KISSsoft special training
Bevel and hypoid gears

KISSsoft AG, A Gleason Company
Rosengartenstrasse 4, 8608 Bubikon, Switzerland
T. +41 55 254 20 50, info@KISSsoft.AG, www.KISSsoft.AG

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Bevel gears

- Straight bevel gear
- skewed bevel gears
- spiral bevel gears
  - without offset (Bevel gears)
  - with offset (Hypoid gears)

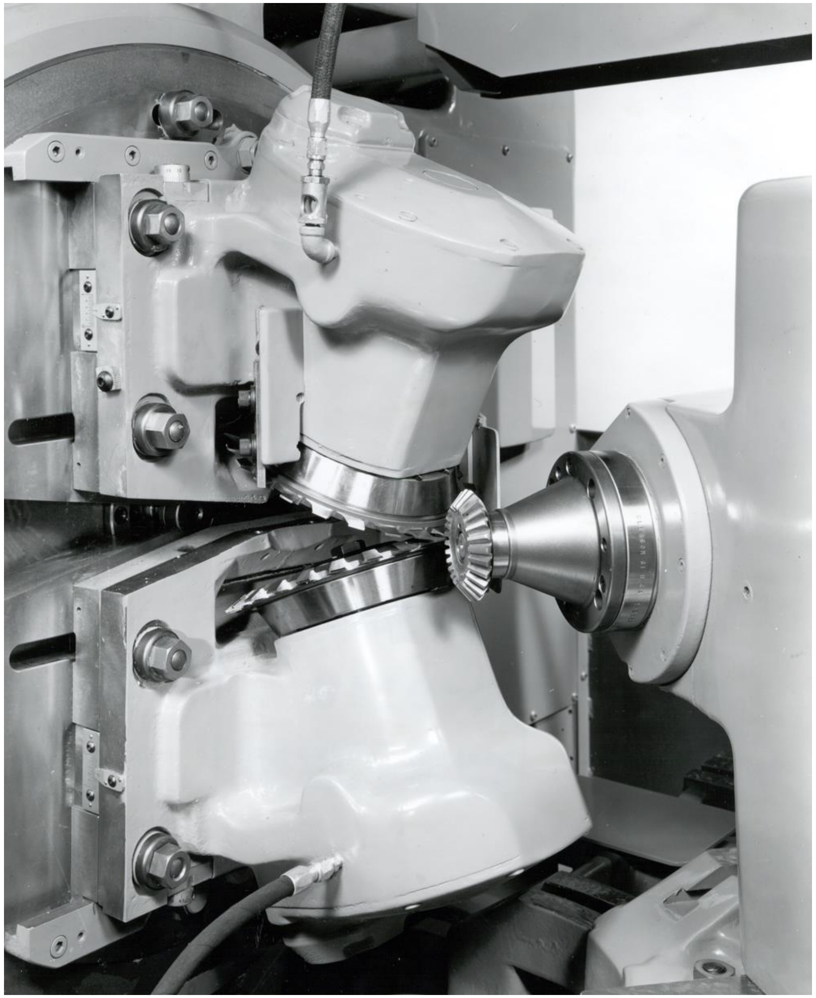
Coniflex

Mechanical machines:

- Generating process only

CNC machines:

- generating and formate process available



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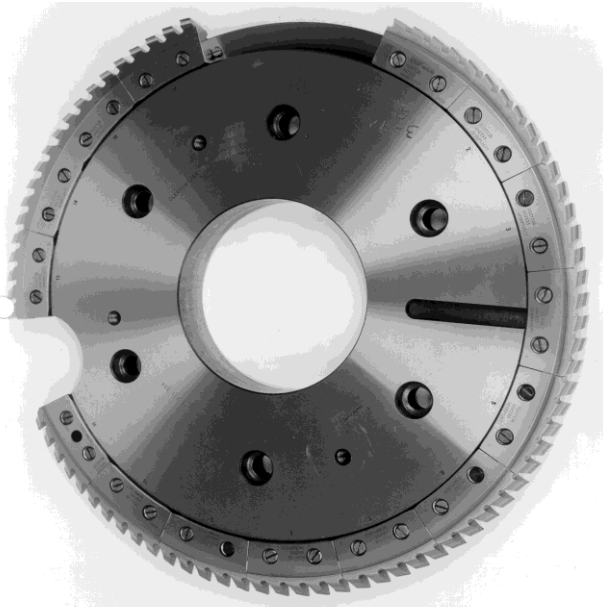
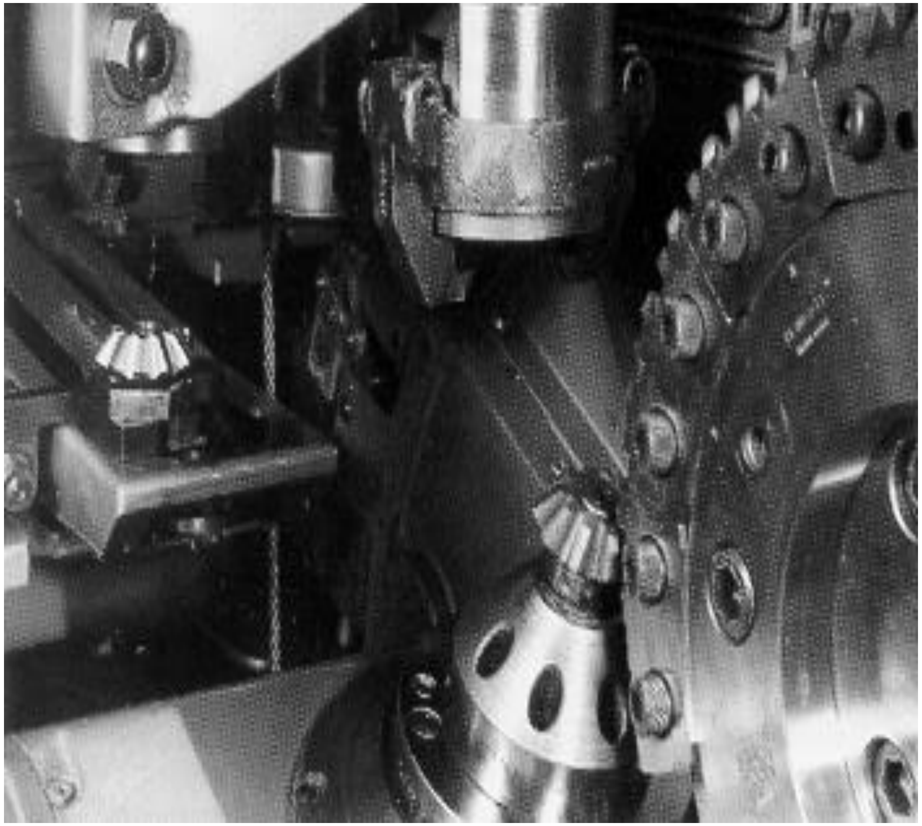
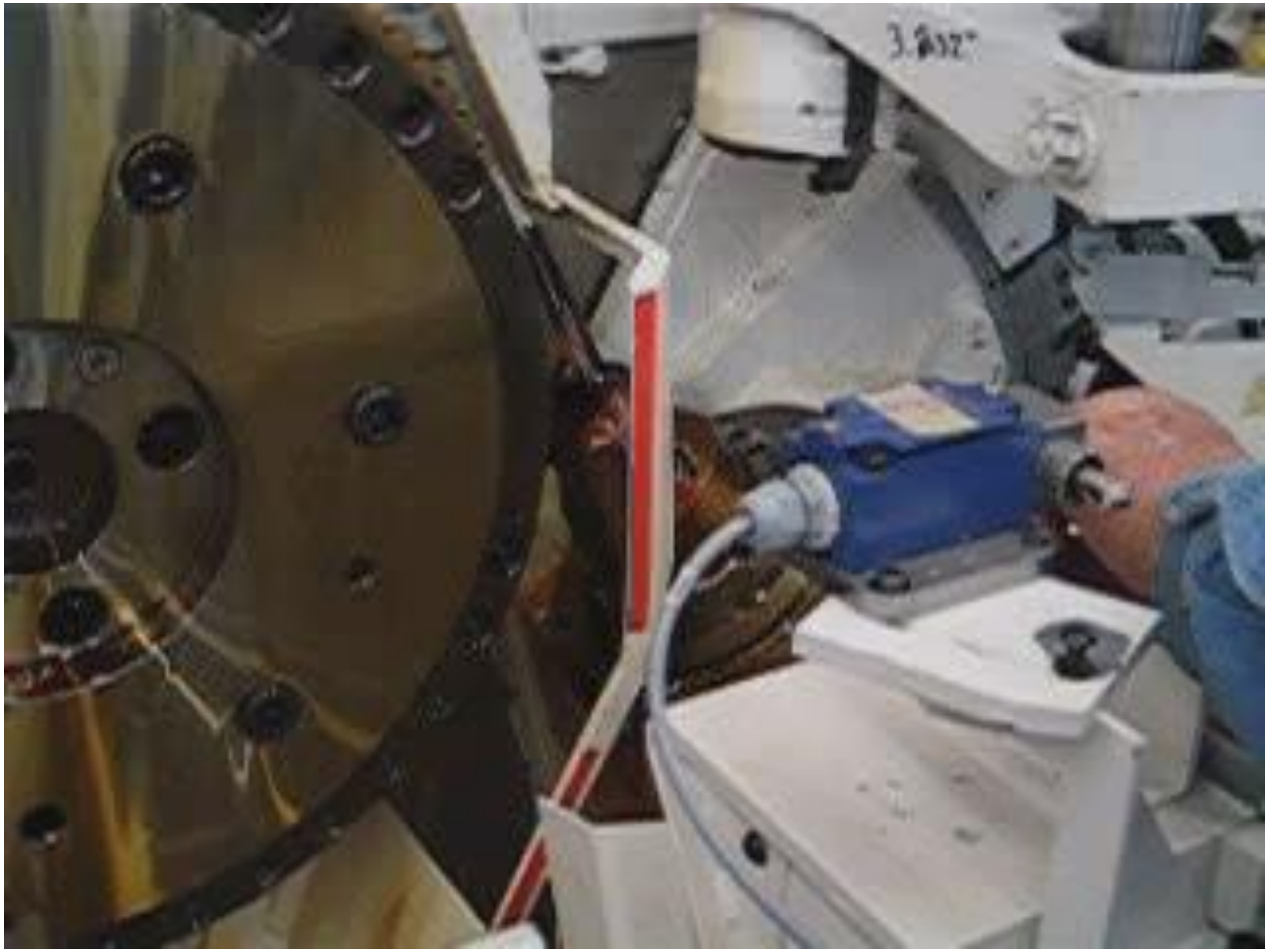




Cutting system straight bevel gears



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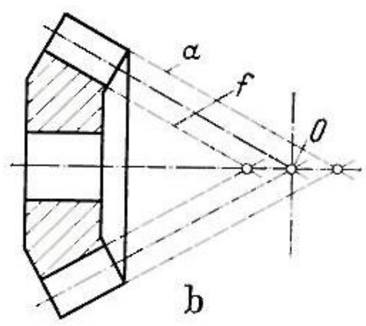
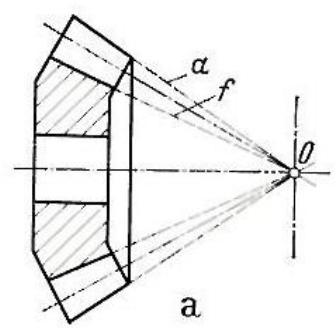


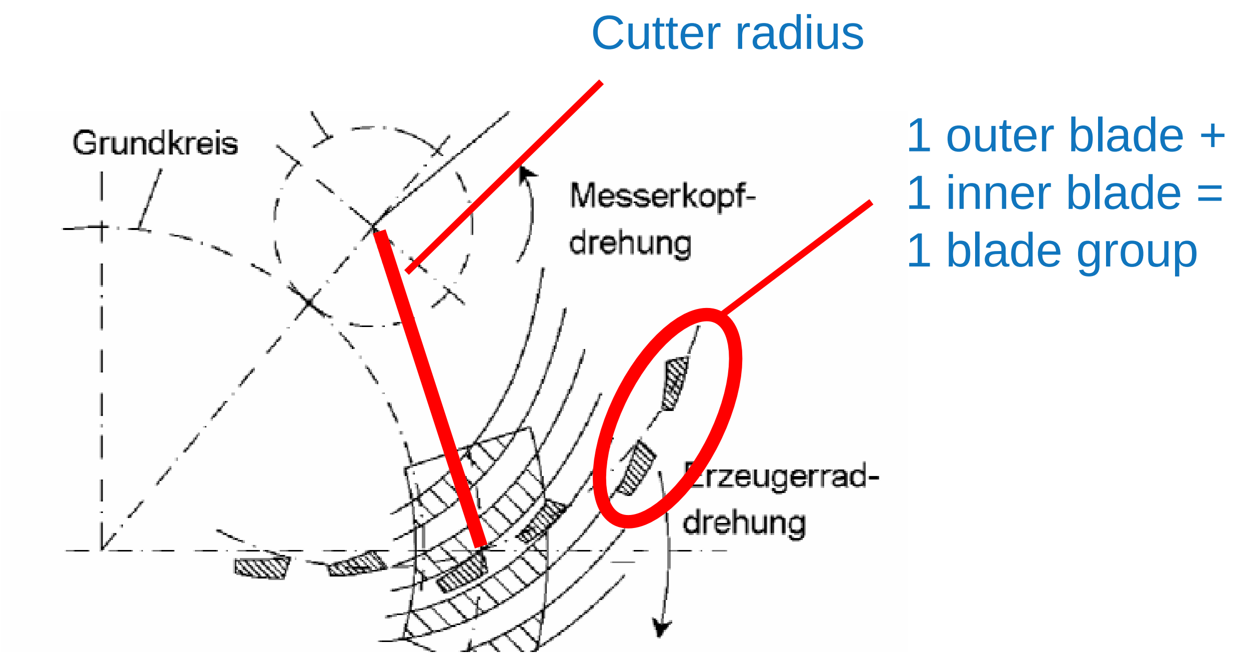


Face Hobbing
(continuous indexing)
constant tooth height

Typical brand names are:
Klingelnberg Palloid®
Klingelnberg Zyklo-Palloid®
Gleason TRI-AC®

Face Milling
(single indexing)
modified tooth height

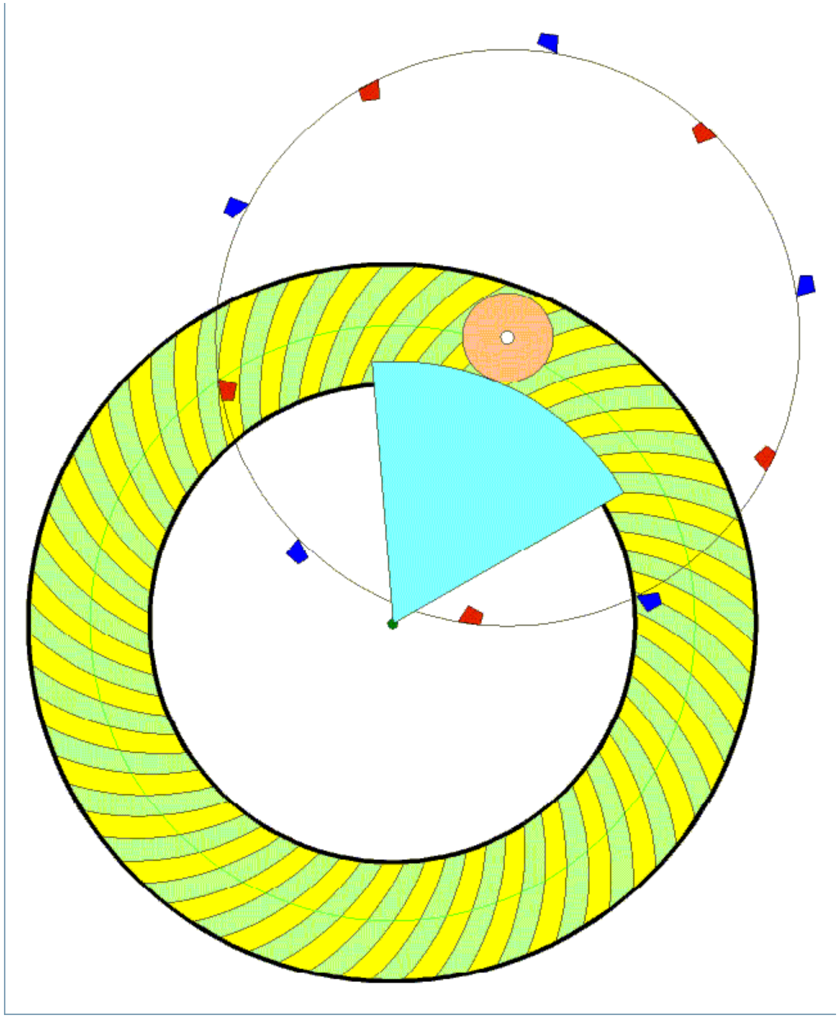
Gleason 5-cut
Gleason Duplex Completing
Klingelnberg ARCON



The workpiece rotates continuously while the cutting tool plunges.

The effective curvature radius is influenced by the number of blade groups and cutter radius.

Face Hobbing



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Face Hobbing - Pinion

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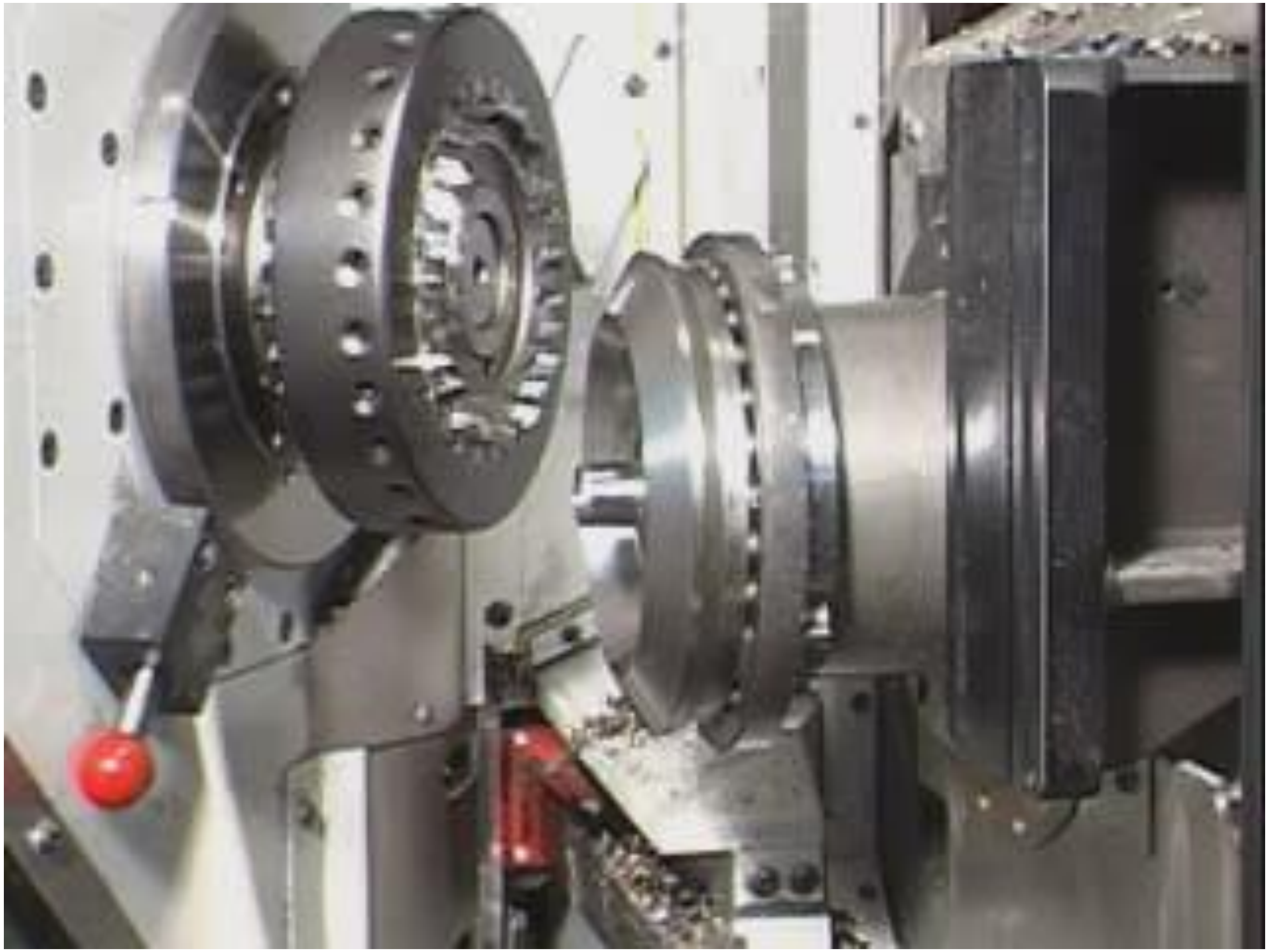
Face Hobbing - Gear



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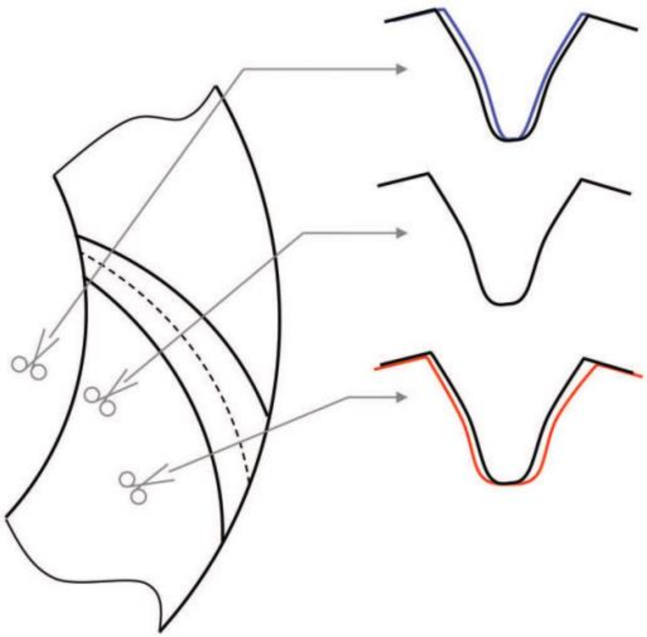

The tooth height is constant

The slot width is varying

The lengthwise curvature is an elongated epicycloid
➔ grinding is not (directly) possible, lapping or skiving (HPG) is applied

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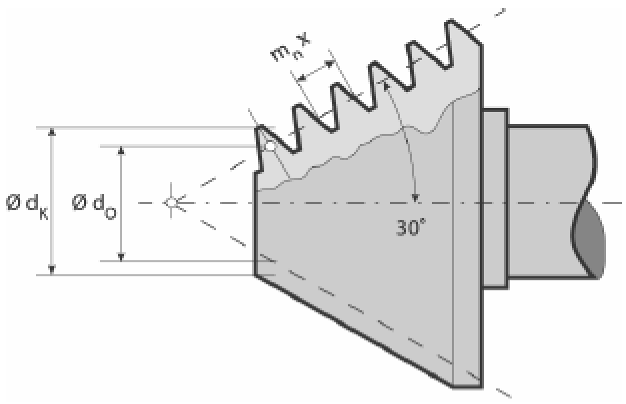
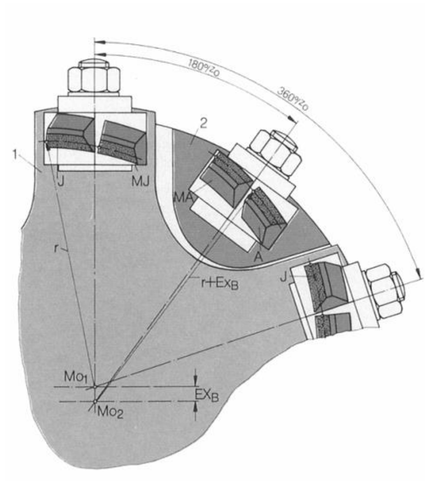
Cutting tools – Universal cutter head





Palloid®

Defined by cutting length SF and diameter dk

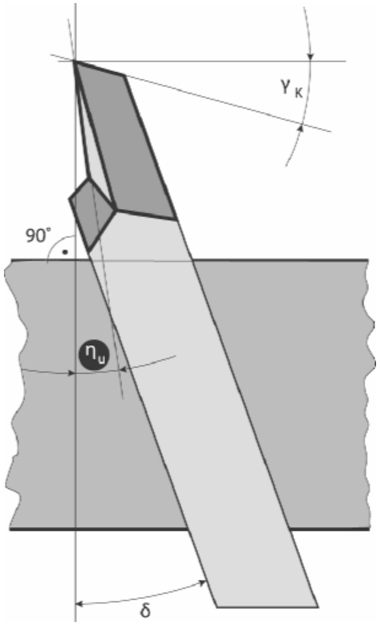
➔ Warning in KISSsoft if the cutter size doesn‘t fit

Zyklo-Palloid®

Defined by blade groups and cutter radius, or Klingelnberg machine type

➔ Sizing of cutter radius in KISSsoft possible

Cutting tools – individual blade design (stick blade system)



Gleason PENTAC® FH
Gleason Spiroform
Gleason TRI-AC®
Gleason Cyclocut™
Klingelnberg SPIRON
Oerlikon FS

Defined by:
blade groups and cutter radius

➔ Sizing of cutter radius in KISSsoft possible

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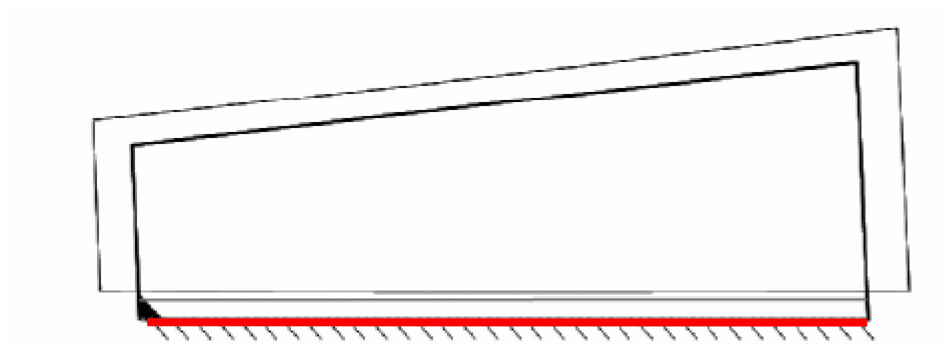
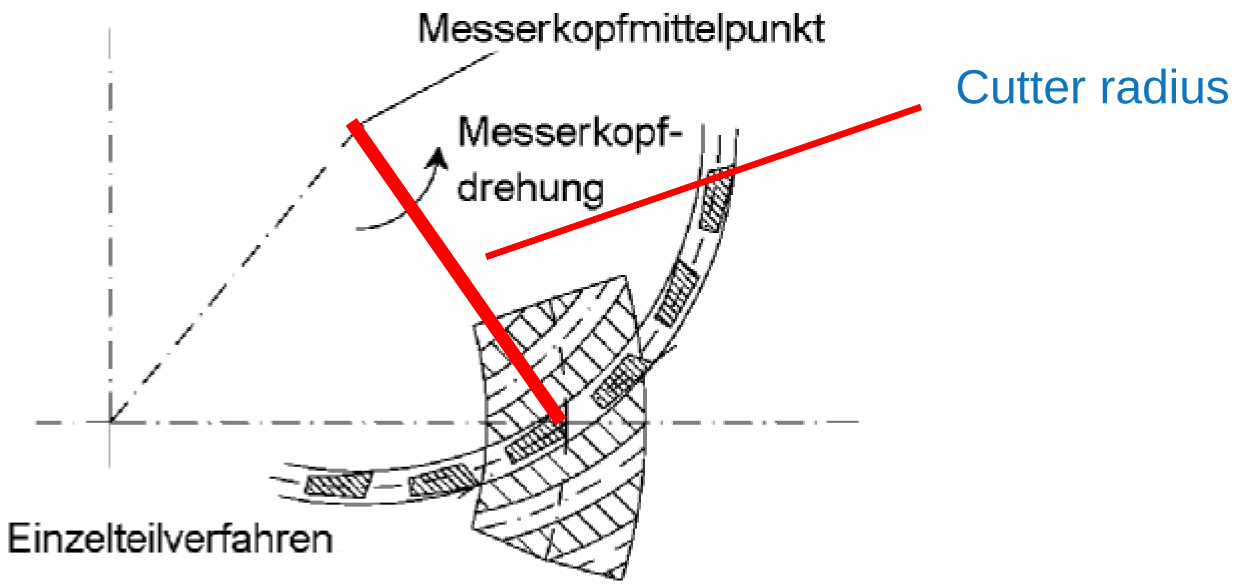
Reference profile



Face hobbing requires only little tilting of the cutter head in order to create the required lengthwise crowning. Zyklo-Palloid® doesn‘t apply tilt at all.

Hence also the root land is flat. There is no risk at the toe or heel side to get interference with the counterpart. The recommended tip clearance c* is:

Face Hobbing :	0.25
Zyklo-Palloid®:	0.25
Palloid :	0.30



The workpiece has no rotation while the cutting tool plunges.

The effective curvature radius is only determined by cutter radius and cutter tilt.

Face Milling - Pinion

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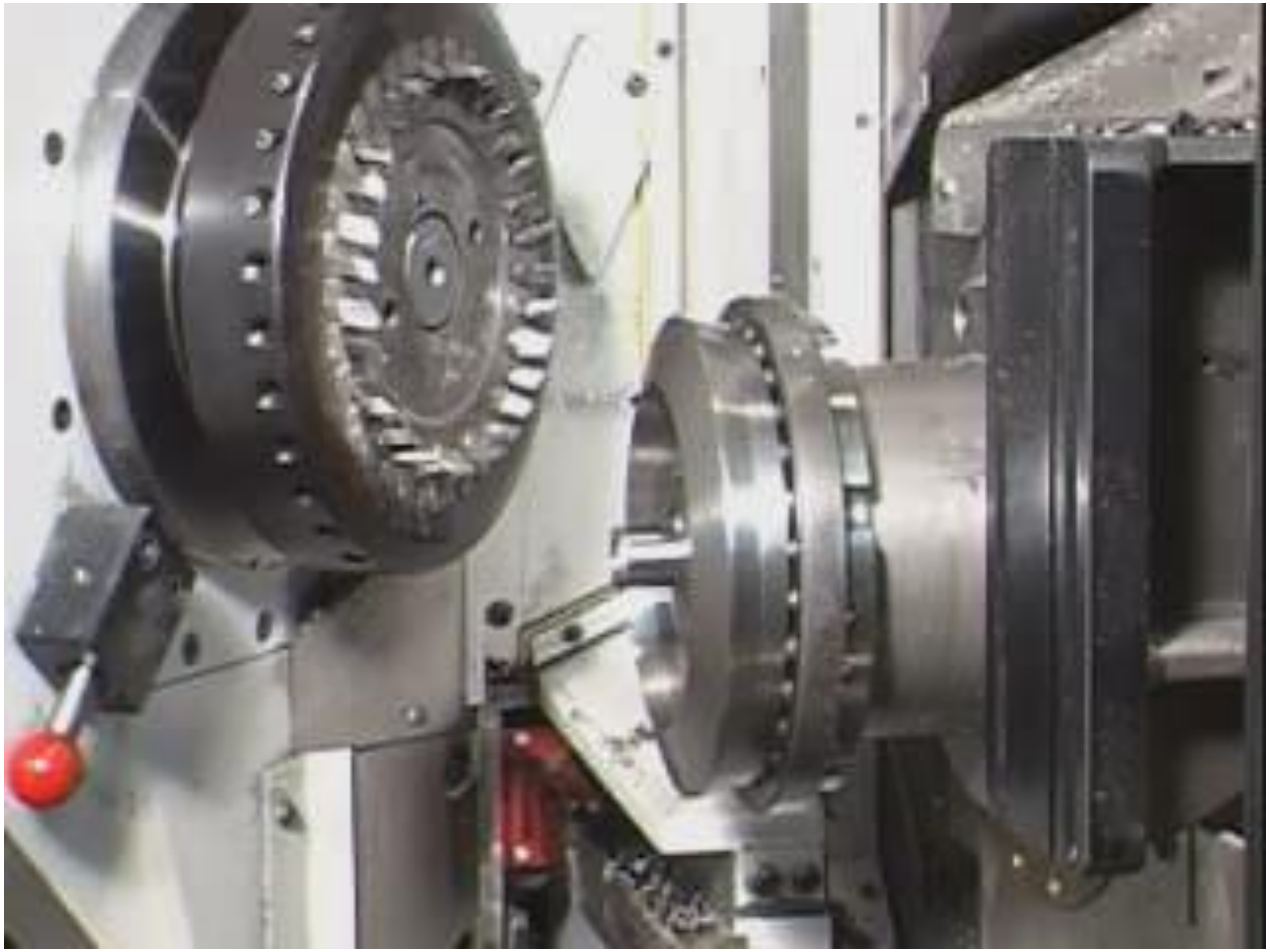
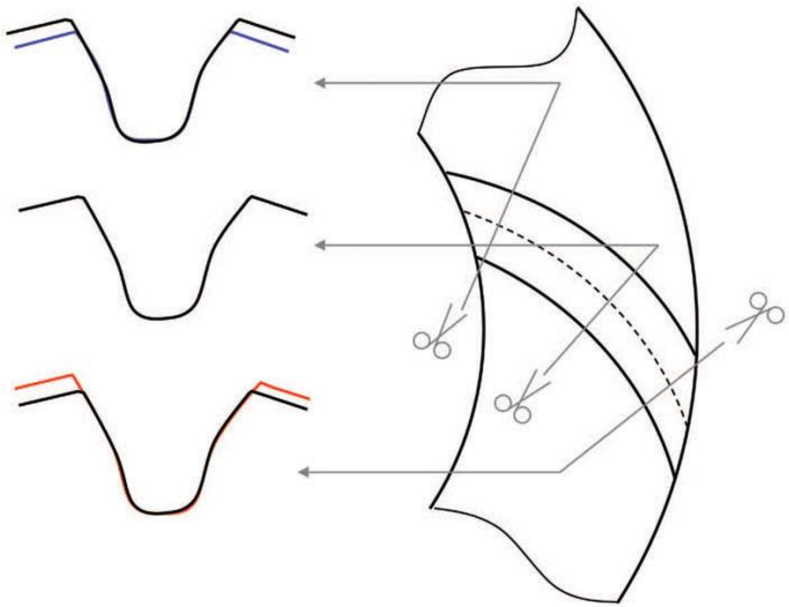
Face Milling - Gear



Geometry



The tooth height is not constant

The slot width is constant (Duplex Completing) or modified (5-CUT FIXED SETTINGS)

The lengthwise curvature is an arc of a circle
➔ grinding or lapping is possible

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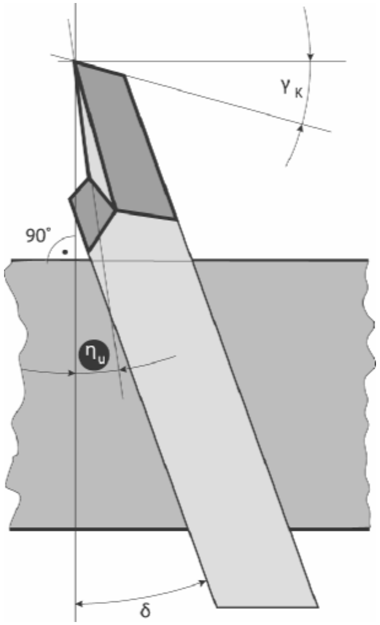
Cutting tools – individual blade design (stick blade system)



Gleason PENTAC® FM
Gleason RSR®
Klingelnberg ARCON

Defined by:
cutter radius

➔ Sizing of cutter radius in KISSsoft possible

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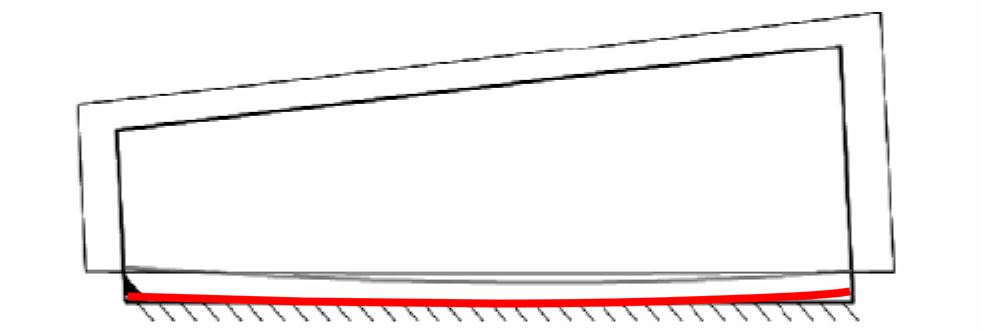
Reference profile



Face Milling requires bigger tilting of the cutterhead in order to create the required lengthwise crowning.

Hence the root land is not flat and there is a higher risk at the toe or heel side to get interference with the counterpart (especially for Formate Gears). The recommended tip clearance c* is:

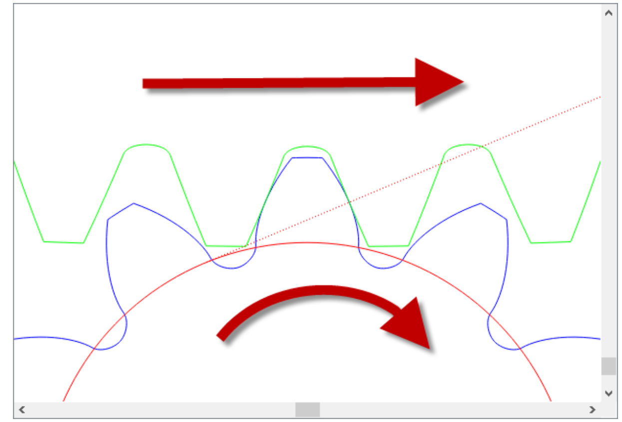
Face Milling (Duplex): 0.35
Face Milling (5 cut): 0.3

Generating process

After plunge process, the tool generates the workpiece by the generating motion.

Application:

- Zyklo-Palloid (both members)
- Pinion (always)
- Ring gear if ratio < 2..2.5

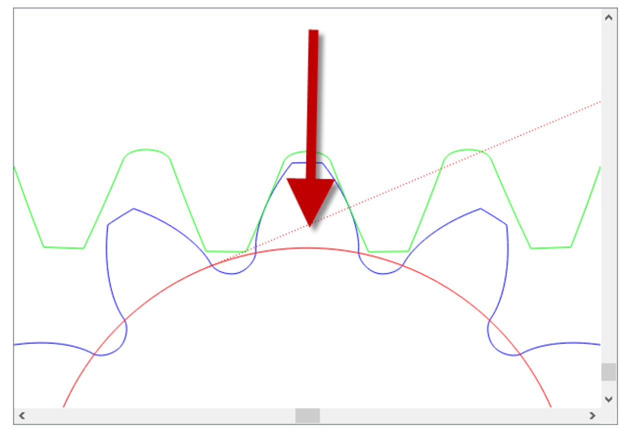


Formate process (Non-Generate)

Only plunge process, no generating process

Application:

- Ring Gear if ratio > 2..2.5



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Tab ‘Process’:

	Gear 1	Gear 2
Manufacture type	generate	generate

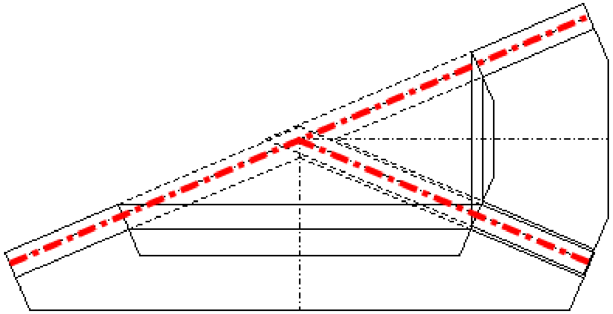
Ranges for achieved quality (samples for d_{e2} = 250 mm):

- Lapping: Quality 8..9
- Grinding: Quality 2..4
- HPG skiving: Quality 3..5

The quality number influences the strength safety values (by $K_{H\alpha}$).

