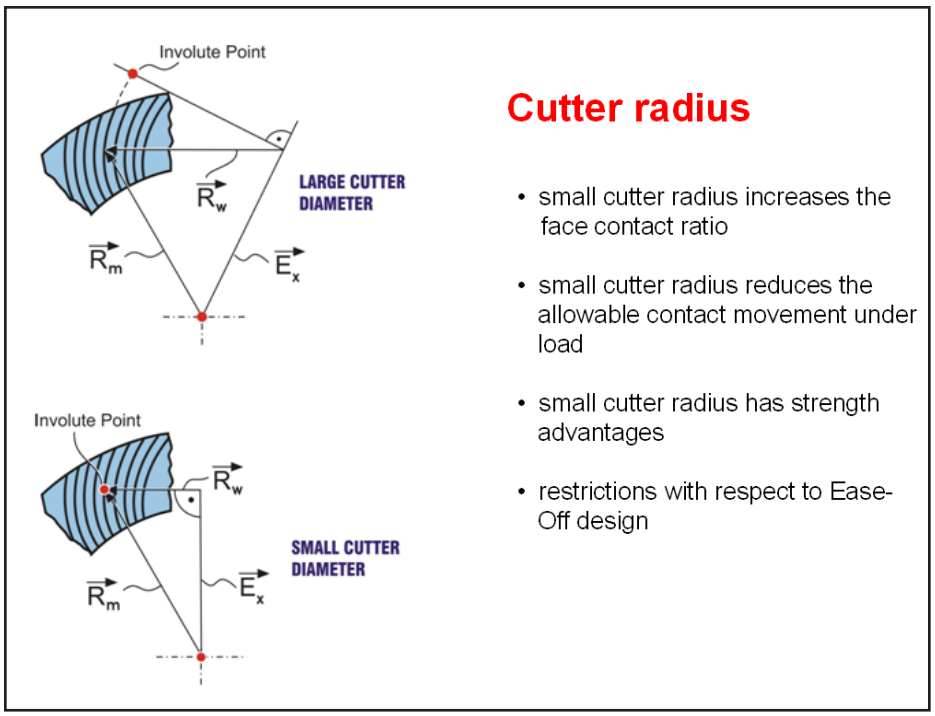

Figure 11 Influence of cutter radius.

In an ideal design the contact pattern without load is positioned towards the toe. A cutter radius should be selected so that a certain contact movement — under load towards the heel — occurs, while at the same time the contact pattern spreads in tooth length direction. In this ideal scenario the contact pattern at nominal load extends over the entire face width — without causing edge contact on heel and toe. The mean point movement towards the heel with increasing load, and the natural opposing of this movement due to the increasing spiral angle, leads to a defined load concentration in the heel area. The load carrying capacity can be significantly increased if the optimal cutter radius is chosen, as the tooth root thickness at the heel is often 50% larger than tooth thickness at the toe. The root bending stresses of bevel and hypoid gears with an optimized cutter radius are quite uniform along the face width; the most important influences of the cutter radius are summarized in Figure 11.

Relationship between involute point and mean cone distance. This section offers a basic explanation of the dependency between the cutter radius and the displacement behavior of spiral bevel and hypoid gears that was discussed in the last section. Load-affected deformations and inaccuracies in the building position of a bevel or hypoid gearset cause a tooth contact movement in the direction of a certain point within or outside of the face width. This point is located where the flank length curvature is equal to the curvature of an involute along the face width. This requirement would be fulfilled in every point along the face width — if the flank line was an involute. Circles or epicycloids fulfill this requirement only at one point, which can lie within or outside of the face width. The base circle radius of this virtual involute is, in the case of face milling, the distance from the generating gear center to the center of the

cutter head. In the case of face hobbing, the base circle of the extended epicycloid is also the base circle of the involute (*see also original text, chapter 2*). The connection between the base circle roll point and any chosen point along the flank line is the radius of curvature at this chosen point. If the vector of the curvature radius is oriented tangential to the base circle, a flank curvature is then generated identical to the involute curvature. Figure 12 can explain the relationships for both bevel gear types — face milling and face hobbing.

