MESYS AG
Technoparkstrasse 1
CH–8005 Zürich
info@mesys.ch
T: +41 44 455 68 00

Engineering Consulting Software AG

MESYS Tutorial: Cylindrical Gear Pair 01

1 Introduction

1.1 Use case

In this tutorial, a helical gear stage is designed. It is used as an output stage in an industrial gearbox. The strength rating with load spectrum is performed and the gearing is optimized to achieve the required safety factors.

1.2 Objective

Tutorial	Property
Suitable for	Users familiar with the tutorial *Single Cylindrical Gear 01*.
Prerequisites	MESYS license (test license, commercial license).
Learning objectives	Use inputs for factors and materials. Understand cylindrical gear pair geometry sizing. Explore output options.
MESYS file	MESYS-Tutorial-Cyl_gearpair_01-ww-v2500.mCGP.

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1.4 Task

Gearbox	Property
Application	Bevel-helical gearbox, e.g. for a conveyor belt
Stages	One bevel gear stage, two cylindrical gear stages, u_{tot} = 1:63
Input and output	Input = electric motor, output via elastically mounted roller
Stage under consideration	Output stage, single helical, u_3 = 3.50
Power	53 kW
Output speed	24 rpm
Output torque	23'800 Nm
Load spectrum	20% of the time at 24 rpm and 23'800 Nm 50% of the time at 24 rpm and 19'800 Nm 30% of the time at 20 rpm and 25'000 Nm
Operation	Non-reversing
Required service life	25'000 h
Lubrication	Oil bath, oil ISO VG 460, mineral oil, maximum temperature 90°C
Materials	Case-hardening steel 20MnCr5
	Case-hardened to 62 HRC surface hardness
	Core hardness 30 HRC, material quality MQ
	Steel housing
Basic rack	ISO 53, Profile A, tip alteration coefficient per standard
Normal pressure angle	α_n = 20°
Gear quality	ISO 1328, A = 7, R41
Tooth modifications	None
Tooth flank roughness	R_{zH} = 7 µm for both gears
Tooth root roughness	R_{zF} = 16 µm for both gears
Tooth thickness deviations	c 25 for both gears
Tip circle diameter tolerance	h7
Center distance tolerance	+-0.15 mm
Inner diameter, pinion	0, pinion shaft
Inner diameter, wheel	80.00 mm
Rating	Face load factor $K_{H\beta}$ = 1.50 Required safety factor tooth root S_{Fmin} = 1.50 Required safety factor tooth flank SHmin = 1.20 Required safety factor scuffing S_{Bmin} = S_{Smin} = 2.00

2 Implementation

2.1 Starting MESYS and Settings

Start MESYS by double-clicking the file *MesysCOM64.exe*. The file is located in the installation directory, typically in *C:\Program Files\MESYS 12-2025*. Start the cylindrical gear pair calculation by clicking the icon below.

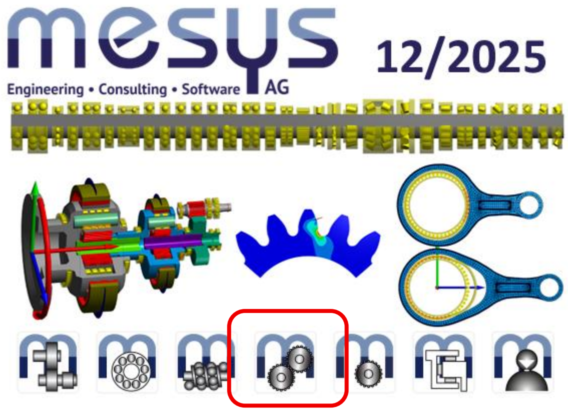


Figure 2-1 Start of cylindrical gear pair calculation in the module selection menu.

Extras/Settings contains relevant predefined settings that are self-explanatory. The selection *Show all messages* should be selected so that, especially at the beginning of using MESYS, errors, warnings, and information are noticed.

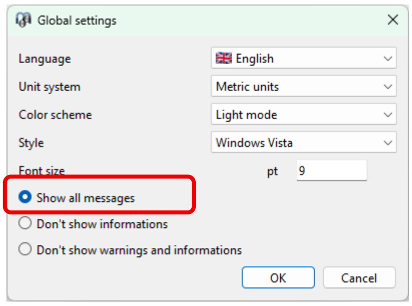


Figure 2-2 *Extras/Settings* with default settings.

The project name and description fields are used to freely annotate the calculation file.

Since the calculation is to be performed with a load spectrum, the corresponding flag Consider load spectrum must be set.

Critical is the selection *Geometry for load capacity calculation = nominal dimension with minimum tooth thickness for YF/YS*. As per ISO 6336-1, section 6.1, the lowest manufacturing profile shift, i.e. the smallest tooth thickness, is to be used for its calculation.

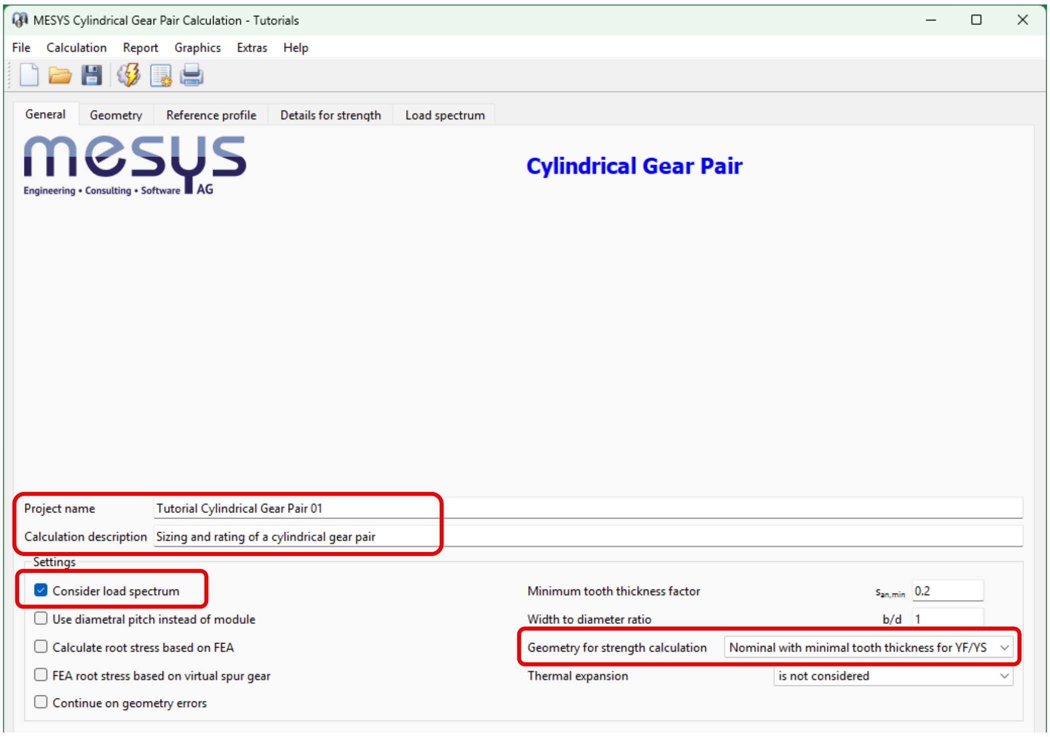


Figure 2-3 Tab General, settings and project description.

2.2 Inputs

The basic rack is selected in the *Basic Rack* tab. It is predefined as *Profile A* and the tip alteration coefficient is applied automatically per standard (flag *Tip Alteration Coefficient* deselect).

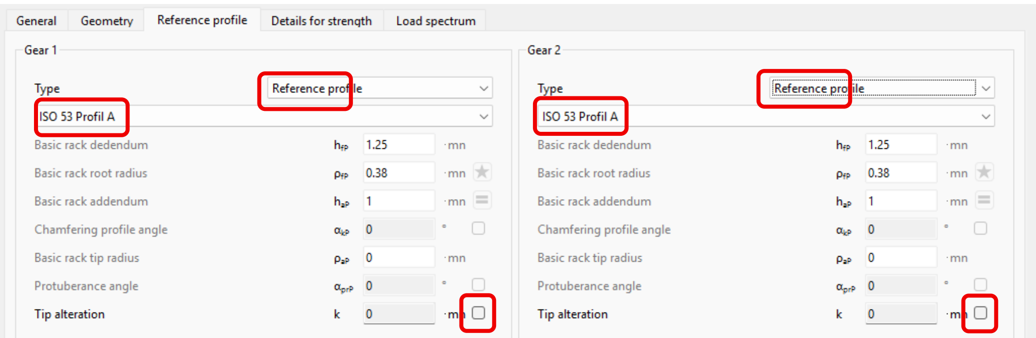


Figure 2-4 Inputs in the *Basic Rack*.

