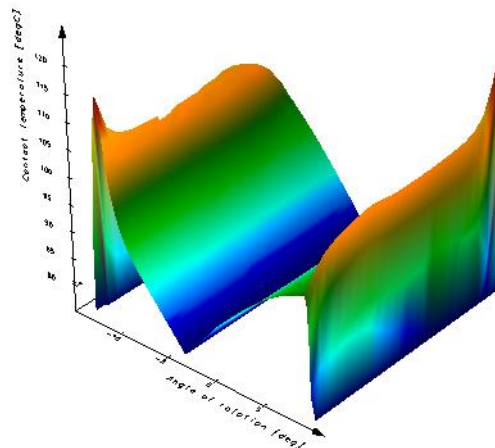
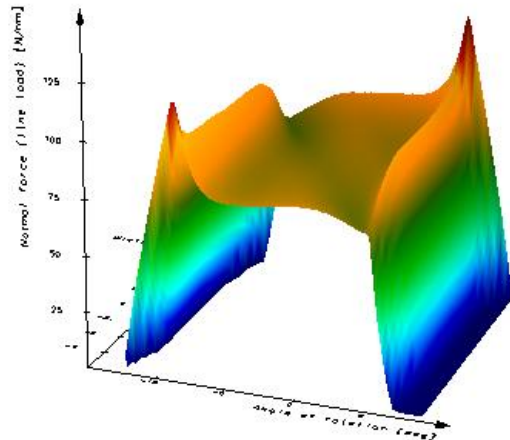
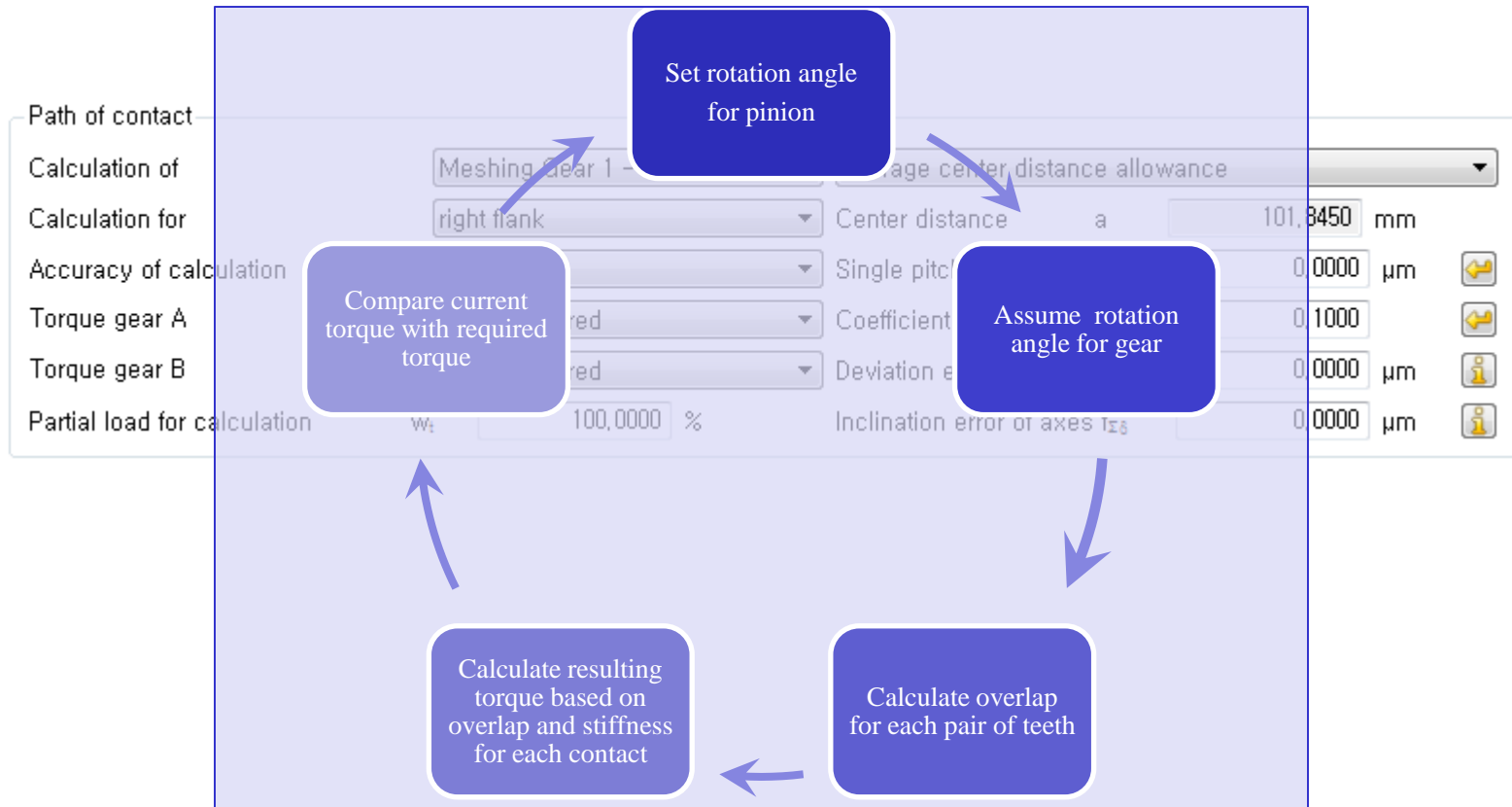