1. Determination of pitch cone parameters (acc. to ISO 23509)

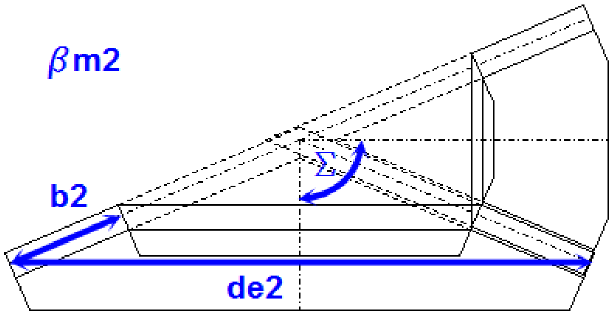
The dimensions at outer pitch cone are independent of straight, helical or spiral bevel type, and independent of cutting method Gleason, Klingelnberg



1. Determination of pitch cone parameters

For the calculation of the pitch cone parameters, a set of initial data is necessary:

- Number of teeth z_1, z_2
- Shaft angle Σ
- Outer pitch diameter of gear (wheel) d_{e2}
- Face width b_2
- Mean spiral angle β_{m2}



These parameters may be determined with the dimensioning calculation based on operating data, e.g. following the Klingelnberg dimensioning calculation procedure.

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1. Determination of pitch cone parameters

Gear ratio u: $$u = \frac{z_2}{z_1}$$

Pinion pitch angle δ_1: $$\delta_1 = \arctan \frac{1}{u}$$

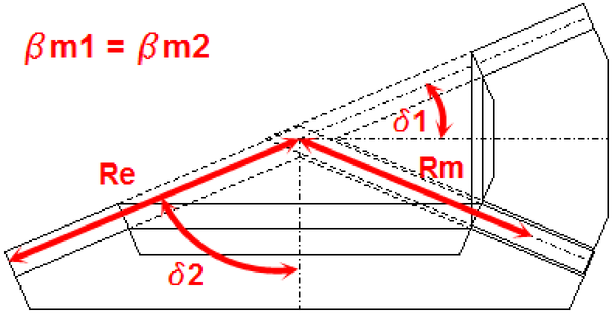
Gear pitch angle δ_2: $$\delta_2 = \Sigma - \delta_1$$

→ „First Aid“ - Formula

Outer cone distance R_e: $$R_{e1,2} = \frac{d_{e2}}{2 \sin \delta_2}$$

Mean cone distance R_m: $$R_{m1,2} = R_{e2} - \frac{b_2}{2}$$

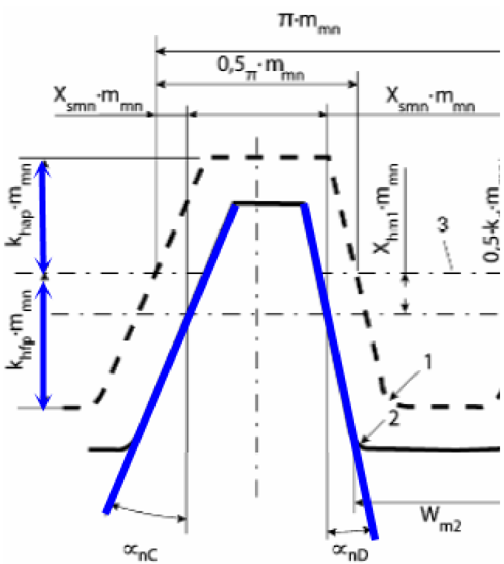
Mean spiral angle β_{m2}: $$\beta_{m1} = \beta_{m2}$$



2. Gear dimensions

A set of additional data is necessary:

- Pressure angle α
- Profile shift coefficient x^*_{hm1}
- Addendum and dedendum factors h^*_{fp}, h^*_{ap}
- Thickness modification coefficient x^*_{smn}
- backlash j_n
- Addendum and dedendum angles $\Theta_{a,f}$



The addendum and dedendum factors and angles are depending on the applied cutting method. Profile shift coefficient is applied based on e.g. avoiding undercut on pinion toe side (inner side).

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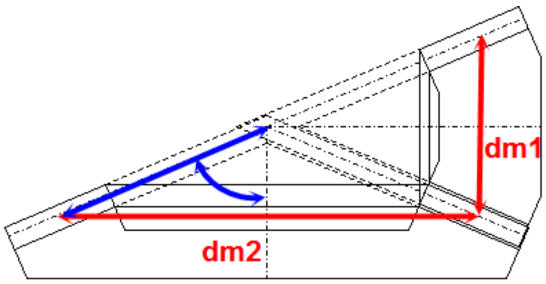
2. Gear dimensions

2.1 Determination of basic data



Mean pitch diameter d_{m1}: $$d_{m1} = 2R_{m1} \sin \delta_1$$

Mean pitch diameter d_{m2}: $$d_{m2} = 2R_{m2} \sin \delta_2$$

Mean normal module m_{mn}: $$m_{mn} = \frac{d_{m2} \cos \beta_{m2}}{z_2}$$

Outer transverse module m_{et}: $$m_{et2} = \frac{d_{e2}}{z_2}$$ → „First Aid“ - Formula

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2. Gear dimensions

2.2 Determination of tooth depth at calculation point

Mean addendum gear h_{am2}:

$$h_{am2} = m_{mn}\left(h^*_{ap} - x_{hm1}\right)$$

Mean dedendum gear h_{fm2}:

$$h_{fm2} = m_{mn}\left(h^*_{fp} + x_{hm1}\right)$$

Mean addendum pinion h_{am1}:

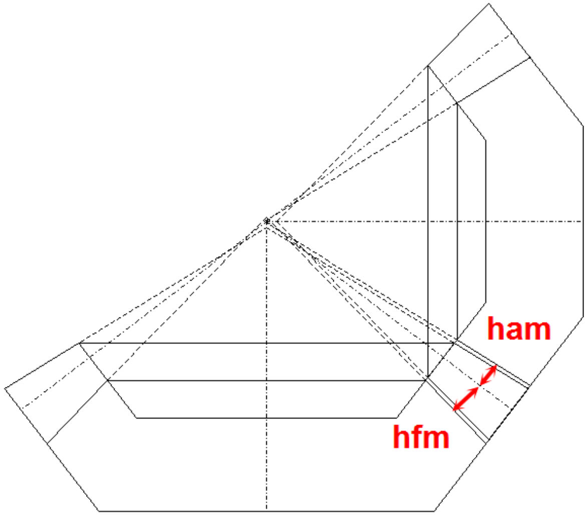
$$h_{am1} = m_{mn}\left(h^*_{ap} + x_{hm1}\right)$$

Mean dedendum pinion h_{fm1}:

$$h_{fm1} = m_{mn}\left(h^*_{fp} - x_{hm1}\right)$$

Clearance c:

$$c = m_{mn}\left(h^*_{fp} - h^*_{ap}\right)$$



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2. Gear dimensions

2.3 Sum of dedendum angles $\Sigma\Theta_f$, addendum and dedendum angles

Standard taper:

$$\Sigma\Theta_{fS} = \arctan\left(\frac{h_{fm1}}{R_{m2}}\right) + \arctan\left(\frac{h_{fm2}}{R_{m2}}\right)$$

Uniform depth :

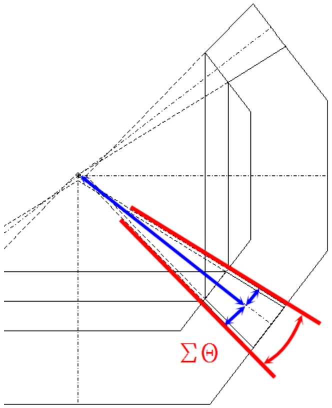
$$\Sigma\Theta_{fU} = 0$$

Duplex taper:
(constant slot width)

$$\Sigma\Theta_{fC} = \left(\frac{90 m_{et}}{R_{e2} \tan\alpha \cos\beta_m}\right)\left(1 - \frac{R_{m2} \sin\beta_{m2}}{r_{c0}}\right)$$

Modified taper:
(modified slot width)

$$\Sigma\Theta_{fM} = \Sigma\Theta_{fC} \qquad \Sigma\Theta_{fM} = 1.3 \cdot \Sigma\Theta_{fS}$$



The sum of dedendum angles $\Sigma\Theta_f$ is calculated depending on the cutting method as well as partially on cutter radius r_{c0}, etc. From sum of dedendum angles $\Sigma\Theta_f$ the addendum angles Θ_a and dedendum angles Θ_f are determined.

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2. Gear dimensions

2.4 Determination of tooth depth:

Outer addendum h_{ae}:

$$h_{ae1,2} = h_{am1,2} + \frac{b}{2}\tan\Theta_{a1,2}$$

Outer dedendum h_{fe}:

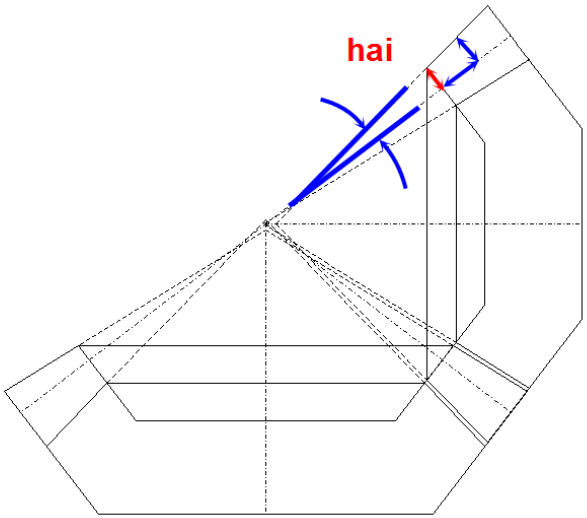
$$h_{fe1,2} = h_{fm1,2} + \frac{b}{2}\tan\Theta_{f1,2}$$

Inner addendum h_{ai}:

$$h_{ai1,2} = h_{am1,2} - \frac{b}{2}\tan\Theta_{a1,2}$$

Inner dedendum h_{fi}:

$$h_{fi1,2} = h_{am1,2} - \frac{b}{2}\tan\Theta_{f1,2}$$



2. Gear dimensions

2.5 Determination of blank dimensions

Outside (tip) diameter d_{ae}:

$$d_{ae1,2} = d_{e1,2} + 2h_{ae1,2}\cos\delta_{1,2}$$

Inside (tip) diameter d_{ai}:

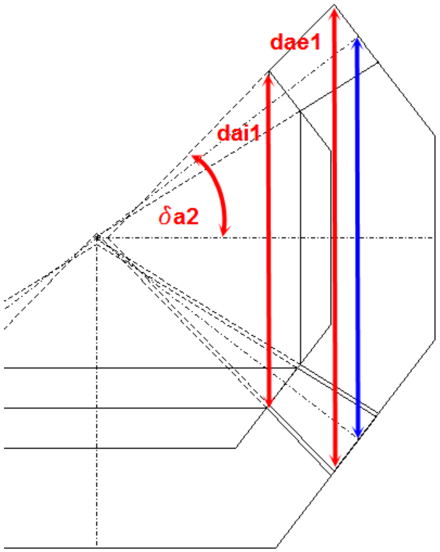
$$d_{ai1,2} = d_{i1,2} + 2h_{ai1,2}\cos\delta_{1,2}$$

Face angle $\delta_{a1,2}$:

$$\delta_{a1,2} = \delta_{1,2} + \Theta_{a1,2}$$

(Root angle $\delta_{f1,2}$:)

$$\delta_{f1,2} = \delta_{1,2} - \Theta_{f1,2}$$



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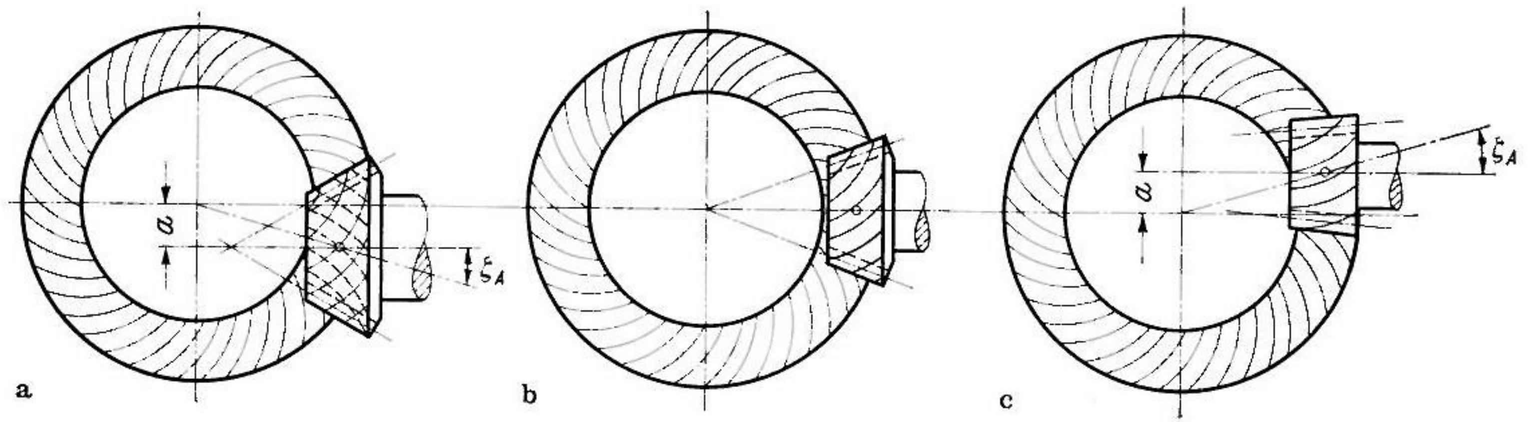


Hypoid gears have an offset between the axes.
The offset leads to a bigger pinion diameter (positive offset) and therefore higher strength. Also the overlap is higher and the gears are quieter.

The offset creates horizontal sliding and therefore higher losses and a higher risk of scuffing.

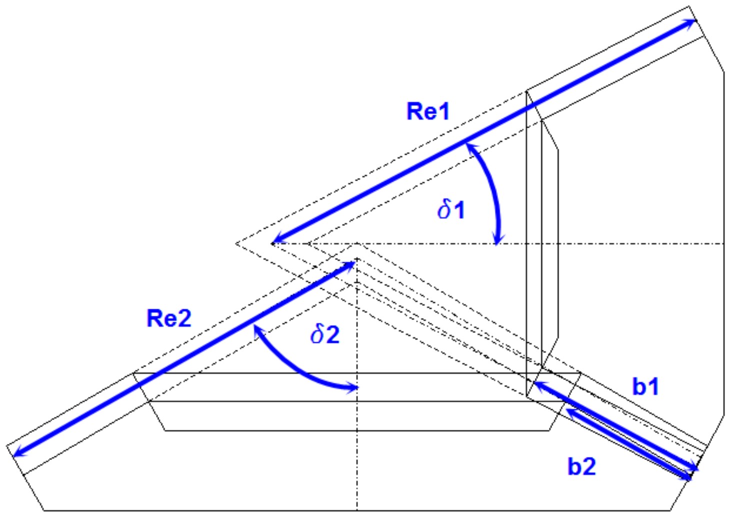
Specialities of hypoid gear geometry are:

The pitch apex don‘t intersect in the crossing point.

The calculation of pitch angles is an iteration, also also depending on i.e. the cutter radius.

Face width of pinion is higher than face width of gear, due to contact ratio.

Spiral angles of pinion and gear are different.



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Limit pressure angle

The limit pressure angle modifies the pressure angle and is required in order to balance the meshing conditions for drive and coast side of hypoid gears.

The limit pressure angle is considered with an influence factor $f_{\alpha lim}$ differently for each cutting method:

$f_{\alpha lim}$ for Face Hobbing:	1
$f_{\alpha lim}$ for Face Milling (Duplex):	0.5
$f_{\alpha lim}$ for Zyklo-Palloid® :	0

➔ in KISSsoft, the factor can be entered under „additional data“

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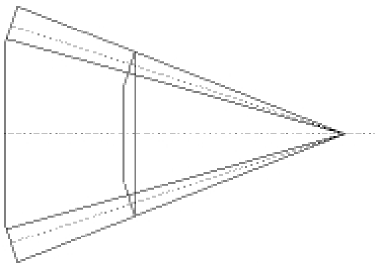
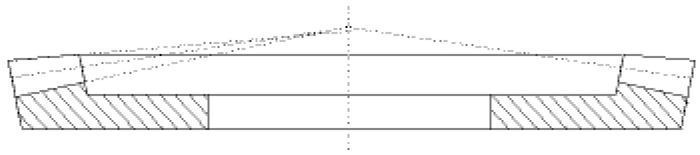
The cone type „standard“ means the geometry where all apex coincide in the crossing point, in one point.

KISSsoft: Standard, fig 1



For a constant tip clearance of pinion tip to wheel root. The face angle of wheel is own input and hence also non-constant tip clearance is possible.

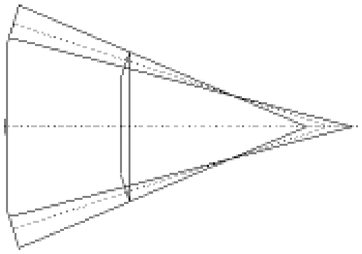
KISSsoft: Standard, fig 4



For own input of individual face and root angles (gear 2). Neither root nor face apex coincide in the crossing point.

KISSsoft: Standard, fig 2

→ **recommended for straight bevel gears**



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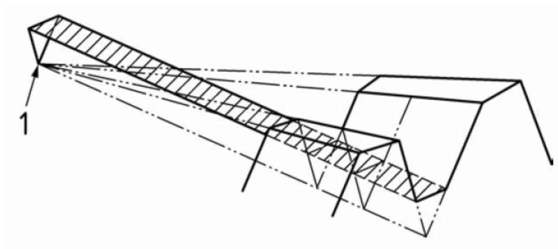
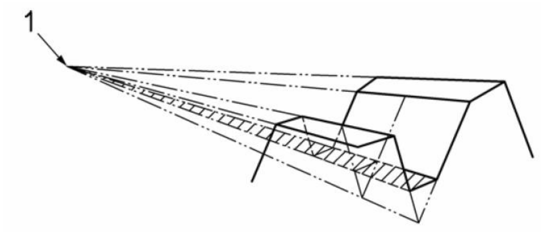
The cone type ‘Duplex’ means the root line is changed, so that the slot width is constant. This is required because the cutting method ‘Completing’ cuts both flanks in one m/c setting.

KISSsoft: constant slot width, fig 2



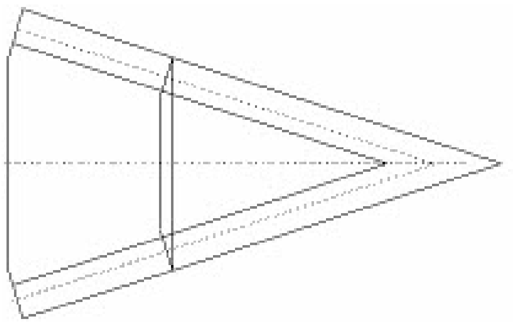
The cone type ‘Tilted root line TRL’ means the slot width is modified. This results from the cutting method ‘5-cut’ where different m/c settings are applied for pinion.

KISSsoft: modified slot width, fig 2



The constant tooth height is applied for face hobbing (Klingelnberg or Oerlikon).

KISSsoft: Uniform depth, fig 3



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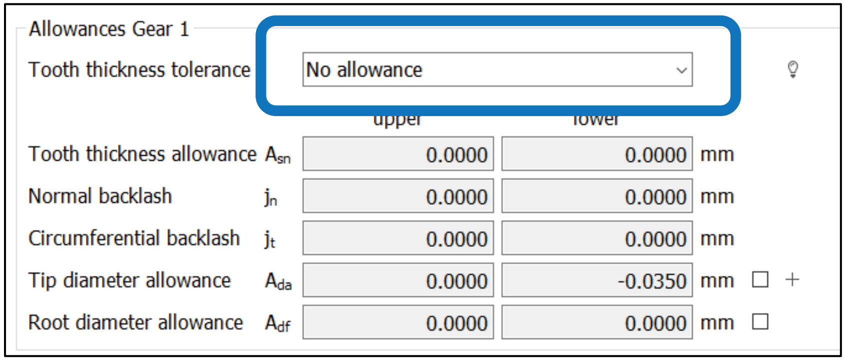
Using a cutting method with **individual blade design**:

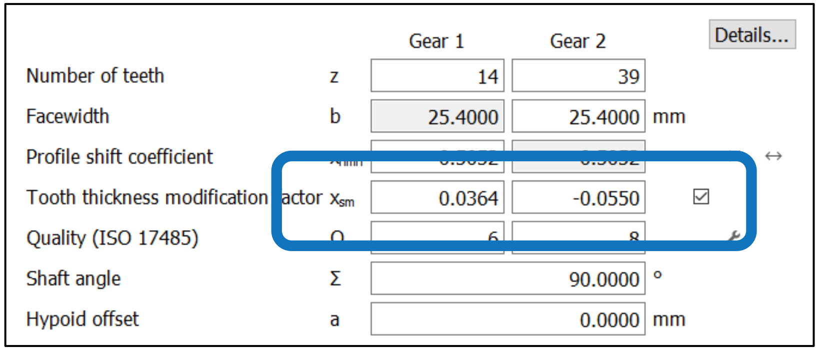
The backlash is introduced by modified blade point widths.

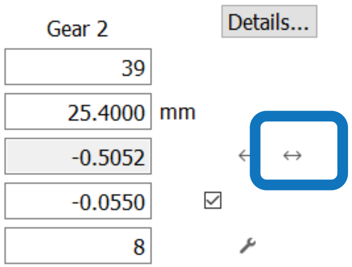
The setting in tab ‘Tolerances’ is ‘no backlash’.

The backlash is defined by the tooth thickness modification coefficient by two different values.

The conversion of backlash into tooth thickness factors may be used.







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Backlash - Assembly

Using a cutting method with **universal tools**:

The backlash is introduced by assembly using larger mounting distance for ring gear. The gears have nominal tooth thickness.

The setting in tab ‘Tolerances’ is ‘no allowance’.

The tooth thickness modification coefficients are equal for pinion and ring gear (+/- values).

In the report, backlash = 0 is shown.

Allowances Gear 1

Tooth thickness tolerance: No allowance

		upper	lower	
Tooth thickness allowance	A_{sn}	0.0000	0.0000	mm
Normal backlash	j_n	0.0000	0.0000	mm
Circumferential backlash	j_t	0.0000	0.0000	mm
Tip diameter allowance	A_{da}	0.0000	-0.0350	mm
Root diameter allowance	A_{df}	0.0000	0.0000	mm

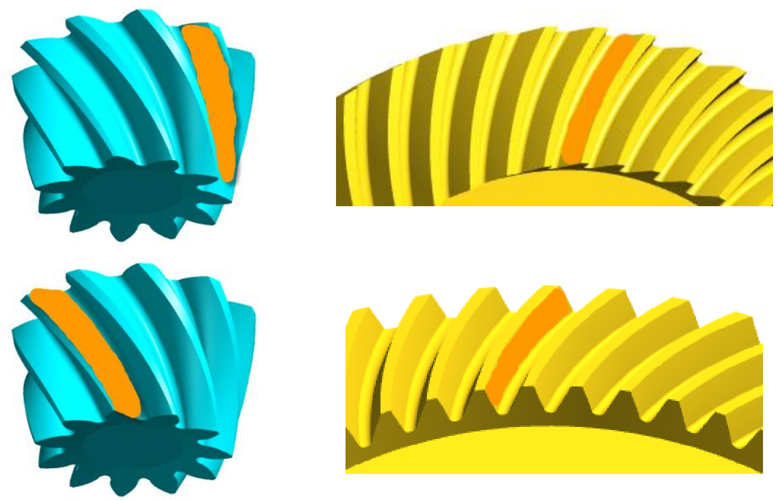
		Gear 1	Gear 2	
Number of teeth	z	11	54	
Facewidth	b	50.0000	50.0000	mm
Profile shift coefficient	x_{hmn}	0.4000	-0.4000	
Tooth thickness modification fact	x_{smn}	0.0300	-0.0300	
Quality (DIN 3965)	Q	6	6	
Shaft angle	Σ	90.0000		°
Hypoid offset	a	0.0000		mm

Details...

		------- Gear 1 ----------------- Gear 2 -------
Circumferential backlash, middle (mm)	[jmt]	-0.000 /-0.000
Circumferential backlash, outside (mm)	[jet]	-0.000 /-0.000
Normal backlash, middle (mm)	[jmn]	-0.000 /-0.000
Normal backlash, outside (mm)	[jen]	-0.000 /-0.000

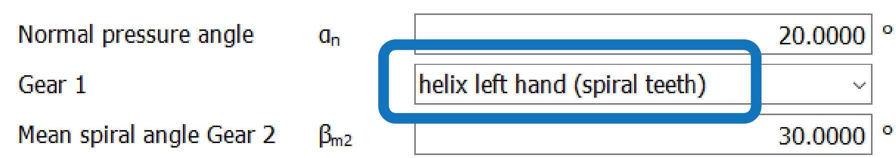
Drive side:
- Pinion concave flank
- Gear convex flank

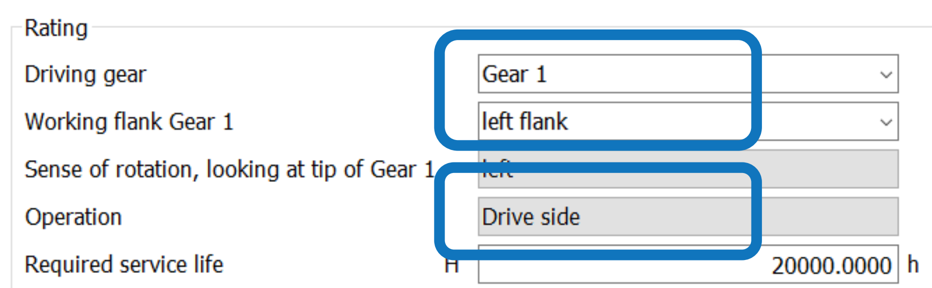
Coast side:
- Pinion convex flank
- Gear concave flank



In KISSsoft, the settings are:
- Tab 'Basic data': Hand of spiral LH / RH
- Tab 'Strength': Driving gear pinion / gear
- Tab 'Strength': Working flank gear 1 left flank / right flank

→ Result: Drive side or Coast side





KISSsoft

Calculation of forces

According to ISO 23509, Annex D

Tangential forces:

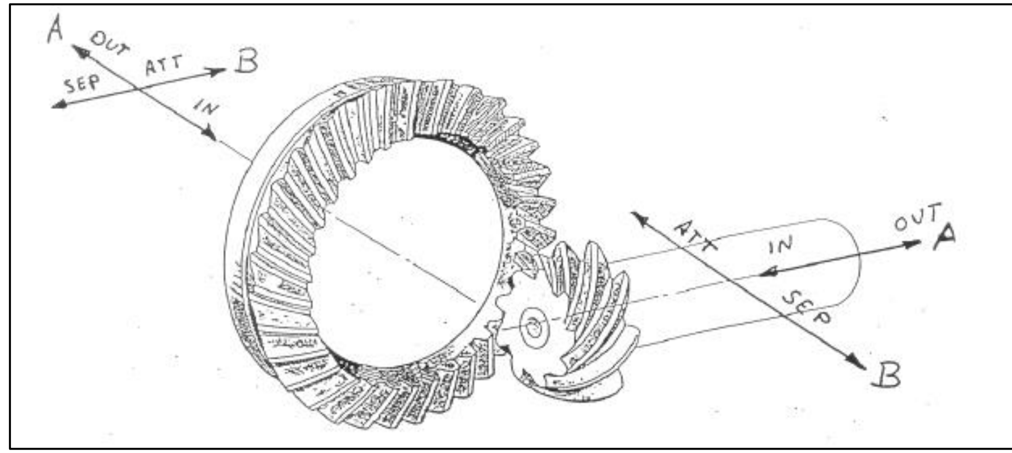
$$F_{\mathrm{mt2}} = \frac{2\,000\,T_2}{d_{\mathrm{m2}}}$$

Radial forces:

$$F_{\mathrm{rad1,D}} = \left(\tan\alpha_{\mathrm{nD}} \frac{\cos\delta_1}{\cos\beta_{\mathrm{m1}}} - \tan\beta_{\mathrm{m1}} \sin\delta_1 \right) F_{\mathrm{mt1}}$$

Axial forces:

$$F_{\mathrm{ax1,D}} = \left(\tan\alpha_{\mathrm{nD}} \frac{\sin\delta_1}{\cos\beta_{\mathrm{m1}}} + \tan\beta_{\mathrm{m1}} \cos\delta_1 \right) F_{\mathrm{mt1}}$$



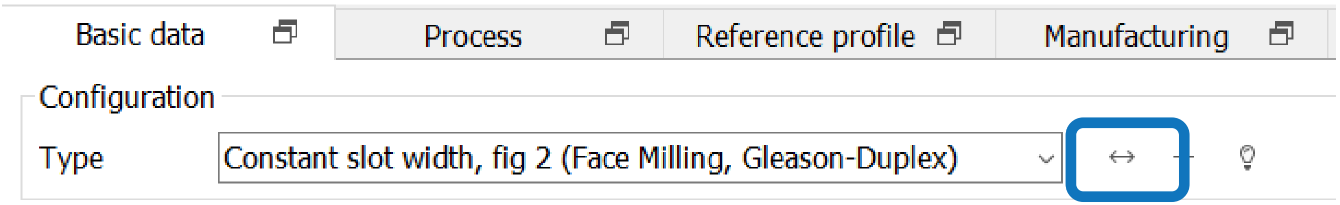
7 General influence factors

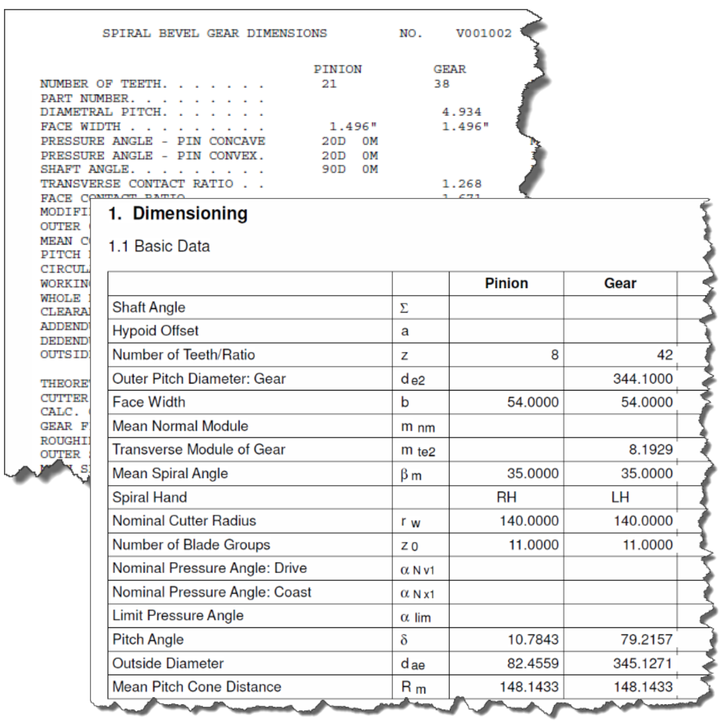
7.1 Forces and circumferential speed

		------- Gear 1 ----------------	Gear 2 -------
Nominal circum. force at pitch circle (N)	[Fmt]	9366.1	11386.2
Nominal circumferential force of virtual cylindrical gear (N)	[Fvmt]		10427.6
Drive side			
Forces calculated according to ISO 23509 with $\alpha_{n\ C/D}$:			
Axial force (N)	[Fa]	12093.4	975.1
Radial force (N)	[Fr]	292.6	10171.6
Normal force (N)	[Fnorm]	15299.0	15299.0
Forces calculated without coefficient of friction			
Axial force (%)	[Fa/Ft]	129.119	10.411
Radial force (%)	[Fr/Ft]	3.124	108.601
Remarks:			
Forces if rotation goes in opposite direction (coast-sided):			
Axial force (N)	[Fa]	-8235.9	8899.1
Radial force (N)	[Fr]	9608.0	-6247.2
Normal force (N)	[Fnorm]	15743.8	15743.8
Axial force (%)	[Fa/Ft]	-87.933	95.014
Radial force (%)	[Fr/Ft]	102.582	-66.700

KISSsoft

Conversion from GLEASON dimension sheet





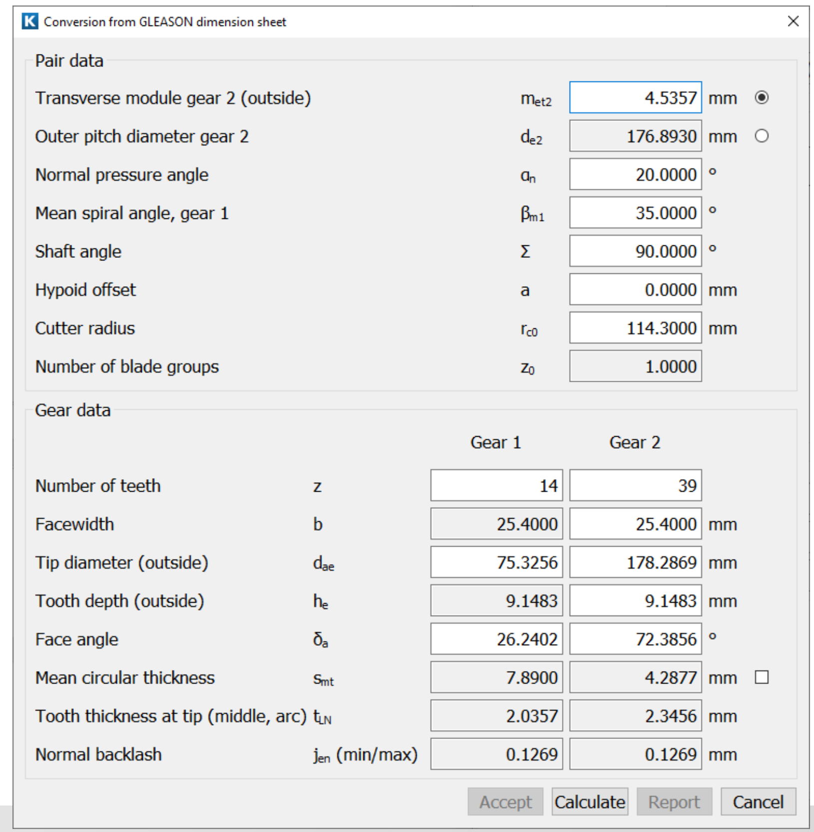
SPIRAL BEVEL GEAR DIMENSIONS NO. V001002

	PINION	GEAR
NUMBER OF TEETH.	21	38
PART NUMBER.		
DIAMETRAL PITCH.		4.934
FACE WIDTH	1.496"	1.496"
PRESSURE ANGLE - PIN CONCAVE	20D 0M	
PRESSURE ANGLE - PIN CONVEX.	20D 0M	
SHAFT ANGLE.	90D 0M	
TRANSVERSE CONTACT RATIO . .		1.268

1. **Dimensioning**

1.1 Basic Data

		Pinion	Gear
Shaft Angle	Σ		
Hypoid Offset	a		
Number of Teeth/Ratio	z	8	42
Outer Pitch Diameter: Gear	d e2		344.1000
Face Width	b	54.0000	54.0000
Mean Normal Module	m nm		
Transverse Module of Gear	m te2		8.1929
Mean Spiral Angle	β m	35.0000	35.0000
Spiral Hand		RH	LH
Nominal Cutter Radius	r w	140.0000	140.0000
Number of Blade Groups	z 0	11.0000	11.0000
Nominal Pressure Angle: Drive	α N v1		
Nominal Pressure Angle: Coast	α N x1		
Limit Pressure Angle	α lim		
Pitch Angle	δ	10.7843	79.2157
Outside Diameter	d ae	82.4559	345.1271
Mean Pitch Cone Distance	R m	148.1433	148.1433

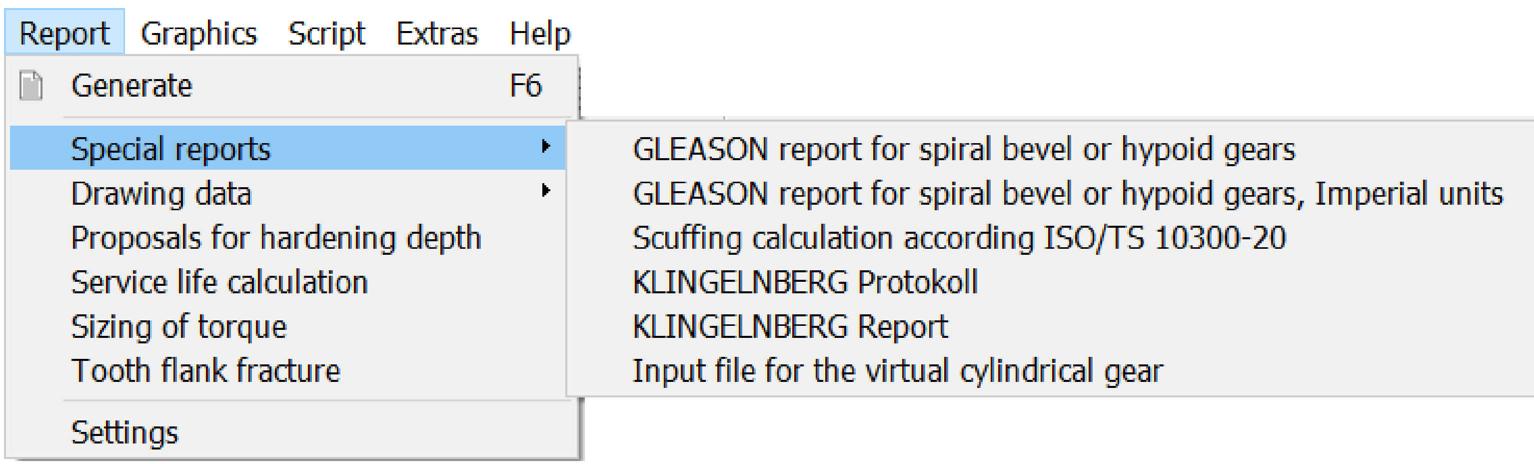


KISSsoft

Special reports

Under reports, several special reports are available:

- Report similar to GLEASON layout
- Report similar to Klingelnberg layout
- Input file for the virtual cylindrical gear