The *Load Capacity Details* tab contains more extensive inputs closely linked to the theory of load capacity verification per ISO 6336. The inputs are explained individually below.

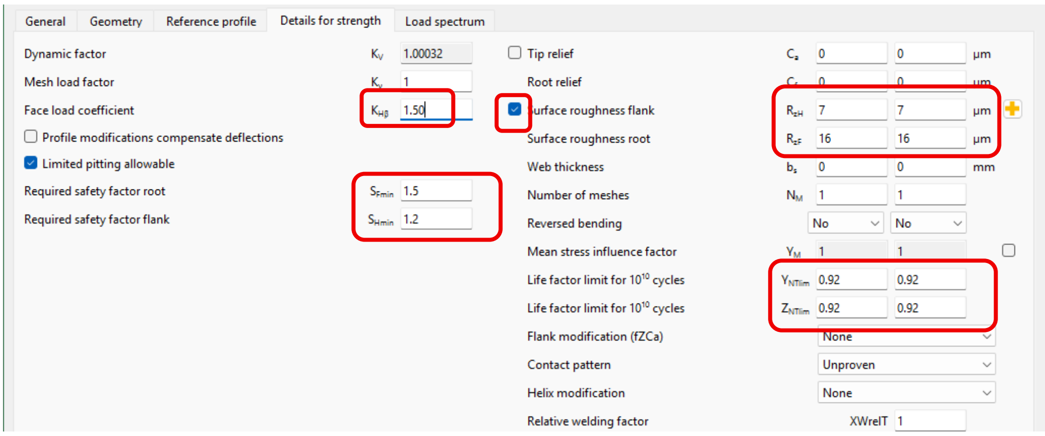


Figure 2-5 Inputs for load capacity calculation in the *Details for strength*. Only deviations from default settings and values are highlighted.

Input	Explanation
Dynamic factor	When flag is not set, calculated per ISO 6336-1, Section 6.
Load sharing factor	Keep at 1.00 as only one load path is considered, see ISO 6336-1, Section 4.2.1.
Face load factor	Enable flag and set to 1.50 per task definition.
Profile modifications compensate for deflections	This flag affects the calculation of the transverse load factor KHα. Since no profile modifications are present, leave the flag disabled.
Limited pitting permitted	This flag affects the S-N curve used to calculate the tooth flank safety factor. Since no pitting is permitted per the task definition, leave the flag disabled.
Required safety factor tooth root	Set to 1.50 per task definition.
Required safety factor tooth flank	Set to 1.20 per task definition.
Required safety factor scuffing	Set to 2.00.
Tip relief	Leave at 0 as none is specified in the task definition.
Root relief	Leave at 0 as none is specified in the task definition.
Tooth flank surface roughness	Enter per task definition.
Tooth root surface roughness	Enter per task definition.
Web width	Since industrial gearboxes typically use solid gears, leave at 0. The input then has no effect.
Number of meshes	This refers to the number of meshes per revolution. Leave at 1, as no multiple meshes are present.
Reverse bending	Since the gearbox is operated non-reversing, select *No*.
Mean stress influence factor	When flag is disabled, is automatically set to 1.00 per the selection Reverse bending *No*, see above
Life factor for 10^{10} load cycles	Set to 0.92, per ISO 9085, Table 6, for mean material quality MQ, for tooth root

Life factor for 10^{10} load cycles	Set to 0.92, per ISO 9085, Table 6, for mean material quality MQ, for tooth flank
Flank modification (f_{ZCa})	Set to *None*, as no flank modifications are present per the task definition.
Contact pattern	Set to *Without verification*, as no verification is performed per the task definition.
Helix modification	Set to *None*, as no modifications are present per the task definition.
Relative welding factor	Leave at 1.00, per ISO/TS 6336-21, Table 4.

Table 2-1 Explanations on the input in tab *Details for strength*.

The load spectrum is now entered in the *Load Spectrum* tab. Using the + button at the bottom right, three rows are added and entered per the task definition. Note that the data in the task definition refers to the output, i.e. gear 2 in the pair. The corresponding selection is in the *Geometry* tab, where *Reference gear = Gear 2* must be selected. The text in the load spectrum changes accordingly to T2 and n2.

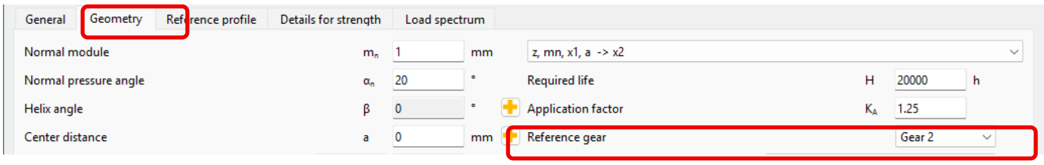


Figure 2-6 Selection *Reference gear = Gear 2* in the tab *Geometry*.

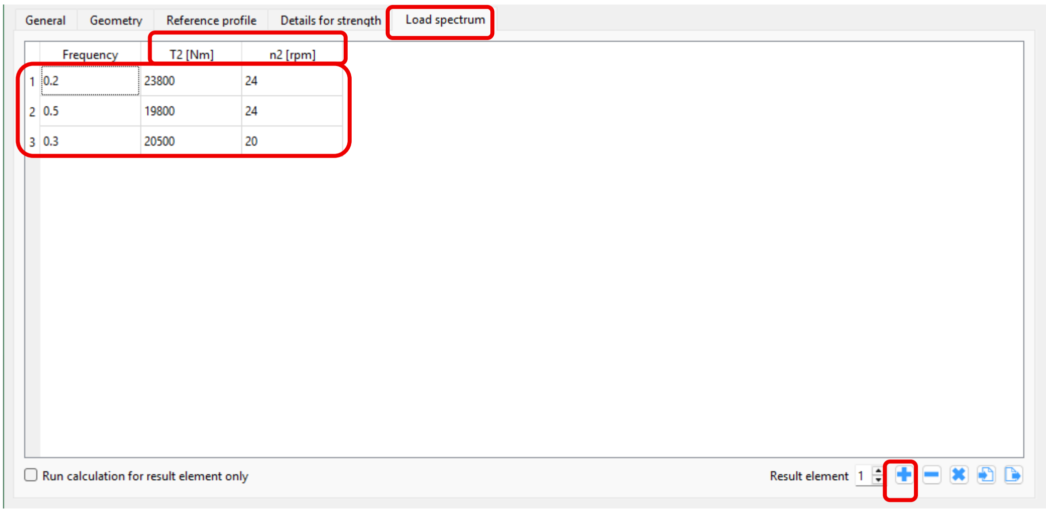


Figure 2-7 Load spectrum input.

2.3 Inputs in the Geometry Tab

The application involves an electric motor drive and output connected to a conveyor belt. Per ISO 6336-1, Tables 4, 5, 6, this results in a recommended application factor per Method B, $K_{A\text{-}B}$ = 1.25. The other specifications are set or selected directly. The reference gear selection was made in the previous step.

When a load spectrum and an application factor K_A > 1.00 are entered, the torques in the load spectrum are multiplied by the application factor K_A. In this tutorial, K_A > 1.00 is used together with the load spectrum. In practice, the load increments are already included in the torque values of the load spectrum; in that case, K_A = 1.00 shall be set.

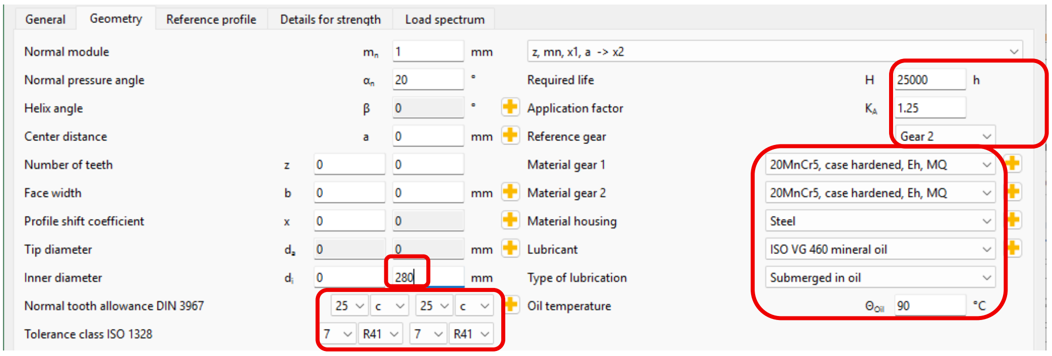


Figure 2-8 Inputs in the Geometry.

The tip circle diameter tolerances and center distance tolerance are entered via the respective + button as follows.