The path of contact calculates

- the normal force during meshing
- the actual path of contact under load
- bearing force
- transmission error and contact stiffness
- stress (contact and bending)
- specific sliding, power loss, heat development
- flash temperature, lubrication film thickness
- micropitting, wear



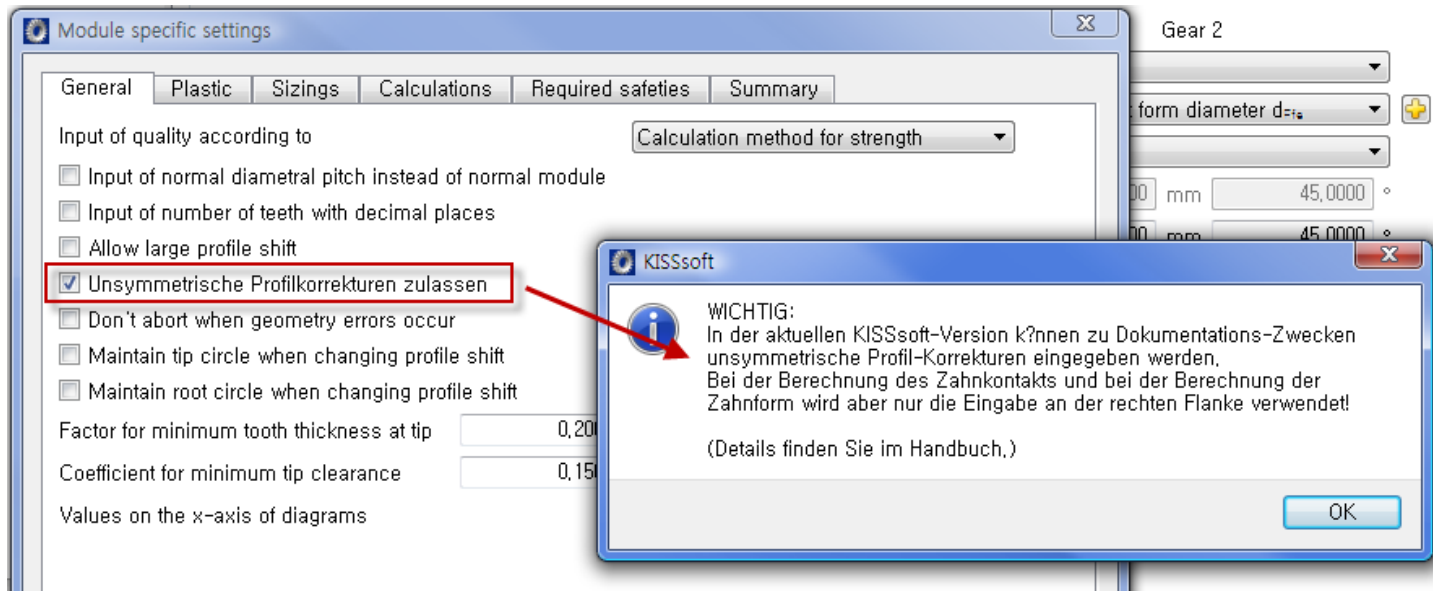


The path of contact under load or transmission error is calculated iterative using local contact stiffness. Helical gears are calculated as slices of spur gears.