Smaller cutter radii are at mid-face or in the toe area, rather than perpendicular to the radial distance (i.e. the base circle single indexing i.e. continuous indexing method). Larger cutter radii lead to involute points at larger diameters. The ratio *Ax*/*Am* is documented in the dimension sheet (see Fig. 1). A value of 1 for *Ax*/*Am* results in an involute point at mid-face of the teeth, meaning that no contact pattern movement under load-affected deformations will occur. This is analogous to the center distance insensitivity of cylindrical gears (*see original text chapter 1*). Although this appears desirable initially, a contact pattern that will not move under any conditions will cause early surface fatigue in the region of the design point (in the flank center). This problem begins to show by evidence of a growing pitting population that can eventually result in flank fracture. The ideal location of the involute point in modern, highly optimized bevel gears is in the middle — between flank center and heel. Recommended values to achieve deflection insensitive gearsets are:

$$1.14 < Ax/Am < 1.2 \text{ and } 0.92 < Ax/Ao < 1.05. \quad (1)$$

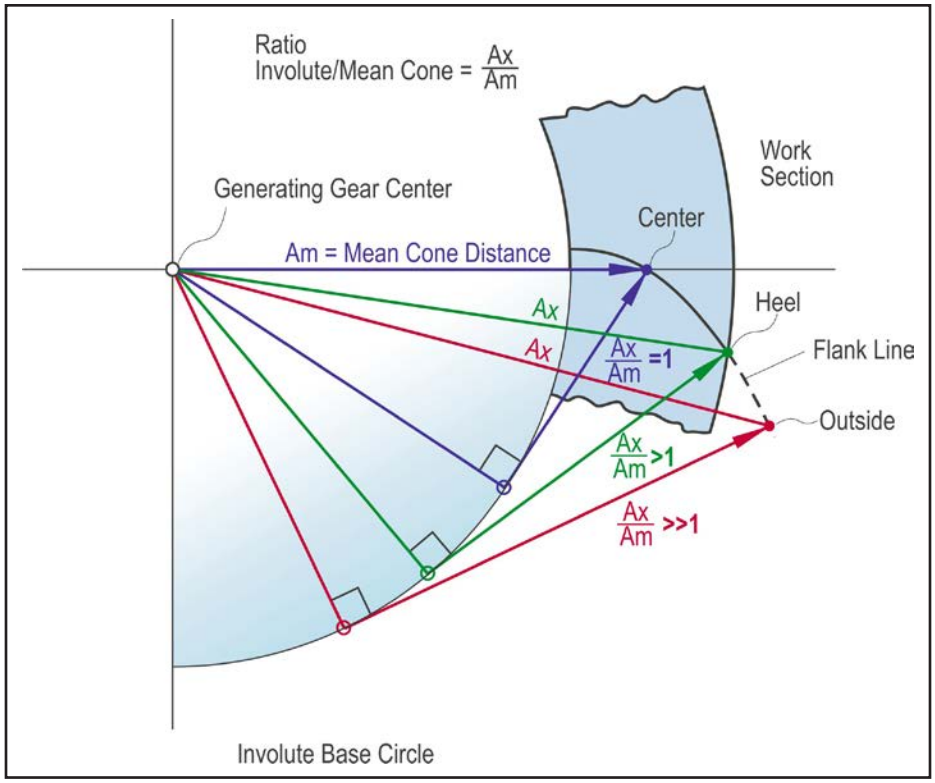

Figure 12 Relationship of involute radius to mean cone distance.