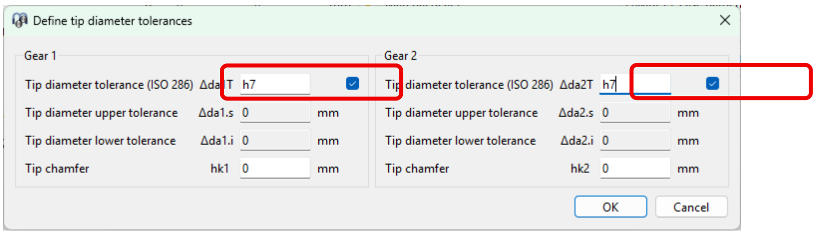


Figure 2-9 Input of tip circle diameter tolerances via the + button at the *Tip circle*.

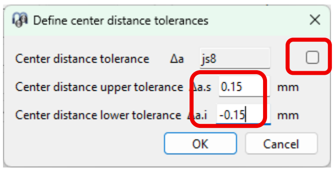


Figure 2-10 Input of center distance tolerance via the + button at the *Center distance*.

2.4 Sizing

The gear geometry is unknown and to be sized, i.e. number of teeth, face width, module, etc. The target gear ratio of the stage is known as $u_3 = 3.50$. Based on experience, the number of teeth of the pinion is chosen as $z_1 = 13$. Lower tooth numbers tend to cause undercut, and the number is a prime. The number of teeth of the wheel follows as $z_2 = z_1 * u_3 = 46$ (rounded up). The helix angle is set to $\beta = 12°$ (for this, first define that the gear is helical and left hand, using "+" button), the normal module $m_n = 22.00$ mm, the face width estimated at 250 mm for the pinion and 240 mm for the wheel, based on similar stages. For the profile shifts, starting values of $x_1 = 0.50$ and $x_2 = -0.50$ are used. The center distance follows.

Accordingly, the sizing method selection *z, mn, x1, x2 -> a* is chosen, i.e. the center distance is calculated from the inputs.

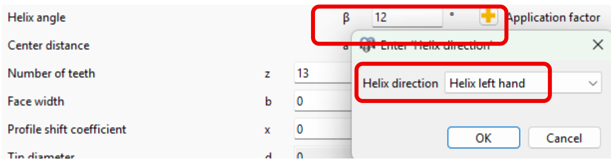


Figure 2-11Definition of gear as left hand helical gear.

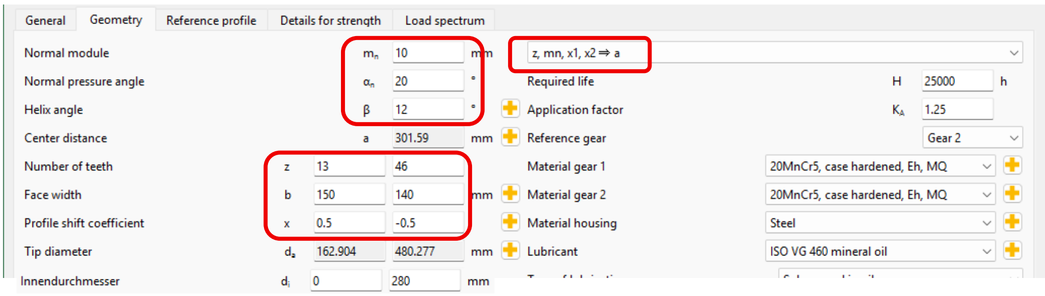


Figure 2-12Selection of sizing method (top right) and input of known parameters.

The calculation is now executed, and the center distance follows.

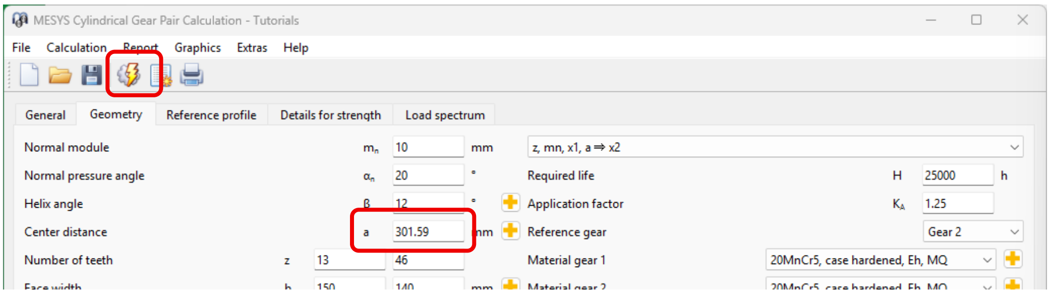
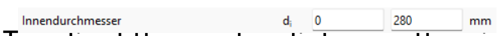


Figure 2-13 Run calculation and resulting center distance.



To adjust the center distance, the sizing method is changed to *z, mn, x1, a -> x2*. I.e. x_1 and a are specified and x_2 is calculated. With a = 301.00, after recalculation x_2 = -0.56. The safety factors for the tooth flank, tooth root and scuffing calculation are displayed in the *Results overview* window.

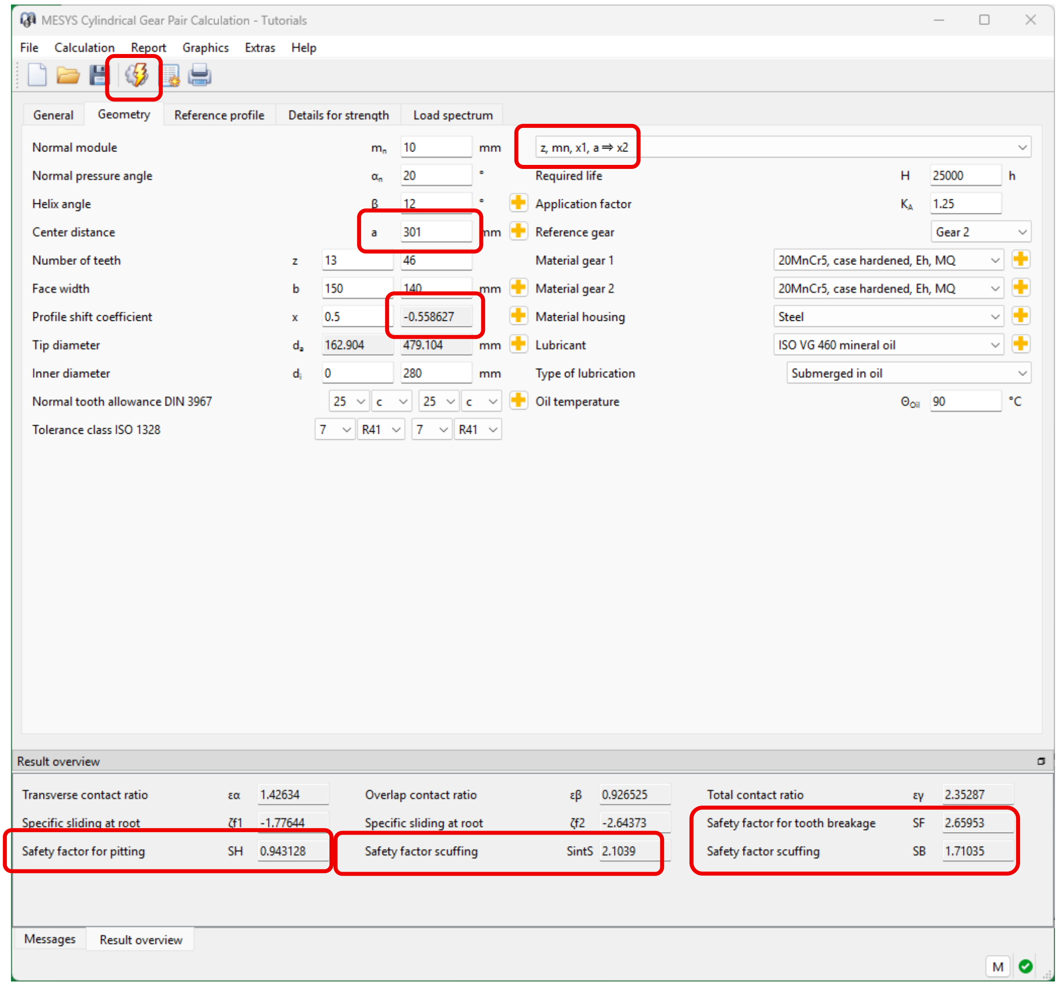
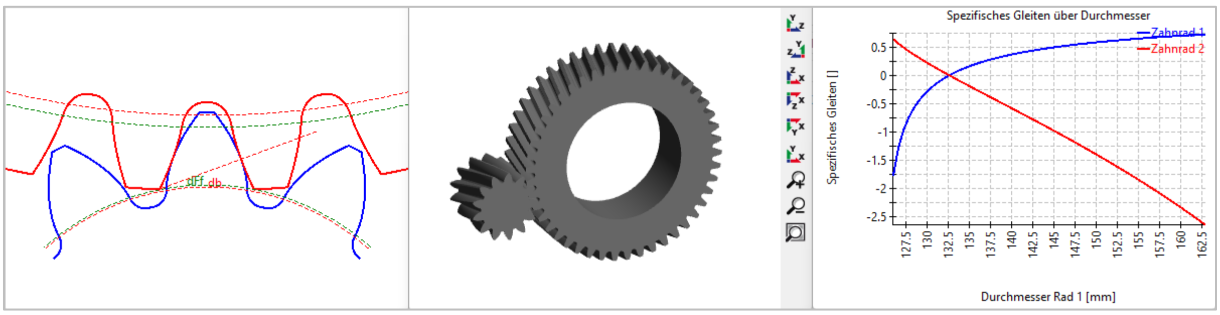


Figure 2-14 Entered center distance a = 301.00 mm, calculated profile shift x_2 = -0.56, safety factors.