The term “Path of Contact” will be changed to “Contact Analysis” in 2011 version to better represent the calculation features.

The path of contact calculation is always done for right flank contact.

In 2011 version, the “Flank” in the Modification tab will be removed, and will be shown only when “Use asymmetric profile correction” option is checked.

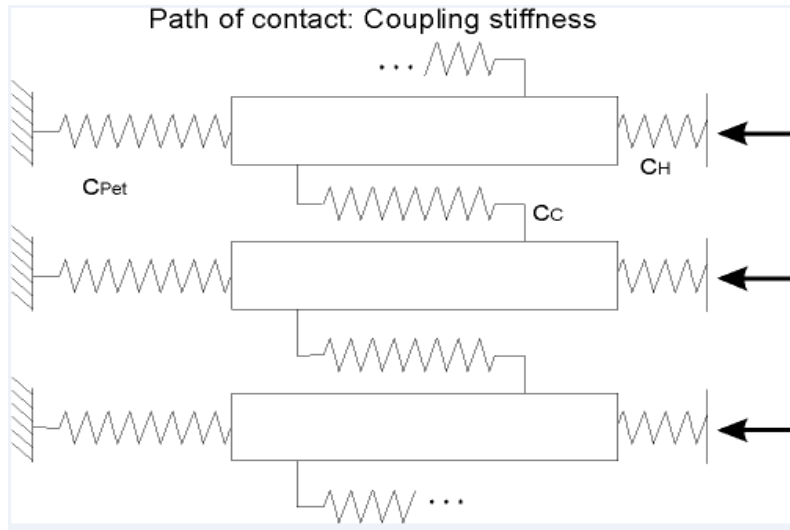


The message says,

“In the current KISSsoft version, asymmetrical profile corrections can be entered only for documentation purposes. Only the right flank will be used for the contact analysis and the tooth form calculation!”

What's new in version 04-2010?

- Coupling between the slices



$C_{Pet} = C_Z + C_{RK} = \text{Stiffness Tooth root following Peterson [N/m/mm]}$
($C_Z = \text{Stiffness from bending and shear deformation following Petersen}$)
($C_{RK} = \text{Stiffness from deformation through rotation in the gear blank}$)

$C_H = \text{Stiffness from Herztian flattening following Petersen}$

$C_C = \text{Coupling stiffness} = 0.04 * (A_{sec})^2 * C_{Pet}$

A_{sec} : Number of slices

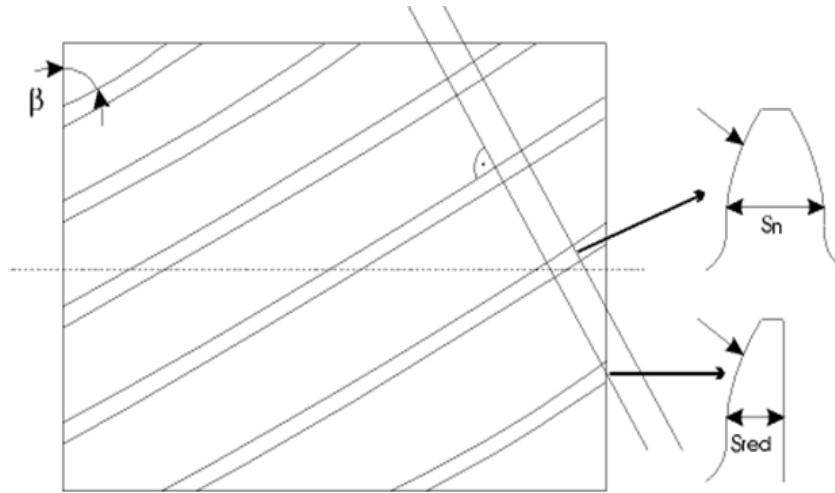
0.04: Empirical factor, approved through comparing calculations with FEM

$(A_{sec})^2$ is logical because: Different number of slices must give same overall result; C is /mm

What's new in version 04-2010?