KISSSOFT REPORT ~ GLEASON DATA SHEET

only for Gleason spiral bevel gears

	PINION	GEAR
NUMBER OF TEETH	11	54
PART NUMBER	0.000.0	0.000.0
MODULE		6.681
FACE WIDTH	50.00	50.00
PRESSURE ANGLE	20D 0M	
SHAFT ANGLE	90D 0M	
TRANSVERSE CONTACT RATIO .	1.244	
FACE CONTACT RATIO	1.592	
MODIFIED CONTACT RATIO . .	2.020	
OUTER CONE DISTANCE		184.09
MEAN CONE DISTANCE		159.09
PITCH DIAMETER	73.49	360.76
CIRCULAR PITCH		20.99
WORKING DEPTH		10.00
WHOLE DEPTH	11.25	11.25
CLEARANCE	1.25	1.25
ADDENDUM	7.00	3.00
DEDENDUM	4.25	8.25
OUTSIDE DIAMETER	87.21	361.96
PITCH APEX TO CROWN . . .	178.98	33.80
MEAN CIRCULAR THICKNESS .	11.44	6.70
OUTER NORMAL TOP LAND . .	-	-
MEAN NORMAL TOP LAND . . .	-	-
INNER NORMAL TOP LAND . . .	-	-
PITCH ANGLE	11D30M	78D29M
FACE ANGLE OF BLANK . . .	11D30M	78D29M
ROOT ANGLE	11D30M	78D29M
DEDENDUM ANGLE . . .	0D 0M	0D 0M
OUTER SPIRAL ANGLE		37D51M
MEAN SPIRAL ANGLE		30D 0M
INNER SPIRAL ANGLE		21D50M
HAND OF SPIRAL	LH	RH

```
VERSION=4.2;
KSOFTVERSION=03/2014;
MODULE=Z012;
RechSt.ZahnZNachK=1;
RechSt.xs_OwnInput= 0;
RechSt.xs_Active= 1;
ZS.Geo.mn= 5.0000;
ZS.Geo.beta= 0.523599;
ZS.Geo.alfn= 0.349066;
ZP[0].a= 813.37259;
ZP[0].AXTolID=10240;
ZR[0].z= 11.2259;
ZR[0].b= 42.5000;
ZR[0].x.nul= 0.4000;
ZR[0].ZDTolID=10010;
ZR[0].As.E= 0.0000;
ZR[0].As.i= 0.0000;
ZR[0].Tool.RefProfile.DBID=10400;
ZkegR[0].XS= 0.0300;
```

1. Dimensioning

1.1 Basic data

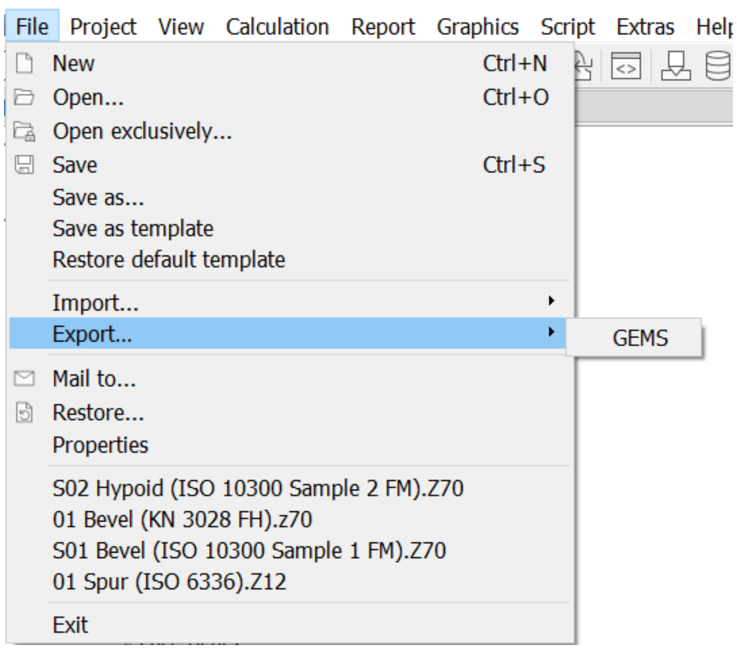
		Pinion	Gear		
Shaft Angle	Σ			90.0000	Deg.
Hypoid Offset	a			0.0000	mm
Number of Teeth/Ratio	z	14	39	2.7857	
Outer Pitch Diameter: Gear	de2		176.8930		mm
Face Width	b	25.4000	25.4000		mm
Mean Normal Module	mnm			3.2133	mm
Transverse Module of Gear	mte2		4.5357		mm
Mean Spiral Angle	ßm	35.0000	35.0000		Deg.
Spiral Hand		left	right		
Nominal Cutter Radius	rw	114.3000	114.3000		mm
Number of Blade Groups	z0	1.0000	1.0000		
Nominal Pressure Angle: Drive	αNv1			20.0000	Deg.
Nominal Pressure Angle: Coast	αNx1			20.0000	Deg.
Limit Pressure Angle	αlim			-0.0000	Deg.
Pitch Angle	δ	19.7468	70.2532		Deg.
Outside Diameter	dae	75.3256	178.2869		mm
Mean Pitch Cone Distance	Rm	81.2726	81.2726		mm

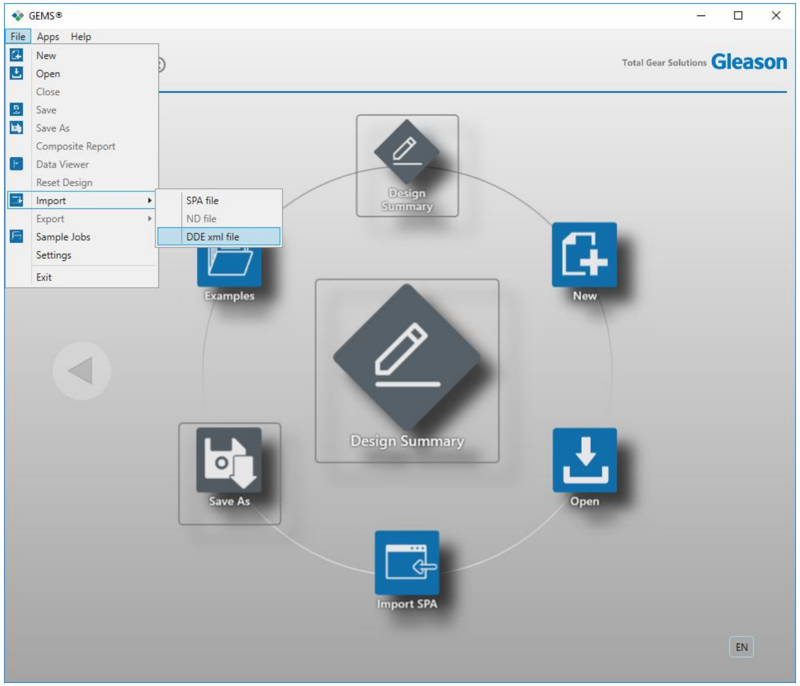
KISSsoft

Interface between KISSsoft and GEMS

- For bevel and hypoid gears
- Macro geometry of gears
- Load data

- Check for cutter head size
- Check for final blank geometry
- Check for blade design (radius, ..)

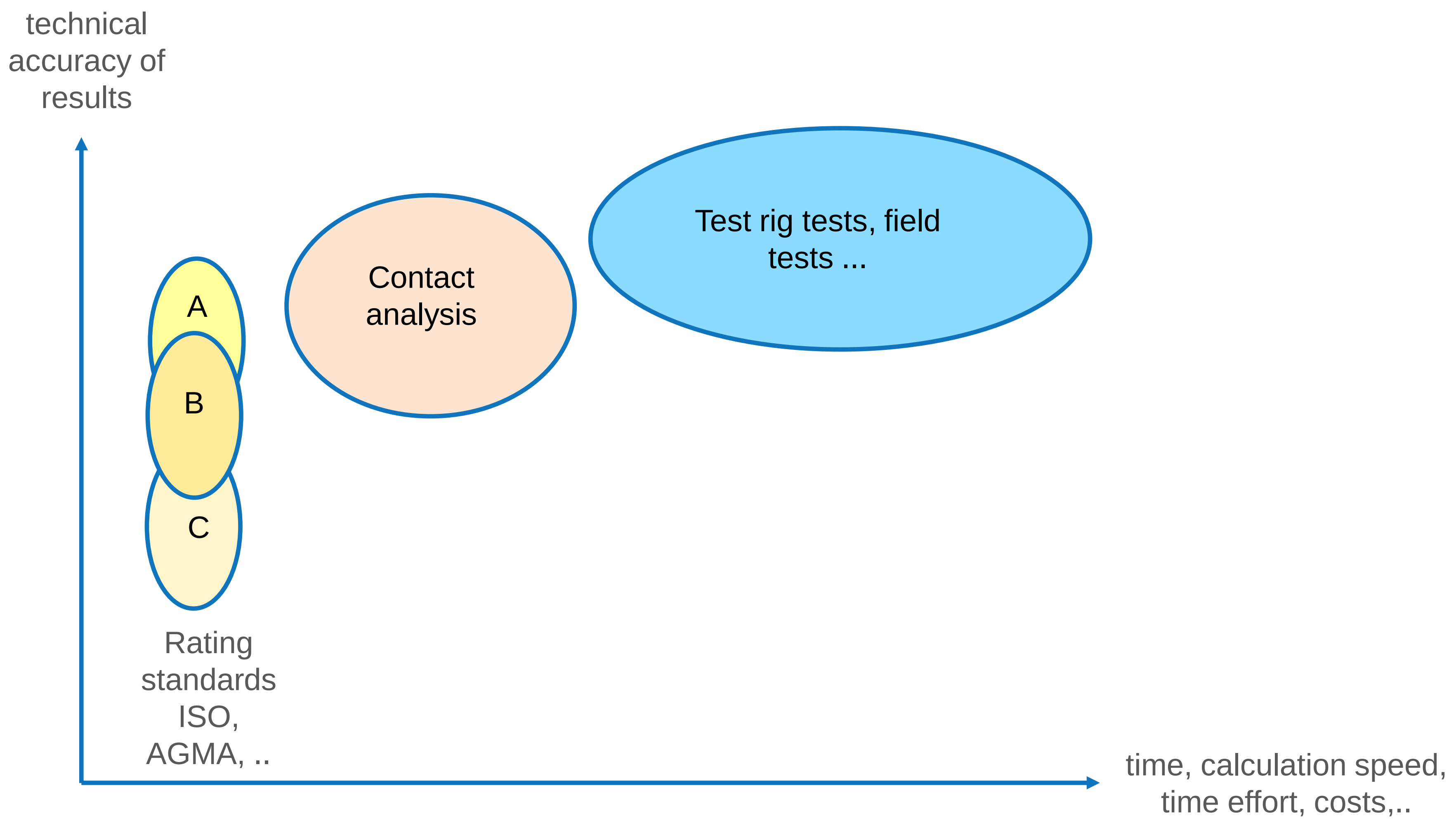




KISSsoft

Strength calculation

Strength and life time rating methods



KISSsoft

	Rating standard	Contact analysis	Test rig tests, field tests
Source	ISO, AGMA, ...	KISSsoft, GEMS, FEM, ..	Customer site
Life time information, safety value	✓	✗ (stresses, stress distribution)	✓
Macro geometry	✓ (varied, optimization)	✓ (defined, 1 gear pair)	✓ (defined)
Micro geometry (incl. tolerances & misalignments)	✗ (method A)	✓ (varied, optimization)	✓ (defined)

KISSsoft

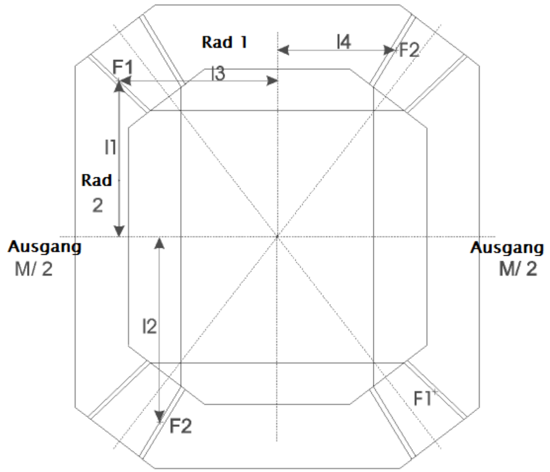
Method	Bevel gear	Hypoid gear	Root stress	Pitting	Scuffing	Flank fracture	Wear	EHT	Micro Pitting
DIN 3991	[green]	[red]	[green]	[green]	[green]	[red]	[red]	[red]	[red]
ISO 10300 (2001)	[green]	[red]	[green]	[green]	[red]	[red]	[red]	[red]	[red]
ISO 10300 (2014)	[green]	[green]	[green]	[green]	[red]	[red]	[red]	[red]	[red]
AGMA 2003 (B97, C10)	[green]	[red]	[green]	[green]	[red]	[red]	[red]	[red]	[red]
DNV 41.2 (2012)	[green]	[green]	[green]	[green]	[green]	[green]	[red]	[green]	[red]
Plastic gear Niemann / Winter	[green]	[light green] not in KISSsoft	[green]	[green]	[light green] not in KISSsoft	[red]	[light green] not in KISSsoft	[red]	[red]
Plastic gear VDI 2545	[green]	[red]	[green]	[green]	[red]	[red]	[red]	[red]	[red]
Klingelnberg Palloid KN 3025, 3026, 3030	[green]	[green]	[green]	[green]	[green]	[red]	[red]	[red]	[red]
Klingelnberg Zyklo-Palloid KN 3028, 3029, 3030	[green]	[green]	[green]	[green]	[green]	[red]	[red]	[red]	[red]
Flank fracture ISO/DTS 10300-4 (2019)	[green]	[green]	[red]	[red]	[red]	[green]	[red]	[red]	[red]
Scuffing ISO/TS 10300-20 (2021)	[green]	[green]	[red]	[red]	[green]	[red]	[red]	[red]	[red]
Micro Pitting ISO/TS 6336-22	[green]	[green]	[red]	[red]	[red]	[red]	[red]	[red]	[green]

Red: not available, green: available and in KISSsoft, light green: available but not in KISSsoft

KISSsoft

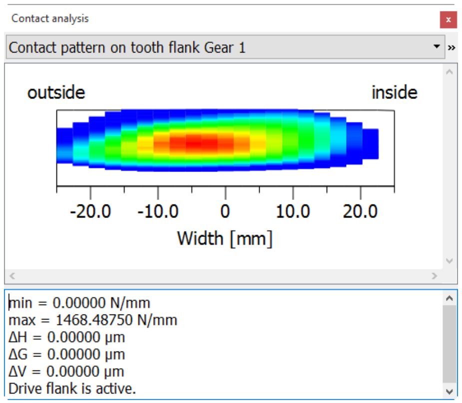
Static calculation

- For bevel gears and differential gears available
- Calculates safety again material values 'yield point' and 'tensile strength'



Contact analysis

- Stress calculation under load
- Considers tooth deformation according to Weber / Banaschek
- Considers displacements of gears
- Considers micro geometry
- Tooth form based on virtual cylindrical gear (no m/c settings)



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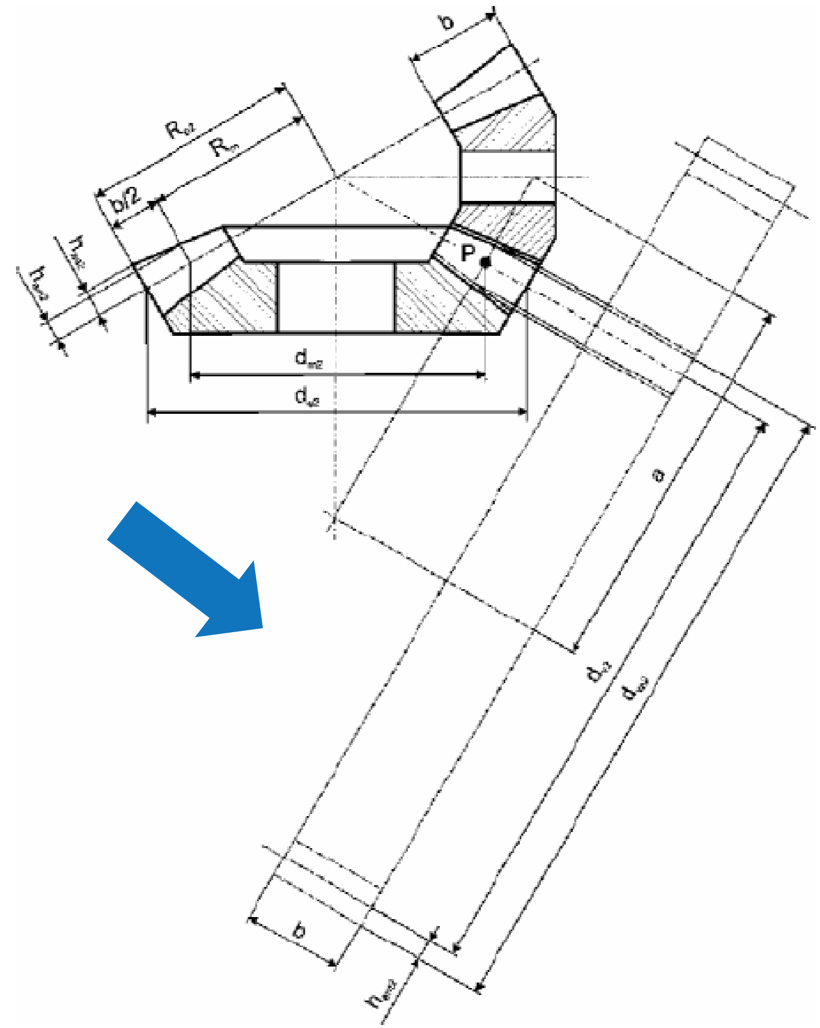
The bevel or hypoid gear geometry is transferred into a virtual cylindrical gear geometry.

The dimensions in middle of face width are used.

For the strength calculation of root and flank the (modified) formulae of cylindrical gears are applied.

Hence there is no difference in loading capacity calculation between Face Hobbing and Face Milling.



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Calculation of surface durability (pitting, acc. to ISO 10300-2, method B1)

The contact stress σ_H is determined on the nominal contact stress σ_{H0}. The bevel gear factor Z_K accounts for the influence of the bevel gear geometry. The hypoid factor Z_{Hyp} accounts for the influence of lengthwise sliding onto the surface durability.

The permissible contact stress σ_{HP} is based on tests from cylindrical gears which provide a wide range of tested materials.

The value of the minimum safety factor for contact stress, S_{Hmin}, should be 1,0.

$$\sigma_H = \sigma_{H0}\sqrt{K_A \cdot K_V \cdot K_{H\beta} \cdot K_{H\alpha}} \le \sigma_{HP}$$

$$\sigma_{H0} = \sqrt{\frac{F_n}{l_{bm}\rho_{rel}}} \cdot Z_{M-B} \cdot Z_E \cdot Z_{LS} \cdot Z_K$$

$$\sigma_{HP} = \sigma_{H\lim} \cdot Z_{NT} \cdot Z_X \cdot Z_L \cdot Z_V \cdot Z_R \cdot Z_w \cdot Z_{Hyp}$$

KISSsoft

Calculation of tooth root strength (acc. to ISO 10300-3, method B1)

The tooth root stress σ_F is determined on the nominal tooth root stress σ_{F0}. The bevel spiral angle factor Y_{BS} accounts for smaller values of contact lines l_{bm} compared to the total face width and their inclination. The cutter head size is considered in the factor $K_{F\beta}$.

The permissible tooth root stress σ_{FP} is based on tests from cylindrical gears which provide a wide range of tested materials.

The value of the minimum safety factor for tooth root stress, S_{Fmin}, should be ≥ 1,3 for spiral bevel gears. For bevel gears where βm ≤ 5°, S_{Fmin} should be ≥ 1,5.

$$\sigma_F = \sigma_{FO} \cdot K_A \cdot K_V \cdot K_{F\beta} \cdot K_{F\alpha} \leq \sigma_{FP}$$

$$\sigma_{FO} = \frac{F_{mt}}{b \cdot m_n} \cdot Y_{Fa} \cdot Y_{Sa} \cdot Y_{\varepsilon} \cdot Y_{BS} \cdot Y_{LS}$$

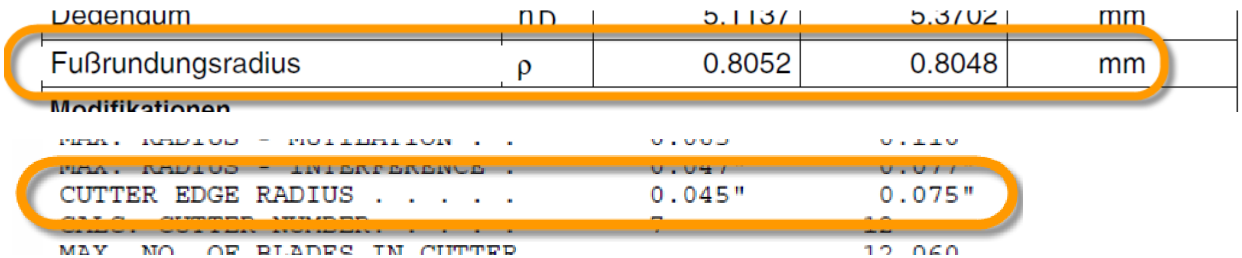
$$\sigma_{FP} = \sigma_{F\lim} \cdot Y_{ST} \cdot Y_{NT} \cdot Y_{\delta relT} \cdot Y_{RrelT} \cdot Y_X$$

KISSsoft

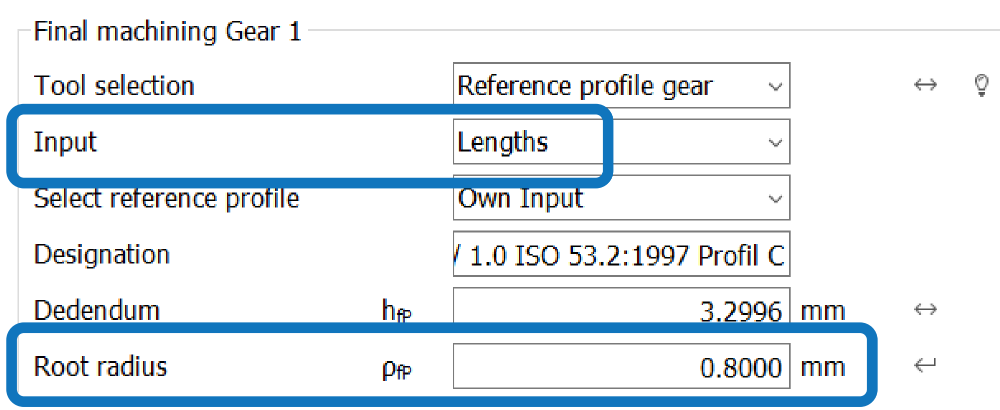
Root radius coefficient

- Enter the cutter edge radius from summary
- Switch input mode to ‘length’

GLEASON or KLINGELNBERG Summary:



KISSsoft:



KISSsoft

$$K_{H\beta} = K_{H\beta\text{-be}} * 1.5$$

General factors

Application factor	K_A	1.1000
Dynamic factor	K_v	1.0203
Transverse load factor	$K_{H\alpha}$	1.1606
Mounting factor (Load distribution modifier)	$K_{H\beta\text{-be}}$	1.1000

Mounting factor according to ISO 10300-1			
Verification of contact pattern	**Mounting conditions of pinion and gear**		
Contact pattern is checked:	**neither member cantilever mounted**	**one member cantilever mounted**	**both members lever mounted**
for each gear set in its housing under full load	1.00	1.00	1.00
for each gear set under light test load	1.05	1.10	1.25
for a sample gear set and estimated for full load	1.20	1.32	1.50

NOTE: Based on optimum tooth contact as evidenced by results of a contact pattern test on the gears in their mountings.

KISSsoft

Effective face width b_{eff}

The default value acc. to ISO 10300 for b_{eff} is = 0.85*b

➔ This is to be changed by the user with the effective contact pattern width

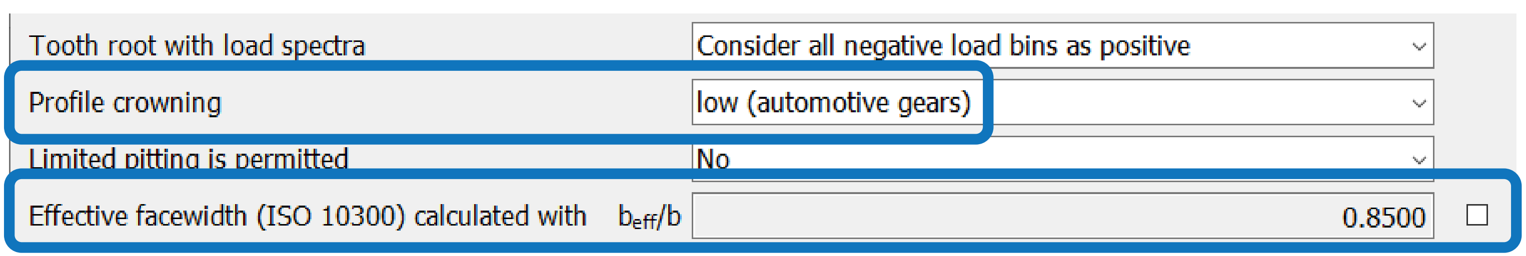
Overlap contact ratio ε_β

The overlap contact ratio ε_β in ISO 10300:2014 depends on the effective face width b_{eff}

➔ direct comparison with ISO 10300:2001 is not possible (unless b_{eff} = 1)

Profile crowning

Two settings available for ‘high’ and ‘low’ profile crowning



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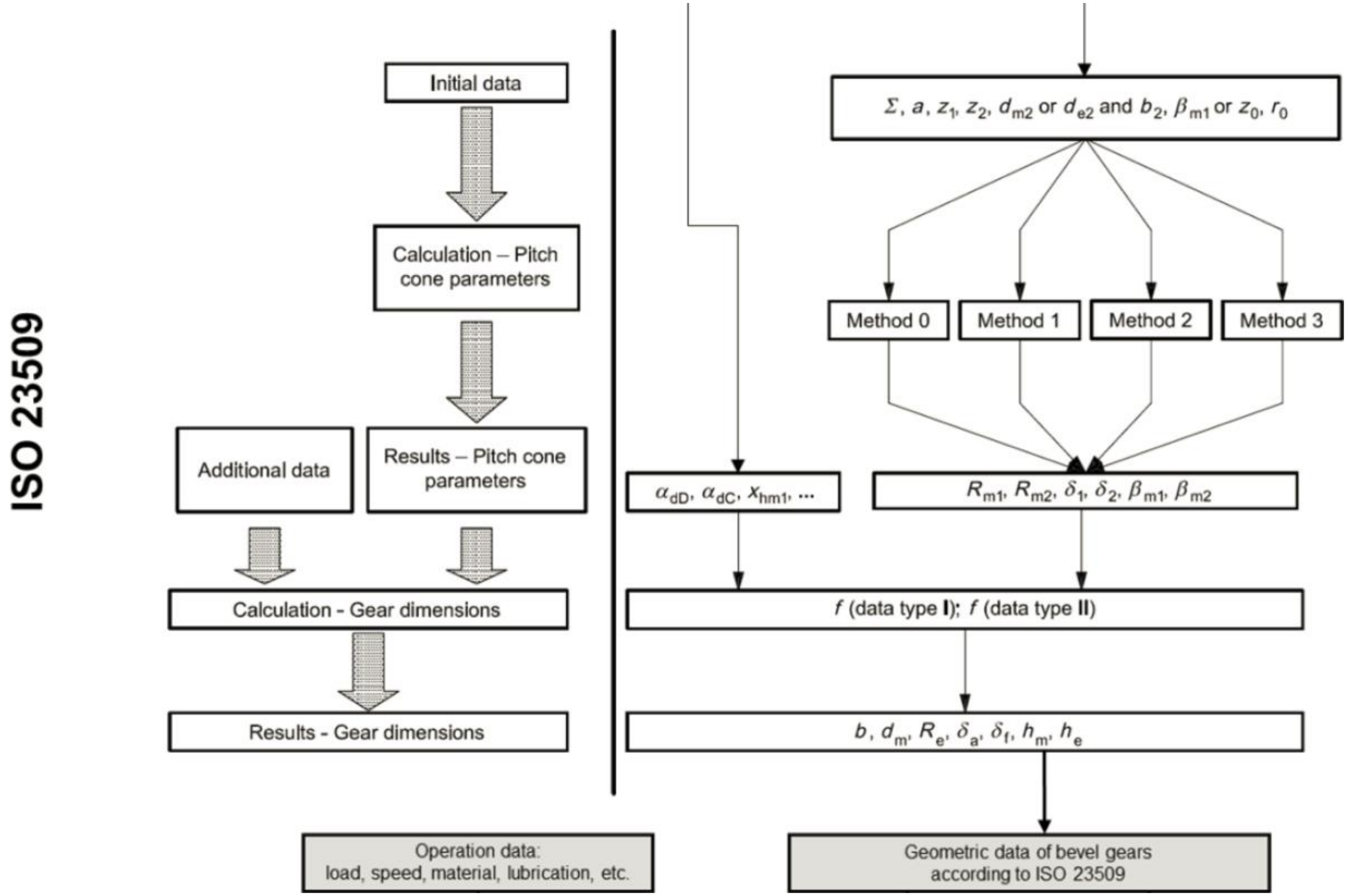
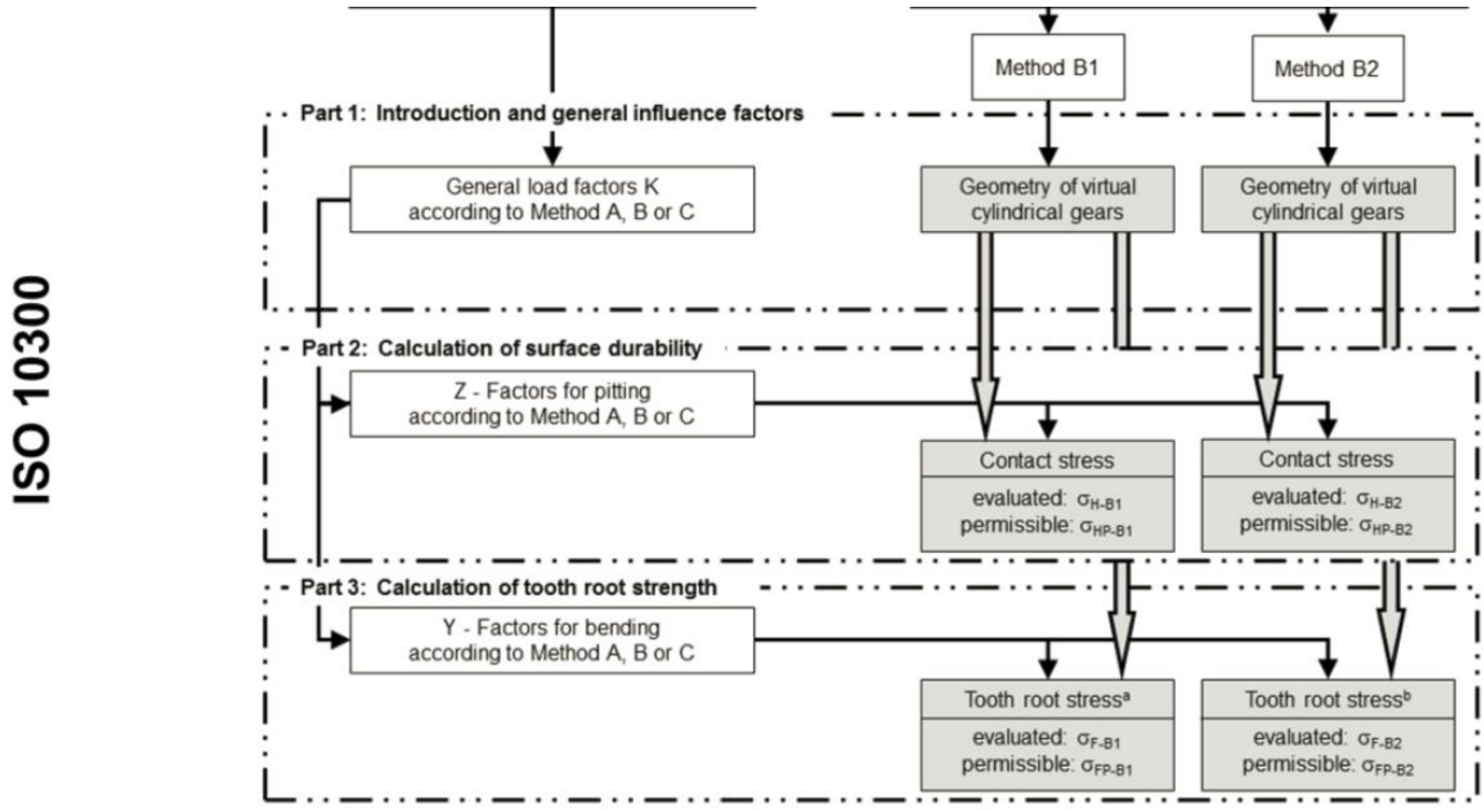
Geometry calculation:



KISSsoft

Strength rating:



For all four examples shown in the ISO/TR 10300-30, the flank and root safety factors S_H and S_F are calculated in KISSsoft and compared to reference calculations in the ISO/TR 10300-30.

KISSsoft

5.1.1 Fundamental contact stress formula

The fundamental contact stress formula for pitting resistance of bevel gear teeth is:

$$s_c = C_p \sqrt{\frac{2T_P}{Fd^2 I} K_o K_v K_m C_s C_{xc}} \quad \text{...(1)}$$

$$\sigma_H = Z_E \sqrt{\frac{2000 T_1}{b d_{e1}^2 Z_I} K_A K_v K_{H\beta} Z_x Z_{xc}} \quad \text{...(1M)}$$

The formula for permissible contact stress number is:

$$s_{wc} = \frac{s_{ac} C_L C_H}{S_H K_T C_R} \quad \text{...(2)}$$

$$\sigma_{HP} = \frac{\sigma_{H\lim} Z_{NT} Z_W}{S_H K_\theta Z_Z} \quad \text{...(2M)}$$

KISSsoft

The I (Z_I) factor is calculated using the following formula:

$$I = \frac{s\,\rho_o\,\cos\psi\,\cos\phi}{F\,d\,C_i\,m_{NI}}\,\frac{P_d}{P_m}$$

$$Z_I = \frac{g_c\,\rho_{yo}\,\cos\beta_m\,\cos\alpha_n}{b\,d_{e1}\,Z_i\,\varepsilon_{NI}}\,\frac{m_{mt}}{m_{et}}$$

15.2.1 Spiral bevel asymmetry

Since spiral bevel gear teeth are not symmetrical in the lengthwise direction, the stresses will differ between the concave and convex sides of the tooth. Normally one calculates the stresses on the concave side of the pinion tooth and convex side of the gear tooth since these are the usual driving surfaces. For bi-directional operation of bevel gears, a complete analysis would require calculating the geometry factors independently for each side of the tooth.

5.2.1 Fundamental bending stress formula

The fundamental formula for bending stress of gear teeth is:

$$s_t = \frac{2T_P}{F\,d}\,\frac{P_d\,K_o\,K_v}{1}\,\frac{K_s\,K_m}{K_x\,J} \qquad \text{...(5)}$$

$$\sigma_F = \frac{2000\,T_1}{b\,d_{e1}}\,\frac{K_A\,K_v}{m_{et}}\,\frac{Y_x\,K_{H\beta}}{Y_\beta\,Y_J} \qquad \text{...(5M)}$$

Pinion geometry factor:

$$J_P = \frac{Y_{KP}}{m_{NJ}\,K_i}\,\frac{r_t}{r}\,\frac{F_{eP}}{F}\,\frac{P_d}{P_m} \qquad \text{...(25)}$$

$$Y_{J1} = \frac{Y_{K1}}{\varepsilon_{NJ}\,Y_i}\,\frac{r_{myo1}}{r_{mpt1}}\,\frac{b_1'}{b}\,\frac{m_{mt}}{m_{et}} \qquad \text{...(25M)}$$

Pinion geometry factor in AGMA depends on drive and coast operating conditions.

Fatigue limit sat (SigmaFlim) is different for bevel gears than for cylindrical gears.

The formula for permissible bending stress number is:

$$s_{wt} = \frac{s_{at}\,K_L}{S_F\,K_T\,K_R} \qquad \text{...(6)}$$

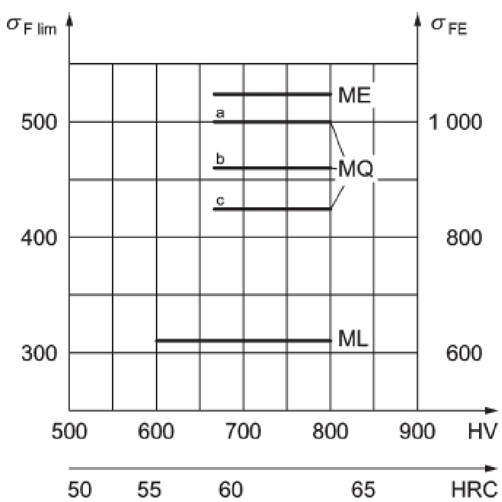
$$\sigma_{FP} = \frac{\sigma_{F\,\lim}\,Y_{NT}}{S_F K_\theta Y_z} \qquad \text{...(6M)}$$

The bending stress number is different in AGMA for cylindrical gears (AGMA 2001-D04) than to bevel gears (AGMA 2003-C10).

AGMA 2001, grade 2 (65'000 lb/in^2, 450 N/mm^2) corresponds well to ISO quality MQ (b) with σ_{Flim} = 460 N/mm^2. So, grade 2 in AGMA 2003 corresponds with ISO MQ (b).

Bending stress number (allowable), s_{at} ($\sigma_{F\,lim}$) lb/in^2 (N/mm^2)		
Grade 1[1)]	Grade 2[1)]	Grade 3[1)]
See figure 10	See figure 10	
12 500 (85)	13 500 (95)	--
22 500 (154)	--	--
30 000 (205)	35 000 (240)	40 000 (275)

AGMA 2003-C10



ISO 6336-5

Allowable bending stress number[2)], s_{at} lb/in^2		
Grade 1	Grade 2	Grade 3
see figure 9	see figure 9	--
45 000	55 000	--
22 000	22 000	--
55 000	65 000 or 70 000[6)]	75 000

AGMA 2001-D04

KISSsoft

Definition

The Q-factor is the most frequently used factor in the Gleason Dimension-Sheet.

The factor, Q, is used to calculate the dynamic **bending stress in the tooth root fillet**, where it is maximum.

The bending stress (psi) for each member is found by multiplying its factor, Q by its applied torque (lb-in).

Source: Bending stresses in Bevel Gear Teeth, Gleason Works, 1981

$$Q = \frac{2P_d \quad K_s}{F \; D \; J}$$

P_d = large end diametral pitch (transverse)
K_s = size factor. See Fig. 2.
F = actual face width in inches for corresponding member. This may be different on the two members.
D = large end pitch diameter in inches for corresponding member.
J = geometry factor for corresponding member. See Figs. 4A through 4J or Appendix A.

$$S_t = \frac{T \; Q \; K_o \; K_m}{K_v \; K_x}$$

Q = strength factor for corresponding member.
K_o = overload factor. See Table 1.
K_m = load distribution factor. See Table 2 or Fig. 3.
K_v = dynamic factor. See Fig. 1.
K_x = cutter radius factor. See Chart Appendix E.

Conversion between units

In the metric system, when the Q-factor is multiplied by the pinion torque in Nm, the result has to be multiplied by 0.61 in order to obtain a result in N/mm².

Torque [Nm] * Q * 0.6102327 = bending stress [N/mm²]

The conversion of Nm in inch-lbs requires a factor of 8.85075. This factor multiplied by the Q-factor results in PSI, which has to be converted with the factor 0.0068947 into N/mm²:

Torque [Nm] * 8.85075 * Q * 0.0068947 = bending stress [N/mm²]

KISSsoft

Q-Factor in KISSsoft

$$Q = \frac{2P_d \quad K_s}{F \; D \; J}$$

When using the AGMA 2003, the Q-factor is calculated based on Eq. 5.