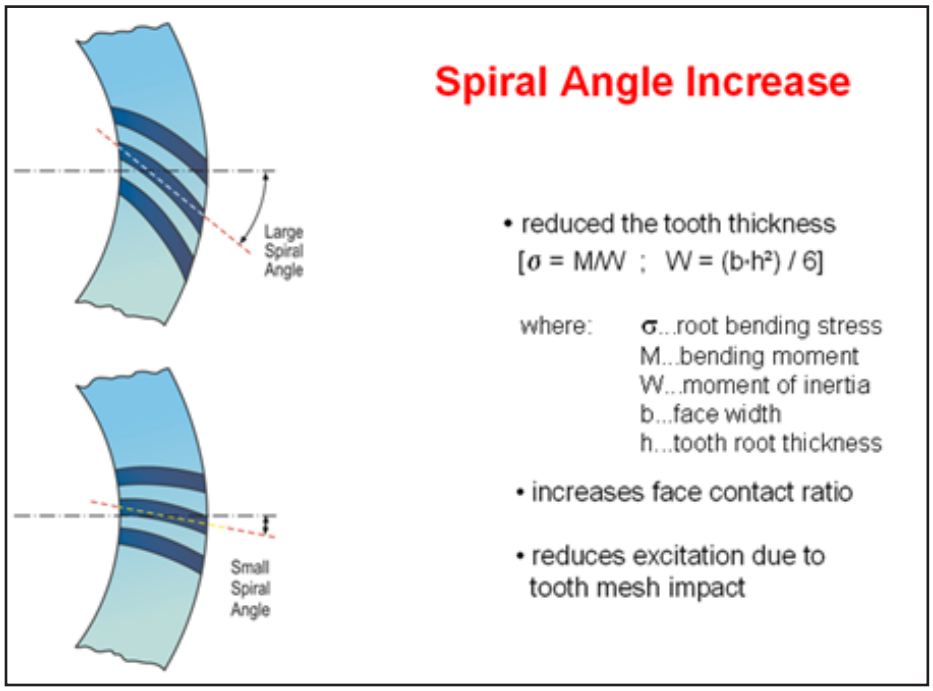

Figure 13 Influence of spiral angle.

Spiral angle. The spiral angle is, per definition, at the center of the face width. A large spiral angle reduces tooth thickness and increases the maximal-possible face contact ratio. The reduced tooth thickness has a quadratic influence on the root bending stress, which reduces the root strength significantly. In contrast, the face contact ratio increases by the tan_{β} of the introduced spiral angle, resulting in a load sharing of additional tooth pairs. Theoretically, the latter effect increases the root bending strength more than the reduction that occurred due to the smaller tooth thickness (Fig. 13). The load carrying capacity of a straight bevel gearset can theoretically increase with the introduction of 30° spiral angle by 18% (*see also original text chapter 4.3.4*).

But the reality is that only a small percentage of the load is transmitted by the neighboring teeth due to the flank crowning. Non-hard-finished bevel gearsets that undergo high deflections in their operation exhibit more load carrying capacity with small spiral angles. Highly optimized, ground bevel gearsets benefit more from the face contact ratio with respect to the load sharing between several teeth. In this case spiral angles between

30° and 35° show optimal results. A larger spiral angle promotes a smooth, rather than an abrupt, tooth engagement and offers more elasticity that results in quieter-running transmissions.

Cutting method. Continuously manufactured bevel gears have a parallel tooth depth; their unrolled flank line is an epicycloid (Fig. 14, top). Continuous cut (face hobbed) bevel gears that are lapped after heat treatment show very good conditions for smooth tooth engagement and quiet operation. Face hobbed flank surfaces feature generating flats that cross the contact lines of the two meshing flanks under an angle. This allows the lapping compound to be present in the contacting zone and to abrasively remove the multitude of contacting points between contact line and generating flats.

Face hobbed bevel gears with a large number of blade groups show a "natural" insensitivity towards load-affected deformations (similar effect as small cutter radii). A simulation of the displacement can help in finding a favorable combination between the number of blade groups and the cutter radius Face hobbed bevel gearsets can be lapped with very good results, but cannot be ground because of the epicyclic flank form and the slot width taper. Face milled bevel gearsets have their generating flats oriented parallel to the contacting lines between pinion and gear, which causes the lapping compound to be "wiped off" and the lapping effect to be diminished. Lapping of face milled bevel gears often increases the magnitude of the generating flats because the contacting lines slide over each generating flat simultaneously, resulting in high-frequency rolling noise.

Face milled bevel gearsets (Fig. 14, bottom) are therefore not well-suited for lapping, and yet allow application of modern and highly precise grinding methods for their hard finishing (*see also original text, chapter 4*). This means that the advantages of grinding are only available for bevel and hypoid gears manufactured in a single-indexing process, which explains today's split of the hard finishing methods (face hobbing → lapping, face milling → grinding).