- Decreased stiffness at tooth ends

Tooth bending rigidity in helical gears may be decreased at tooth ends, see figure:



$$C_{Pet}^* = \text{Reduced coupling stiffness at border} = C_{Pet} * (S_{red}/S_n)^{0.5}$$

S_n : Normal tooth thickness

S_{red} : Reduced tooth thickness at tooth ends

Exponent 0.5 was evaluated by comparing with FEM and with LVR calculations.

- Revised calculation of tooth stiffness

Tooth stiffness following Peterson is calculated based on the effective tooth form in normal section (in earlier KISSsoft-versions tooth form was based on transverse section reduced by $\cos(\text{helix angle})$).

What's new in version 04-2010?

- Face load factor $K_{H\beta}$ not any more added to stresses

As due to the coupling between the slices, the load distribution calculation over the face width is improved and non-parallelism of the axis can be introduced.

Thus, $K_{H\alpha}$, $K_{H\beta}$ (ISO) and K_M (AGMA) Factors are not multiplied to the displayed stresses. Displayed stresses are still multiplied with the application factor K_A and the dynamic factor K_V .

- Hertz stress

Calculation of Hertz stress is based on the Hertzian law of the contact of two cylinders. This gives realistic results in most situations. A problem encountered is when the contact is on a corner of the flank (as for example corner at the tip diameter, corner at the beginning of a linear profile modification, corner at the beginning of an undercut), then the radius of curvature becomes very small, the Hertz stress calculated has a high peak. This is not a realistic issue, because the part of the flank near to the corner will be joined in the contact. An algorithm to check the joining flank parts and increase the radius of curvature is included. If high peaks still remains, we recommend to add a realistic radius to the corners and use circular profile modifications instead of linear ones.

- 3D Graphics display along the face width introduced

Where you can see the differences?

- Graphics result menu

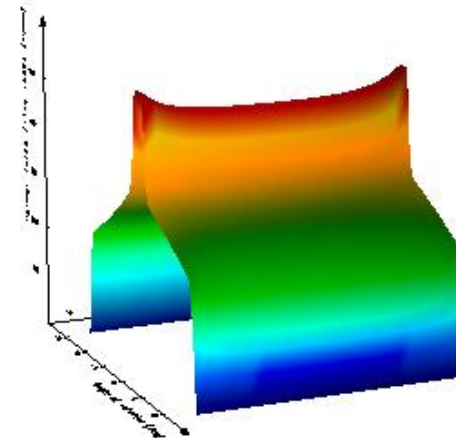
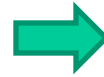
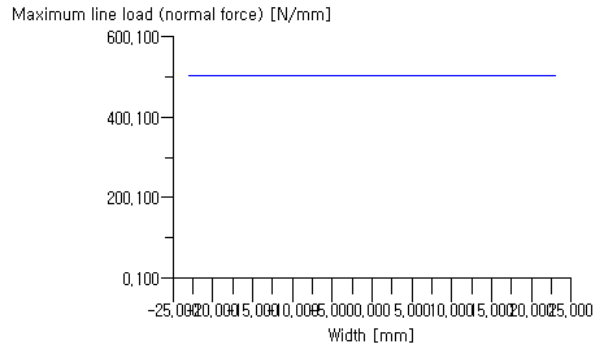
- Transmission Error
 - Normal force curve
 - Maximum normal force over facewidth
- Torque curve
- Stiffness curve
- Bearing force curve
- Bearing force curve in %
- Direction of the bearing forces
- Kinematics
- Specific sliding gear A
- Specific sliding gear B
- Power loss
- Heat development
- Heat development gear A
- Heat development gear B
- Flash temperature (ISO TR 6336-7)
- Lubricating film (ISO TR 6336-7)
- Specific film thickness (ISO TR 6336-7)
- Stress curve
- Maximum stress over facewidth
- Stress curve gear A
- Stress curve gear B
- Wear gear A
- Wear gear B