The Q-factor may differ to the Q-factor from the Gleason calculation.

The Q-factor is listed only in the special report.

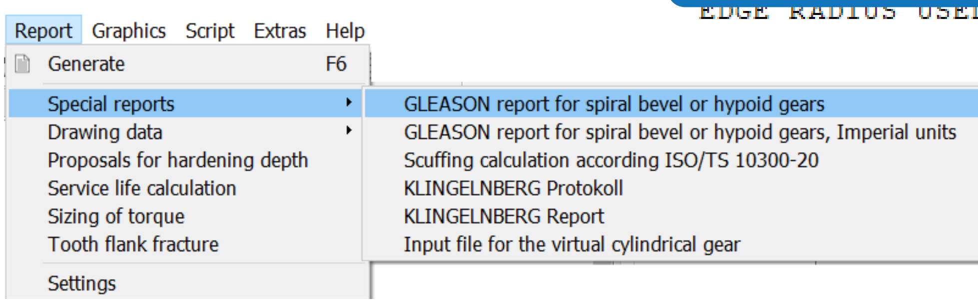
5.2.1 Fundamental bending stress formula

The fundamental formula for bending stress of gear teeth is:

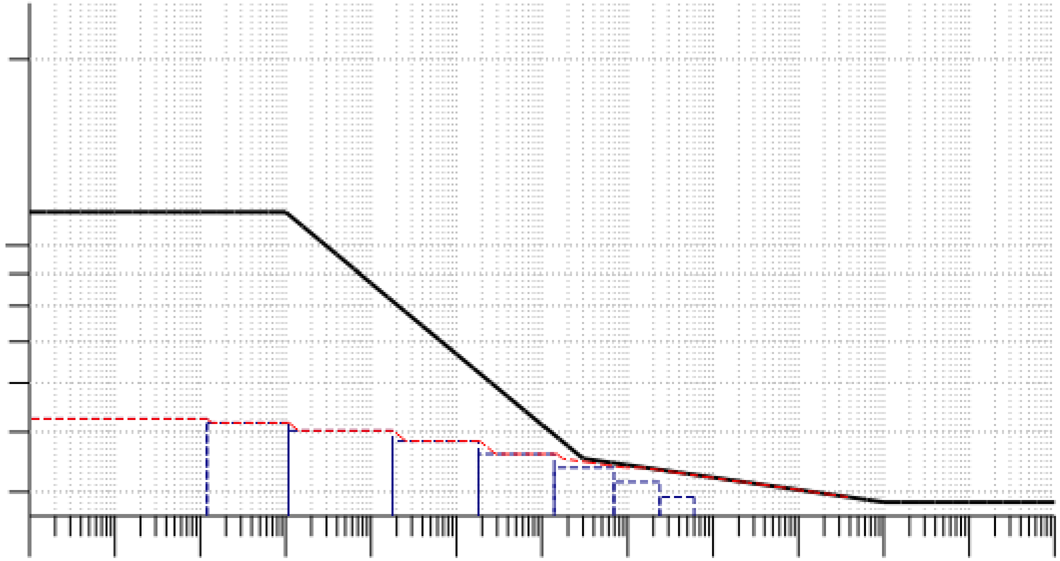
$$s_t = \frac{2T_P}{F\,d}\,\frac{P_d\,K_o\,K_v}{1}\,\frac{K_s\,K_m}{K_x\,J} \qquad \ldots(5)$$

$$\sigma_F = \frac{2000\,T_1}{b\,d_{e1}}\,\frac{K_A\,K_v}{m_{et}}\,\frac{Y_x\,K_{H\beta}}{Y_\beta\,Y_J} \qquad \ldots(5M)$$

```
STRENGTH FACTOR - Q                    1.98841          0.35590
```

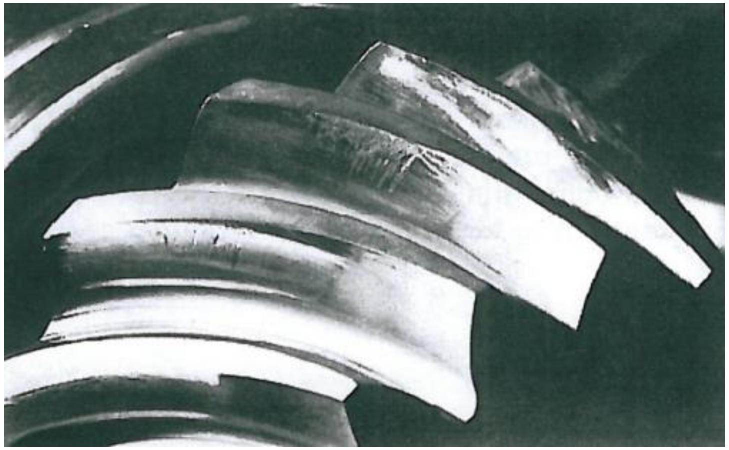


Miner’s rule

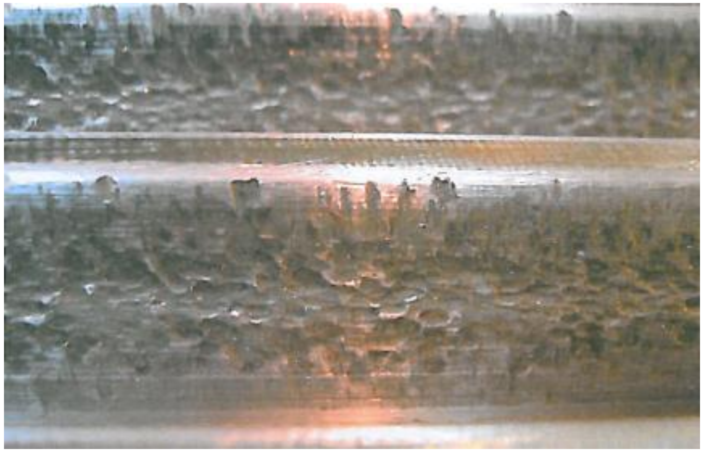


$$\frac{n_1}{N_{f1}} + \frac{n_2}{N_{f2}} + \cdots + \frac{n_i}{N_{fi}} = 1$$

- Scuffing is a severe form of adhesive wear that can result in progressive damage to the gear teeth.
- Scuffing is not a fatigue phenomenon and may occur instantaneously.
- Rating against scuffing should be done for all steps in the load spectrum individually.
- Rating method is available in ISO/TS 10300-20, ISO/TR 13989, DIN 3991 or AGMA 925-A03.



Mild scuffing: often self-healing



Hot severe scuffing

- Flash temperature method predicts the probability of scuffing by comparing the maximum contact temperature with the critical scuffing temperature (ISO/TS 10300-20).
- The contact temperature θ_{Bmax} is the sum of the bulk temperature and the flash temperature calculated by Blok.

$$\theta_{B\max} = \theta_{Mi} + \theta_{fl\max}$$

- The scuffing temperature θ_S is calculated from the test gear or provided from field measurement.

$$\theta_S = \theta_{MT} + X_W \cdot \theta_{fl\max T}$$

- Safety factor

$$S_B = \frac{\theta_S - \theta_{oil}}{\theta_{B\max} - \theta_{oil}}$$

$\theta_{M(T,\ i)}$	Bulk temperature of test gear (T) or interfacial (i)
X_W	Structural factor
θ_{Oil}	Oil temperature, in ˚C
$\theta_{flmax(T)}$	Maximum flash temperature, in ˚C, of test gear (T)

KISSsoft

- Integral temperature method predicts the probability of scuffing by comparing the mean value of the contact temperature along the path of contact (integral temperature) with the critical scuffing temperature.
- The integral temperature is the sum of the bulk temperature and the weighted mean of the integrated values of flash temperature along the path of contact.

$$\theta_{\text{int}} = \theta_M + C_2 \cdot \theta_{fla\,\text{int}}$$

C_2 : Weighting factor from experiments, (= 1.5 for cylindrical gears)

- The scuffing temperature should be provided for the scuffing tests (see ISO 14635, FZG test procedures) or from field measurement.
- Safety factor

$$S_{\text{int}\,S} = \frac{\theta_{\text{int}\,S}}{\theta_{\text{int}}}$$

$\theta_{\text{int}\,S}$: Scuffing integral temperature, in °C

KISSsoft

Local calculations over the path of contact

the parameters are calculated at 11 positions over the path of contact. The smallest safety value is considered as the critical safety value.

Stresses and velocities

local contact stress, Sliding and sum velocity, Lubricating film thickness, Coefficient of friction, ..

Occurring contact temperature

is composed from the bulk temperature (determined by the power losses) and the flash temperature

Permissible contact temperature

includes the limit temperature from scuffing test, the influence of contact temperature and contact time

Run in

running-in of gear pair has a high beneficial influence for high scuffing safety.

Load stage of lubrication

the load stage of lubrication according to FZG step load test is required for the calculation.

Profile crowning ‘high’ and ‘low’

high profile crowning leads to reduced load at tip and root of tooth.

Structure factor

considers several typical remedies such as phosphate or copper plated gears.

Material	X_W
Through-hardened steel	1,00
Phosphate-coated steel*	1,25
Copper-coated steel*	1,50
Bath or gas nitride steel	1,50
Hardened carburized steel, with austenite content:	
- less than average	1,15
- average (10% to 20%)	1,00
- greater than average	0,85
Austenite steel (stainless steel)	0,45

*for phosphate- and copper-coated steels use running-in factor X_E for not run-in surfaces

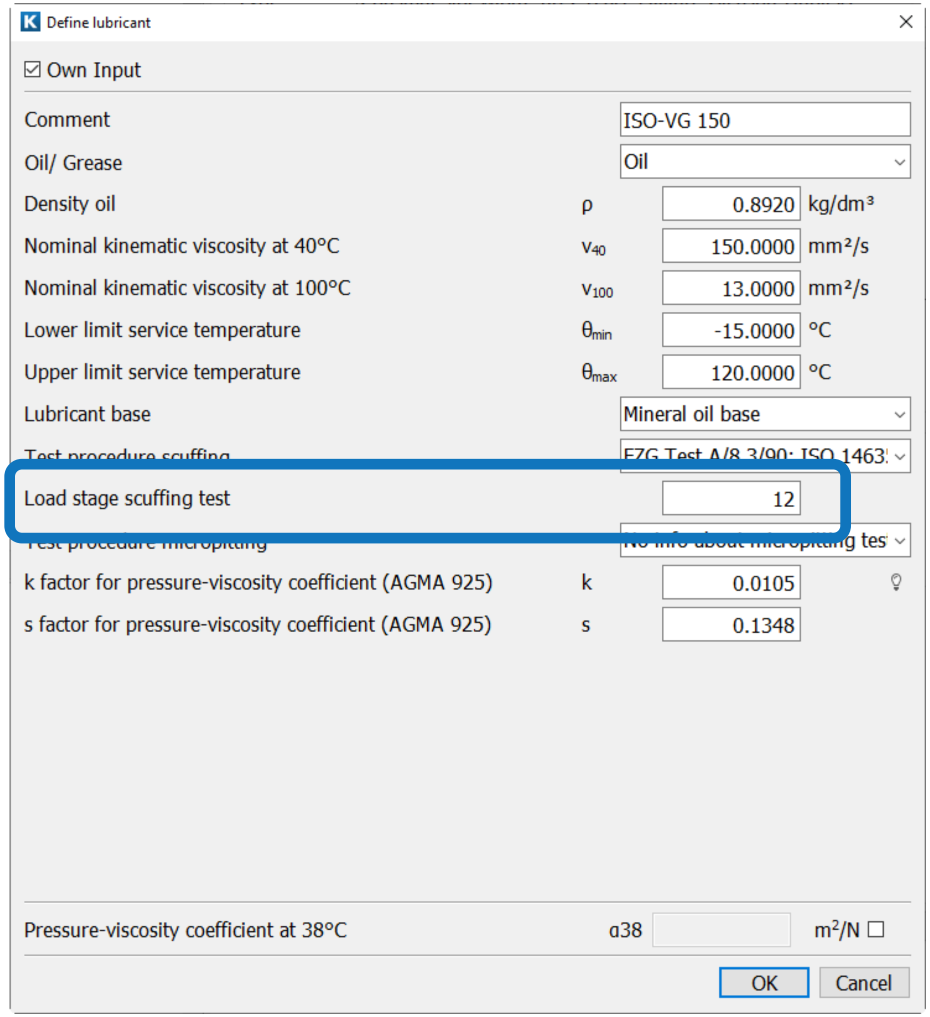
KISSsoft

Lubrication with scuffing test data

The calculation requires the data from the load stage scuffing test

The oil type ‘GL5’ typically has a load stage of approx. 14 (Pinion torque of 800 Nm).

Further research is currently done regarding testings of oil type ‘GL5 ‘.



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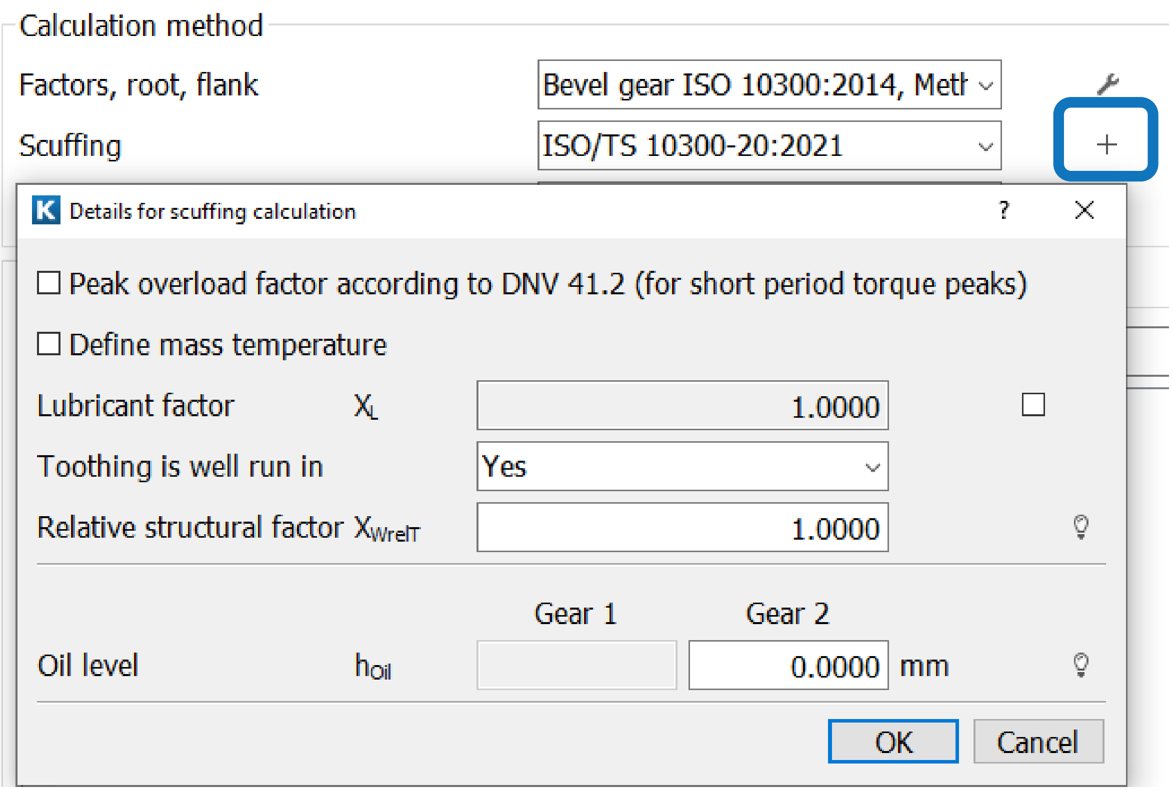
Several settings for Scuffing



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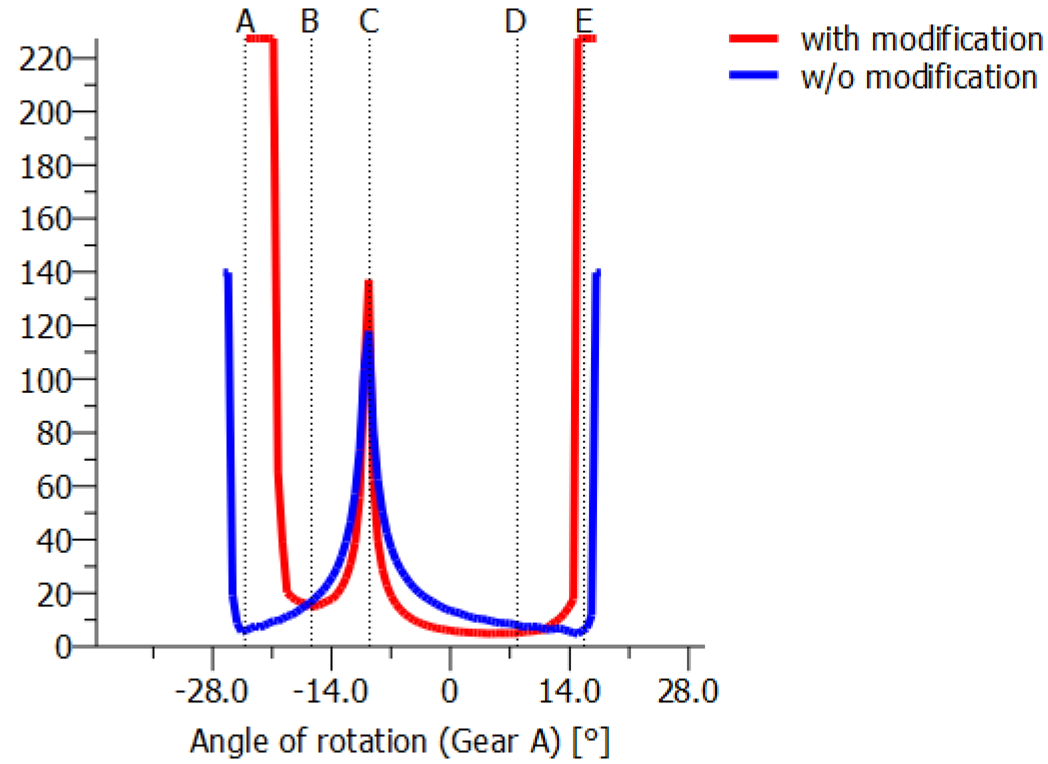
Flank modification

With profile modification, the risk of scuffing can be reduced, as the pressure at begin and end of meshing is reduced.

With the contact analysis, the scuffing is rated over the complete path of contact.



Method according to Annast

- Investigation done on bevel gears
- Published in 2002

Method according to ISO/DTS 10300-4

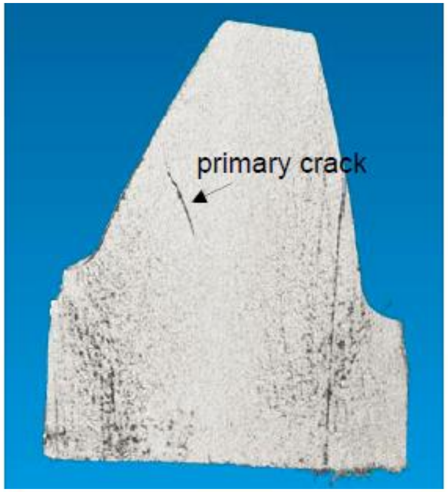
- Investigation done on cylindrical gears
- Basis thesis is from Dr. Witzig, Munich
- Thesis published in 2012
- Adaption for bevel gears directly possible
- ISO technical report currently under process

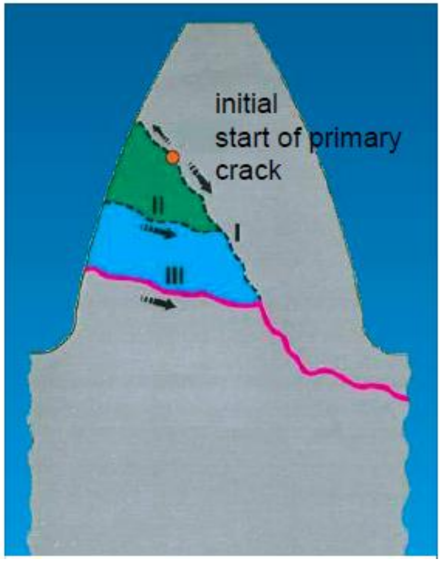
Figure 1 Flank breakage and pitting on wheel.

Figure 2 Flank breakage — two different wheel:

KISSsoft

- Failure initiated below the surface due to shear stress, crack often at approximately half the height of the tooth
- Failure initiated typically at or below case-core interference
- Crack starter often is a small non-metallic inclusion
- Crack propagates from starting point towards surface and towards inside of tooth, later being larger crack at approximately 45deg angle to the flank surface
- Often, no indication of surface fatigue like micropitting or pitting are observed even if tooth flank fracture occured





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An **effective shear stress τ_{eff}(y)** is compared to a local material strength τ_{per}(y) where (y) is the material depth.

A **material exposure $A_{FF,x}$(y)** is calculated for each depth (y) and each point x on the flank.

$$S_{FF} = \frac{1}{A_{FF,\max} + c_2} + c_2 \geq S_{FF,\min}$$

$$A_{FF,x}(y) = \frac{\tau_{eff}(y)}{\tau_{per}(y)} + c_1$$

$$c_2 = 0{,}2$$

$$A_{FF,\max} = \max(A_{FF,x}(y))$$

Method A: stress distribution based on contact analysis, stress is known for each point x on the whole flank

Method B: for some specific points in the contact

KISSsoft

Tooth flank fracture, effective stress

The **effective shear stress τ_{eff}(y)** is a function of the shear stress due to external loads and the residual shear stress (and some interaction of the two)

$$\tau_{eff}(y) = \tau_{eff,L}(y) - \Delta\tau_{eff,L,RS}(y) - \tau_{eff,RS}(y)$$

τ_{eff}	local occurring equivalent stress state
$\tau_{eff,L}$	equivalent stress state due to external load, without consideration of residual stresses
$\tau_{eff,L,RS}$	influence of the residual stresses on the equivalent stress state
$\tau_{eff,RS}$	quasi-stationary residual stress state

Residual stress profile $\sigma_{RS}(y)$

The residual stress profile is used for both parameters, the influence of the residual stresses on the local equivalent stress state $\Delta\tau_{eff,L,RS}$ and the quasi-stationary assumed residual stress state $\tau_{eff,RS}$

$$\Delta\tau_{eff,L,RS,Y}(y) = K_1 \cdot \frac{|\sigma_{RS}(y)|}{100} \cdot 32 \cdot \tanh(9 \cdot y^{1,1}) - K_2 \qquad \tau_{eff,RS} = \sqrt{\frac{2}{15}} \cdot |\sigma_{RS}(y)|$$

Method A requires a measured residual stress curve. Method B describes a procedure to calculate the residual stress depth profile $\sigma_{RS(y)}$ from the hardness depth profile HV(y):

$$\sigma_{RS}(y) = -1{,}25 \cdot (HV(y) - HV_{core}) \qquad \text{for } (HV(y)-HV_{core}) \le 300$$

$$\sigma_{RS}(y) = 0{,}2857 \cdot (HV(y) - HV_{core}) - 460 \qquad \text{for } (HV(y)-HV_{core}) > 300$$

where

HV_{core}	is the core hardness;
$HV(y)$	is the local hardness at the material depth y (hardness depth profile).

KISSsoft

The hardness depth profile HV(y) can either be measured (method B) or calculated (method C) according to two methods Thomas and Lang.

Also the calculation of the local material strength τ_{per} requires the hardness depth profile.

Method according to Lang

$$HV(y^*) = HV_{core} + \left(HV_{surface} - HV_{core}\right) \cdot f(y^*)$$

$$f(y^*) = 10^{(a+b \cdot y^*) \cdot y^*}$$

$$y^* = \frac{y}{CHD}$$

$$a = -0{,}0381$$

$$b = -0{,}2662$$

Method according to Thomas

$$HV(y) = a_a \cdot y^2 + b_a \cdot y + c_a$$

$$HV(y) = a_b \cdot y^2 + b_b \cdot y + c_b$$

$$HV(y) = \mathrm{HV_{core}}$$

KISSsoft

The **local material strength $\tau_{per}(y)$** is calculated as follows

$$\tau_{per}(y) = K_{\tau,per} \cdot K_{material} \cdot HV(y)$$

$K_{\tau,per}$ is the conversion factor;

$K_{material}$ is the material factor;

$HV(y)$ is the hardness at the material depth y (hardness depth profile).

Material factor:

Table 2 — Values for $K_{material}$

Case hardened steels R_m~	$K_{material}$		
	Tooth thickness in mm		
	$3 < s_{t,B-D} \leq 10$	$10 < s_{t,B-D} \leq 40$	$40 < s_{t,B-D}$
min. 800 N/mm²	1,00	0,90	0,70
min. 900 N/mm²	1,13	1,00	0,90

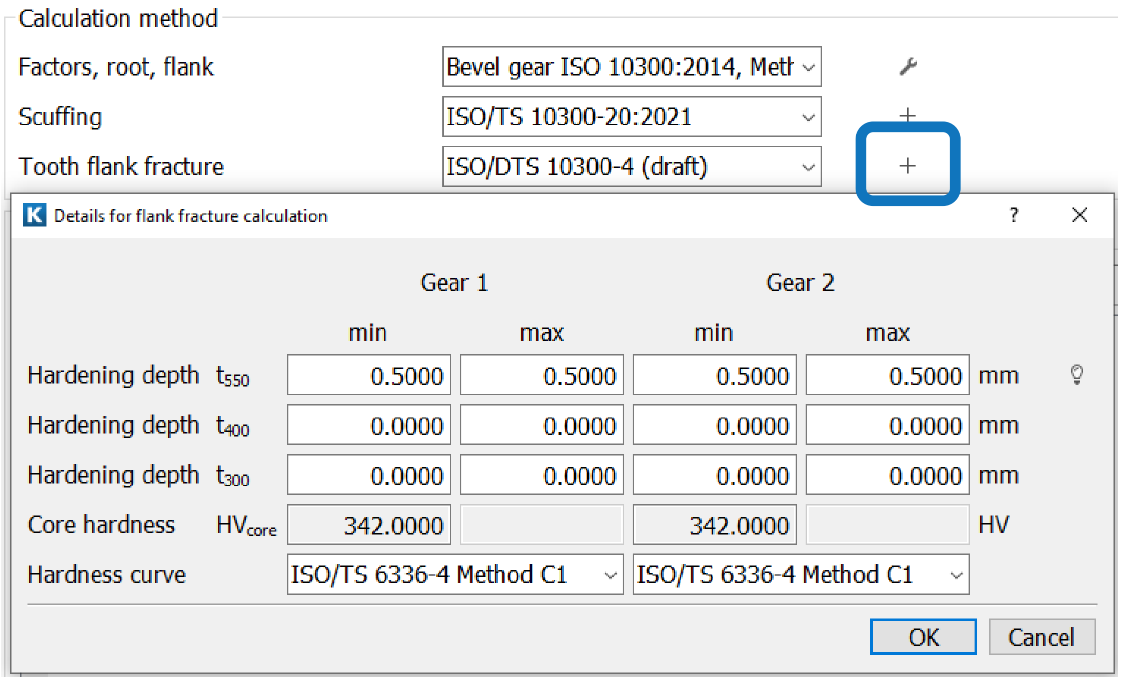
Conversion factor:

$$K_{\tau,per} = 0{,}4$$

KISSsoft

Settings for Tooth flank fracture in KISSsoft

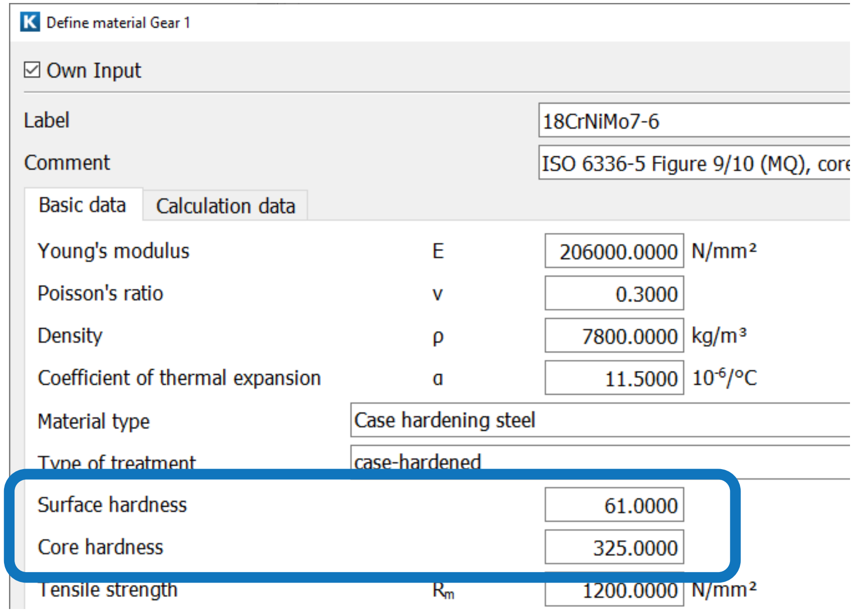
Selection of calculation method:



The calculation of hardness course needs:

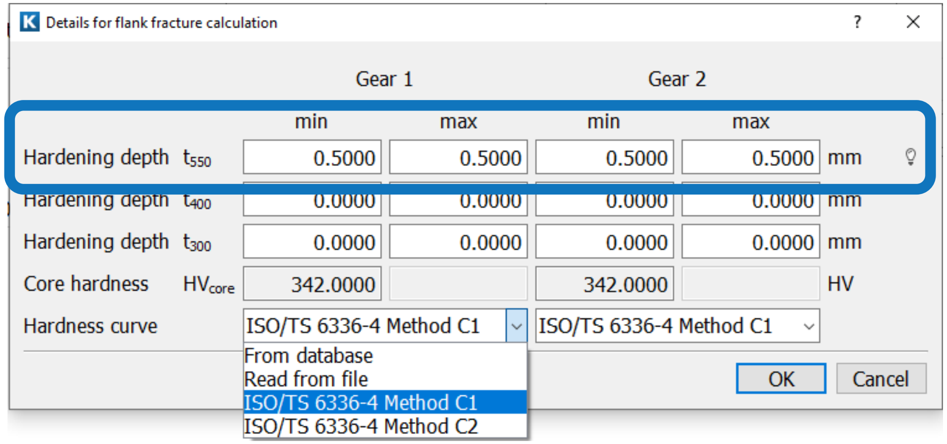
- Surface hardness
- Core hardness
- Case hardening depth

The surface hardness is entered in ‘material properties’.



The core hardness is entered in ‘Define Material’.

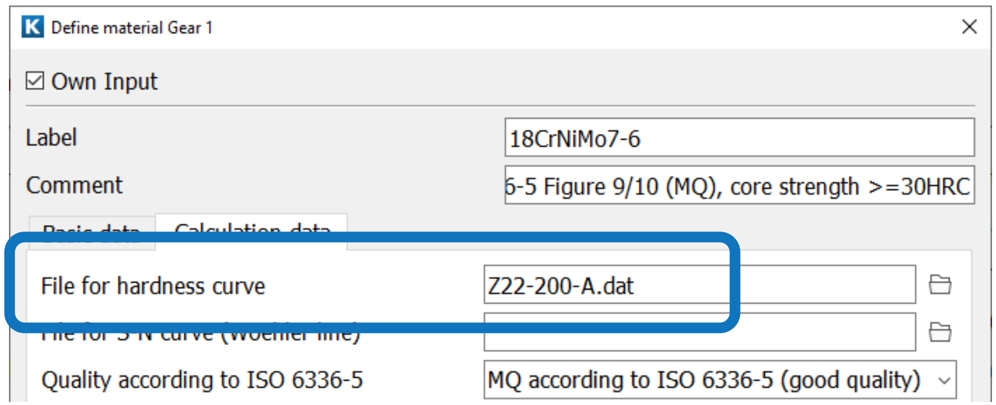
The case hardening depth CHD is entered in ‘Details for flank fracture’.

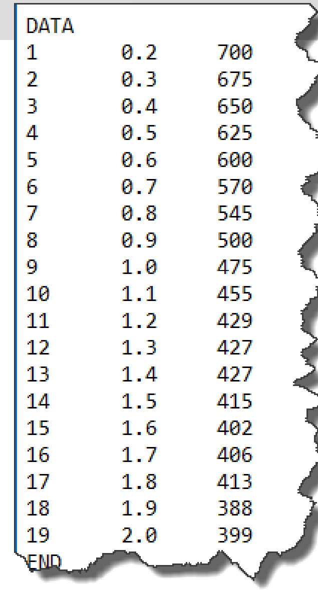


KISSsoft

Measured hardness curve

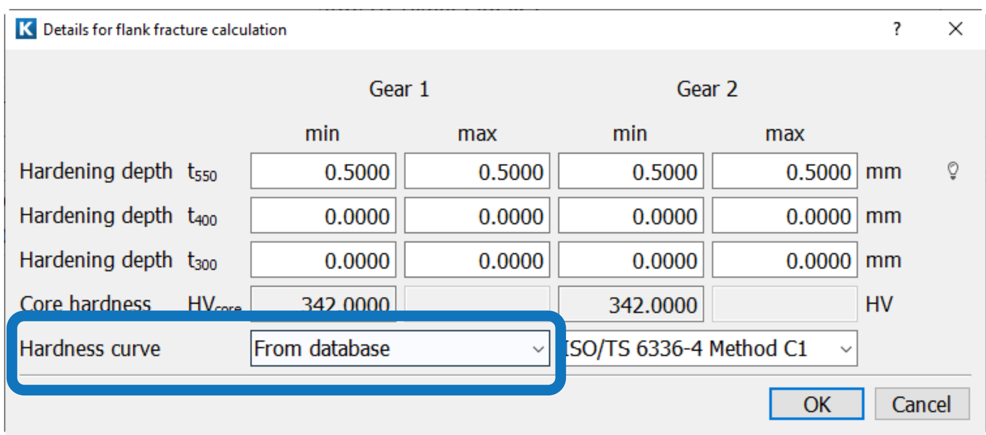
The file with measured hardness data is added to the material properties, or added in Tab ‘Tooth flank fracture’.

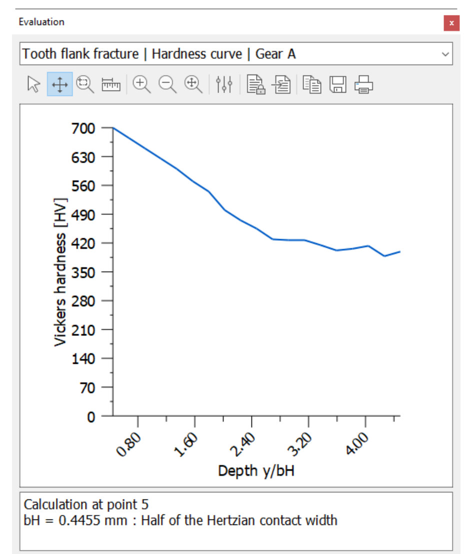



DATA

1	0.2	700
2	0.3	675
3	0.4	650
4	0.5	625
5	0.6	600
6	0.7	570
7	0.8	545
8	0.9	500
9	1.0	475
10	1.1	455
11	1.2	429
12	1.3	427
13	1.4	427
14	1.5	415
15	1.6	402
16	1.7	406
17	1.8	413
18	1.9	388
19	2.0	399

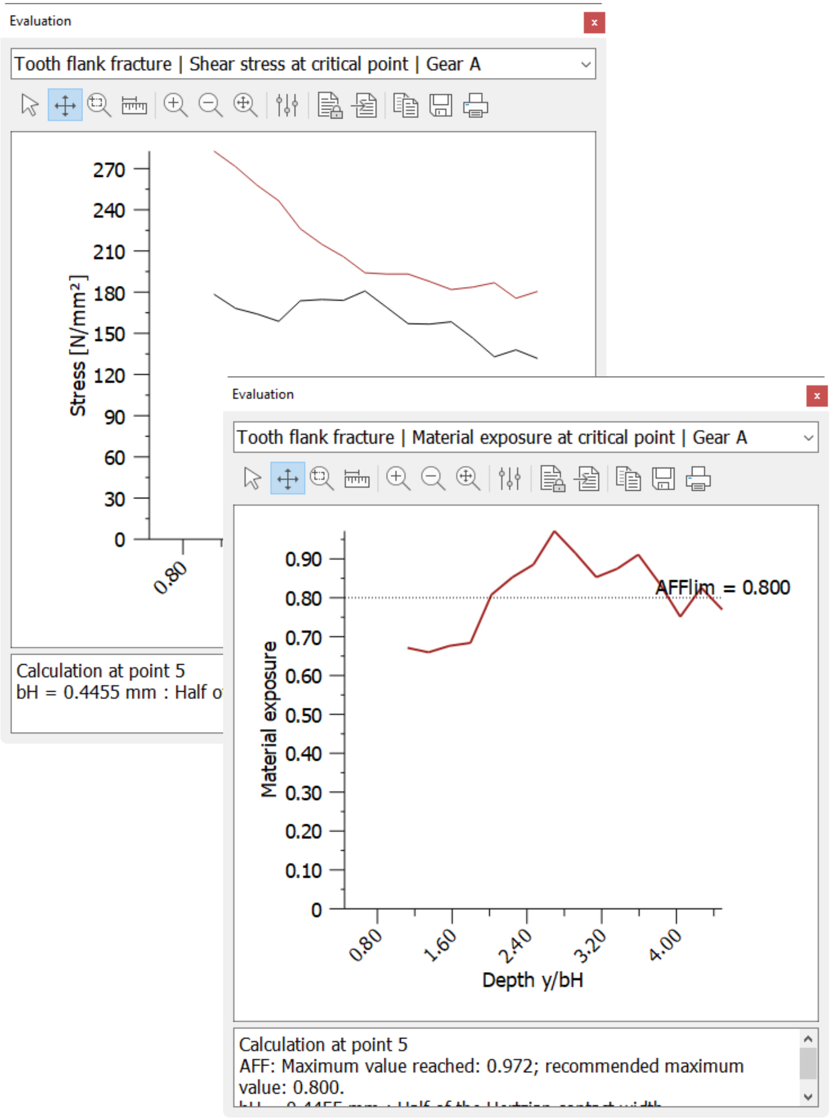
END





In the graph ‘evaluation – tooth flank fracture’ the results are shown for:

- Hardness curve (measured or calculated)
- Equivalent stress state τ_{eff}
- Material shear strength τ_{per}
- Material exposure A_{FF} is the maximum permitted exposure (0.8), red curve to the right → too high exposure



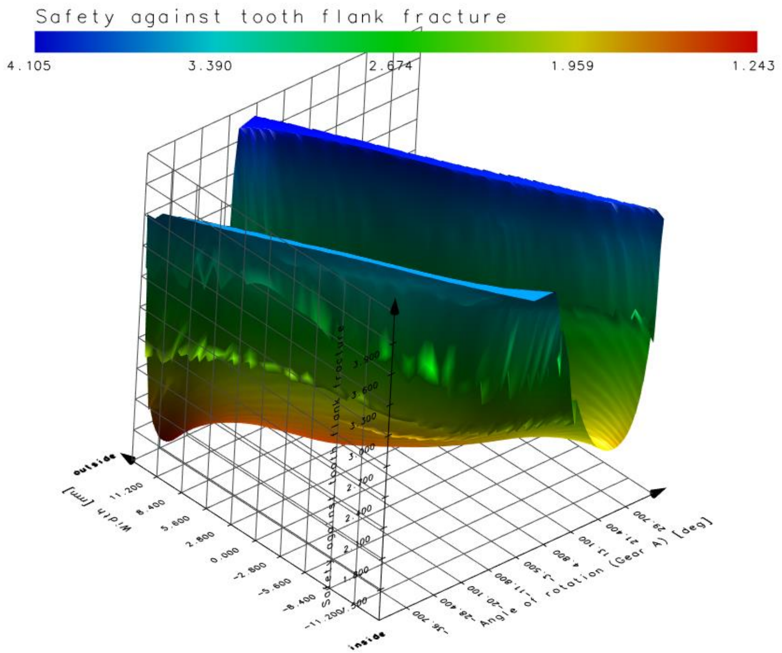
KISSsoft

Using the contact analysis, the safety for TFF is calculated along method A.