2.5 Load capacity verification

The tooth flank safety factor with S_H = 0.94 does not meet the requirements, nor does the scuffing safety factor S_B = 1.71 which is too low. The scuffing safety factor S_{intS} = 2.10 and the tooth root safety factor S_F = 2.66 meet the requirements or exceed them respectively.

Via the menu *Graphics* various graphs can be displayed for visualization of the gear and its properties.



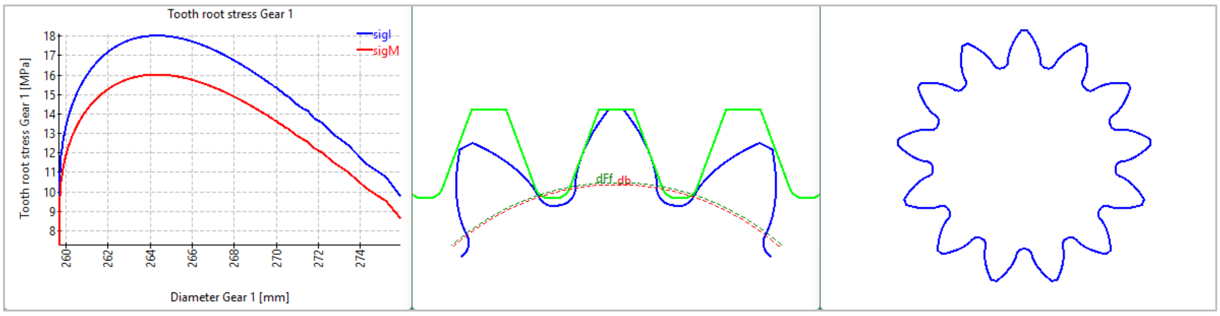
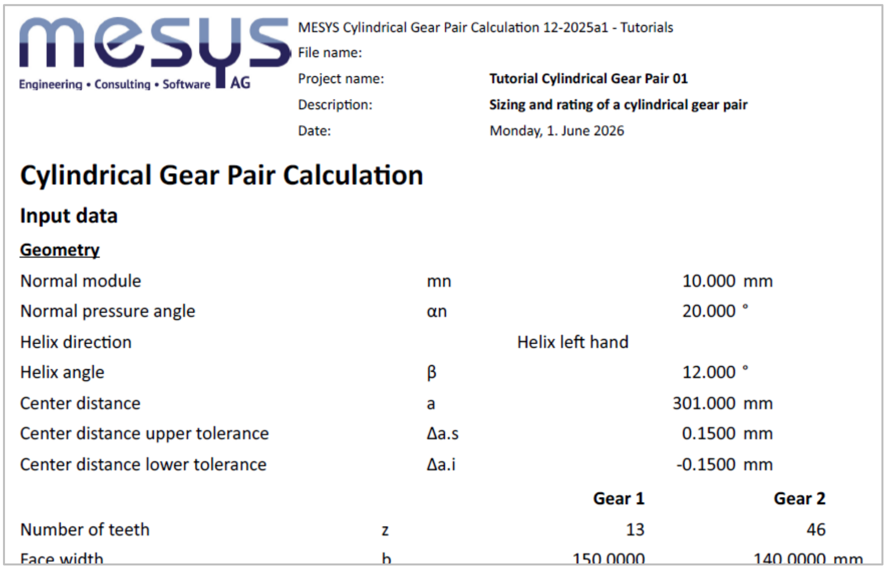


Figure 2-15 Various graphs for the mesh, the gears and their properties.

The detailed calculation report is generated via F6 or the menu *Report/Show report*.



MESYS Cylindrical Gear Pair Calculation 12-2025a1 - Tutorials

File name:	
Project name:	**Tutorial Cylindrical Gear Pair 01**
Description:	**Sizing and rating of a cylindrical gear pair**
Date:	Monday, 1. June 2026

Cylindrical Gear Pair Calculation

Input data

Geometry

Normal module	mn	10.000	mm
Normal pressure angle	αn	20.000	°
Helix direction		Helix left hand	
Helix angle	β	12.000	°
Center distance	a	301.000	mm
Center distance upper tolerance	Δa.s	0.1500	mm
Center distance lower tolerance	Δa.i	-0.1500	mm

		Gear 1	Gear 2	
Number of teeth	z	13	46	
Face width	b	150.0000	140.0000	mm

Figure 2-16 Calculation report.

3 Optimizations

3.1 Scuffing

In the calculation of the scuffing safety factor per the flash temperature criterion, S_B, profile modifications are considered. Per the task definition, no modifications are present; accordingly, the input for tip relief C_a = 0 μm and root relief C_f = 0 μm for both gears in tab *Details for Strength*. This leads to a high flash temperature at the beginning and end of the mesh. The corresponding graph is found in the menu Graphs/Contact temperature. It shows that the contact temperature rises to about 240°C, making the lubricant film unstable and the scuffing risk high.

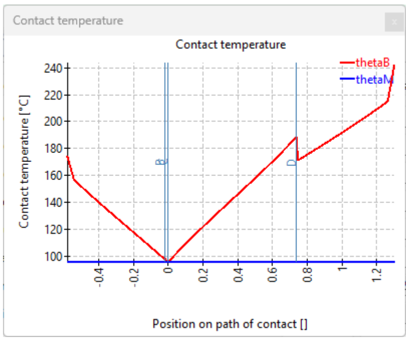


Figure 3-1Contact temperature graph, without tip relief.

In the report, in the section *Scuffing - Flash temperature* the following can be found: *Optimal tip relief*. This is the profile modification that leads to the lowest contact temperature. A tip relief of 50 μm is added each on the pinion and wheel and run the calculation again, the calculated safety factor increases from S_B = 1.06 to S_B = 1.82 and the contact temperature drops to approx. 230°C.

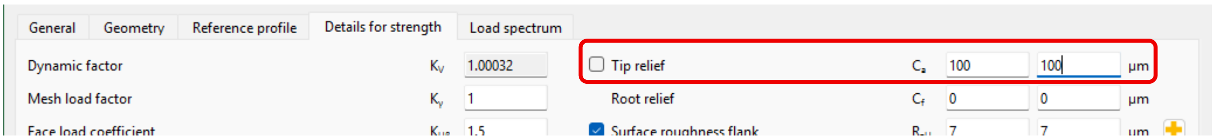


Figure 3-2 Input of tip relief in tab *Details for strength*.

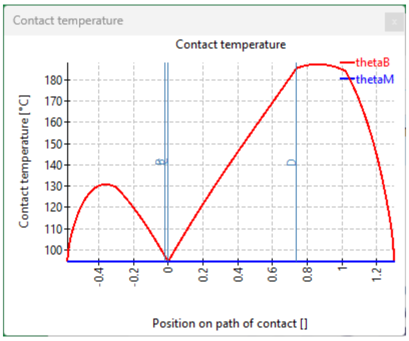


Figure 3-3 Contact temperature graph, with tip reliefs.

This has significantly reduced the contact temperature. Alternatively, or additionally, the load-carrying capacity of the lubricant can be increased by using a lubricant with a higher scuffing load stage. The scuffing load stage can be changed in the *Geometry*, via the + button at the *Lubricant, Lubricant* = *Own input* input. Modern lubricants reach scuffing load stage 14.

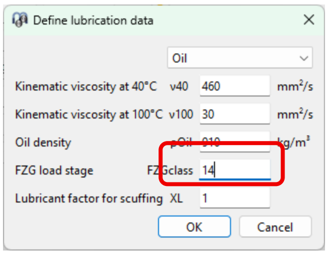


Figure 3-4 Increase of scuffing load stage from 12 to 14.