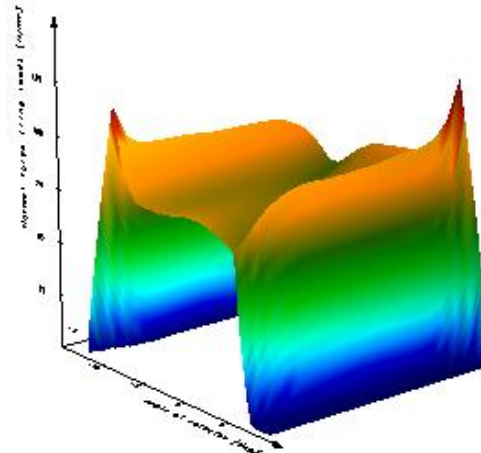
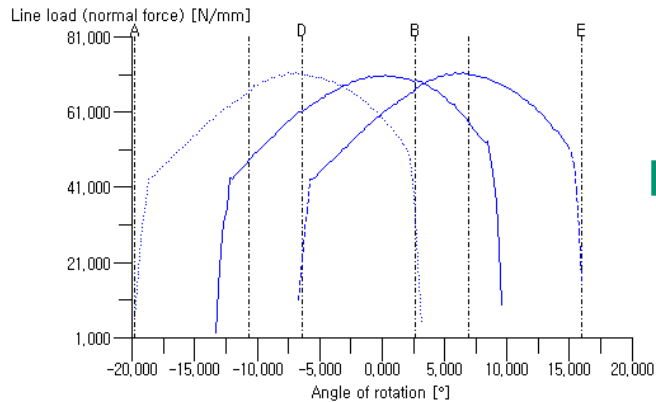
- Transmission Error
 - Normal force curve (line load) ▶
 - 2D
 - 3D
 - Torque curve
 - Stiffness curve
 - Bearing force curve
 - Bearing force curve in %
 - Direction of the bearing forces
 - Kinematics
 - Specific sliding per gear ▶
 - Gear A
 - Gear B
 - Power loss ▶
 - Heat development ▶
 - Heat development per gear ▶
 - Flash temperature (ISO TR 15144) ▶
 - Lubricating film (ISO TR 15144) ▶
 - Specific film thickness (ISO TR 15144) ▶
 - Safety against micropitting (ISO TR 15144) ▶
 - Stress curve ▶
 - Stress curve per gear ▶
 - Wear per gear ▶

Where you can see the differences?

- Spur gear pairs with different face widths



- Helical gear pairs



- Gear pairs with deviation and/or inclination errors of axes

Normal force curve (line load)