This considers the flank and lead modifications, gear misalignment and load distribution under applied torque.

The result is shown in the graphic ‘Graphics - Contact analysis - Safety against tooth flank fracture’

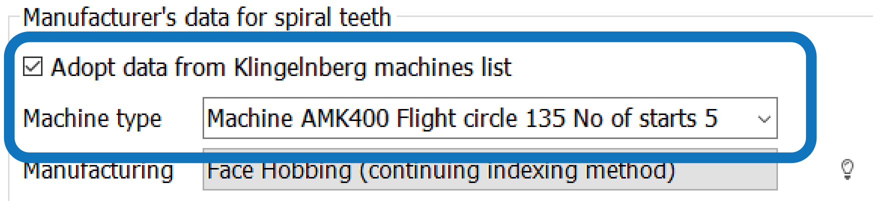
The material exposure $A_{FF,x}(y)$ is calculated for each point x on the flank and hence the safety factor SFF is also shown for each point on the flank.



KISSsoft

Zyklo-Palloid® Cutting method from Klingelnberg

In the tab 'Process', the list of cutter heads is available, if the cone type 'Uniform depth, Face Hobbing, Klingelnberg' is selected.



According to Klingelnberg

Machine	Cutter flight radius	Normal module
FK41B	25 ... 30 30	0.25 ... 1.60 (0.25) 0.75 ... 1.60
AMK400	55 100 135 170	(1.2) 2.0 ... 4.0 2.4 ... 5.5 3.5 ... 8.0 (3.5) 6.5 ... 9.0 (12)
KNC40	30 55 75 100 135	1.0 ... 1.6 1.2 ... 4.0 2.0 ... 4.5 2.4 ... 5.5 3.5 ... 8.0
	55	(1.2) ... (4.0)

Several checks as defined in KN 3028:

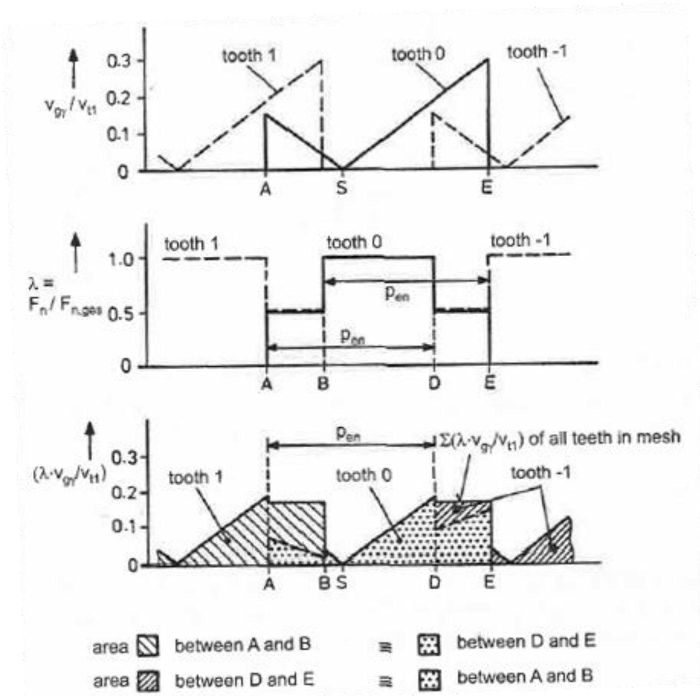
- Machine size (machine distance)
- Interference at inner side
- Etc.

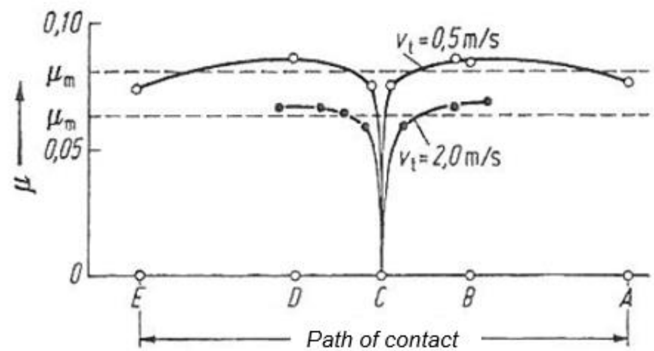
No calculation of machine settings in KISSsoft.

KISSsoft

$$\eta_{ZP} = 1 - H_V \cdot \mu_{mZ}$$

$$H_V = \frac{1}{p_e \cos\alpha} \int_x \frac{F_t(x)}{F_n} \frac{v_g(x)}{v_t} dx$$





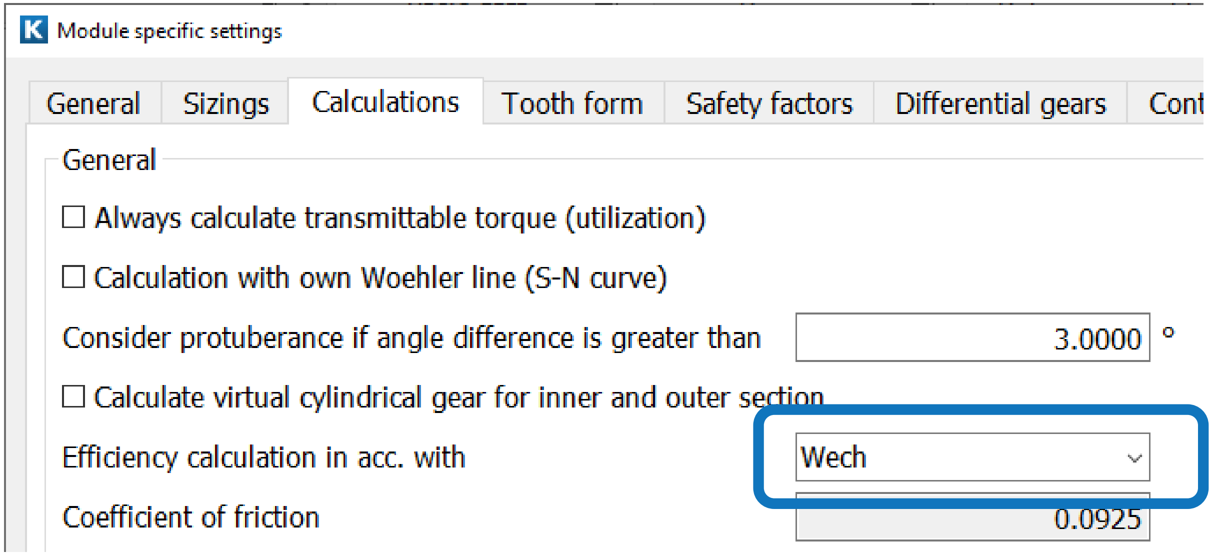
H_V Tooth mesh loss factor

μ_m Mean friction coefficient

In KISSsoft, the efficiency calculation is available for various methods:

- Niemann
- Wech
- ISO 10300-20
- Own input

The calculation according to Wech is basically recommended.

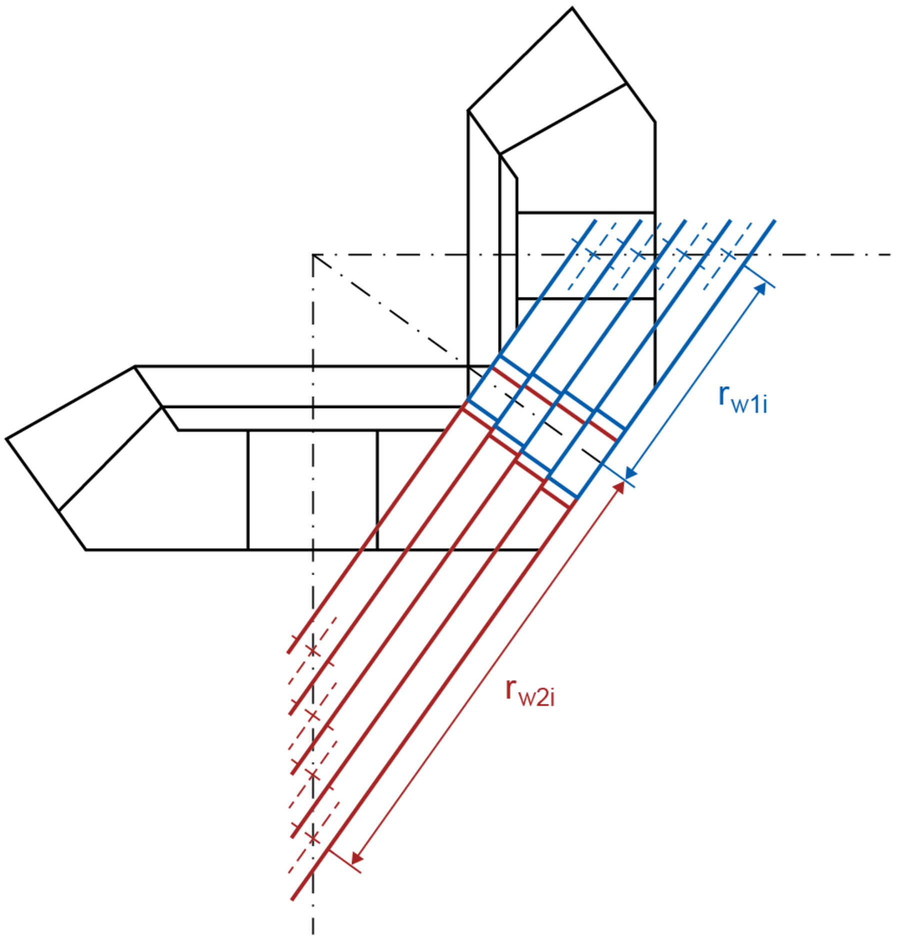


KISSsoft

Contact analysis

Theoretical model

- The face width is splitted into several virtual cylindrical gears
- Each virtual cylindrical gear has it's own tooth form with:
    - Operating pitch diameter, centre distance, etc.
    - Operating load, etc.



KISSsoft

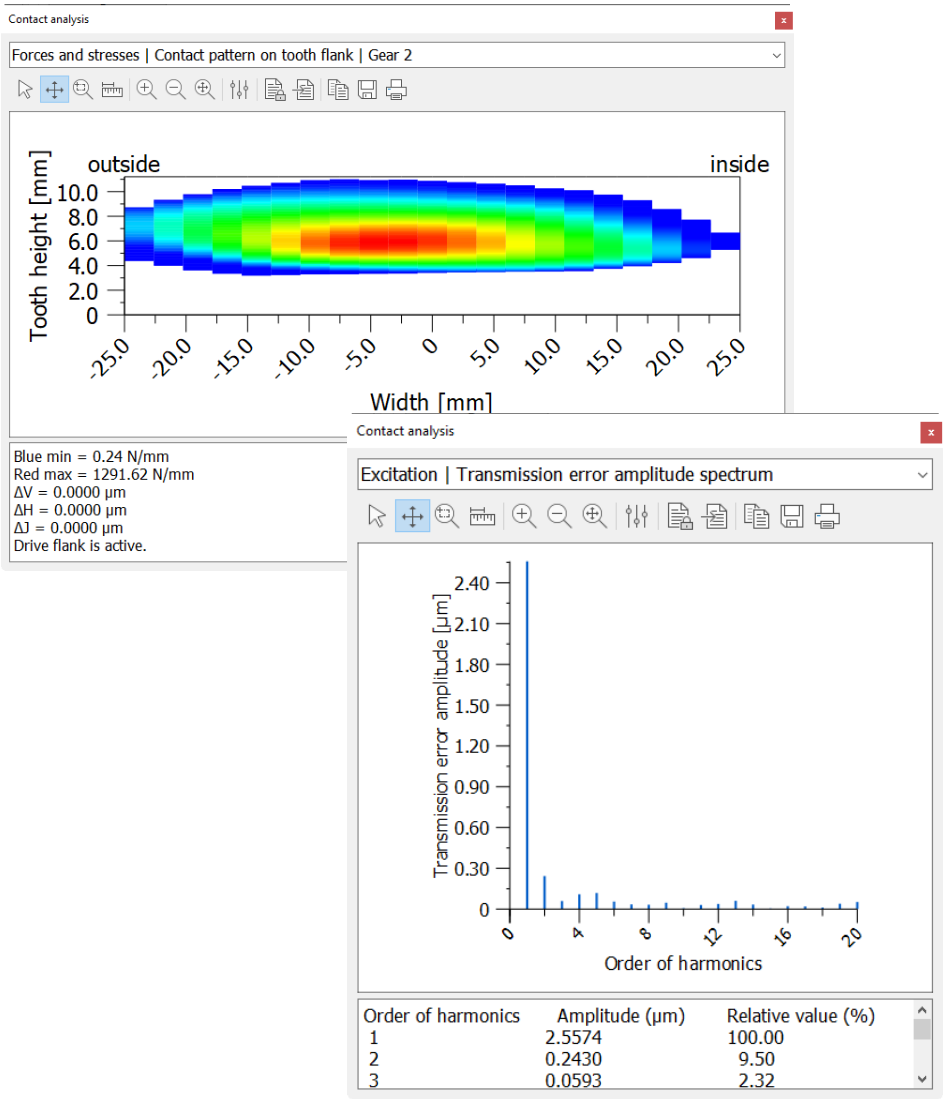
Contact analysis - Evaluation

Graphics:

- Contact pattern on tooth flank on ring gear (based on line load)
- Transmission error (TE)
- FFT of TE
- Scuffing
- etc.

Report:

- Flank pressure
- Peak-to-peak transmission error
- etc.



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Contact analysis

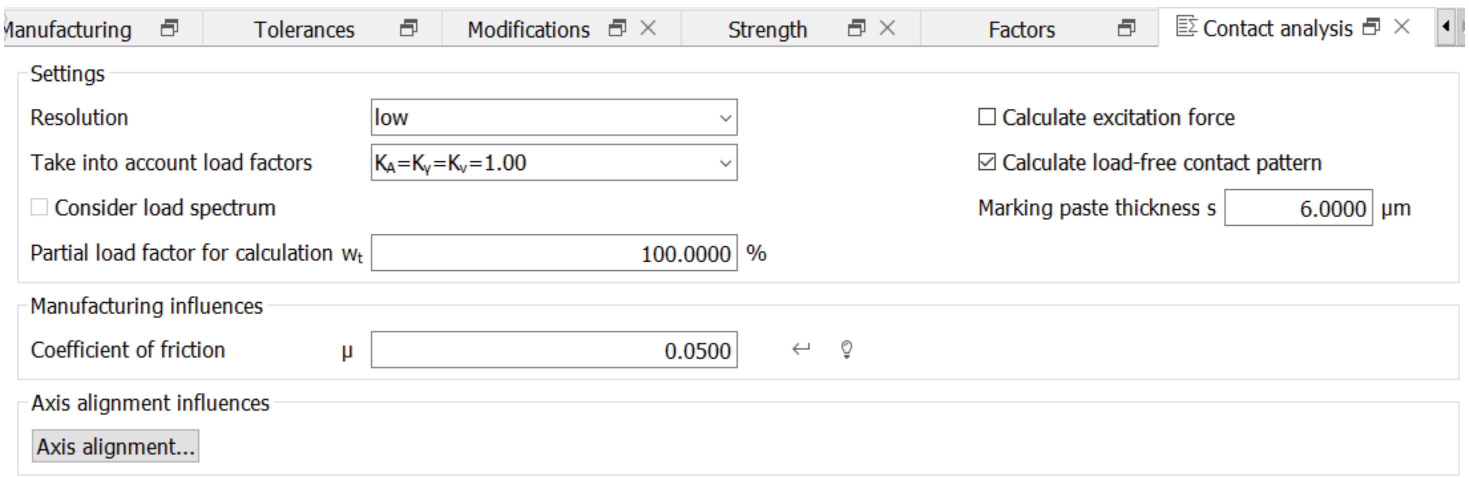
- several settings for load stages

Coefficient of friction

- with sizing acc. to literature

Axis alignment

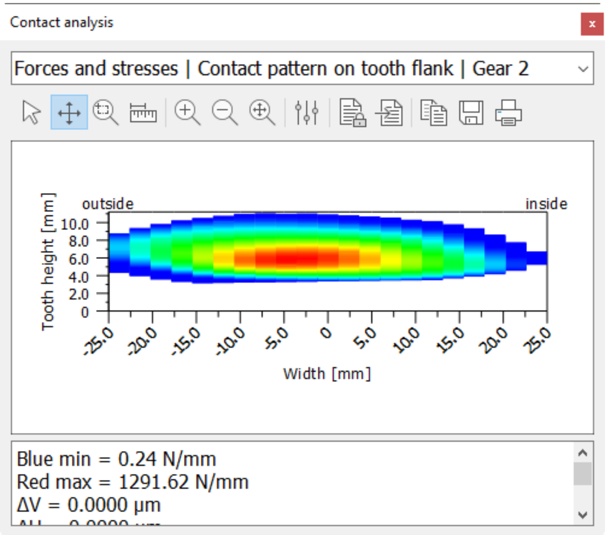
- with definition own input and from shaft files

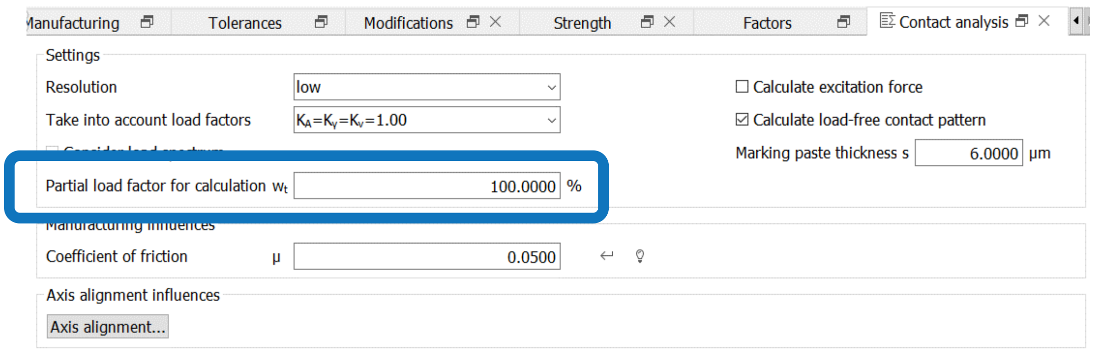


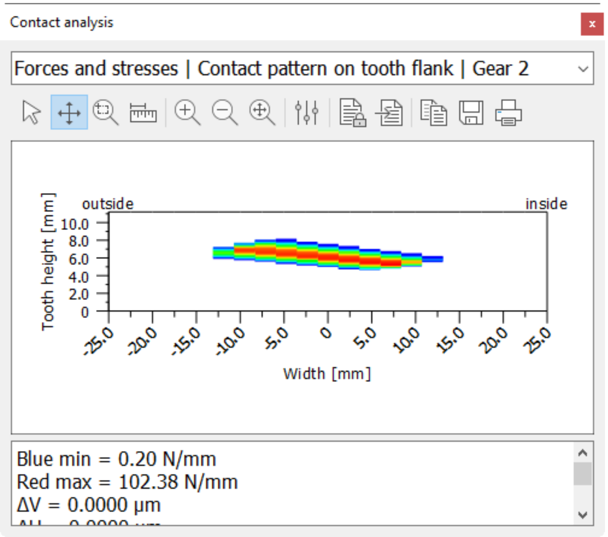
Operating load

The contact analysis is performed:

- Under full load: 100%,
- Under partial load, as on the roll testing machine: the torque is approx. 10 Nm, ≈ 3%
- Consider load spectra
- Consider overload factors K_A, K_Y







KISSsoft

Differences between KISSsoft and GEMS contact analysis

KISSsoft contact analysis

- straight, helical and spiral bevel gears
- using Weber/Banaschek
- flank form based on mathematical approach (planar involute)
  → deviations to m/c settings approach
- no evaluation of Ease off, path of contact
- calculation of root stresses, Hertzian pressure
- safeties for scuffing and flank fracture according to ISO
- nominal position (VH under testing)

GEMS contact analysis

- spiral bevel and hypoid gears
- using FEM
- flank form based on machine settings
- evaluation of Ease Off, path of contact
- calculation of root stresses, Hertzian pressure
- root stresses evaluated with S-N curve
- including the EPG, VH misalignm.

→ **The contact analysis in KISSsoft is not a replacement of GEMS!**

KISSsoft

Part I:

Macro geometry

Define:
general factors, material,
calculation methode

↓

Analysis of
critical gear set

↓

Required safeties
in KISSsoft

↓

Sizing of new gear set

ISO 10300-1:

> Supplied gears or assembled gear drives should have a minimum safety factor for contact stress $S_{H,min}$ of 1,0. The minimum bending stress value $S_{F,min}$ should be 1,3 for spiral bevel including hypoid gears, and 1,5 for straight bevel gears or those with $\beta_m \leq 5°$.
>
> The minimum safety factors against pitting damage and tooth breakage should be agreed between the supplier and customer.

Forging differential bevel gears:

Required safeties are typically much lower due to strengthening effect by the webbings, S_{min} = 0.4 ... 0.8

Note, that the required safeties also influence the damage values.

11. SERVICE LIFE, DAMAGE

Required safety for tooth root	[SFmin]		1.40
Required safety for tooth flank	[SHmin]		1.00
Service life (calculated with required safeties):			
System service life (h)	[Hatt]		52.521
Tooth root service life (h)	[HFatt]	1.305e+004	52.5
Tooth flank service life (h)	[HHatt]	132.7	520.4

Damage calculated on the basis of the required service life [H] (5000.0 h)

F1%	F2%	H1%	H2%
38.32	9519.93	3768.96	960.72

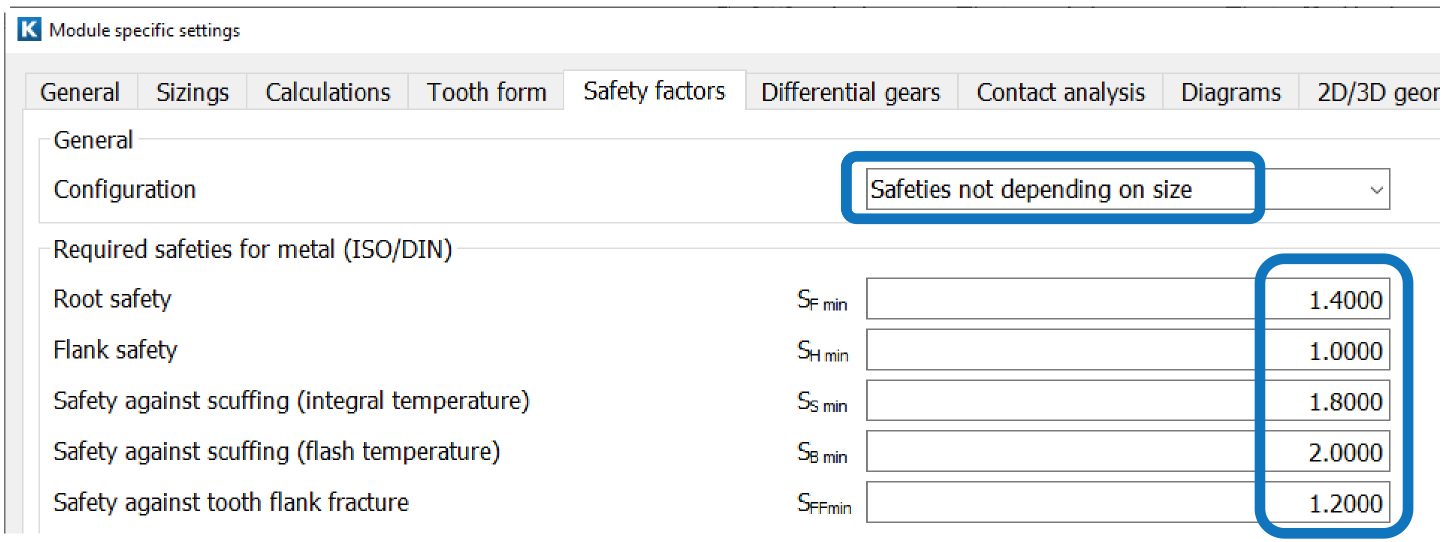
Damage calculated on basis of system service life [Hatt] (52.5 h)

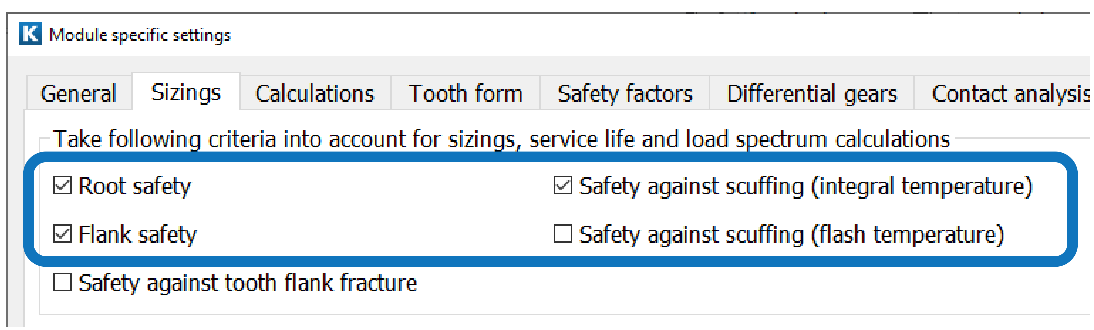
F1%	F2%	H1%	H2%
0.40	100.00	39.59	10.09

KISSsoft

Required safeties in KISSsoft

In KISSsoft, the settings are defined in the tabs ‘Safety factors’ and ‘Sizings’



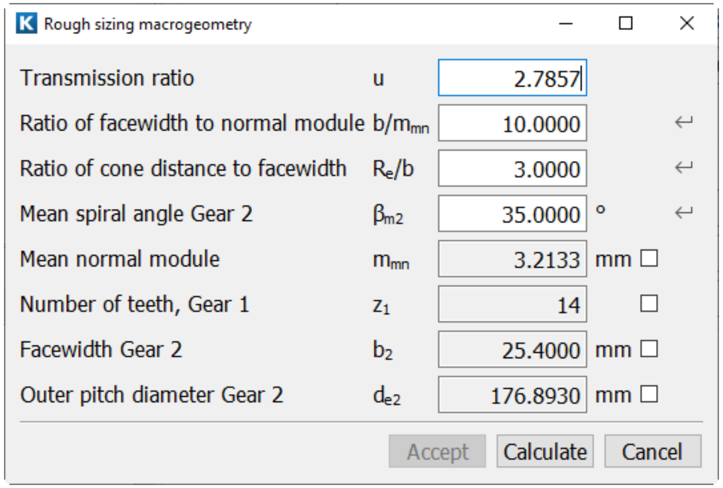


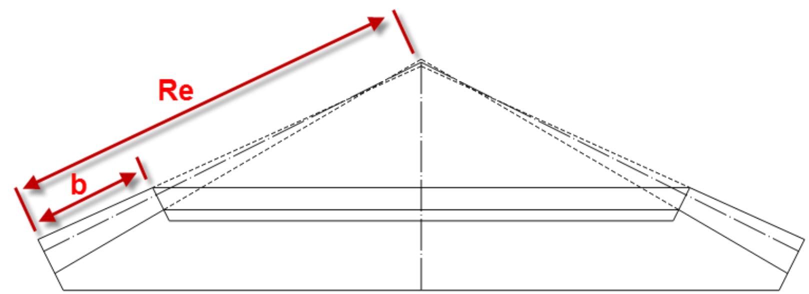
The rough sizing functionality provides a first design. It follows basically the Klingelnberg sizing formulae and considers the parameters b/mn and R_e/b.

The parameter b/m_{mn} is usually in the range of 7…12. The smaller the value, the higher the rooth strength.

Note: in the FVA project 411, root failure was intentionally „produced“ with b/m_{mn} = 10 and flank failure with b/m_{mn} = 7.2

The parameter R_e/b is usually 3 for spiral and hypoid gears.





KISSsoft

Whine

- Mesh frequency and it’s harmonics
- Ghost frequencis (frequencies goes u with increasing speed)
- Profile excited
- May occour at both heavy and light loads

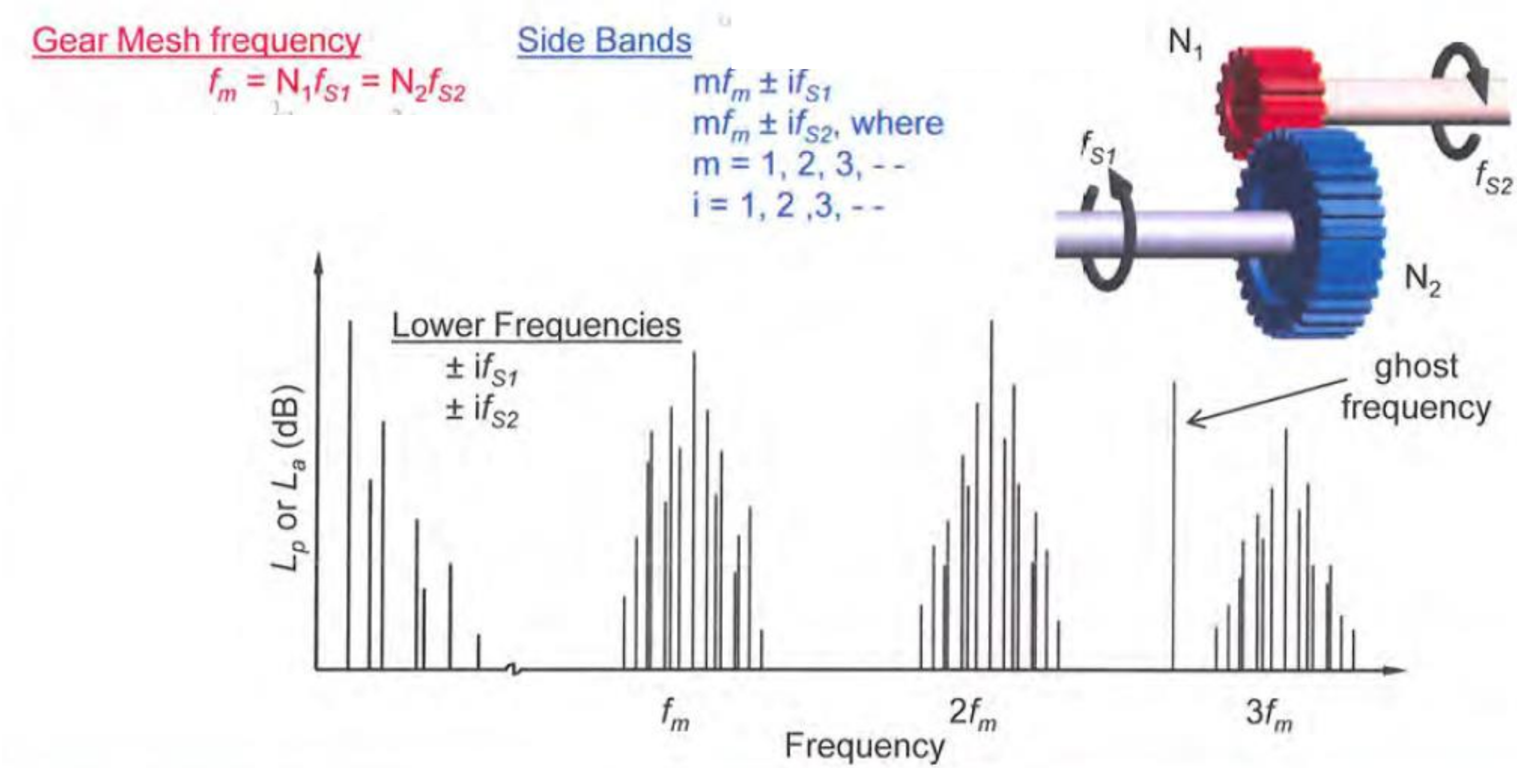
Rattle

- Externally excited
- Lights load
- No distinct frequencies

Gimmick sounds

- Clicks from nicks
- Squeals of plastic gears

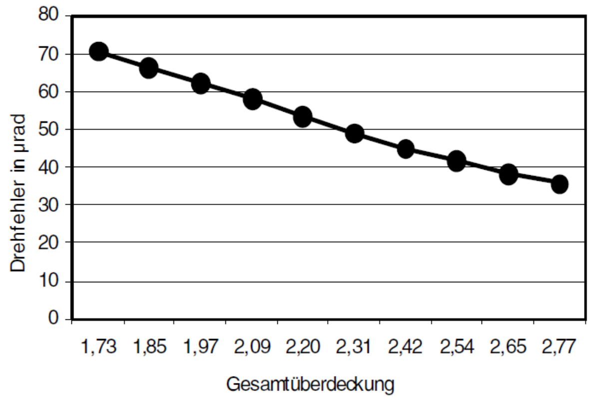
Typical frequency spectrum for Whine



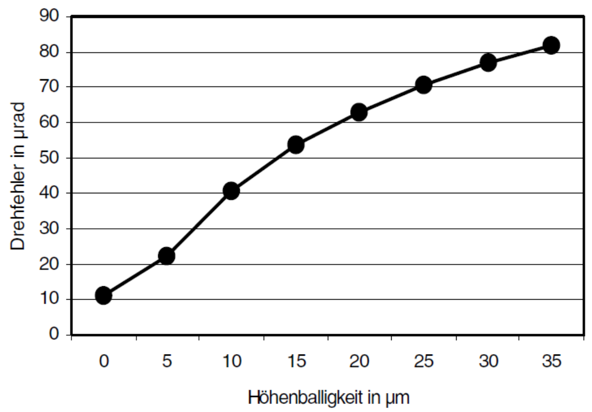
KISSsoft

For noise optimization, macro geometry, micro geometry and manufacturing have to be considered.

For macro geometry, the **contact ratio** (Gesamt-überdeckung) is a main parameter. Note, that the contact ratio is calculated differently between the calculation standards.



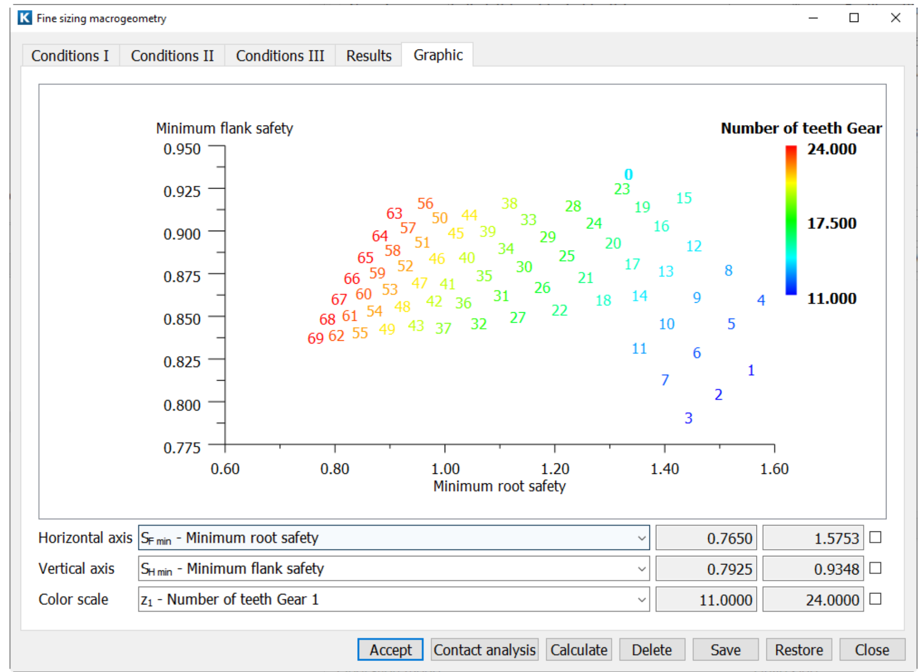
For micro geometry, the **Peak-to-peak transmission error** (Drehfehler) is minimized by low crowning. Note, that for strength performance (i.e. scuffing) crowning is required.



For manufacturing, high pitch and runout quality, as well as the micro structure, are to be considered.

The fine sizing functionality provides a variant calculation within given min/max values for the macro geometry.

As a result, many parameters can be compared as i.e. root safety, flank safety, efficiency, axial and radial forces, etc.

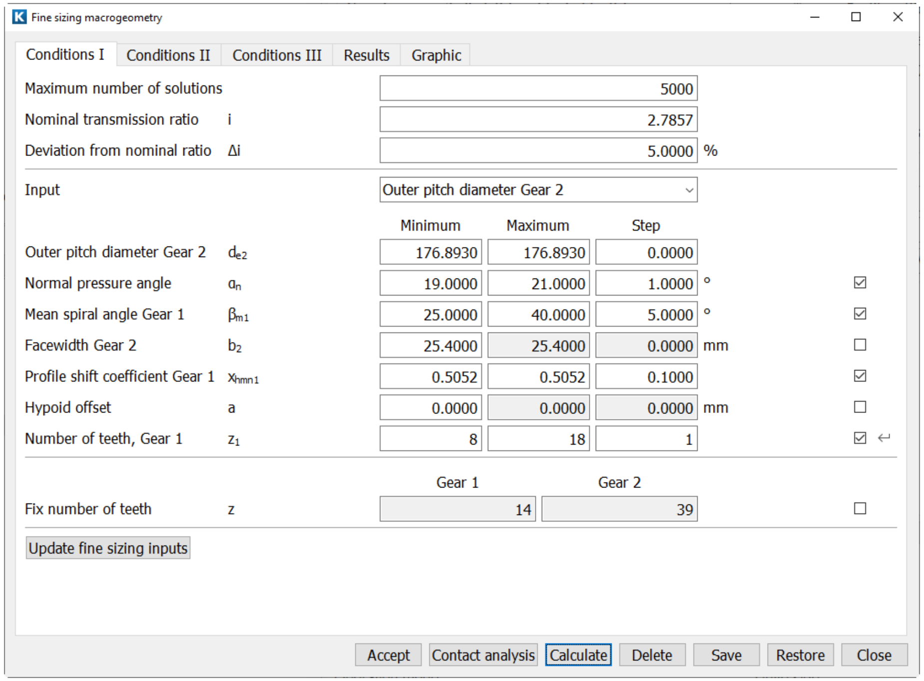


KISSsoft

In conditions I, parameters as spiral angle, face width, number of teeth, etc. are varied.