When the calculation is run again, the scuffing safety factor S_B increases to S_B = 3.69 and meets the requirement.

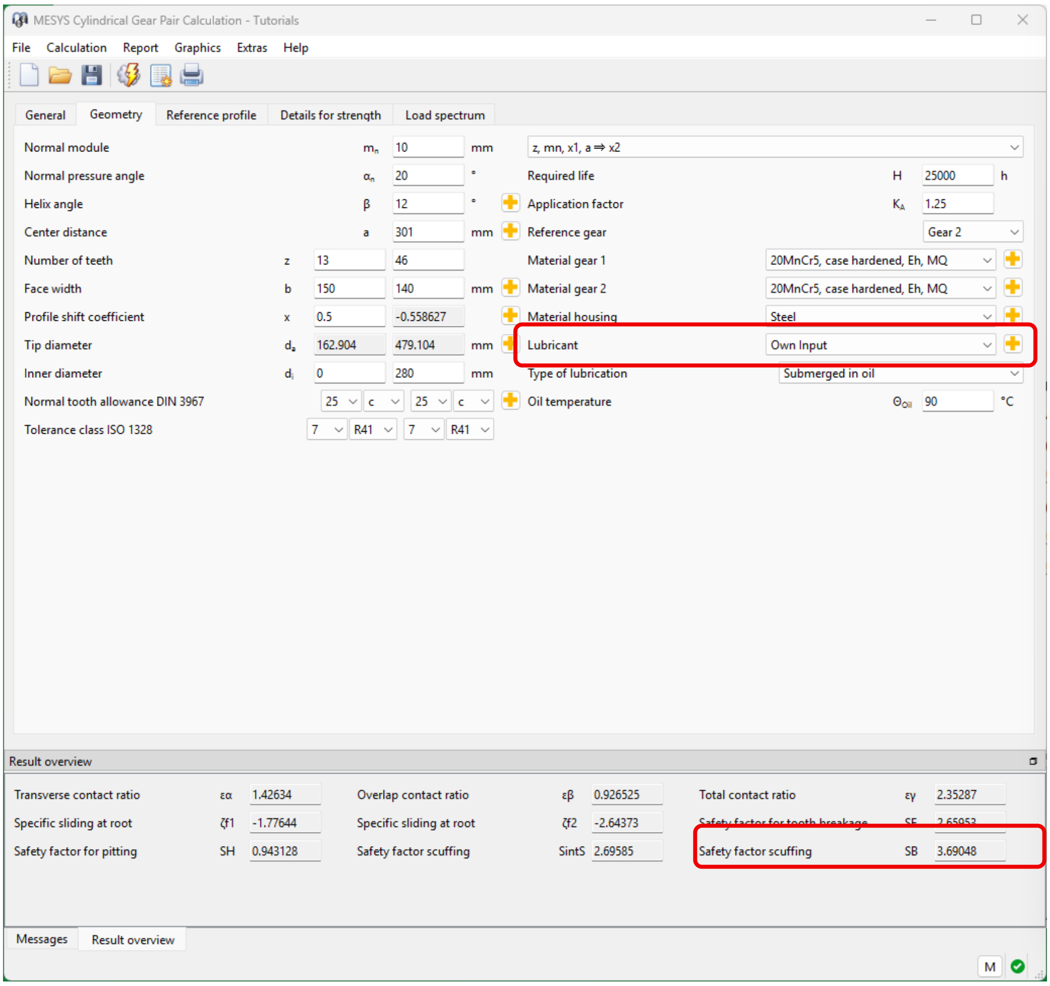


Figure 3-5 Scuffing safety factor S_B, after increase of scuffing load stage from 12 to 14, input *Lubricant* = *Own input*.

3.2 Tooth flank load capacity

The calculated tooth flank safety factor can be increased by making the gear larger. This approach is trivial; within the scope of this software tutorial, three other effects are demonstrated

1. Reduction $K_{H\beta}$.

2. Flank modification, factor f_{ZCa}.
3. Modification of the S-N curve.

The face load factor $K_{H\beta}$ = 1.50 may be conservatively chosen; it is set to $K_{H\beta}$ = 1.25. Accordingly, it is assumed that modifications are designed computationally via contact analysis, and thus f_{ZCa} = 1.00 may be set (ISO 6336-2, Table 3). Third, it is accepted that after some time a degree of pitting appears on the tooth flanks (compare S-N curve (1) per Figure 6 in ISO 6336-2 with S-N curve (2)).

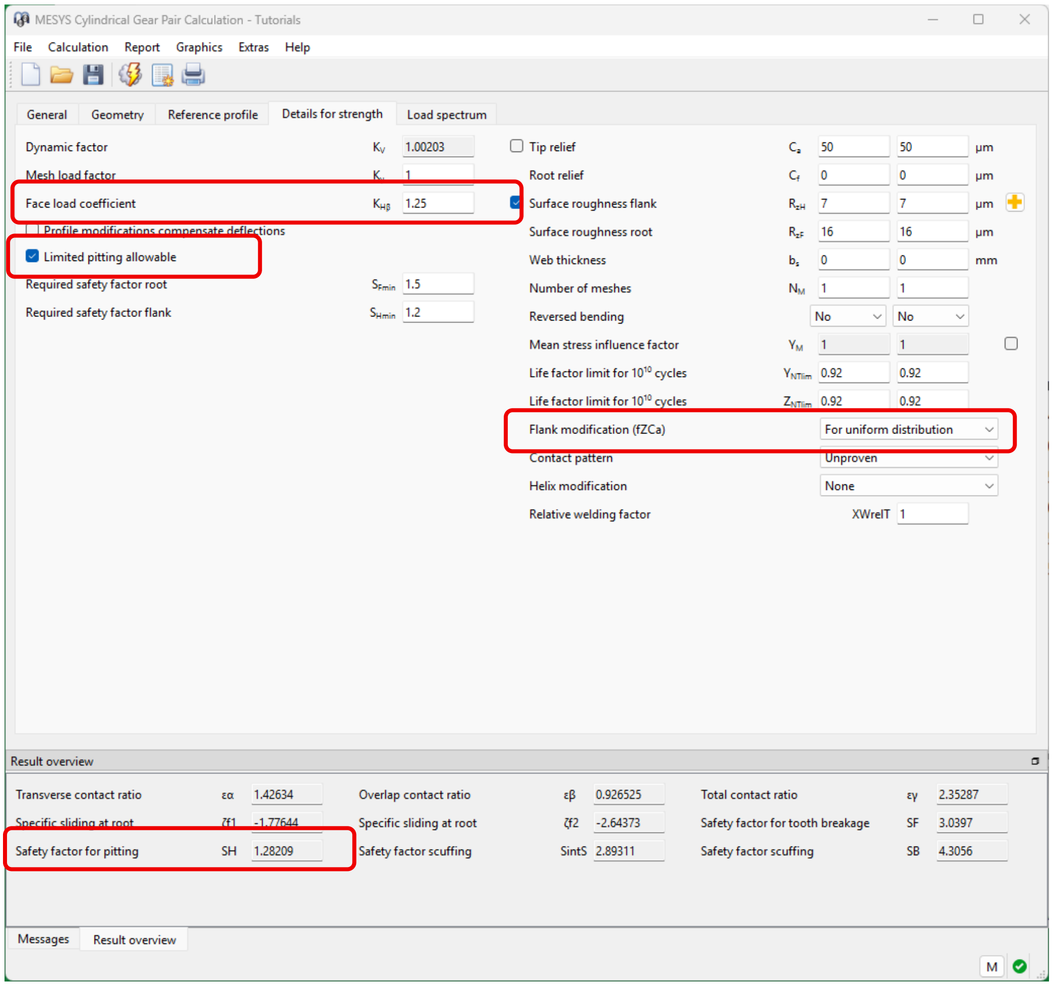


Figure 3-6 Changed inputs, influence on tooth flank safety factor (and others).

The safety factor for pitting now satisfies the requirement with S_H = 1.28 > S_{Hmin} = 1.20.

The file *MESYS-Tutorial-Cyl_gearpair_01-ww-v2500.mCGP* reflects this status.

4 Further gear sizing procedures

4.1 Initial sizing

Above, the number of teeth, module and tooth width were determined from experience with similar gears. These empirical values are not always available, and the desire is to obtain a suggestion for the size of the gearing from the loads.

This is implemented in MESYS by selecting *T, u = > z, mn, x, b, a* for the design strategy. The inputs are then

- The load, required service life, K-factors, the load spectrum.
- Angle of engagement and bevel angle, tolerances and inner diameter.
- Material and lubrication.

Enter the desired translation, here u = 4.50. Number of teeth and module are now grayed out and still come from the calculations above.