Calculation of path of contact

Save the graphics

Print the graphics

Lock the graphics

Show property window

Save the resulting curve

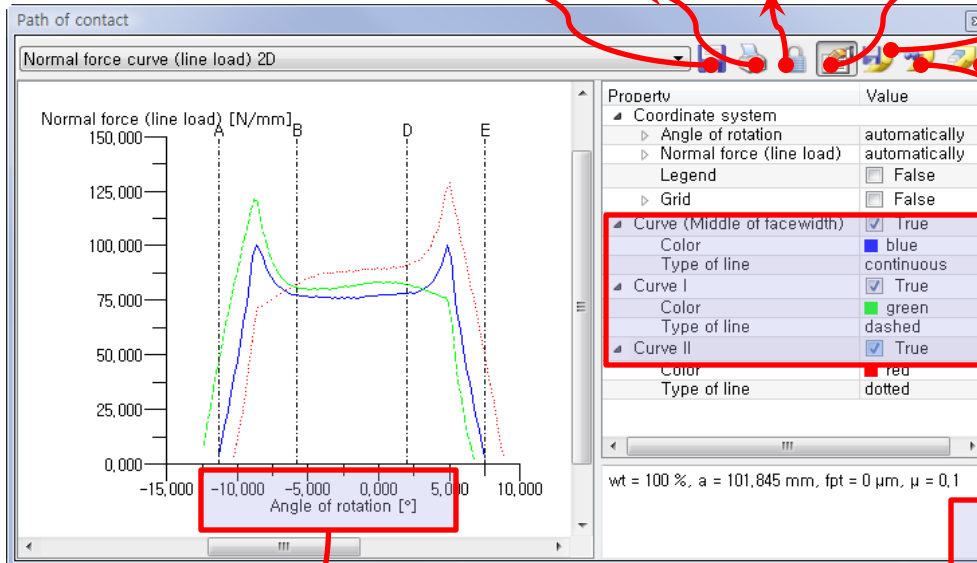
Clear the locked curve

Lock the resulting curve

Setting for resulting curve

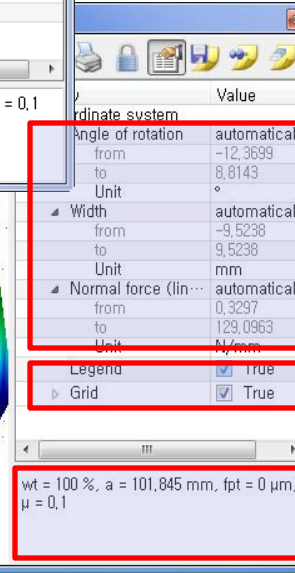
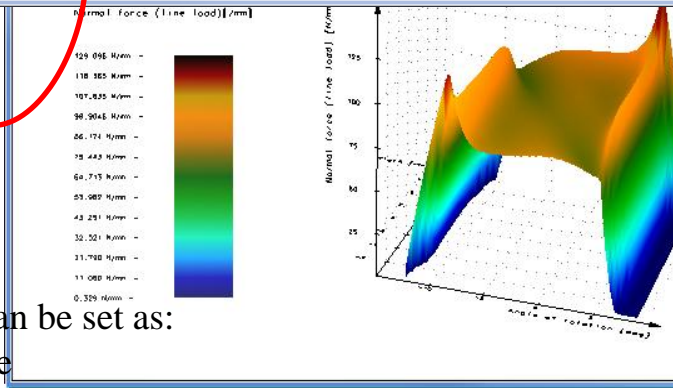
Setting for axis format

Setting for background



Horizontal axis can be set as:

1. Rolling angle
2. Length (path of contact)
3. Diameter (gear A)
4. Angle of rotation



Comments window:
Important settings and the calculation results will be shown

Transmission error is defined as the variation of rotation angle $\Delta\phi_2 = \phi_2 - u \cdot \phi_1$.

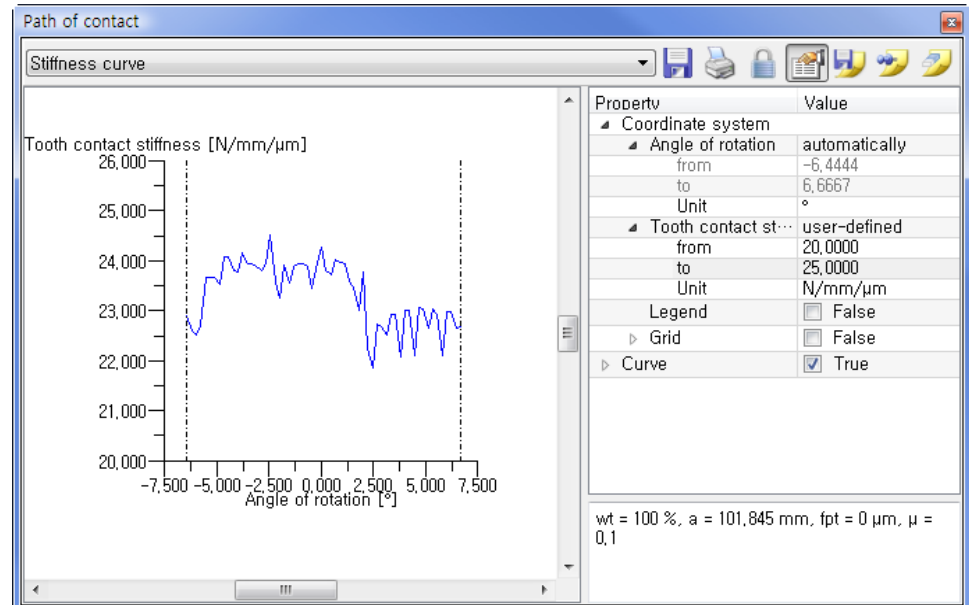
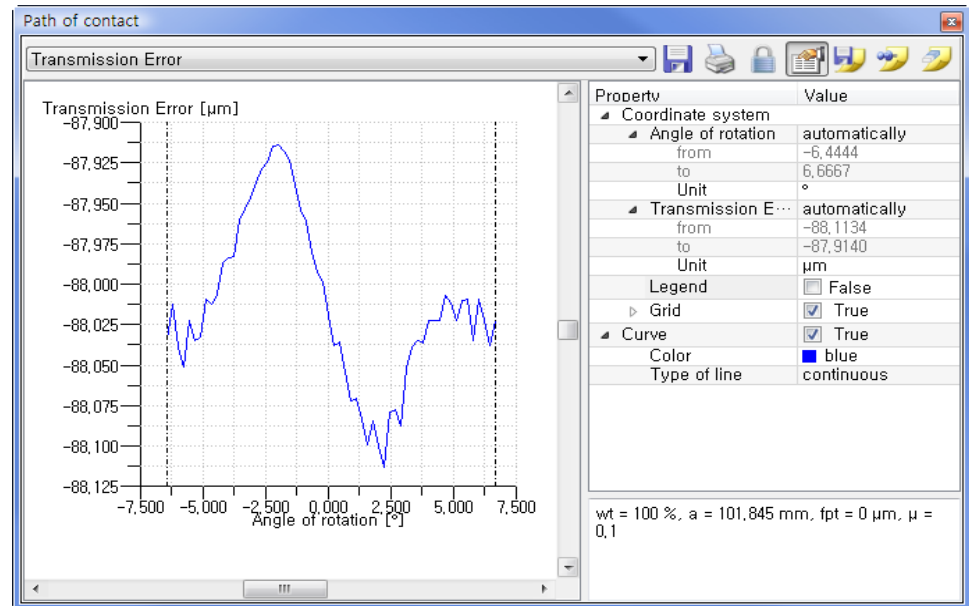
It is usual to show as length on the pitch circle $\Delta j = \Delta\phi_2 \cdot d_w / 2$.