Typical settings:

- Increase the ‘Maximal No. of solutions’ to ca. 5000.
- Select ‘Outer pitch diameter’
- Enter min / max values and step for the required parameters

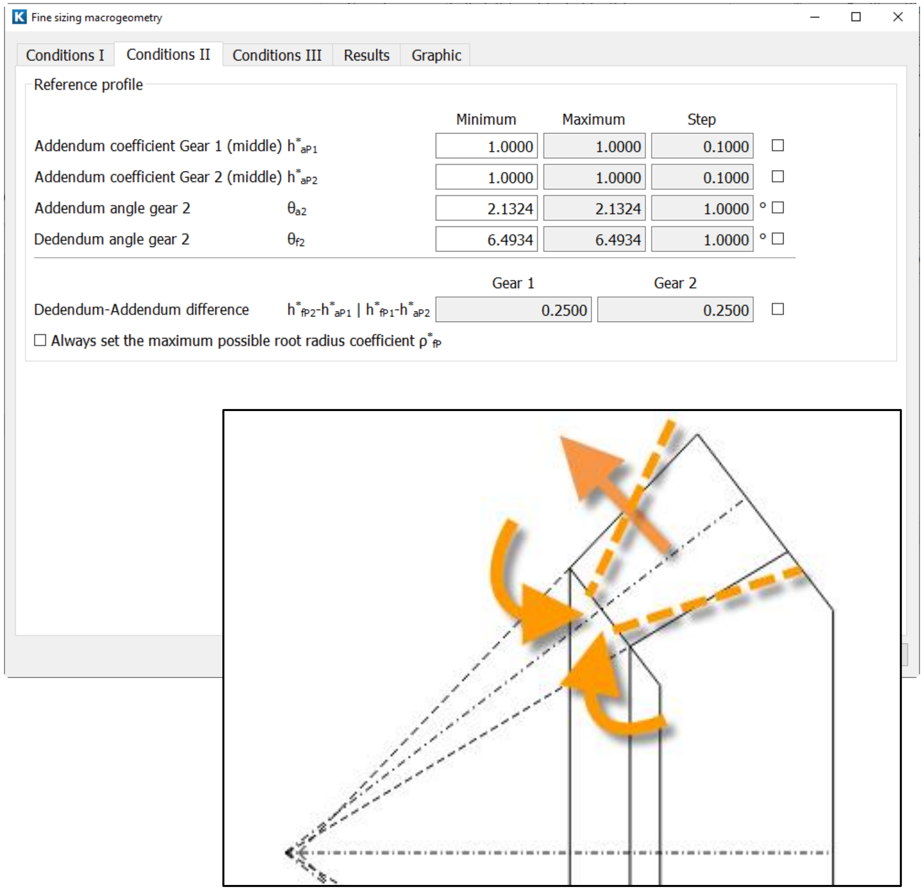


KISSsoft

In conditions II, parameters for deep teeth form and different cone angles are varied.

Typical settings:

- For spiral bevel gears, only ‘Addendum coefficient gear 1 and 2’ are varied.
- For differential bevel gears, also ‘Addendum and Dedendum angle gear 2’ may be varied.

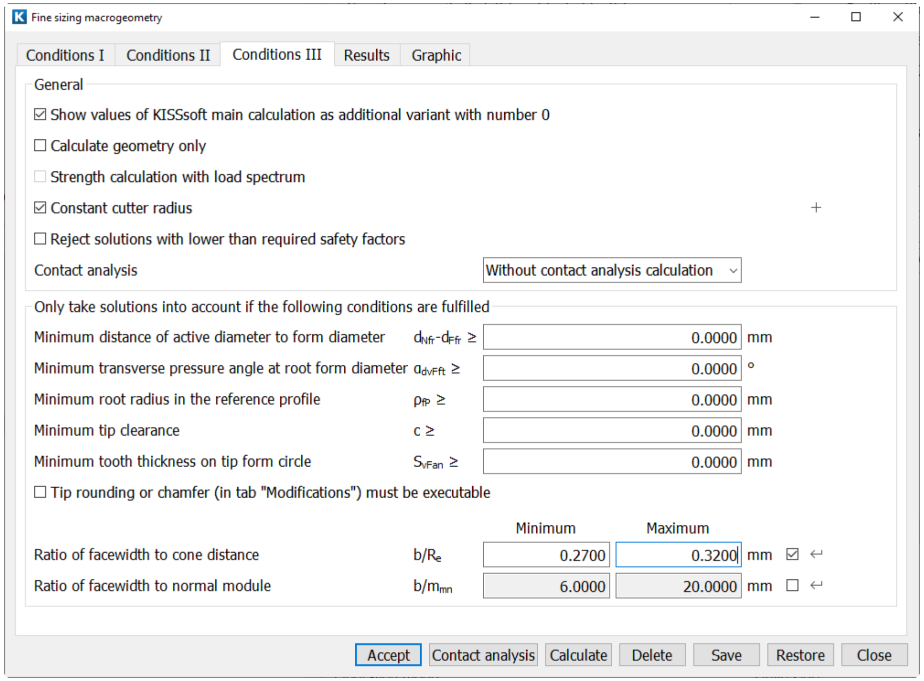


KISSsoft

In conditions III, various options and settings can be defined.

Typical settings:

- Ratio face width to cone distance (for Gleason gears) is set to 0.27 to 0.32
- Ratio cone distance to face width (for Klingelnberg gears) is set to 2.8 to 3.3
- No check for b/mn
- For differential bevel gears, many special geometry conditions may be applied.



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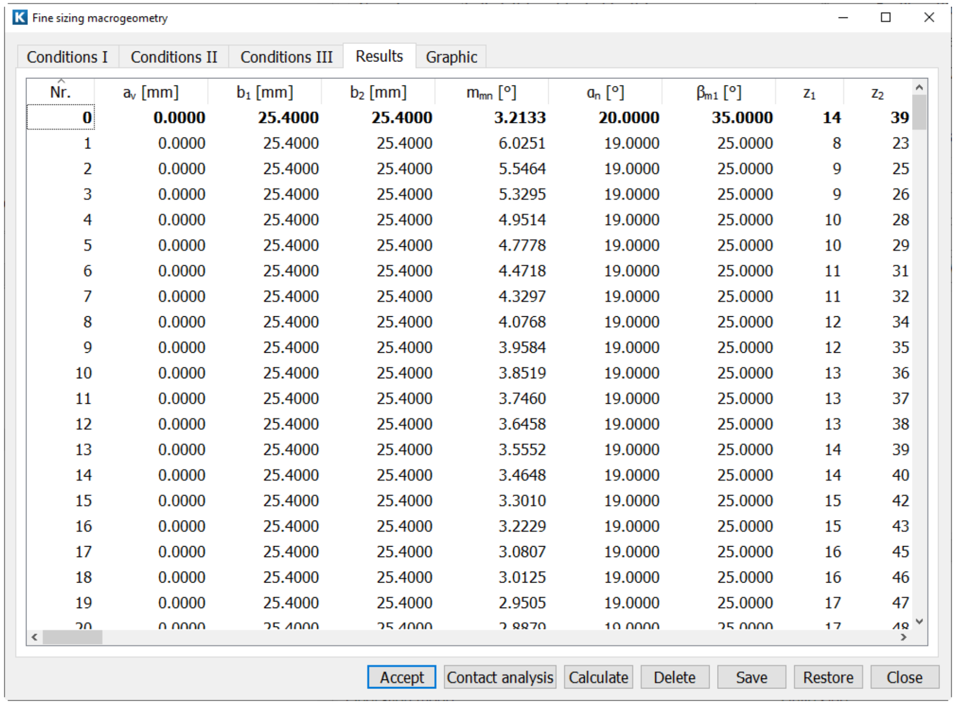
In tab 'Results', all the solutions are displayed. The initial solution **0** is displayed bold.

Typical settings:

- Right click into the solution field in order to show / hide gear and design parameters
- Click on the coloumn title in order to sort the complete list according to a required parameter
- Select solutions in order to delete unsuitable variants



Nr.	a_v [mm]	b_1 [mm]	b_2 [mm]	m_{mn} [°]	α_n [°]	β_{m1} [°]	z_1	z_2
0	**0.0000**	**25.4000**	**25.4000**	**3.2133**	**20.0000**	**35.0000**	**14**	**39**
1	0.0000	25.4000	25.4000	6.0251	19.0000	25.0000	8	23
2	0.0000	25.4000	25.4000	5.5464	19.0000	25.0000	9	25
3	0.0000	25.4000	25.4000	5.3295	19.0000	25.0000	9	26
4	0.0000	25.4000	25.4000	4.9514	19.0000	25.0000	10	28
5	0.0000	25.4000	25.4000	4.7778	19.0000	25.0000	10	29
6	0.0000	25.4000	25.4000	4.4718	19.0000	25.0000	11	31
7	0.0000	25.4000	25.4000	4.3297	19.0000	25.0000	11	32
8	0.0000	25.4000	25.4000	4.0768	19.0000	25.0000	12	34
9	0.0000	25.4000	25.4000	3.9584	19.0000	25.0000	12	35
10	0.0000	25.4000	25.4000	3.8519	19.0000	25.0000	13	36
11	0.0000	25.4000	25.4000	3.7460	19.0000	25.0000	13	37
12	0.0000	25.4000	25.4000	3.6458	19.0000	25.0000	13	38
13	0.0000	25.4000	25.4000	3.5552	19.0000	25.0000	14	39
14	0.0000	25.4000	25.4000	3.4648	19.0000	25.0000	14	40
15	0.0000	25.4000	25.4000	3.3010	19.0000	25.0000	15	42
16	0.0000	25.4000	25.4000	3.2229	19.0000	25.0000	15	43
17	0.0000	25.4000	25.4000	3.0807	19.0000	25.0000	16	45
18	0.0000	25.4000	25.4000	3.0125	19.0000	25.0000	16	46
19	0.0000	25.4000	25.4000	2.9505	19.0000	25.0000	17	47
20	0.0000	25.4000	25.4000	2.8870	19.0000	25.0000	17	48

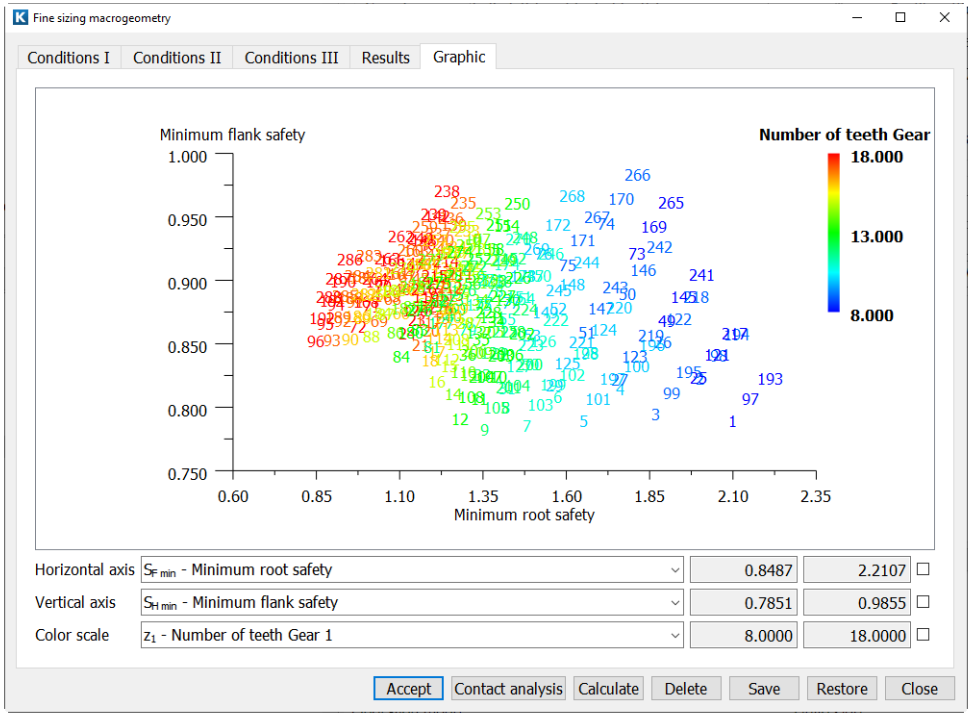
KISSsoft

Fine Sizing: Graphics

In tab ‘Graphics’, 3 parameters can be visualized in one graph.

Typical settings:

- For horizontal and vertical axis, the design criteria are selected, as S_{Hmin}, S_{Fmin}, efficiency, etc.
- For color scale, the gear parameter are selected, as number of teeth, normal module, spiral angle, offset, etc.



KISSsoft

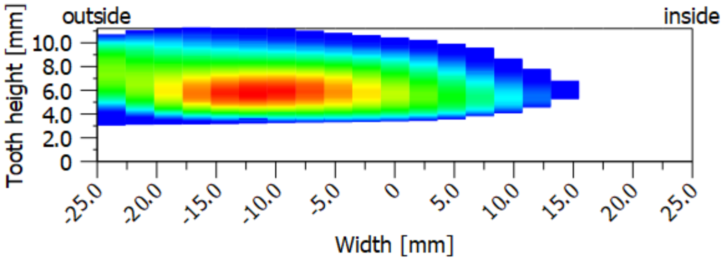
Part II
Micro geometry

Tooth flank modifications are applied in order to achieve gear optimizations in strength and reduce noise.

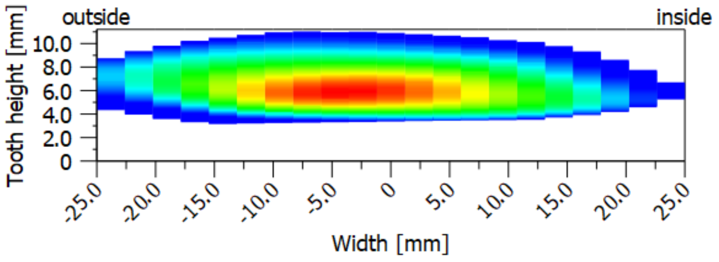
Considering the EPG displacements, bevel gears are typically manufactured with default length crowning.

The available modification types are:

- Spiral and profile angle
- Lengthwise and profile crowning
- Twist
- Topological modification



Contact pattern including misalignments - without modifications



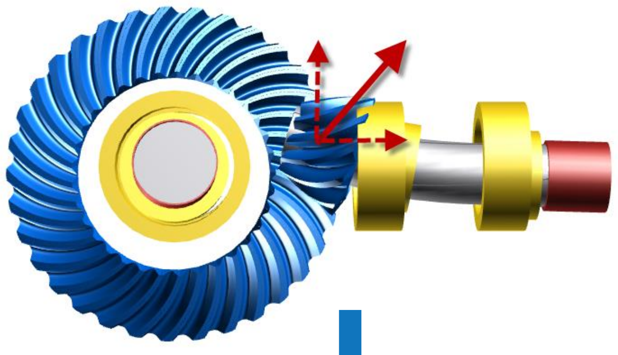
Contact pattern including misalignments - with modifications

Under load, pinion and ring gear are displaced due to tooth normal forces, which cause deformation of shafts and bearings.

Also temperature influences have to be considered.

The relative position between pinion and gear is considered in the loaded contact analysis – Axis alignment.

The contact pattern is typically moved due to the VHJ displacements.

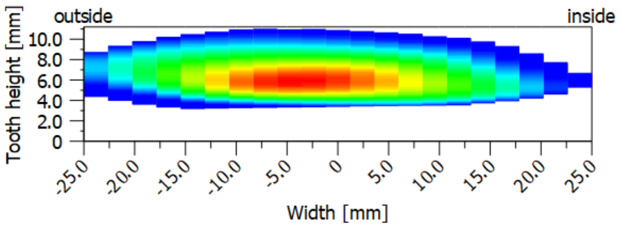


ΔH / ΔP

ΔV / ΔE

ΔJ / ΔG

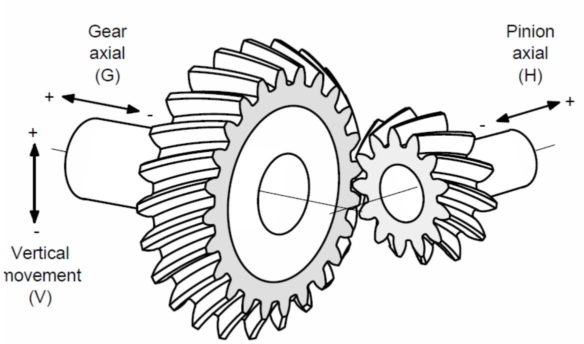
$\Delta \Sigma$



KISSsoft

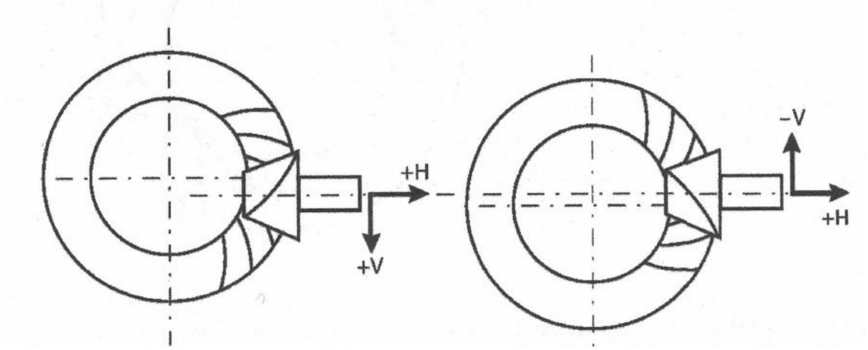
ISO/TR 10064-6

- VHG



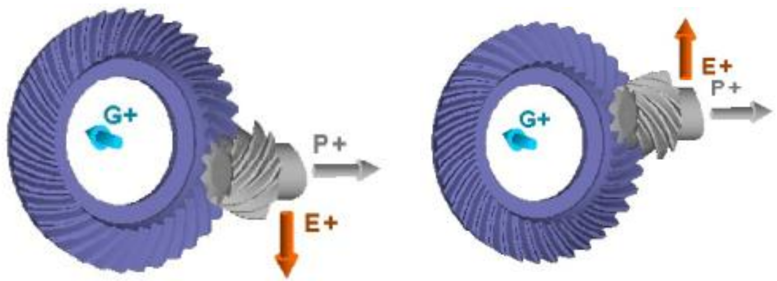
Klingelnberg (Literature)

- VHJ: Vertical, Horizontal
  → in KISSsoft



Gleason (Literature)

- EPG: Excenter, Pinion, Gear
  E+ means offset increasing, so the direction depends on hand of spiral

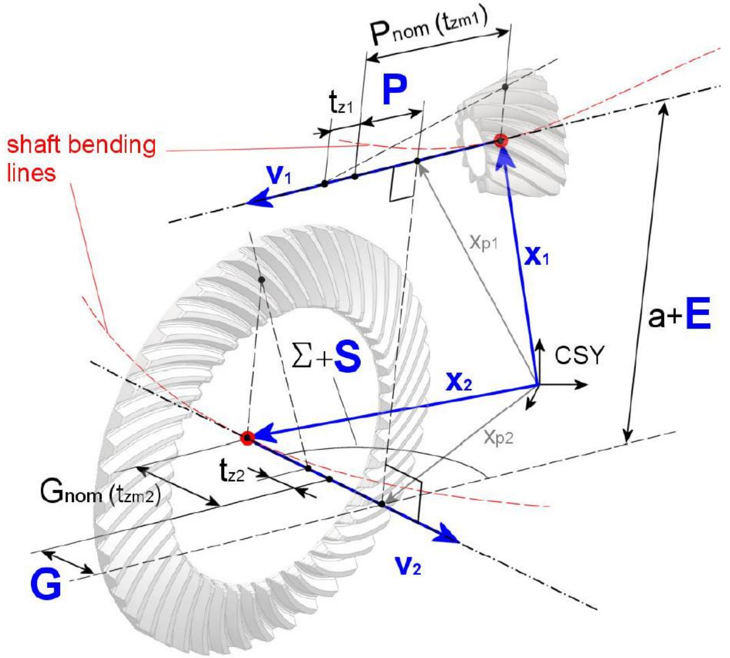


Calculation by approach of vectors

$$\mathbf{S} = a\cos\left(\frac{\bar{v}_1 \bullet \bar{v}_2}{\|\bar{v}_1\| \cdot \|\bar{v}_2\|}\right) \cdot \frac{180}{\pi} - \Sigma \qquad \mathbf{E} = \frac{\left|(\bar{x}_1 - \bar{x}_2) \bullet (\bar{v}_1 \times \bar{v}_2)\right|}{\|\bar{v}_1 \times \bar{v}_2\|} - a$$

$$\frac{\left\|(\bar{x}_{p1} - \bar{x}_2) \times \bar{v}_2\right\|}{\|\bar{v}_2\|} = E + a \quad \text{, where } \bar{x}_{p1} = \bar{x}_1 + (P_{nom} + \mathbf{P}) \cdot \bar{v}_1$$

$$\frac{\left\|(\bar{x}_{p2} - \bar{x}_1) \times \bar{v}_1\right\|}{\|\bar{v}_1\|} = E + a \quad \text{, where } \bar{x}_{p2} = \bar{x}_2 + (G_{nom} + \mathbf{G}) \cdot \bar{v}_2$$



KISSsoft

V displacement

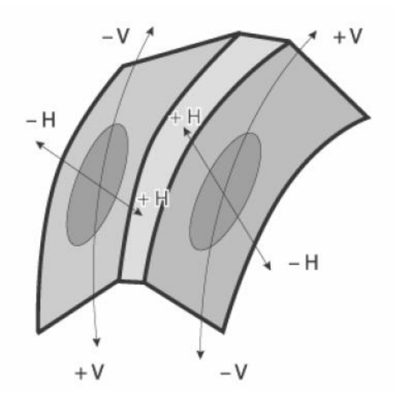
- theoretical displacement is along the face width
- independent of the cutter head size

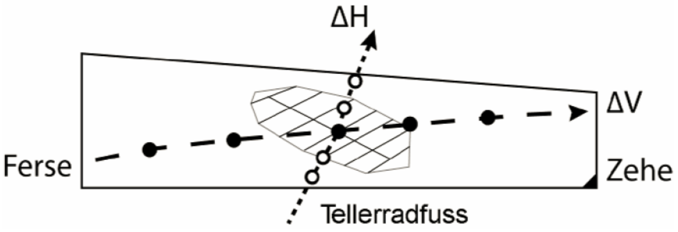
H displacement

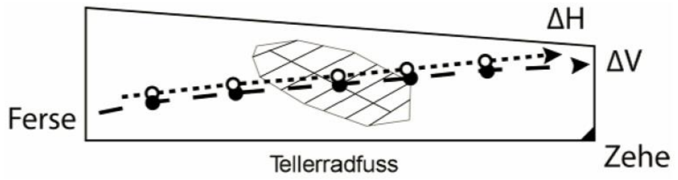
- theoretical displacement is
  a) for small cutter heads: along the face width
  b) for large cutter heads: in profile direction

Properties of small cutter designs:

- Advantage: contact pattern remains in central position under load (see 'cutter head factor')
- Disadvantage: when lapping, the flank can't be lapped fully.





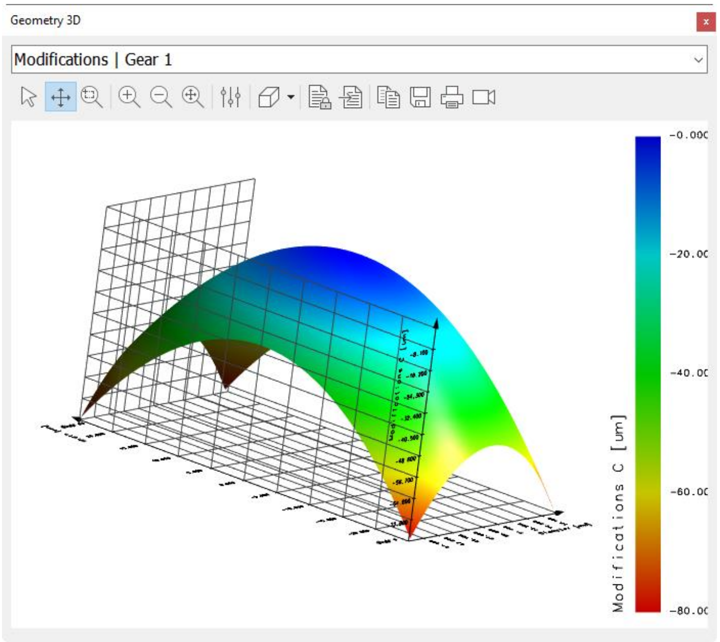


KISSsoft

Modifications are defined in the tab ‘modifications’ and verified in the 3D graph.

Typical modifications and values are:

- Lengthwise crowning
  face width / 1000, for pinion
- Profile crowning, diameter-centered
  0.005 * normal module, for pinion and ring gear
- Spiral angle
  for TCP position optimization



Additional modifications

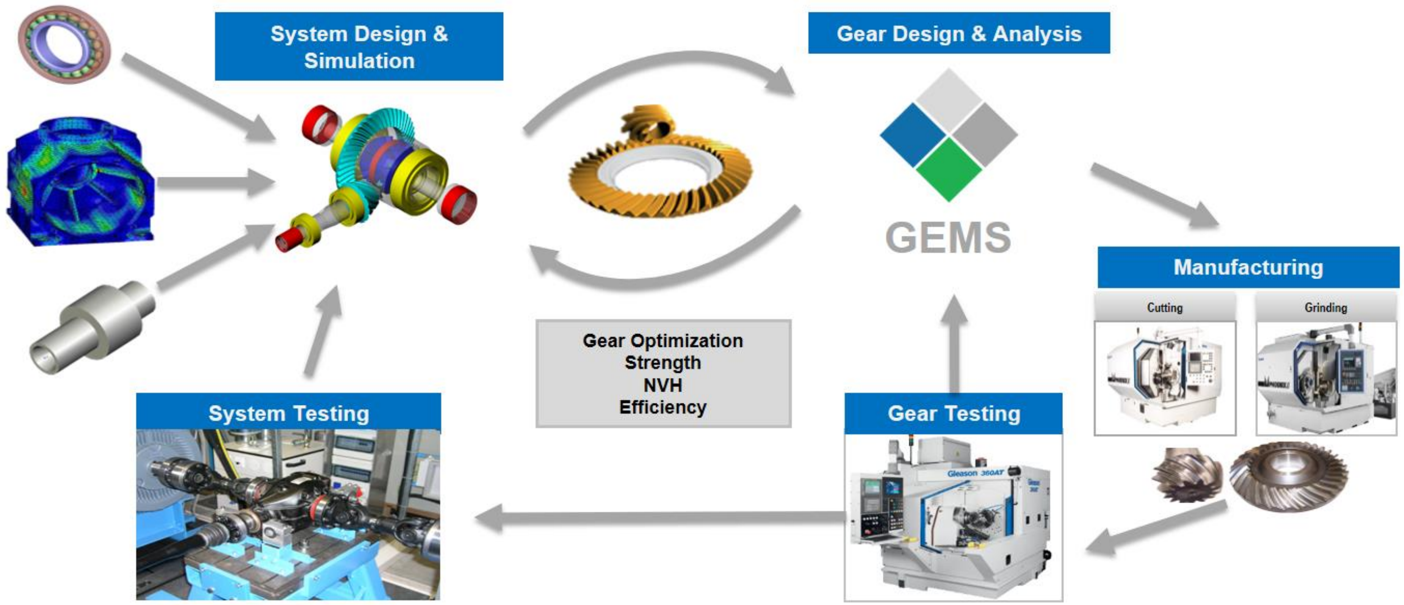
Variant for calculation: No variant defined

Gear	Flank	Modification type	Value [µm]	Factor 1	Factor 2	Status
Gear 1	both	Profile crowning, diameter-centered	30.0000			active
Gear 1	both	Flank line crowning	50.0000	1.0000		active
Gear 2	both	Profile crowning, diameter-centered	30.0000			active

KISSsoft

Production of bevel gears

In KISSsoft / KISSsys the gear design using fine sizing is done.
The macro geometry including load data is transferred to GEMS®
The contact pattern is developed and LTCA is calculated.
The results are transferred back to KISSsys



KISSsoft

Sample for design of bevel gear set (industrial application, Face Milling, ground)

Macro geometry in KISSsoft

- Face width to outer cone distance (b/Re): 0.33
- Spiral angle (beta): 30°
- Gear radius: from GEMS

Cutter head size (ratio involute / outer cone)

- Ratio involute / outer cone : (0,95)..1,15

Base design of TCA in GEMS (final ground design)

- Lengthwise crowning: face width / 700 (standard industrial gearbox)
- Profile crowning: defined by blade curvature, 800 mm
- Contact pattern position: slightly to toe

Blade design (both members)

- Protuberance pre-cutting: straight toprem 4.5º, height by letter
- Protuberance grinding: no protuberance, or blended toprem

KISSsoft

Macro geometry in KISSsoft

- Face width to outer cone distance (b/Re): 0.33
- Spiral angle pinion (beta): 40..50º
- Tooth root radius: from GEMS
- Offset: usually 10..15% of d_2

Cutter head size:

- Ratio involute / outer cone : 0,90..(1,1)

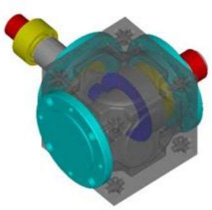
Base design of TCA in GEMS (pre-manufacturing)

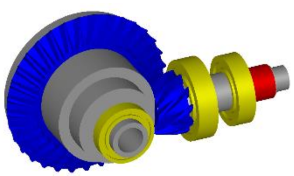
- Lengthwise crowning: face width / 800
- Profile crowning: defined by blade curvature, 500 mm
- Transmission error: approx. 40 µrad (after lapping 20..30 µrad)

Blade design (pinion only):

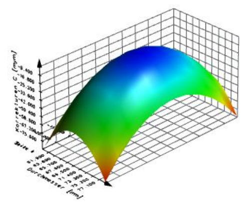
- Protuberance (pre-cutting): straight toprem 4.5º, height by letter

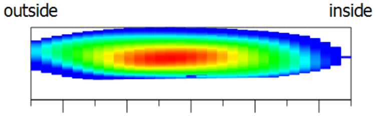
The process of manufacturing with 5-axis machines is as follows:

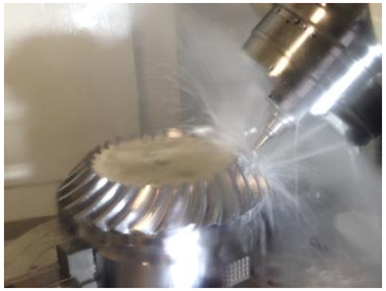


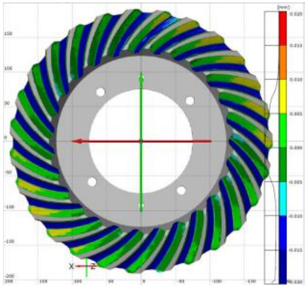


ΔV	-233.9915	µm
ΔH	199.7815	µm
ΔJ	387.1006	µm
ΔΣ	-0.0146	°






Source: Euklid CAD/CAM AG


Source: GOM mbH

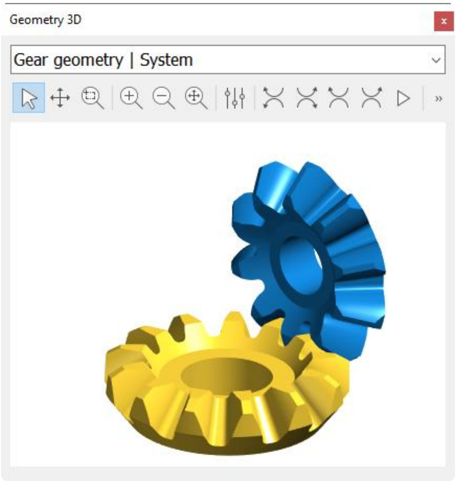
KISSsoft

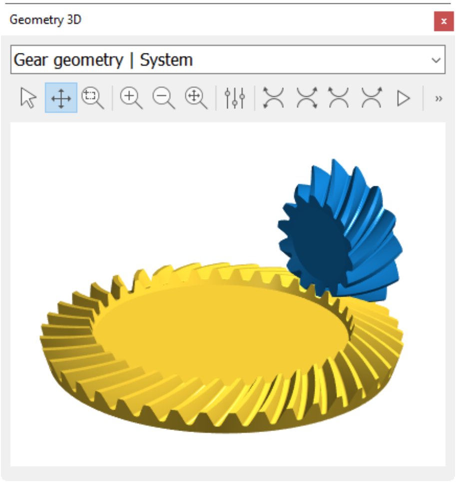
For straight, helical and spiral bevel gears 3D models can be generated and exported as STEP files.

Modifications are possible with length and profile crowning, pressure and spiral angle and twist.

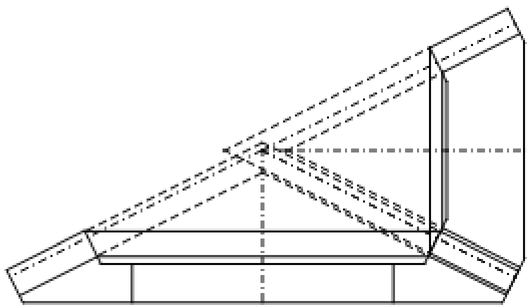
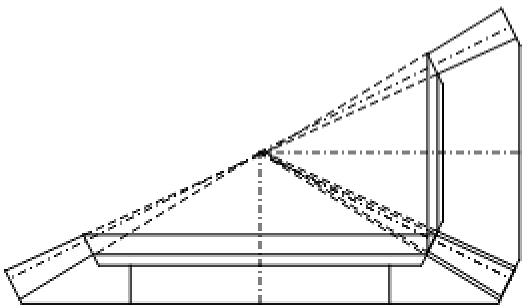
With contact line check the manufactured contact is checked and optimized.

The STEP models can be used for 5-axis milling cutting method.



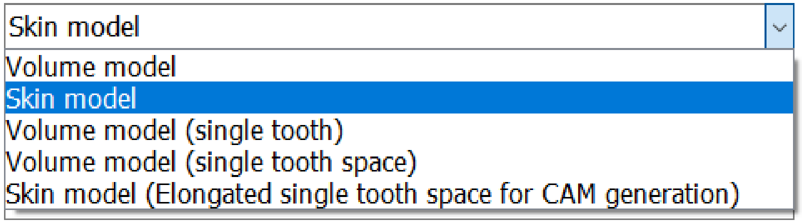


KISSsoft

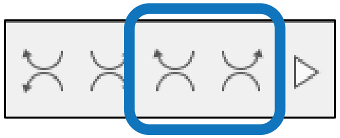
	Face Hobbing	**Face Milling**
		
Tooth height:	Parallel	tapered
Lengthwise:	Elongated epicycloid	straight, arc of a circle

Differences to conventional bevel gears: Profile shape is planar involute, non-conjugating contact of ring gear and pinion for Klingelnberg

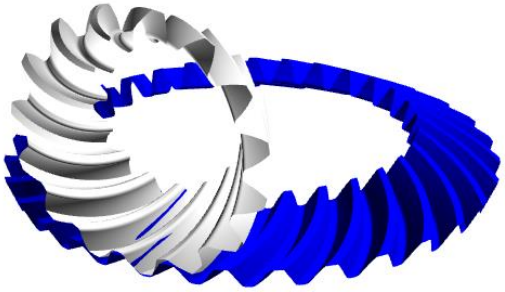
Checking of contact lines is possible using the option ‘Skin model’.



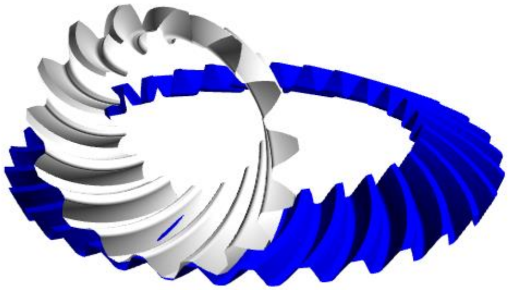
Rotate independently one gear according to the hint in the message box, so that a smallest contact line is shown due to intersection of the two models.



By rotation of the gears, all the contact lines can be checked.