Face milled bevel gearsets ground after heat treatment can be finished with nonlinear flank modifications that can be adapted to the precise load-affected displacements, and also achieve rolling with minimized mesh impact.

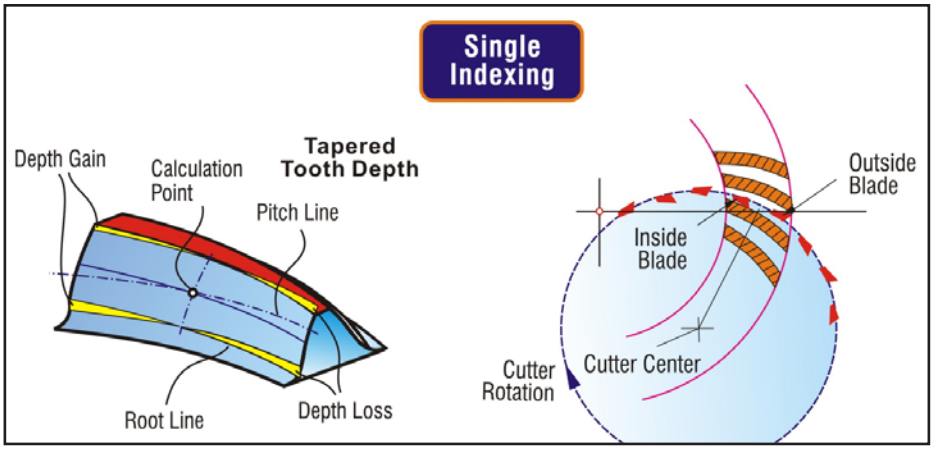

Figure 14 Influence of cutting method.

Limitations in Dimension Sheet Data Alterations

The discussed basic gear data offers many possibilities to improve the strength and rolling behavior of bevel gearsets. Yet, in many cases the parameters are given from an existing transmission whose requirements have somewhat changed. An example is a vehicle that received a higher-powered engine, but the dimension of the axle drive unit cannot increase in size. Another example is a suddenly noticed gear noise that must be eliminated or reduced. It might occur because of an alteration in the vehicle components that influence or change the acoustic transmission path.

For strength optimization, new or improved basic gear data are generally required. An exception is the case where the gear basic data are already optimal and the gearset cannot be enlarged, which however features an Ease-Off that has not been optimized with modern flank modifications. In this case, a significant strength increase using suitable flank form optimizations can be achieved. For noise optimizations the alteration of the basic gear data is not possible in 9 of 10 cases because of time and cost restraints. If the basic parameters have been changed, then requalification of the gearset on a test rig and in a vehicle is required. This is true even if the reason for the change was only based on unacceptable noise emission.

Sophisticated flank optimizations generally have a neutral or positive influence on the strength of a gearset. It is widely accepted in the industry that a gearset does not have to be re-qualified for strength after an Ease-Off optimization, as long as the basic gear parameters remained the same. For certain small improvements or changes, the rule that is taught in engineering around the world should be applied, i.e. — Whenever possible, the proven base geometry must be preserved and only gradual changes, in order to achieve the required improvements, should be implemented. This rule is the key to successful products and short developmental times.

Optimization by flank form modifications. Flank form modifications are deviations of a flank pair from their conjugate condition (*see also originaal text chapter 4*). Conjugate flank pairs are not practically applicable since the conjugate characteristic will vanish in the presence of loads and component tolerances. Instead, edge contact occurs with unfavorable load concentrations, together with a "saw tooth-shaped motion error."

Bevel and hypoid gearsets require Ease-Off in profile and length direction, beginning at the tooth center with zero and increasing outwards towards the boundaries of the teeth. Until now circular flank crowning was applied in nearly all cases; Figure 15 shows the three commonly applied elements of crowning design.