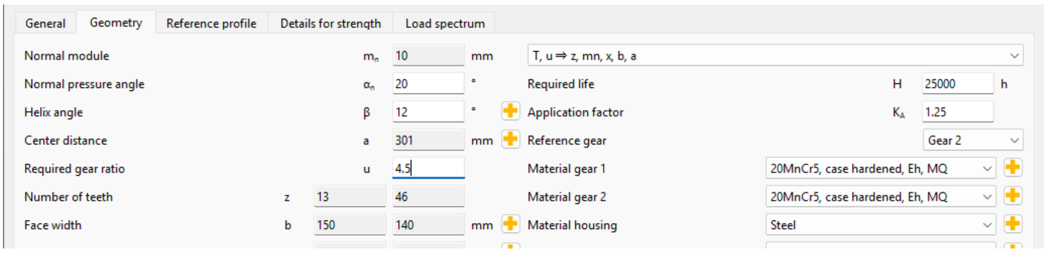
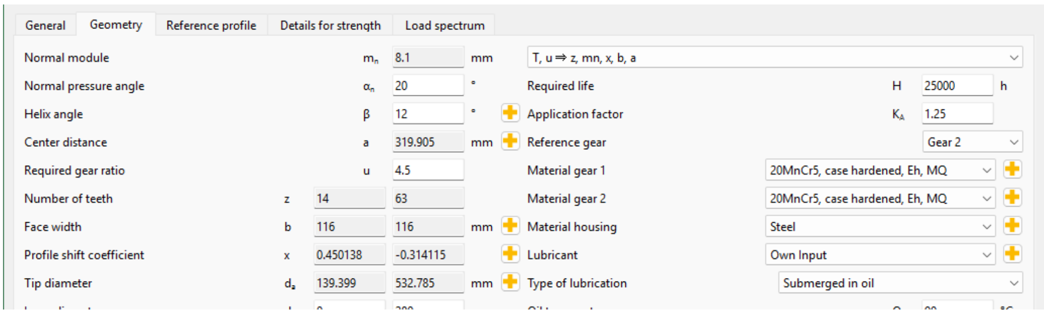


Figure 4-1 Sizing the gears from the required gear ratio and the given torque, etc.

If the calculation is now executed, a gear is defined that meets the required safety factors S_{Hmin} and S_{Fmin}. Module, center distance and tooth width can be adjusted afterwards. This allows specifications from production, for example, to be considered.



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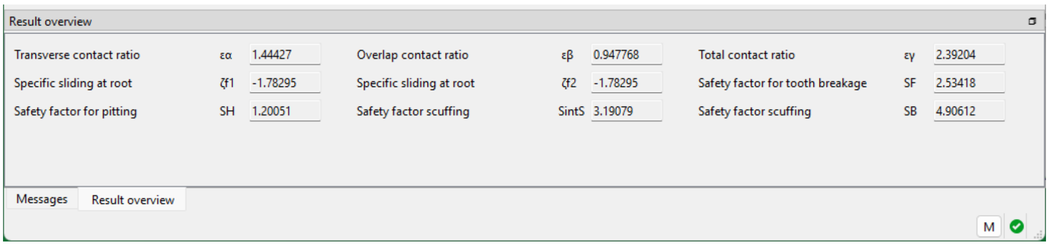


Figure 4-2 Resulting gear geometry, requirements on flank and root safety factor are met.

If the sizing method is changed to *z, mn, x1, a = > x2*, the module, center distance and width can be adjusted. After the calculation, the corresponding collateral can be found in the results overview.

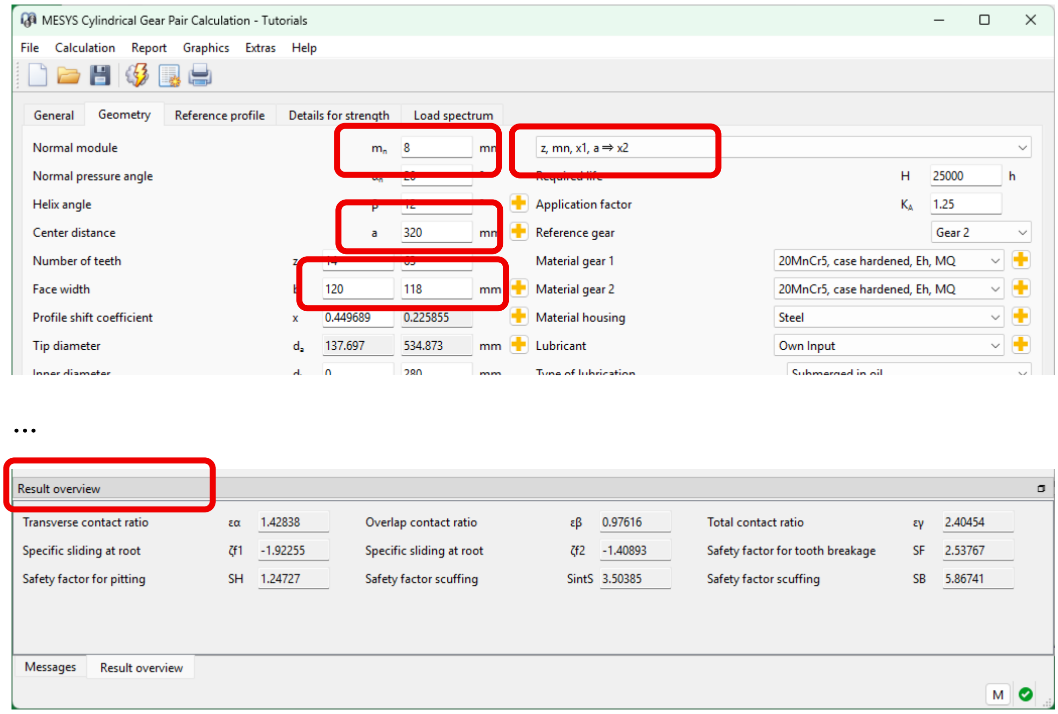


Figure 4-3 Modified sizing and results overview.

4.2 Sizing of module and face width

Open file *MESYS-Tutorial-Cyl_gearpair_01-ww-v2500.mCGP*.

A typical question in gear design, especially in industrial gearboxes, is the question of module and tooth width if the number of teeth and the center distance are given. In practice, this is the case if the housing already exists, along with the required gear ratio. From experience, a minimum number of teeth, e.g. $z_1 = 11$, is often required for the pinion. With a required translation, e.g. u = 3.20, we find the number of teeth on the gear as $z_2 = 35$. The center distance in the housing is given as a = 290.00 mm. The helical angle is chosen as β = 20° assuming we design the first stage rotating at higher speed.

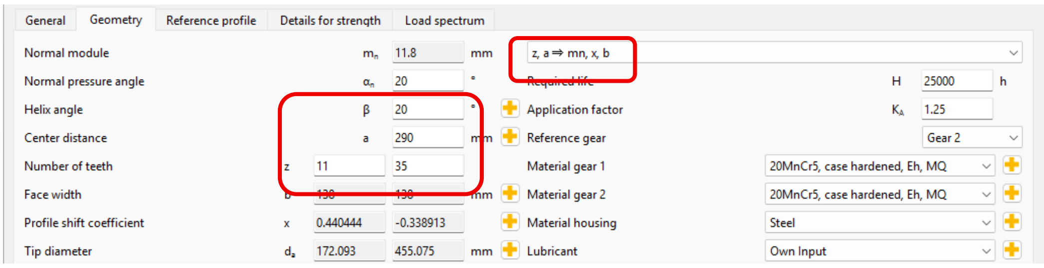
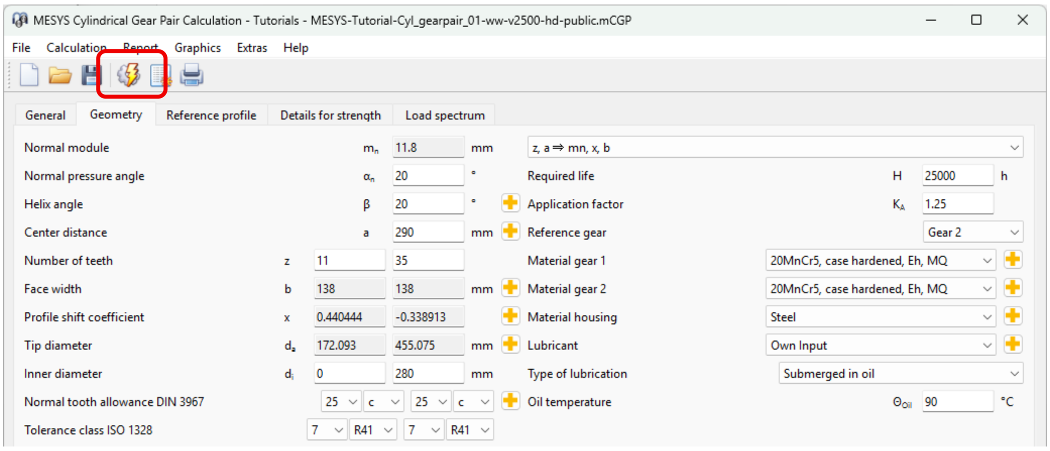


Figure 4-4 Input of number of teeth, center distance, helix angle. Selection of sizing method.