Initial tooth contact

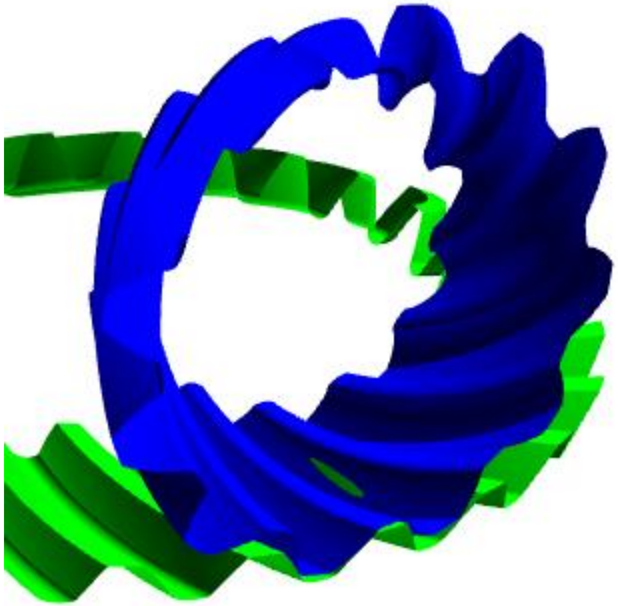


Tooth contact with crowning

KISSsoft

Verification of contact pattern

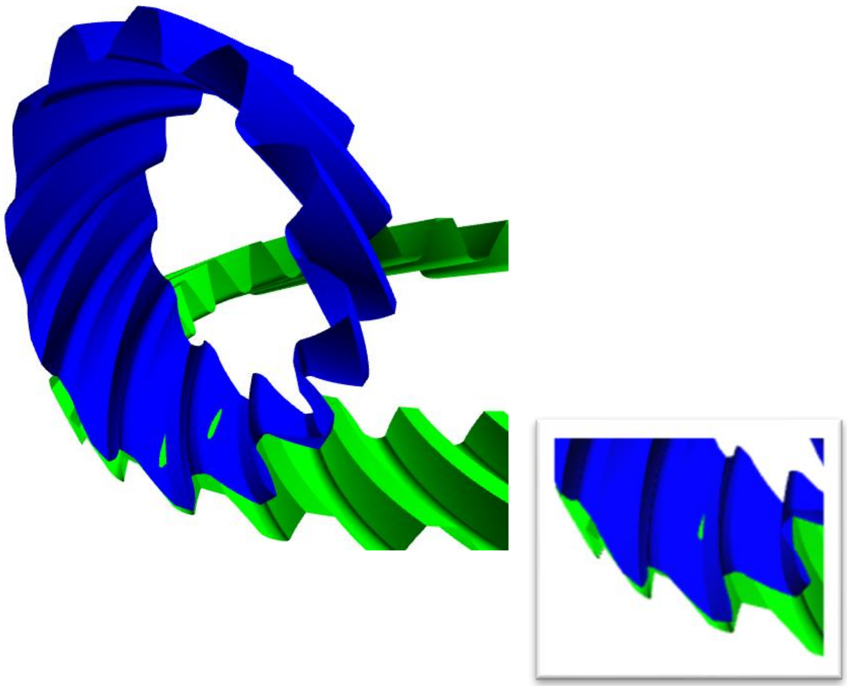
Drive side



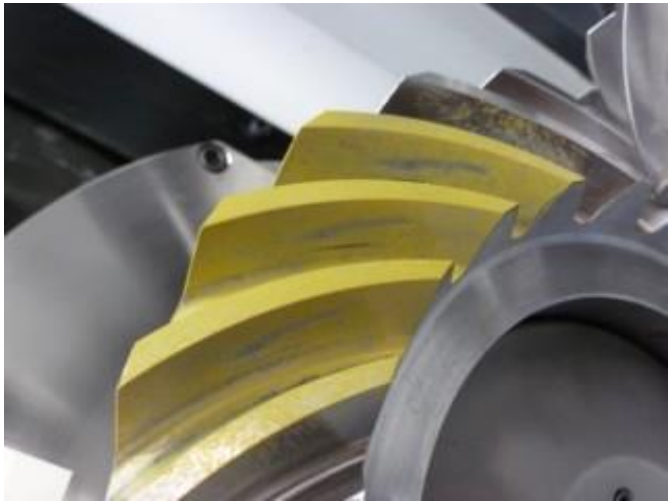
KISSsoft

Roll tester

Coast side



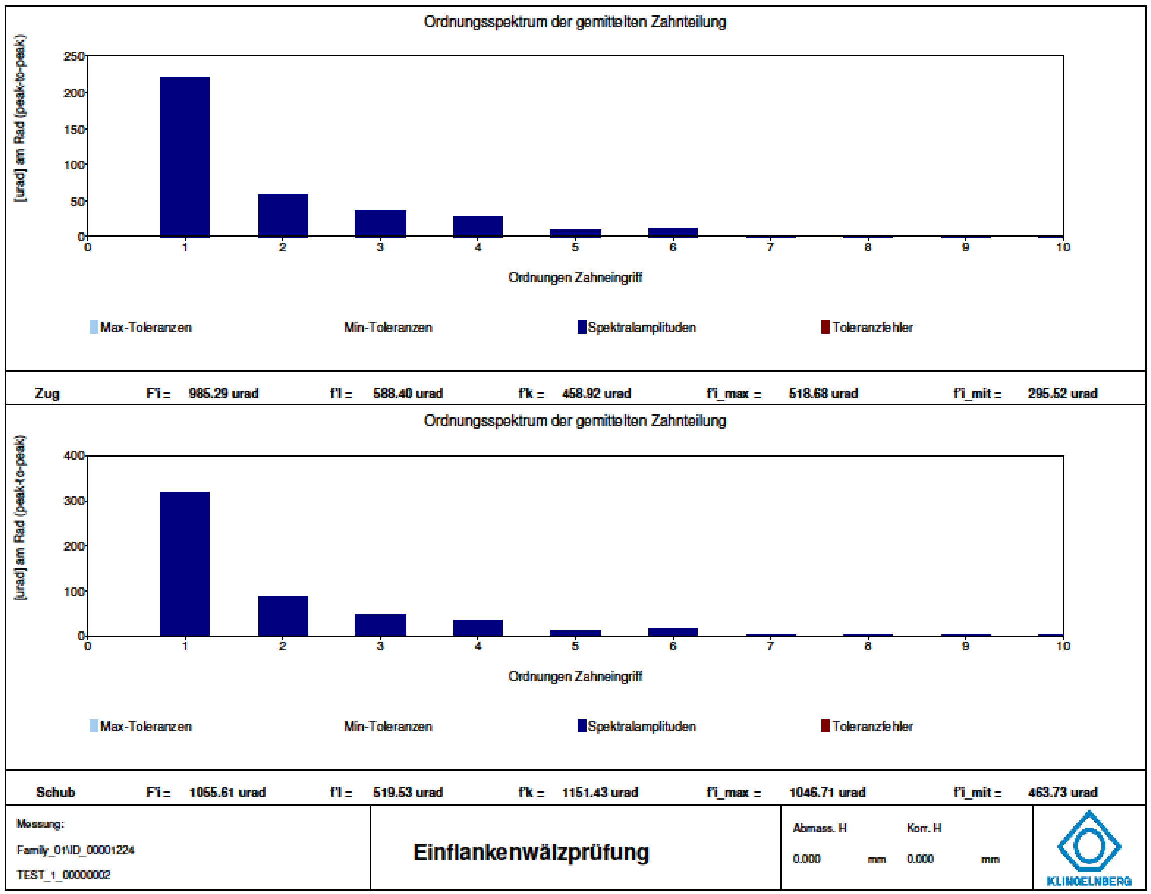
KISSsoft
(note: interference line in root)



Roll tester
(note: interference line in root)

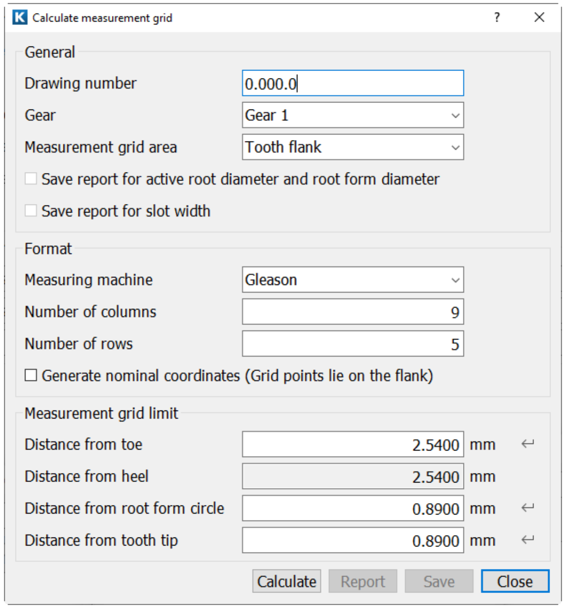
KISSsoft

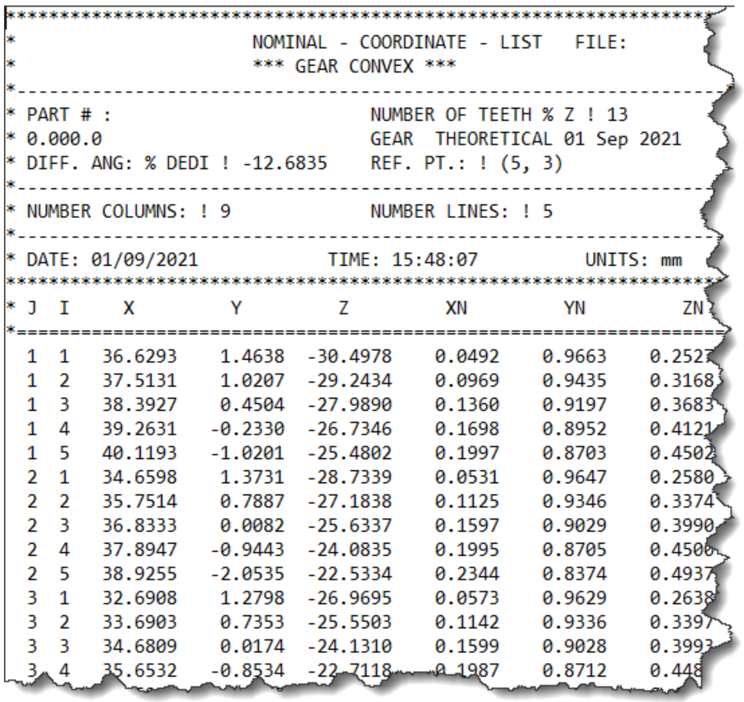
Spectra of meshing



For topological measurements of gear flank, the nominal data are calculated in KISSsoft ‘Calculation – Measurement grid export’.

The data are provided in the definition and format of Klingelnberg and Gleason measuring machines.





NOMINAL - COORDINATE - LIST FILE:
*** GEAR CONVEX ***

PART # : 0.000.0 — NUMBER OF TEETH % Z ! 13

GEAR THEORETICAL 01 Sep 2021

DIFF. ANG: % DEDI ! -12.6835 — REF. PT.: ! (5, 3)

NUMBER COLUMNS: ! 9 — NUMBER LINES: ! 5

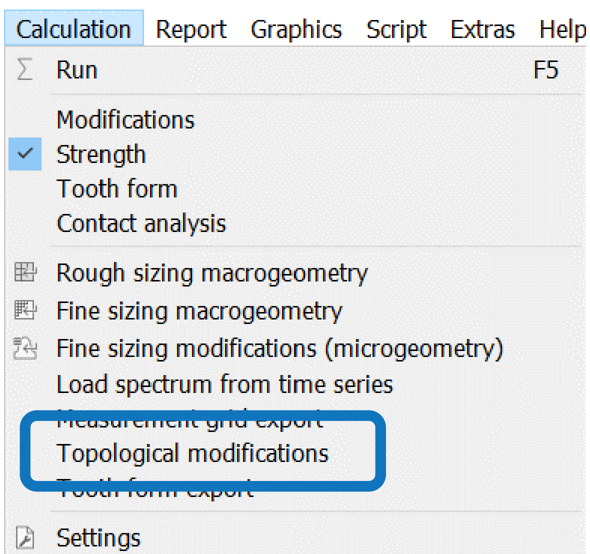
DATE: 01/09/2021 — TIME: 15:48:07 — UNITS: mm

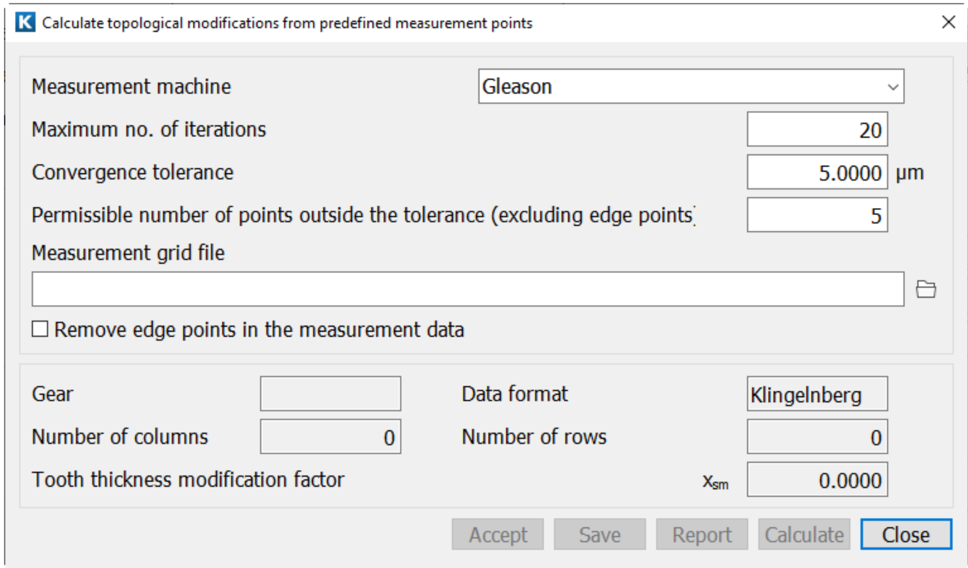
J	I	X	Y	Z	XN	YN	ZN
1	1	36.6293	1.4638	-30.4978	0.0492	0.9663	0.2527
1	2	37.5131	1.0207	-29.2434	0.0969	0.9435	0.3168
1	3	38.3927	0.4504	-27.9890	0.1360	0.9197	0.3683
1	4	39.2631	-0.2330	-26.7346	0.1698	0.8952	0.4121
1	5	40.1193	-1.0201	-25.4802	0.1997	0.8703	0.4502
2	1	34.6598	1.3731	-28.7339	0.0531	0.9647	0.2580
2	2	35.7514	0.7887	-27.1838	0.1125	0.9346	0.3374
2	3	36.8333	0.0082	-25.6337	0.1597	0.9029	0.3990
2	4	37.8947	-0.9443	-24.0835	0.1995	0.8705	0.4500
2	5	38.9255	-2.0535	-22.5334	0.2344	0.8374	0.4937
3	1	32.6908	1.2798	-26.9695	0.0573	0.9629	0.2638
3	2	33.6903	0.7353	-25.5503	0.1142	0.9336	0.3397
3	3	34.6809	0.0174	-24.1310	0.1599	0.9028	0.3993
3	4	35.6532	-0.8534	-22.7118	0.1987	0.8712	0.448

KISSsoft

It is possible to add topological modification as ‘grid data’, in order to adopt the geometry of any bevel gear into the KISSsoft 3D model.

The measurement data can be provided to KISSsoft in the typical format of GLEASON or KLINGENBERG measuring machines.

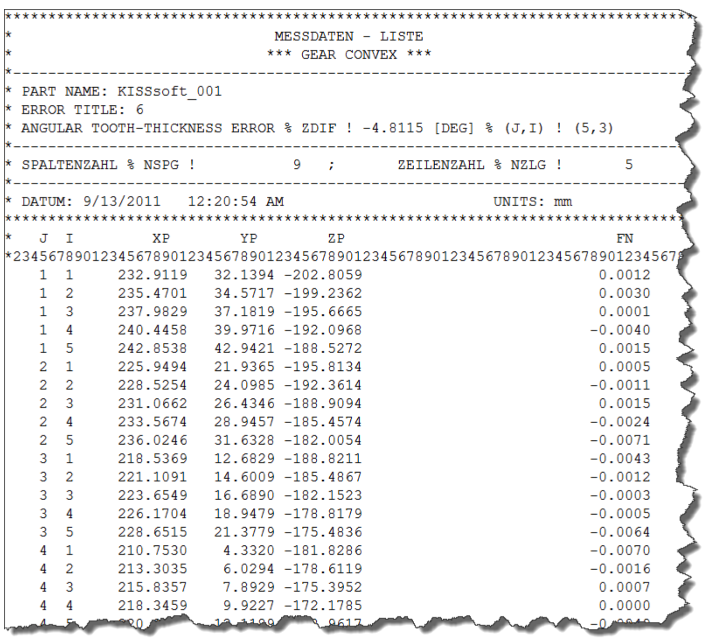




KISSsoft

Using measuring data of an i.e. existing bevel gear, the flank topology of the KISSsoft bevel gear is modified accordingly.

Finally, the modified KISSsoft model contains the same flank geometry as the existing bevel gear and can be exported for milling.



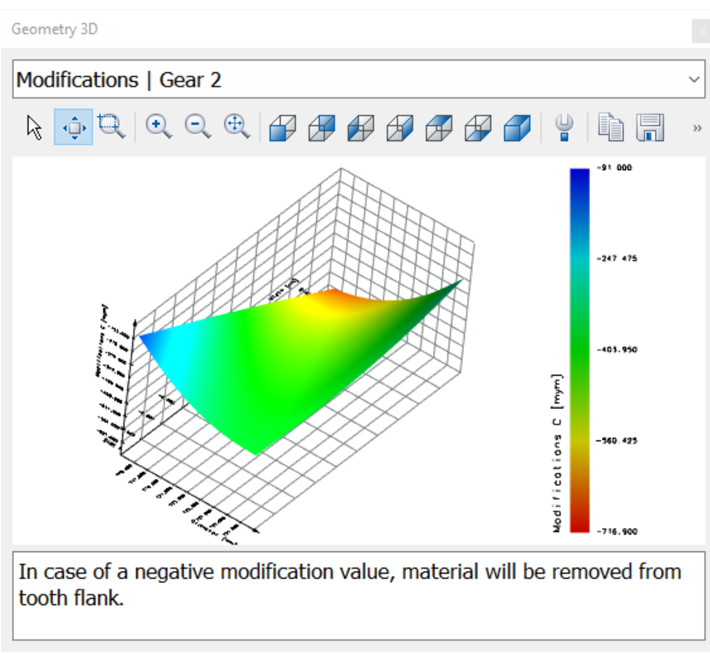
MESSDATEN - LISTE
*** GEAR CONVEX ***

PART NAME: KISSsoft_001
ERROR TITLE: 6
ANGULAR TOOTH-THICKNESS ERROR % ZDIF ! -4.8115 [DEG] % (J,I) ! (5,3)

SPALTENZAHL % NSPG ! 9 ; ZEILENZAHL % NZLG ! 5

DATUM: 9/13/2011 12:20:54 AM UNITS: mm

J	I	XP	YP	ZP	FN
1	1	232.9119	32.1394	-202.8059	0.0012
1	2	235.4701	34.5717	-199.2362	0.0030
1	3	237.9829	37.1819	-195.6665	0.0001
1	4	240.4458	39.9716	-192.0968	-0.0040
1	5	242.8538	42.9421	-188.5272	0.0015
2	1	225.9494	21.9365	-195.8134	0.0005
2	2	228.5254	24.0985	-192.3614	-0.0011
2	3	231.0662	26.4346	-188.9094	0.0015
2	4	233.5674	28.9457	-185.4574	-0.0024
2	5	236.0246	31.6328	-182.0054	-0.0071
3	1	218.5369	12.6829	-188.8211	-0.0043
3	2	221.1091	14.6009	-185.4867	-0.0012
3	3	223.6549	16.6890	-182.1523	-0.0003
3	4	226.1704	18.9479	-178.8179	-0.0005
3	5	228.6515	21.3779	-175.4836	-0.0064
4	1	210.7530	4.3320	-181.8286	-0.0070
4	2	213.3035	6.0294	-178.6119	-0.0016
4	3	215.8357	7.8929	-175.3952	0.0007
4	4	218.3459	9.9227	-172.1785	0.0000

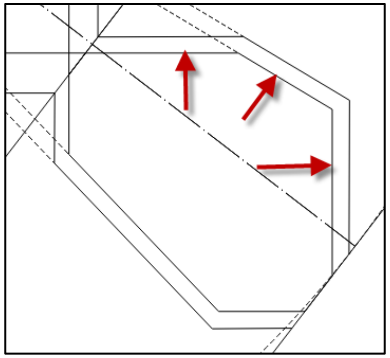


KISSsoft

The forging process has many more degrees of freedom in the design than for conventional, milled bevel gears.

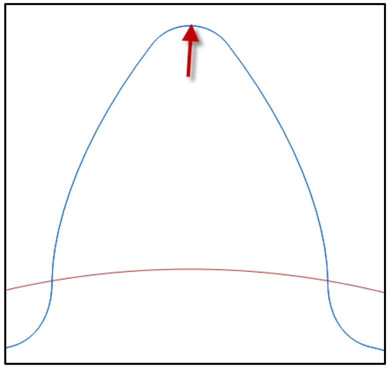
Source: mav.industrie.de

On the other hand, there are a number of additional restrictions, which the engineer has to consider, and also the calculation software has to be able to consider these points.



Webbings, no jamming of gears

KISSsoft calculates the root alteration based on tip alterations of the counter gear, and the required tip clearance.



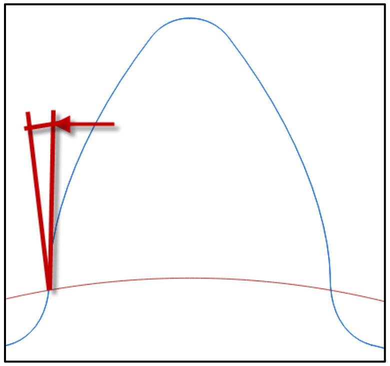
Rounding radius at the tip

in the tab ‘Modification’, the tip rounding radius is entered.

KISSsoft

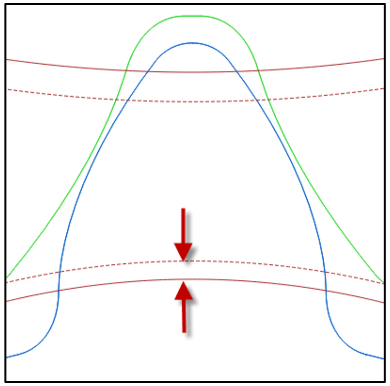
Demoulding of parts is required

the transverse pressure angle at root form circle must be greater than 0 (or any experience value).



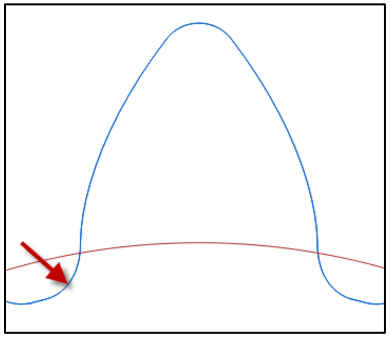
No interference when rolling

the distance of active diameter to form diameter must be greater than 0 (or any experience value).



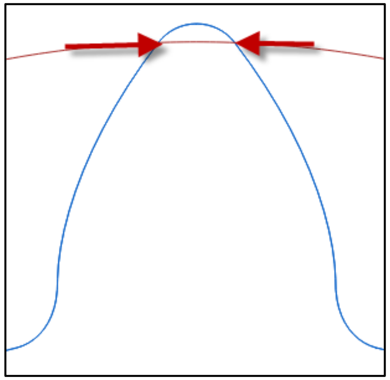
Minimum root radius

a minimum root radius based on a tool radius can be entered.



Tooth thickness at tip form circle

for avoiding through hardening, a minimum thickness is required.



KISSsoft

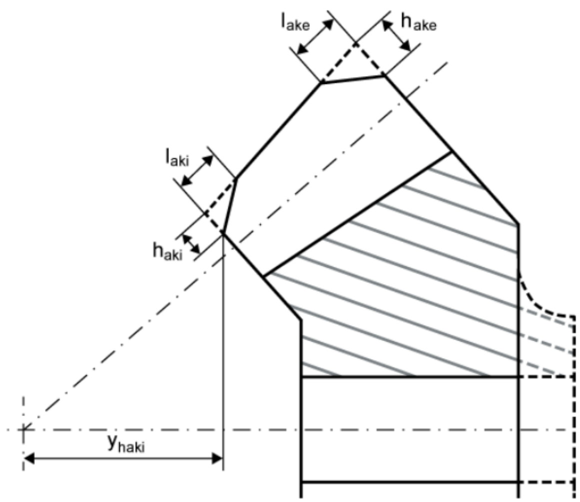
Root alterations are a result of the forging process and lead to higher bending strength.

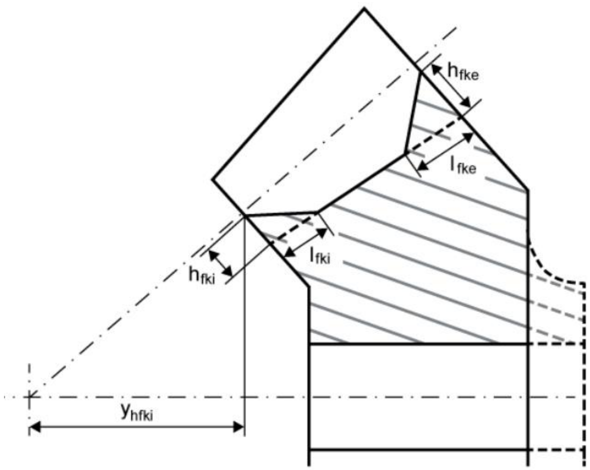
Tip alterations are applied due to space constraints and to avoid jamming with root alterations.

A disadvantage is the reduced contact area and the increased Hertzian pressure.

In KISSsoft, there are two possibilities for sizing the alterations:

- sizing of tip alterations based on space constraints
- sizing of root alterations based on minimum webbing thickness, etc.



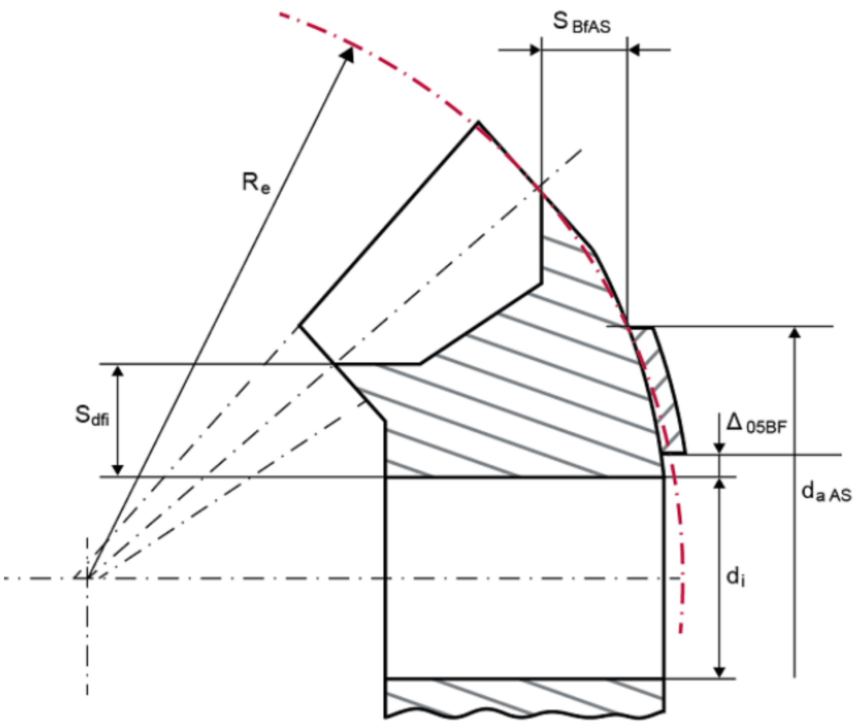


KISSsoft

Sizing of root alterations

In the ‘module specific settings’, the root alteration can be sized according to the given design constraints:

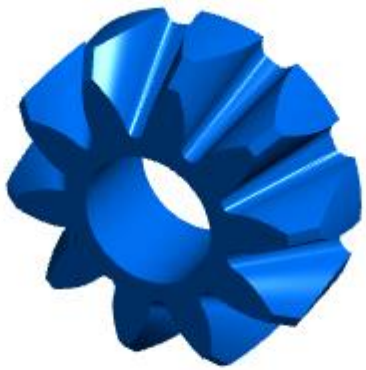
- required distance between bore and webbing at toe
- required distance between bore and washer
- maximum pressure of thrust washer
- required distance between bore and webbing at heel

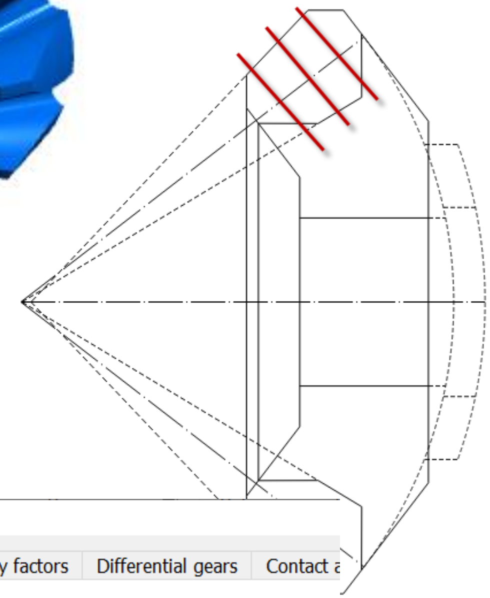


The webbings at the outer side follow to a specific algorithm by considering di, max pressure, daAS, sBfAS.

KISSsoft

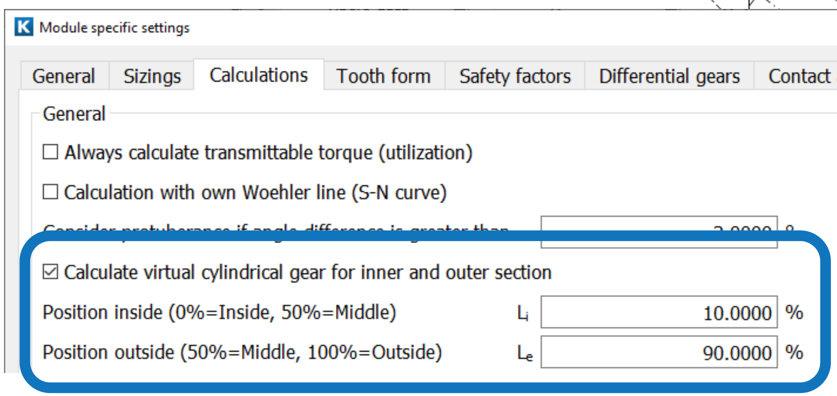
Together with GKN Driveline, the fine sizing was enhanced with special parameters, which are useful to rate the manufacturability of the forging manufacturing process.





The parameters are calculated at 3 positions over the face width: inner, mean and outer side.

- Tip clearance
- maximum root radius
- tooth thickness at tip
- transverse contact ratio, etc.



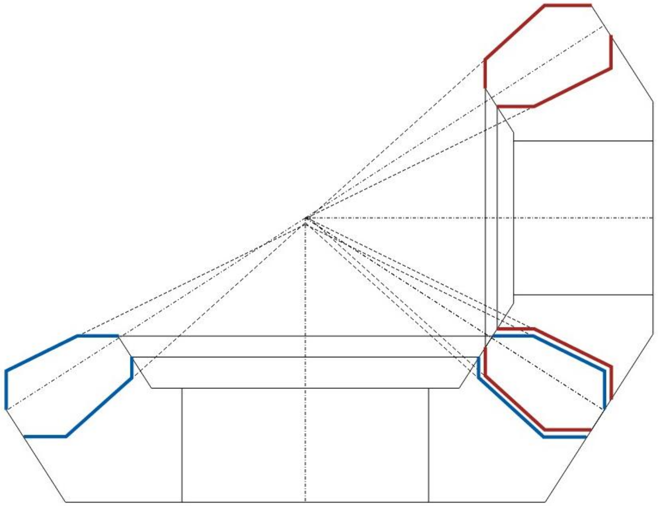
KISSsoft

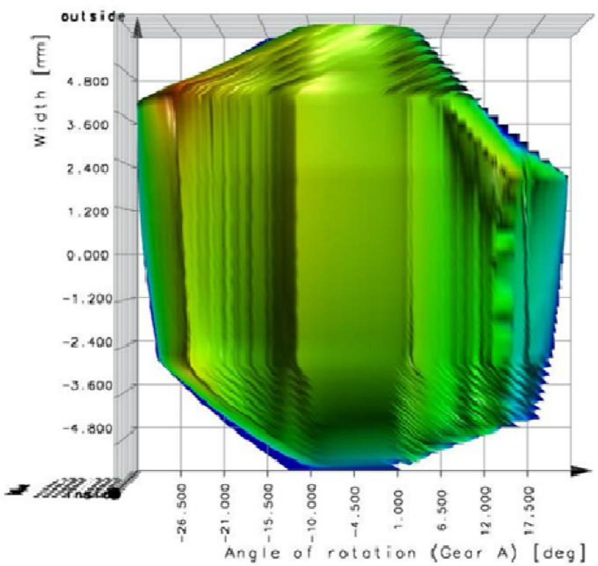
Typically, the differential bevel gears have large blank modifications (webbings).

At outer and inner side, tip and root are alterated. Also, the alterations are depending on the counter gear.

For the strength calculation, the modified blank geometry (tip and root alteration) are considered in the contact analyis.

The contact analysis can be activated within fine sizing.





KISSsoft

Root alterations (webbings)

a) root alteration at heel side

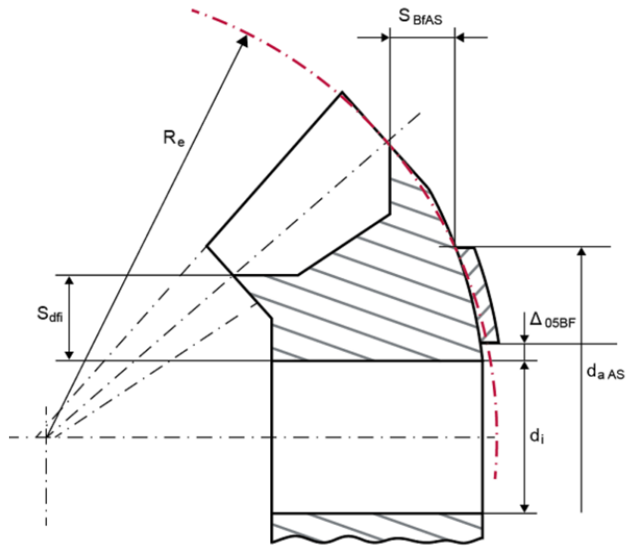
- based on permissible pressure
- based on the bore diameter
- based on required thickness of webbing

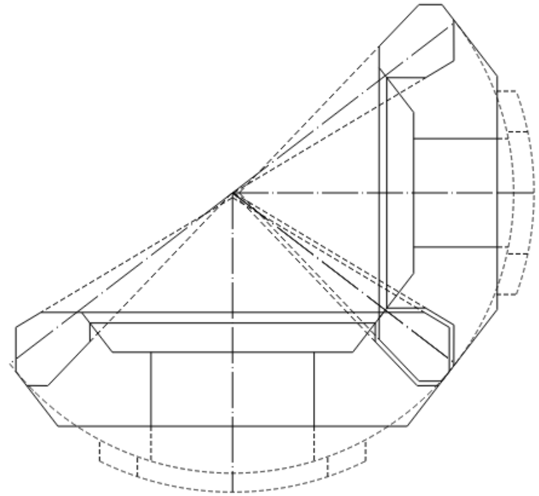
b) root alteration at toe side

- based on the bore diameter
- based on required thickness to bore

Tip alterations

the tip alteration is determined automatically, so that the required tip clearance is achieved.

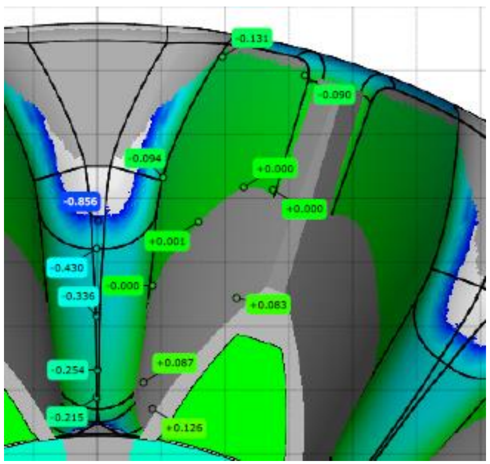


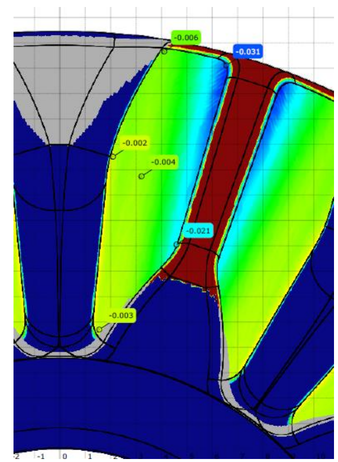


KISSsoft

Forged differential bevel gears have different definition for profile shift and tooth thickness allowance than conventional manufactured bevel gears.

The models can be adjusted, that the value for profile shift and tooth thickness allowance is constant over the face width:



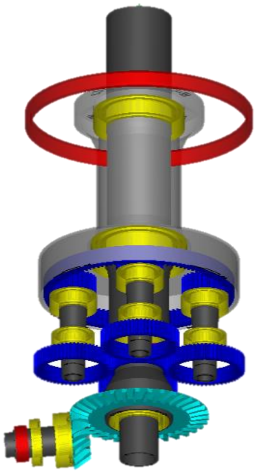


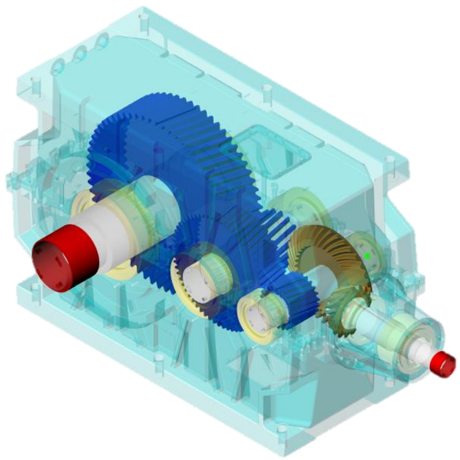
KISSsoft

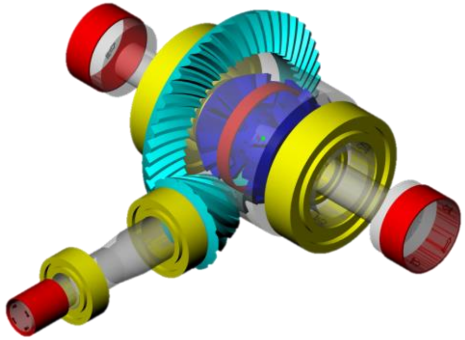
Bevel gears in transmissions

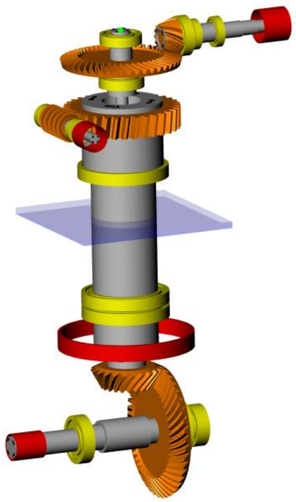
Many applications in KISSsys

- Automotive
- Industrial
- Marine
- Aircraft
- GPK models in KISSsys
- etc.