Profile crowning is a circular removal of the flank profile, which appears in the presentation plane as Ease-Off, like a section of a cylinder with the axis of the cylinder oriented in the direction of the flank line (Fig. 15, left). Length crowning

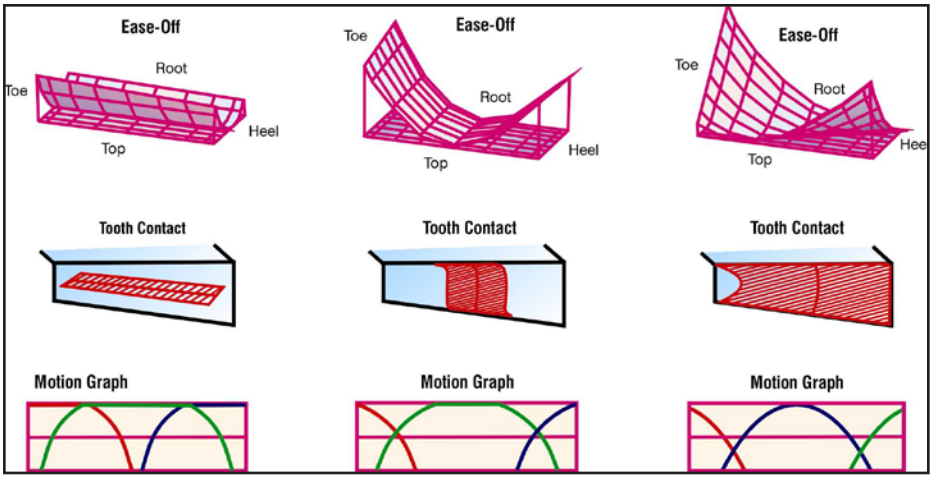

Figure 15 Conventional Ease-Off design elements.

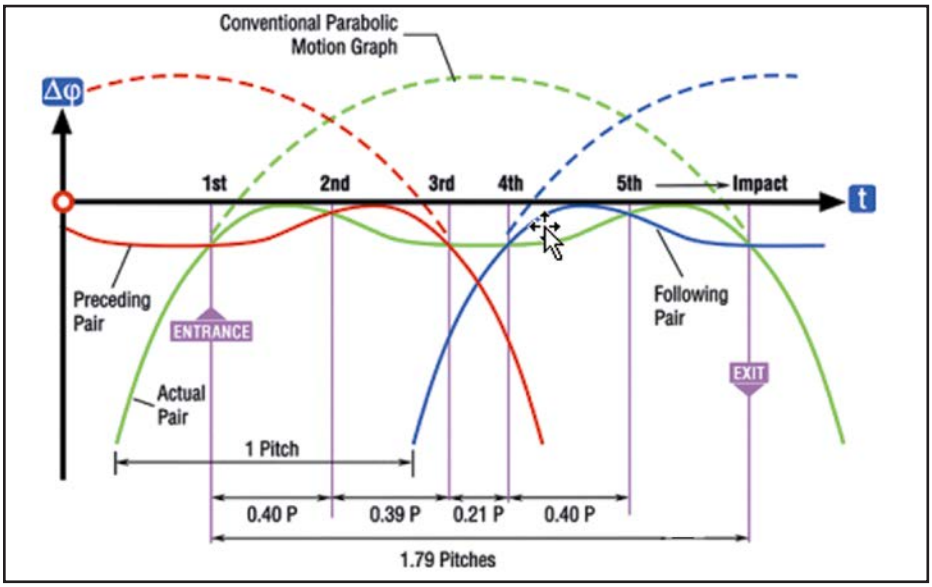

Figure 16 Advanced motion error modifications.

is a circular material removal in flank line direction, which appears in the presentation plane as Ease-Off like a section of a cylinder with the axis of the cylinder oriented in flank line direction (Fig. 15, center). Flank twist is created with a circular material removal that appears in the presentation plane as a section of a cylinder, with the axis of the cylinder oriented in the direction of the contact lines (Fig. 15, right). The figure sequence below the Ease-Off graphics show the tooth contacts and motion errors that result from each particular crowning. When practically applied, bevel and hypoid gears always feature a combination of the three fundamental crowning elements shown in Figure 15.

Today's Gleason bevel and hypoid gear cutting and grinding machines offer the possibility of superimposing higher order motion combinations. These opportunities are utilized by today's flank correction software — e.g., Gleason *UNICAL*, for achieving flank modulations along the path of contact during the generating process, and which are superimposed onto the traditional corrections of Figure 15 in order to achieve nearly any modification of the Ease-Off. This new strategy for Ease-Off design can be realized with Gleason *UNICAL* and is called "selective crowning."