After running the calculation, safety factors meet the requirements and a suitable gear geometry results.



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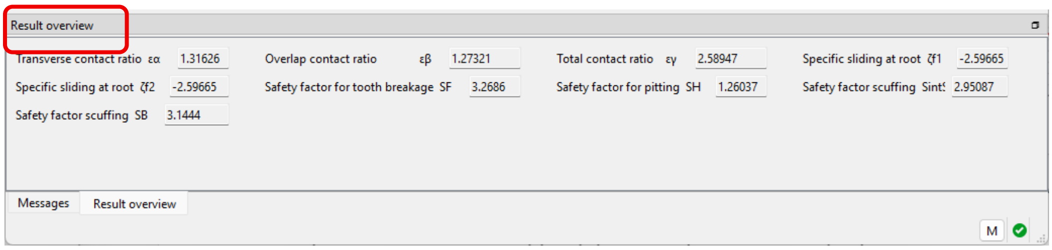


Figure 4-5 Executing the calculation, resulting gear geometry, resulting safety factors.

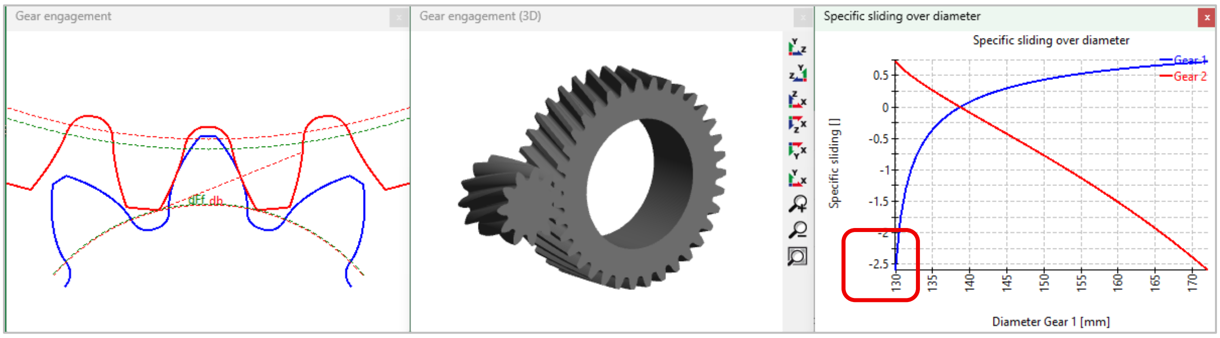


Figure 4-6 Graphics from menu *Graphics/Gear engagement*, *Graphics/CAD/Gear Engagement (3D)*, *Specific sliding over diameter*.

With ζ_{f1} = -2.23, the specific sliding value is somewhat high in absolute terms. MESYS selects the profile shift factors such that $\zeta_{f1} = \zeta_{f2}$ applies. The graphic Graphics/Specific sliding over profile shift shows that a change of the profile shift coefficient for the pinion (gear 1, x-axis) will lead to a worse situation.

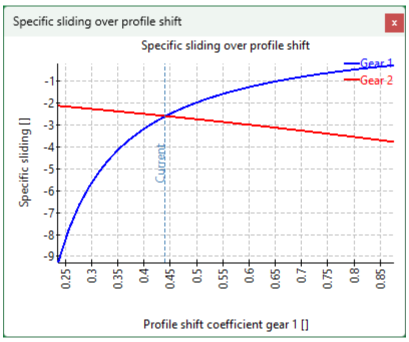
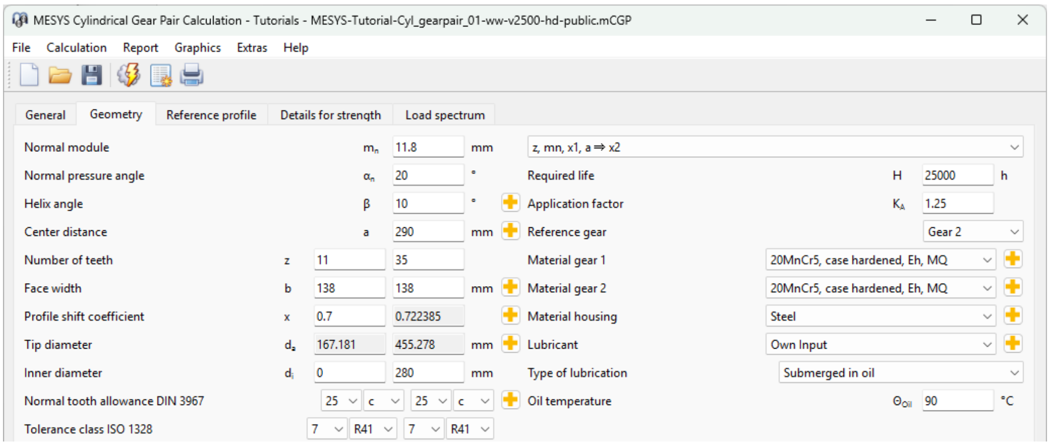


Figure 4-7 Graphic Graphics/Specific sliding over profile shift. Currently, the specific sliding is balanced ($\zeta_{f1} = \zeta_{f2}$) and therefore minimized. A change of profile shift for gear 1 (and therefore also for gear 2 in opposite direction) results in a worse situation.

To reduce the absolute value of ζ_f, increase the profile shift for both gears. Since the center distance is given, the helical angle must be reduced. To do this, the sizing strategy is changed back to *z, mn, x1, a* = > *x2* and with β = 10° and x_1 = 0.70 as input, x_2 = 0.81 results. The safety factors are within the specifications, and the result is very low absolute values for specific gliding.



...

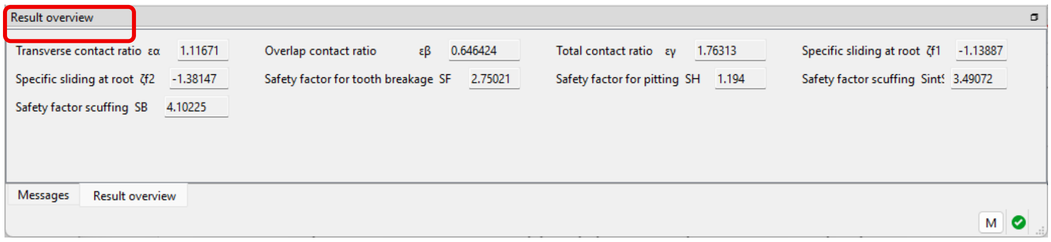


Figure 4-8 Modifying the gearing, center distances remains, helix angle is lowered.

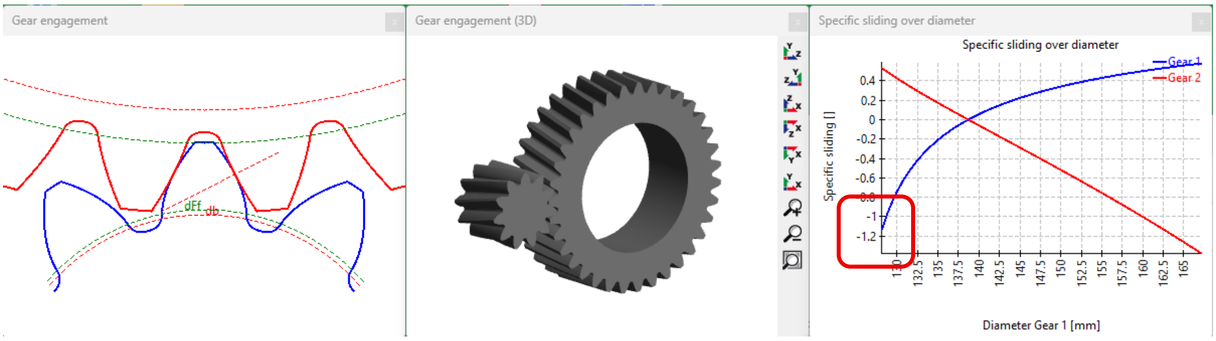


Figure 4-9 Gears with optimized profile shift and low specific sliding.

4.3 Sizing of internal gear

Open file *MESYS-Tutorial-Cyl_gearpair_01-ww-v2500.mCGP*.