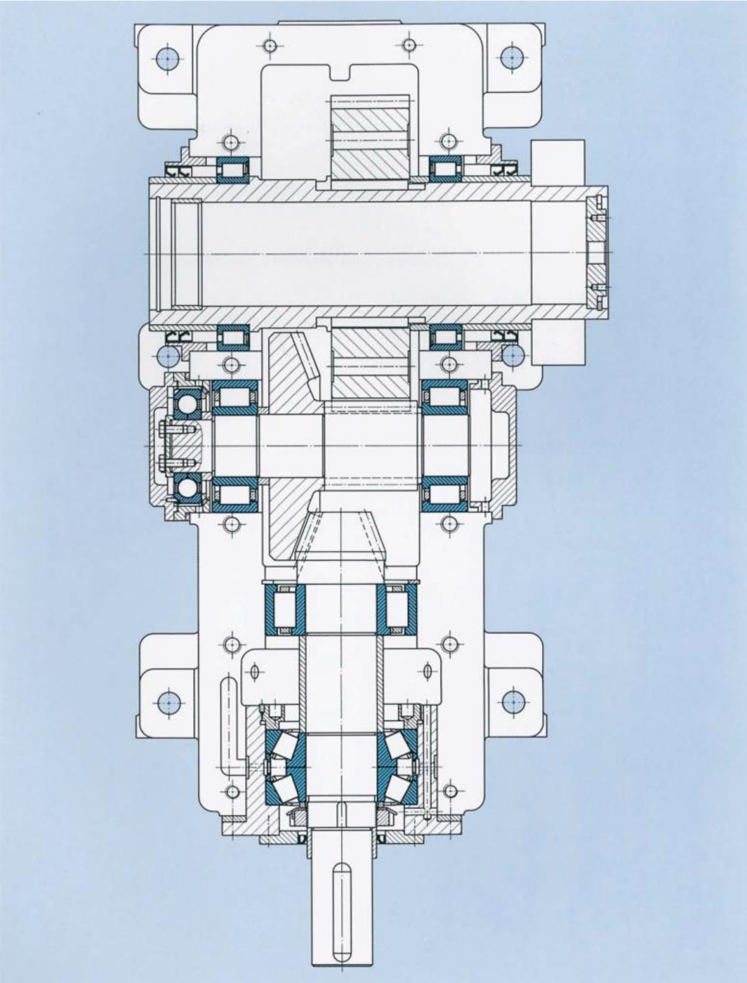


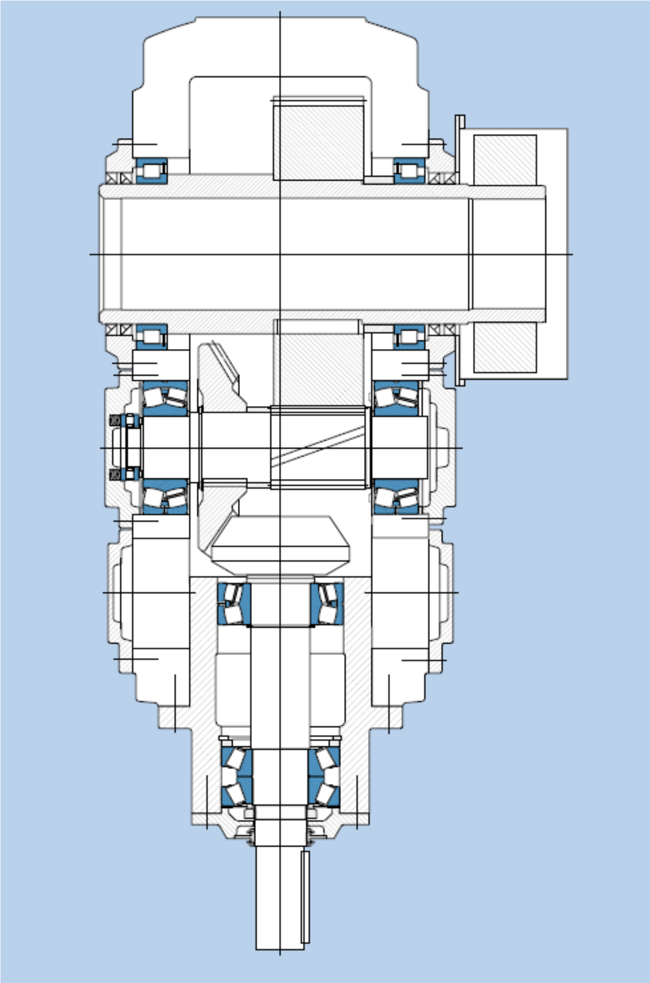




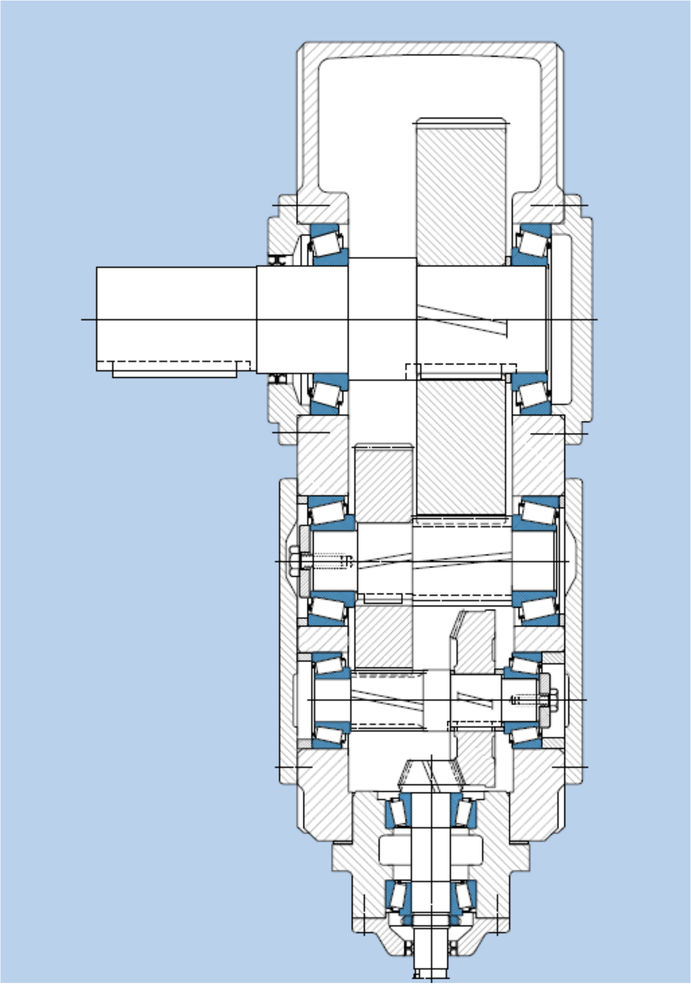


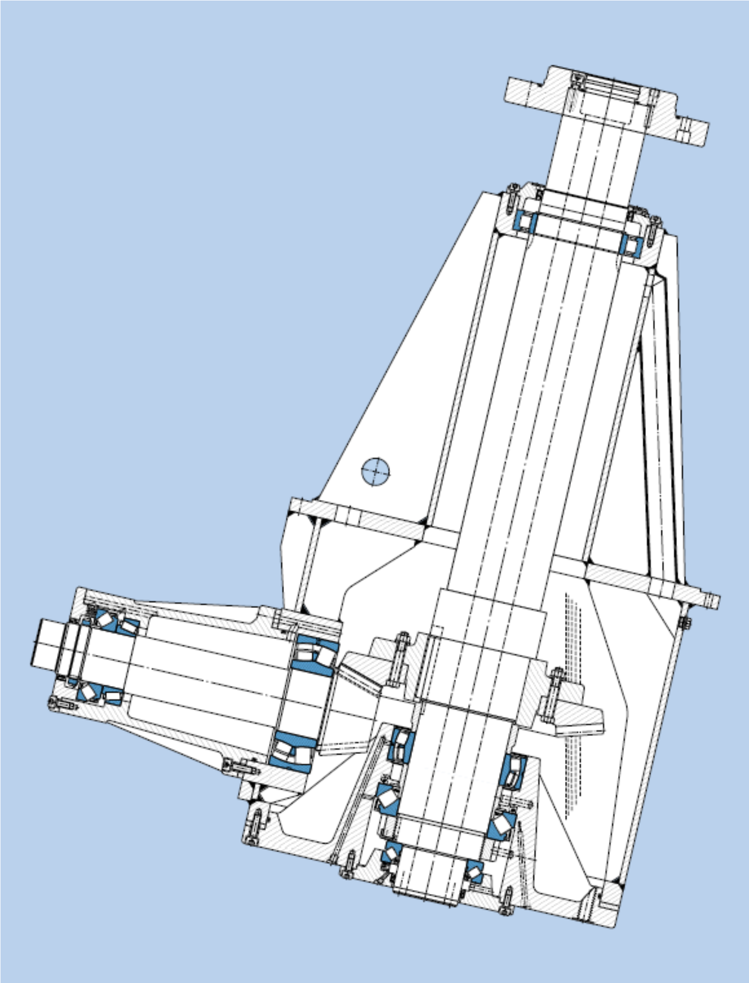
KISSsoft



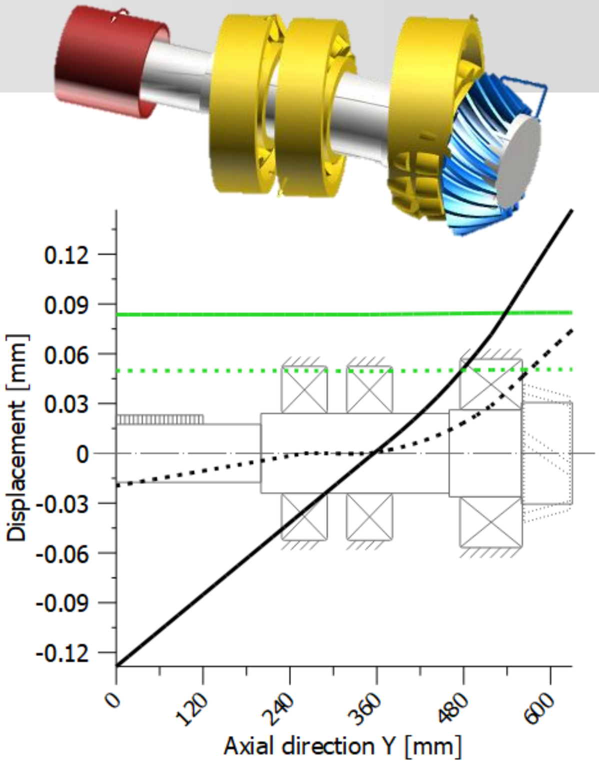


Influence of rolling bearings - arrangements





KISSsoft



Influence from bearing stiffness

- Considers compliance of roller bodies and inner and outer ring due to hertzian contact.
- Activate the bearing stiffness in KISSsys (globally).

CALCULATION METHODS		
Helical gears	ISO 6336:2006 Method B	Disconnected
Bevel gears	Bevel gear ISO 10300:2001, Method B	Disconnected
Worm gears	ISO/TR 14521:2010	Disconnected
Crossed helical gears	Cross helical gear according ISO 6336:2006 and G.Niemann, Method B/C	Disconnected
Face gears	Method ISO 6336:2006-B/ Literature	Disconnected
Shafts	DIN 743:2012	Disconnected
Bearings	Rolling bearing service life from inner geometry (ISO/TS 16281)	Connected
Modified rathing life according ISO 281	No	Disconnected

KISSsoft

Influence from bearing temperatures

- calculates the reduction of bearing clearance
- the inner ring has the same temperature as the shaft
- the outer ring has the same temperature as the casing
- the resulting operating bearing clearance is shown in the report «rolling bearing»

Operating bearing clearance (Results)					
Tolerance field	[-]	Mean value			
Press fit, shaft internal ring					
Shaft speed	[ni]		1700.000	1/min	
Shaft temperature	[Ts]		80.0	°C	
Diameter	[dsoT]		120.072	mm	
Diameter	[dbiT]		120.073	mm	
Interference	[U wi]		0.000	µm	
Embedding	[si]		0.000	µm	
Hertzian pressure	[p_i]		0.000	N/m²	
Effective interference (80.0 (°C))	[U wi_eff]		0.000	µm	
Bearing clearance change	[ΔPdi]		-109.035	µm	
Press fit, hub external ring					
Hub speed	[no]		0.000	1/min	
Hub temperature	[Tn]		60.0	°C	
Diameter	[DboT]		260.102	mm	
Diameter	[DhiT]		260.110	mm	
Interference	[U wo]		0.000	µm	
Embedding	[so]		0.000	µm	
Hertzian pressure	[p_o]		0.000	N/m²	
Effective interference (60.0 (°C))	[U wo_eff]		0.000	µm	
Bearing clearance change	[ΔPdo]		104.049	µm	
Rolling body temperature	[Tw]		70.0	°C	
Rolling body expansion	[ΔD w]		20.125	µm	
Total bearing clearance change	[ΔPd]		-45.235	µm	(ΔPdi + ΔPdo - 2 * ΔD w)
Operating bearing clearance	[P_d]		12.265	µm	(Pd0 + ΔPd)

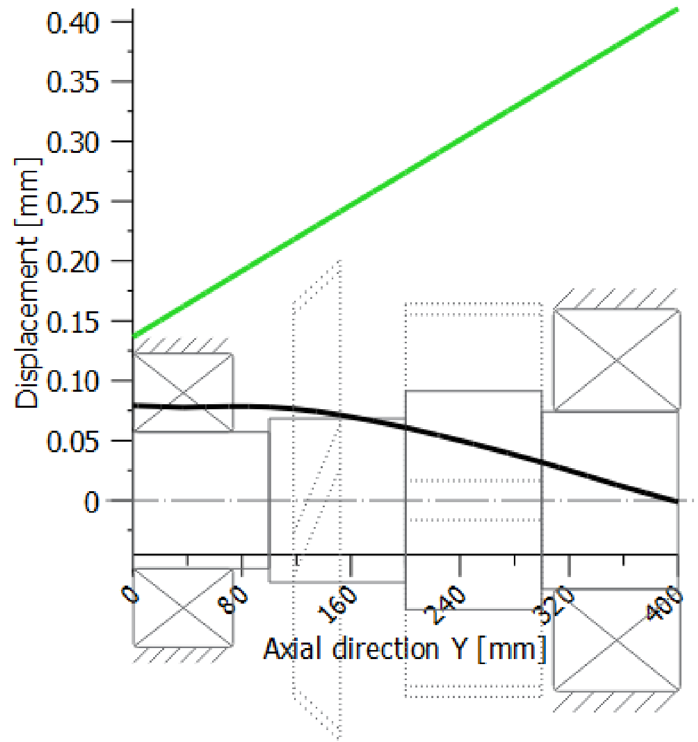
Define the shaft and housing temperatures in KISSsoft (individually) or in KISSsys (globally)

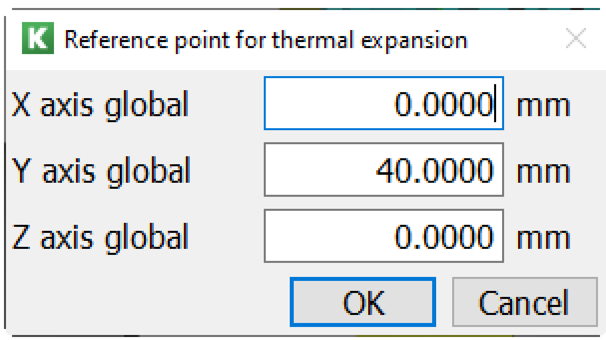
SHAFT SETTINGS			
Reference temperature [°C]		20	Disconnected
Housing temperature [°C]		60	Connected
Shaft(s) temperature [°C]		80	Connected
Gears	Consider gears as mass and as stiffness		Disconnected

KISSsoft

Influence from shaft temperatures

- calculates the thermal elongation of the shaft
- one temperature per shaft possible
- a thermal reference point can be defined, which results in an axial offset of the shaft compared to the casing





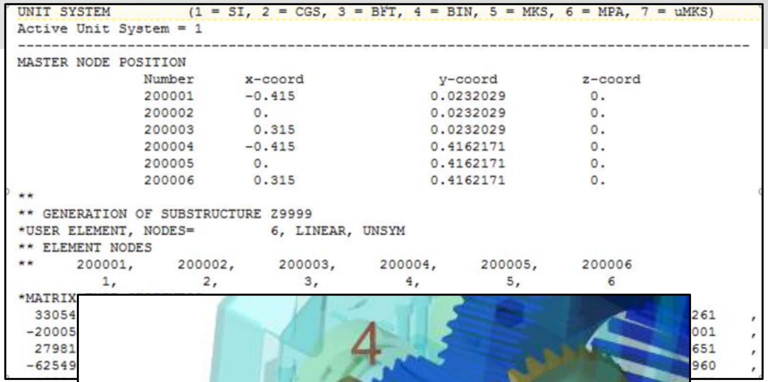
KISSsoft

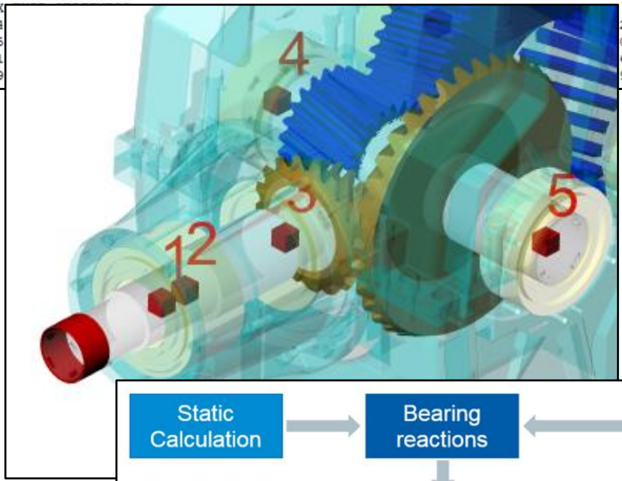
Contact analysis considering housing stiffness from ANSYS, ABAQUS or NASTRAN

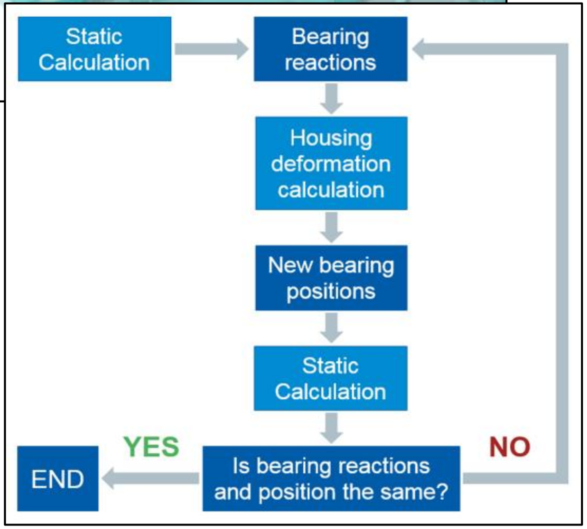
The stiffness matrix is created in FE and imported in KISSsys.

The nodes are positioned at the bearing centers.

The bearing position displacements are calculated iteratively with the shaft calculation.


```
UNIT SYSTEM          (1 = SI, 2 = CGS, 3 = BFT, 4 = BIN, 5 = MKS, 6 = MPA, 7 = uMKS)
Active Unit System = 1
--------------------------------------------------------------------------------------
MASTER NODE POSITION
               Number      x-coord              y-coord          z-coord
               200001      -0.415               0.0232029        0.
               200002       0.                  0.0232029        0.
               200003       0.315               0.0232029        0.
               200004      -0.415               0.4162171        0.
               200005       0.                  0.4162171        0.
               200006       0.315               0.4162171        0.
**
** GENERATION OF SUBSTRUCTURE Z9999
*USER ELEMENT, NODES=         6, LINEAR, UNSYM
** ELEMENT NODES
**      200001,     200002,     200003,     200004,     200005,     200006
           1,          2,          3,          4,          5,          6
*MATRIX
 33054                                                                   261   ,
 -20005                                                                  001   ,
 27981                                                                   651   ,
 -62549                                                                  960   ,
```

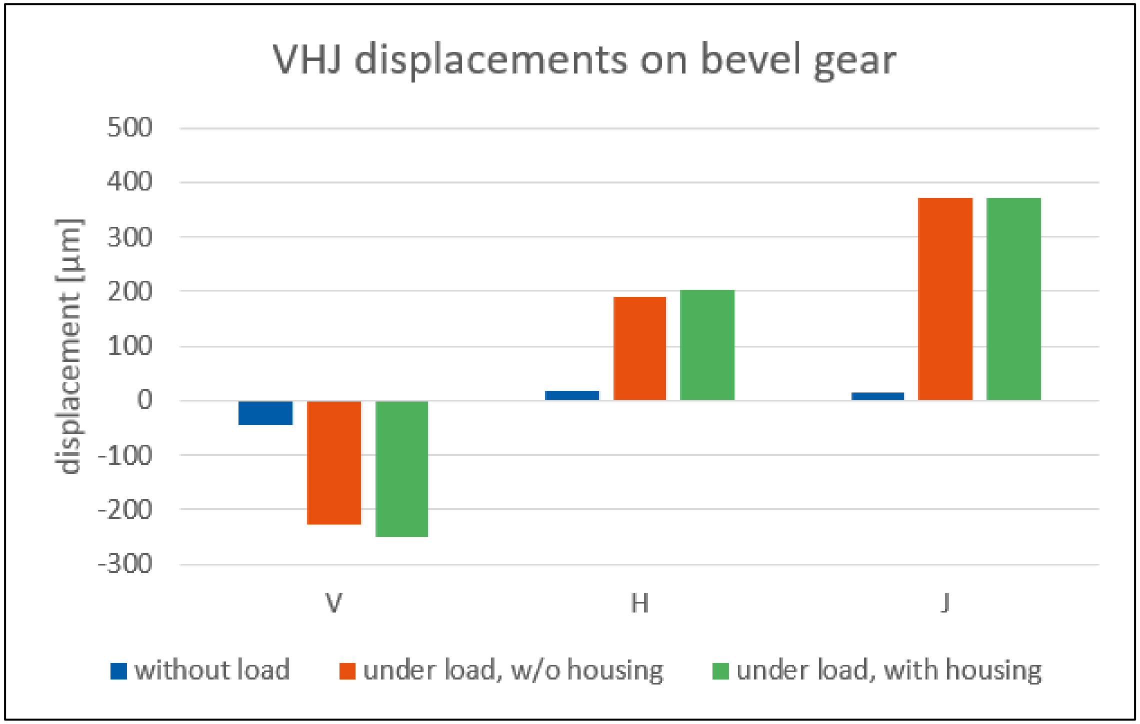




KISSsoft

Investigations on a industrial gearbox showed, that the influence of housing stiffness was rather small.

So, it is not a general need to apply the housing deformation. However, for e.g. vehicle transmissions the housing compliance may be considered.



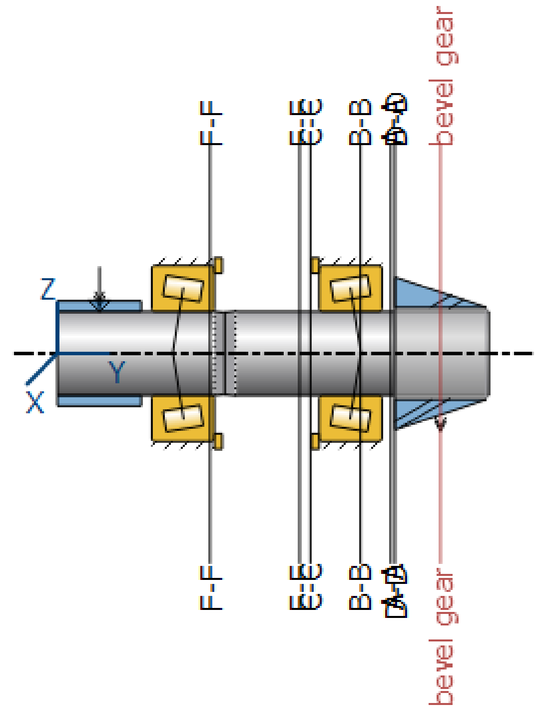
KISSsoft

In the shaft calculation, the displacement of the shaft is shown, what is helpful for understanding the deformation.

With the documentation point, the displacement values at center of bevel gear can be verified.

Elements-editor

Label		bevel gear				
Comment						
Position on shaft	y				91.5000	mm
Position in global system	Y				91.5000	mm
Equivalent stress	σ_V				48.1420	N/mm²
		X	Y	Z	R_{xz}	
Displacement	u	0.0057	-0.0240	0.0264	0.0270	mm
Rotation	r	1.1514	-2.8892	-0.3978	1.2181	mrad
Force	F	-0.3695	-1.1635	1.7088	1.7483	kN
Torque	M	9.2391	-25.0000	-15.0182	17.6325	Nm



KISSsoft

Template «BevelDisplacement»

- shows the displacements of pinion and wheel at middle of face width for each force element and cumulated.
- shows the calculation of EPG and Sigma for each shaft and cumulated.
  → these values are to be used for the LTCA in GEMS

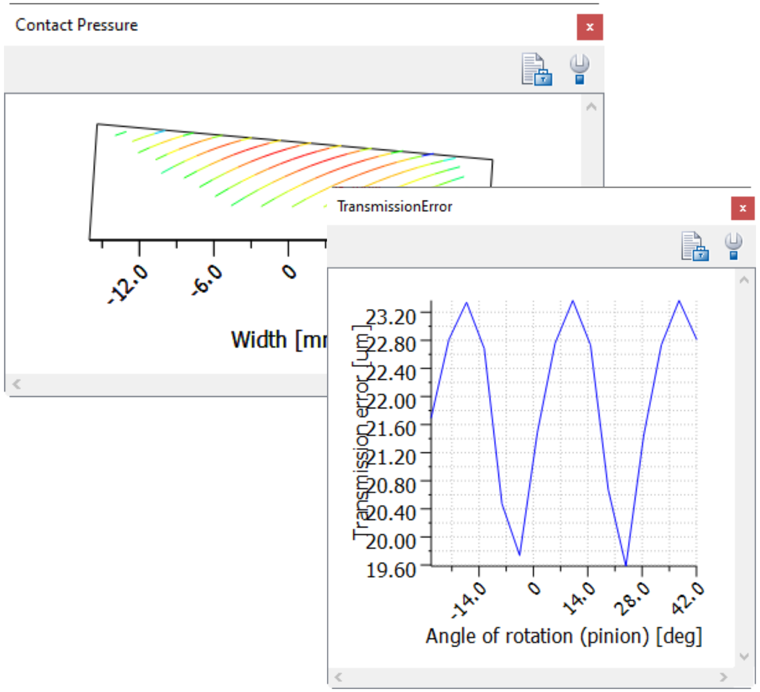
SETTINGS				
Presentation	Draw deflection lines	no		
	Deflection scale	1000		Setup model
	Type	Gleason (EPG)		Calculate
Load on	Coast Side			Export deflection
RESULTS				
Shaft results at middle of facewidth	x	y	z	
Pinion displacement [mm]	0.051002	0.0013805	-0.064243	
Pinion rotation [deg]	-0.020326	-0.12699	-0.0183	
Wheel displacement [mm]	-0.032505	0.22956	0.06791	
Wheel rotation [deg]	-0.0058792	-0.00084374	-0.0096932	
Bevel gear displacements	**E [mm]**	**P [mm]**	**G [mm]**	**S [deg]**
Total	**0.13215**	**-0.033886**	**0.28057**	**-0.0086067**
Pinion	0.064243	-0.0013805	0.051002	-0.0183
Wheel	0.06791	-0.032505	0.22956	0.0096932

KISSsoft

Template «GEMS Interface»

- load cases are defined by the user, as e.g. «Drive 100%», «Coast 100%», etc.
- XML Exchange for transfer of geometry, load & misalignment data to GEMS LTCA, and results from GEMS LTCA into KISSsys
- Results are shown in KISSsys, such as contact pressure, stiffness, transmission error, root stress, etc.

LOAD CASES			
	Definition Dialog	Read from File	Show
XML EXCHANGE			
	Import Data	Export Data	Import Results
RESULTS	Type	Member	Load Case
Generate Graphic	Contact pressure	Wheel	Drive 100%

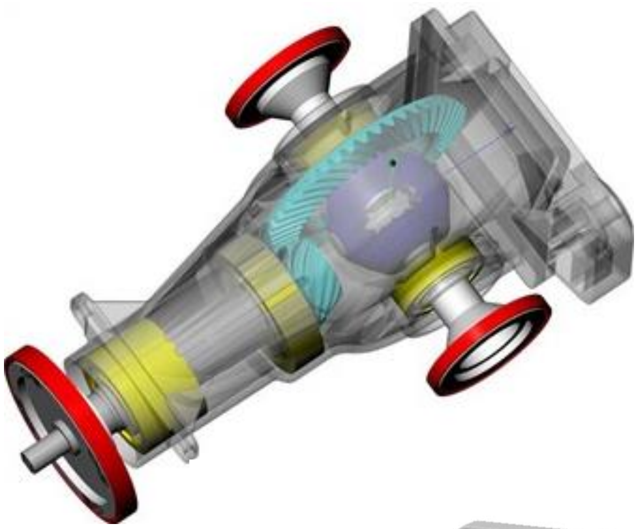


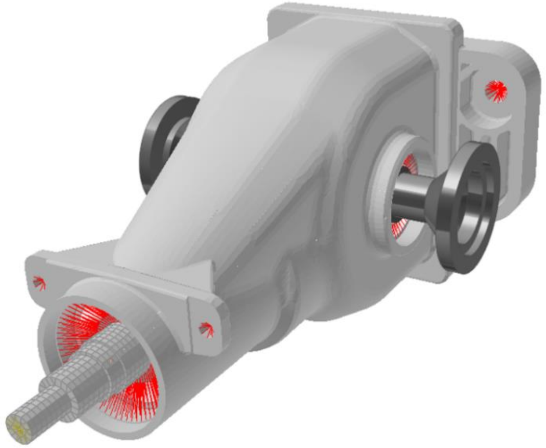
KISSsoft

For investigation of the emmitted noise of a rear axle, several steps are needed.

A) In KISSsys, the static dimensioning of the drivetrain of the rear axle is done.

B) The model of the rear axle is created within the MBS software (e.g. RecurDyn).

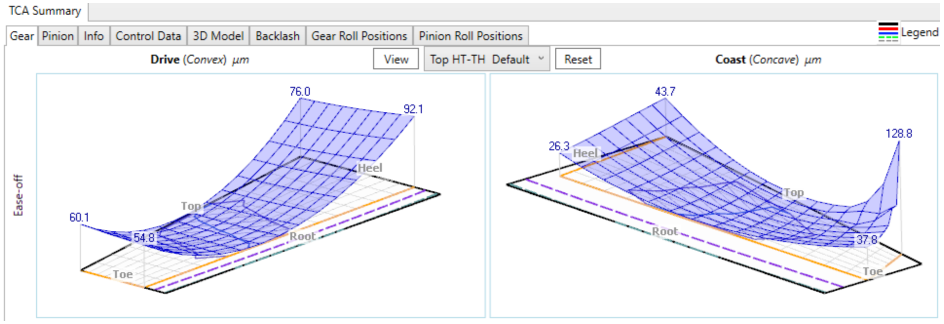




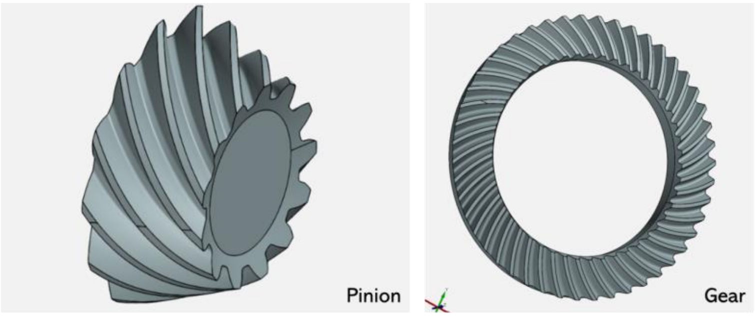
KISSsoft

NVH analysis of a rear axle

C) The bevel or hypoid gear pair is developed and optimized within GEMS.



D) The exact 3D models are exported from GEMS and imported in the MBS software.

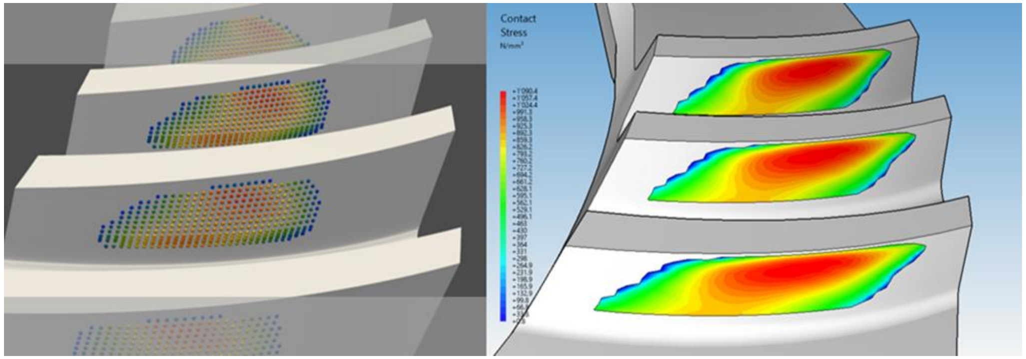


E) The gear contact forces are calculated in the MBS Softwarê.

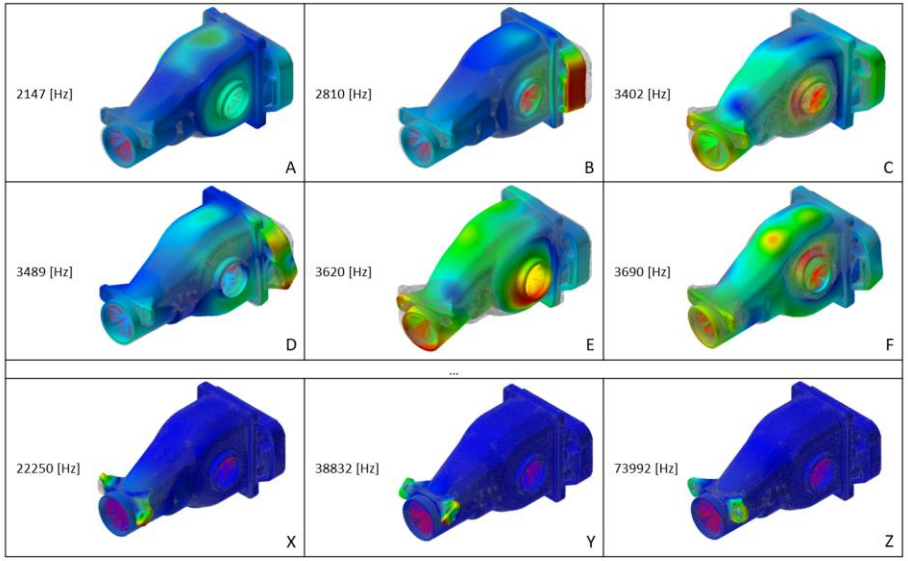
$$f_n = k\delta^{m_1} + c\frac{\dot{\delta}}{|\dot{\delta}|}\left|\dot{\delta}\right|^{m_2}\delta^{m_3}$$

KISSsoft

F) The contact patterns are compared between the MBS software and GEMS, for validation purpose.



G) The structural modes are calculated in the MBS software.



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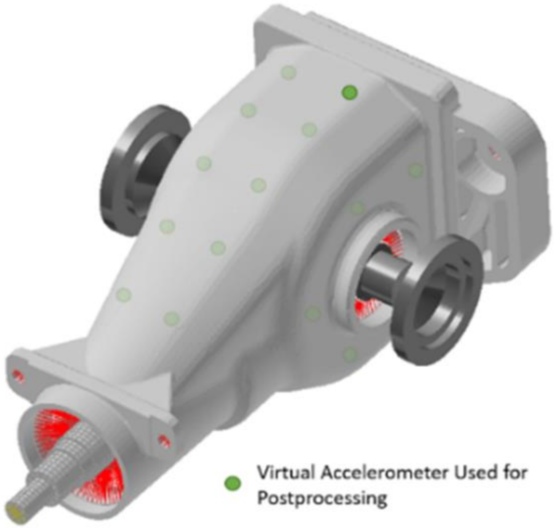
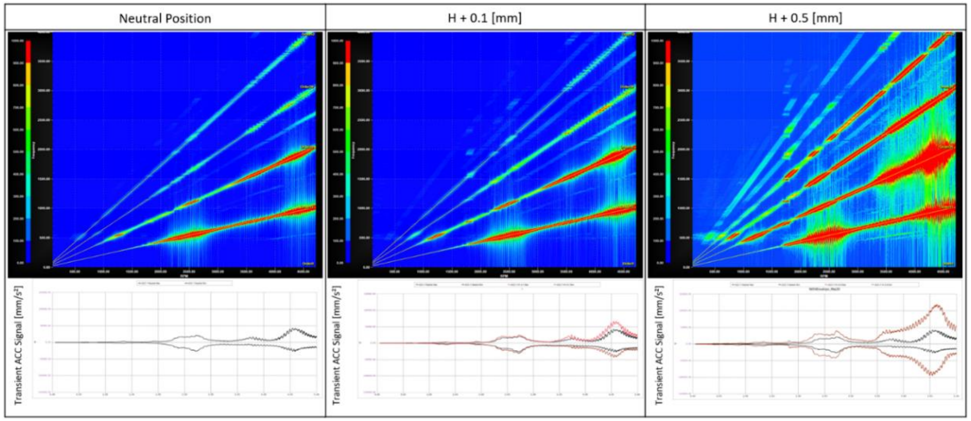
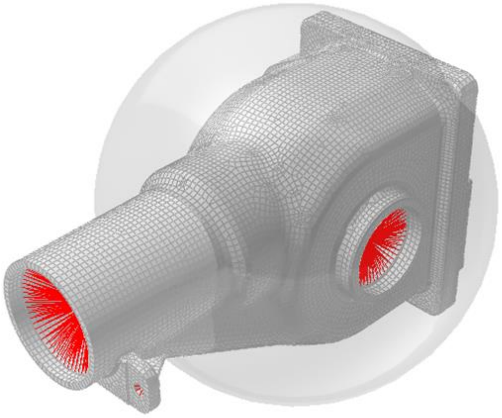
H) The evaluation of the noise emission is done using virtual accelerometers.

I) A speedup is simulated and the pinion mounting distance is variied. The emmitted radiated power is displayed in campbell diagrams and with spherical radiators.







Recommended literature

ISO 23509: Bevel and hypoid gear geometry

ISO 10300: Calculation of load capacity of bevel gears

ISO/DTR 19041: ISO rating system for bevel and hypoid gears — Sample calculations

ISO/TR 10064-6: Code of inspection practice -Part 6: Bevel gear measurement methods

ISO/TR 22849: Design Recommendations for Bevel Gears

ISO/TR 13989: Calculation of scuffing load capacity of cylindrical, bevel and hypoid gears

Klingelnberg: Bevel gears, Edition 2016

Gleason: Gear Encyclopedia, Gleason, Rochester NY, 2008

Efficiency: Untersuchungen zum Wirkungsgrad von Kegelrad und Hypoidgetrieben, Wech, 1987

Scuffing: Zur Fresstragfähigkeit von Kegelrad- und Hypoidgetrieben, Markus Klein, 2012

Flank fracture: Tooth flank fracture – basic principles.., I. Boiadjiev, 2014

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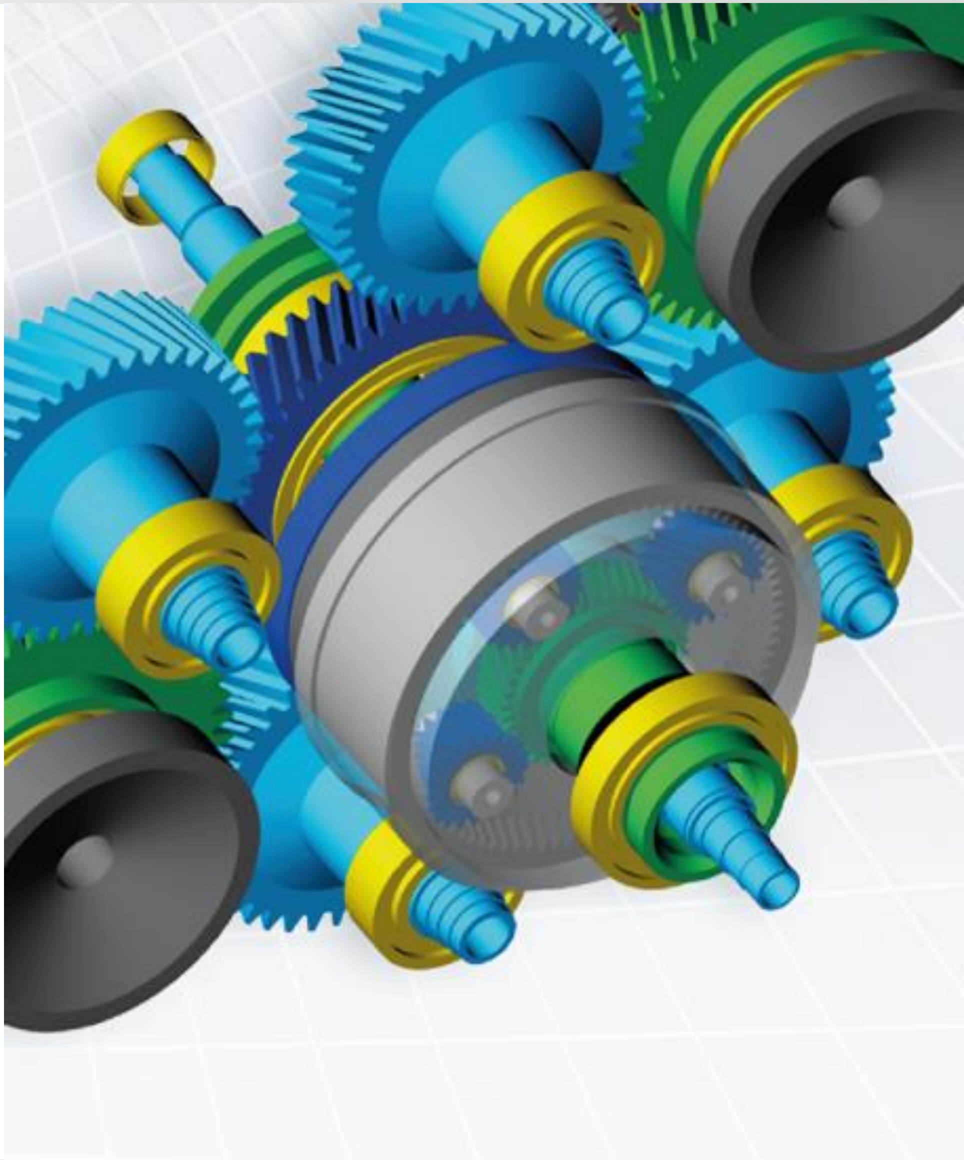
Thank you for your attention!

Sharing Knowledge

KISSsoft AG, A Gleason Company
Rosengartenstrasse 4, 8608 Bubikon, Switzerland
T. +41 55 254 20 50, info@KISSsoft.AG, www.KISSsoft.AG



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