Strength Improvement and Noise Reduction with Ease-Off Modification

Flank form modifications may be used as effective tools in order to achieve noise reduction and strength optimization; these modification possibilities are available today on all Gleason cutting and grinding machines. In a typical case the maximum possible contact ratio from the Dimension Sheet is only achieved under maximal load — and even then the load sharing between adjacent tooth pairs is very unfavorable. With the "right" modification the contact ratio, even under partial load, can be increased. The load sharing of the simultaneously engaged tooth pairs can be arranged so that no tooth pair need be transmitted more than 60% of the torque. 80% or more of the load in traditional bevel and hypoid gear designs is transmitted by single tooth pairs. This indicates that the improvement by certain modern Ease-Off modifications is significant and may result in a 25% increase in root bending strength and a 30% higher surface strength, as compared to gearsets with a standard second order crowning as given in Figure 15.

Higher order modifications can be limited to certain flank sections, thus allowing for adjustment of the correction to meet the different requirements for the entrance and exit zone, as well as the flank center (*see also original text chapter 16*). Figure 16 shows the graphic for a motion transmission generated with a fourth-order flank modification. The effect of a flat motion graph with a wave shape has no disadvantages. The load sharing between adjacent teeth pairs is more uniform and less abrupt than the diagrams in Figure 15 (Ref. 4). The nominal contact ratio for a conventional Ease-Off is equal to one of the higher order Ease-Offs; yet, the higher order Ease-Off utilizes the potential that the nominal contact ratio presents more "intelligently."

For gear optimization with higher order flank surface modifications, it must be noted that gearsets that have already been optimized with a modern tool, such as Gleason *UMC*, that the potential for improvement might already be exhausted. The attempt to achieve even more strength increase with more optimization might result in a turn in the opposite direction. Also, the attempt to address certain noise excitations with additional *UMC* optimizations might reduce the strength of the gearset. It is recommended in any case to conduct finite element calculations parallel to the geometric flank form optimizations in order to track changes in the strength characteristics during the improvement process.

Summary

It is important to realize that the basic gear data of the macrogeometry represent the foundation of the physical properties of a gearset. Flank form modifications can contribute to utilizing those properties completely. A conventional Ease-Off does not achieve the entire strength potential of a gearset. An optimized Ease-Off makes it possible to utilize all of the physical possibilities of a gearset by optimally using existing proportions. An optimal combination between gear basic data and flank form modifications can often enable the transmission of twice the torque and, at the same time, be significantly quieter in comparison with a gearset with less-than-optimal design.

Generally, strength optimization requires a new, basic design. An exception is when the basic design is found to be optimal, but the Ease-Off is conventional and shows significant room for improvement. In this case modern flank form modifications can contribute to a great improvement of the load carrying capacity—and might in fact be the only possibility if the existing gear size cannot be increased. Noise optimization requires in most cases only a flank form modification. An exception is given if the present design already has a modern flank form optimization. In this case, noise optimization will not be easily accomplished.

If an improvement of the noise characteristic via Ease-Off still seems possible, then a finite element calculation of the situation, before and after the optimization, is recommended. In the case where critical stress conditions are discovered, it might be necessary to optimize the gear basic data first in order to find a new foundation that fulfills the strength requirements, and then as a second step to conduct a modern flank form optimization. The elements making the teeth more elastic (*see original text chapters 6.21, 6.24 and 6.26*) should be considered to assure good load sharing and to guarantee a reduction in noise emission.

The possibility of improving the parameters of a given basic gear design should not be treated as a burden, but welcomed as a seldom-occurring opportunity to improve the roll and strength characteristic of a gearset that has not been brought up to date. Nevertheless, the existing design should be used as a basis since it is possible in most cases to adjust the gearset with a number of smaller, but significant, improvements to meet today's stringent requirements. One should not make the mistake of believing that the time-consuming practice of gear design can be saved by certain flank optimizations. In short, the last section can be summarized by the following rule: "The basic gear parameters give the direction for the physical properties of a gearset, while modern flank form optimizations only assure a better access to those properties."

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