In MESYS, the design is also possible for internal gearing. The number of tooth ratio u is then negative. Select sizing option *z1, u, mn, x1, x2 => z2, a* and enter u = -3.50. From experience, x_1 = 0.50 and x_2 = -1.00 is set, the other values are left as they are. The calculation is executed.

If the entry for the ring gear inner diameter (i.e. the absolute largest diameter, the outside) is forgotten, the following message appears. Inner diameter should be set to d_i = 0.00, then it is set to |di| = |df| + 4*m_n.

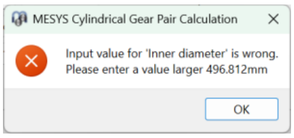


Figure 4-10 Message that the input of the diameter for the internal gear is not suitable.

The reference profile of the internal gear is automatically adjusted, the maximum possible tool tip radius is set.

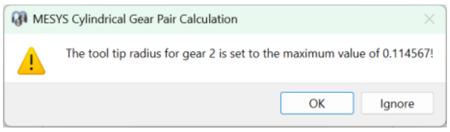


Figure 4-11 Message that the internal gear tool tip radius is set to the maximum possible value.

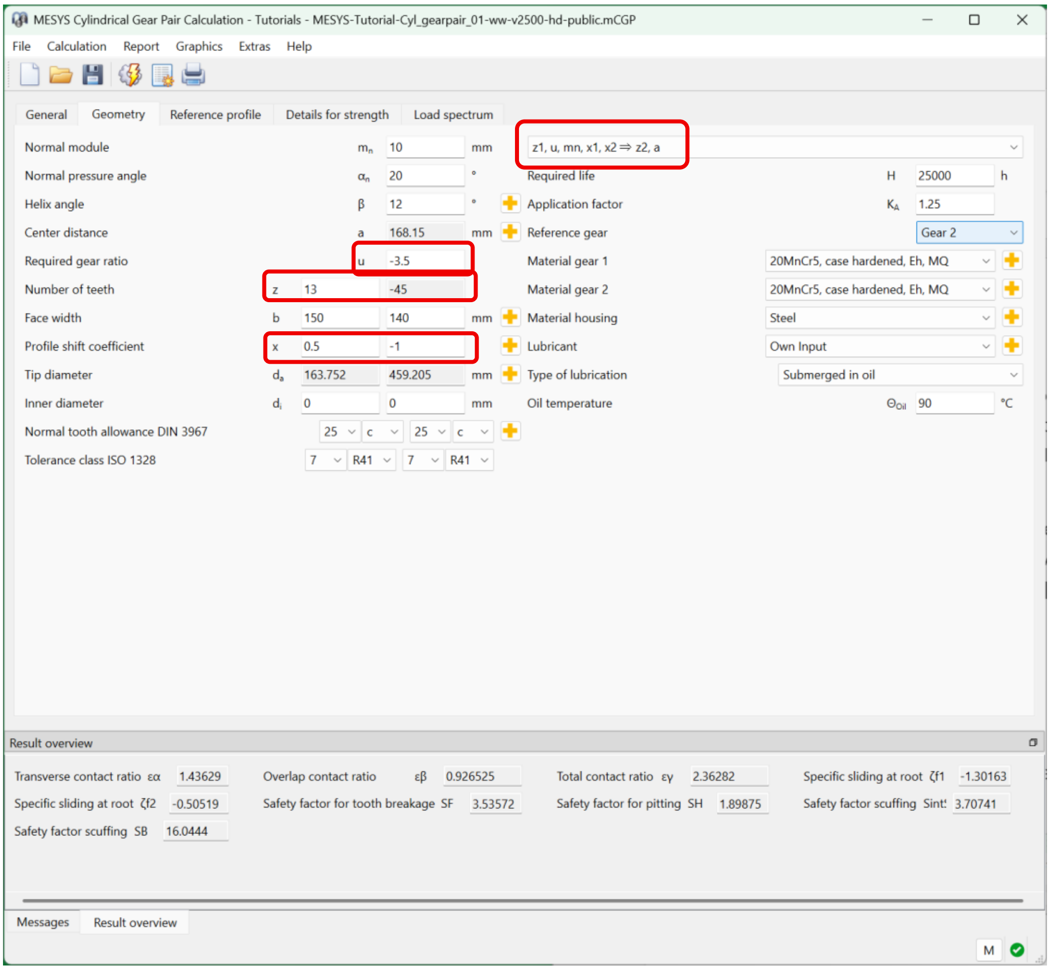


Figure 4-12 Situation and results after the calculation.

Graphics Graphics/Gear engagement and Graphics/CAD/Gear engagement (3D) are opened to visualize the mesh. The thickness of the ring gear is too low, set d_{i2} = 580.00 and repeat calculation.

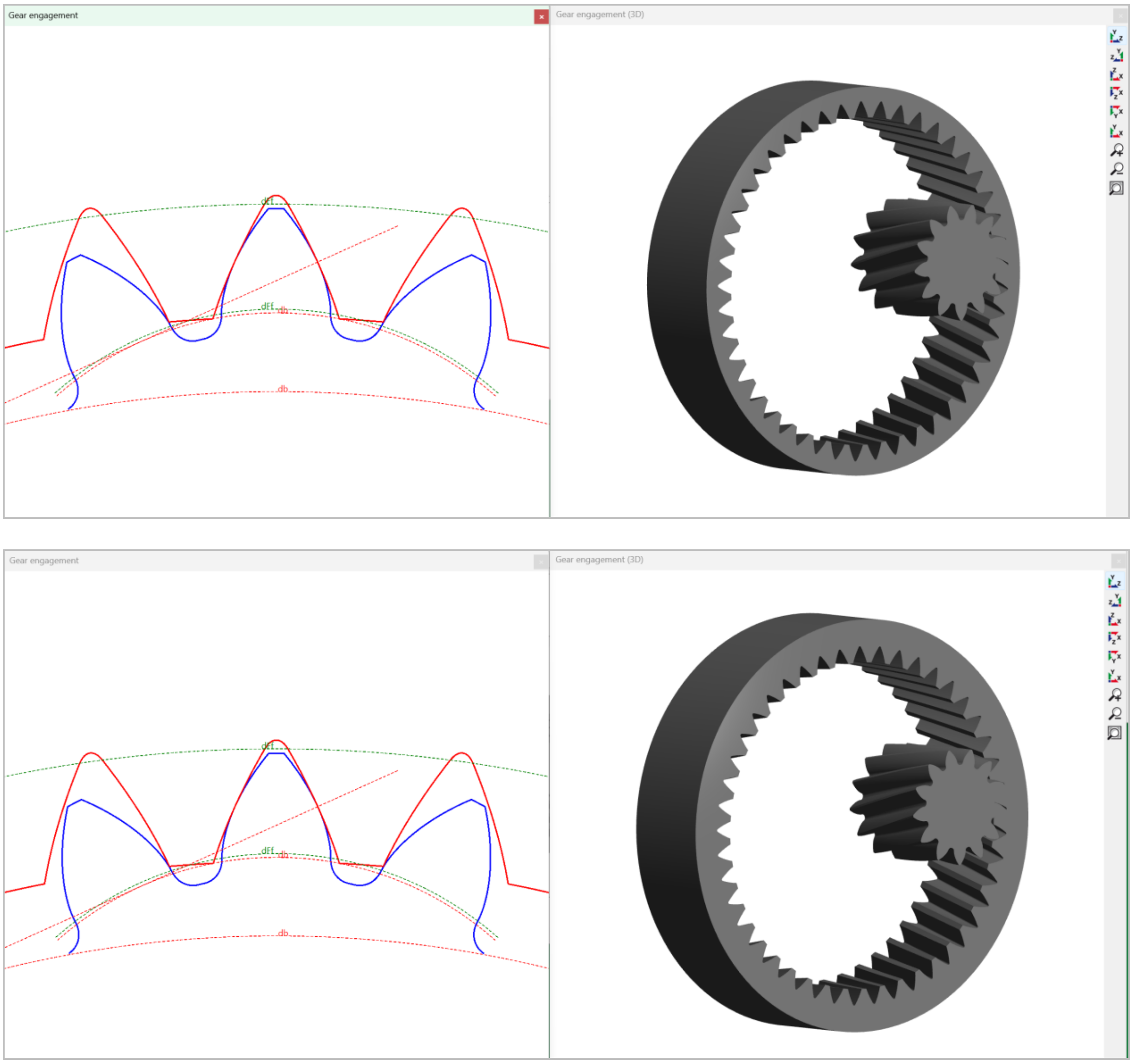


Figure 4-13 Top: Mesh and 3D geometry with original ring gear thickness. Bottom: Ring gear inner diameter set to d_{i2} = 580.00 mm.