The main part of the TE is the result of backlash, then the bending deformation of the teeth.

For each contact a tangential stiffness is calculated $c_i = \Delta F_n / \Delta \delta$

- rotation of the tooth in its base
- bending deformations
- shearing deformations
- Hertz stress

The total tangential stiffness is the sum of these values for contact stiffness over all contacts.



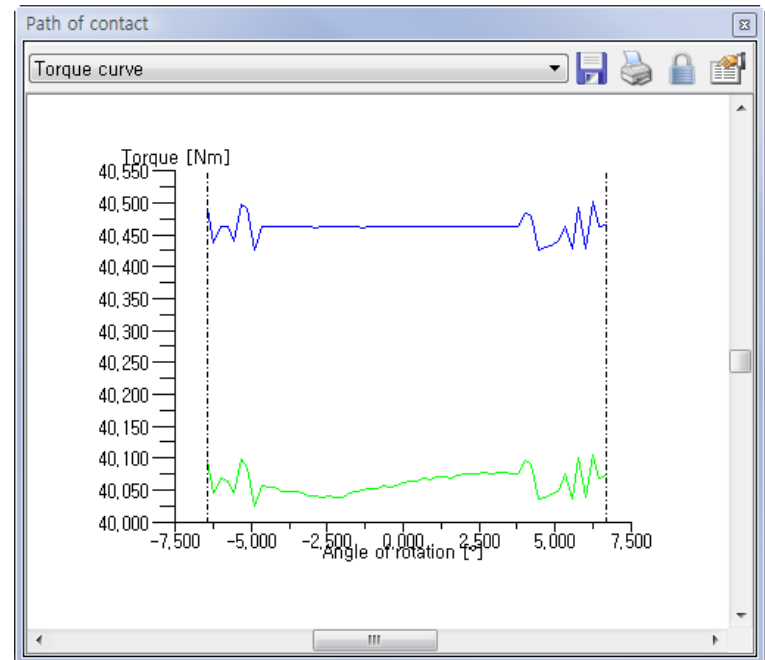
The torque is calculated as

$$T = r \times (F_n \cdot \mathbf{n} + \mu \cdot F_n \cdot \mathbf{t})$$

And the iteration is performed until the torque provided as input is reached. The torque of one gear should be constant in the output, the torque of the mating gear shows the losses.

The diagram shows T_2/u so the losses can be easily identified.

Since one torque is constant, the normal force is not constant if friction is considered.

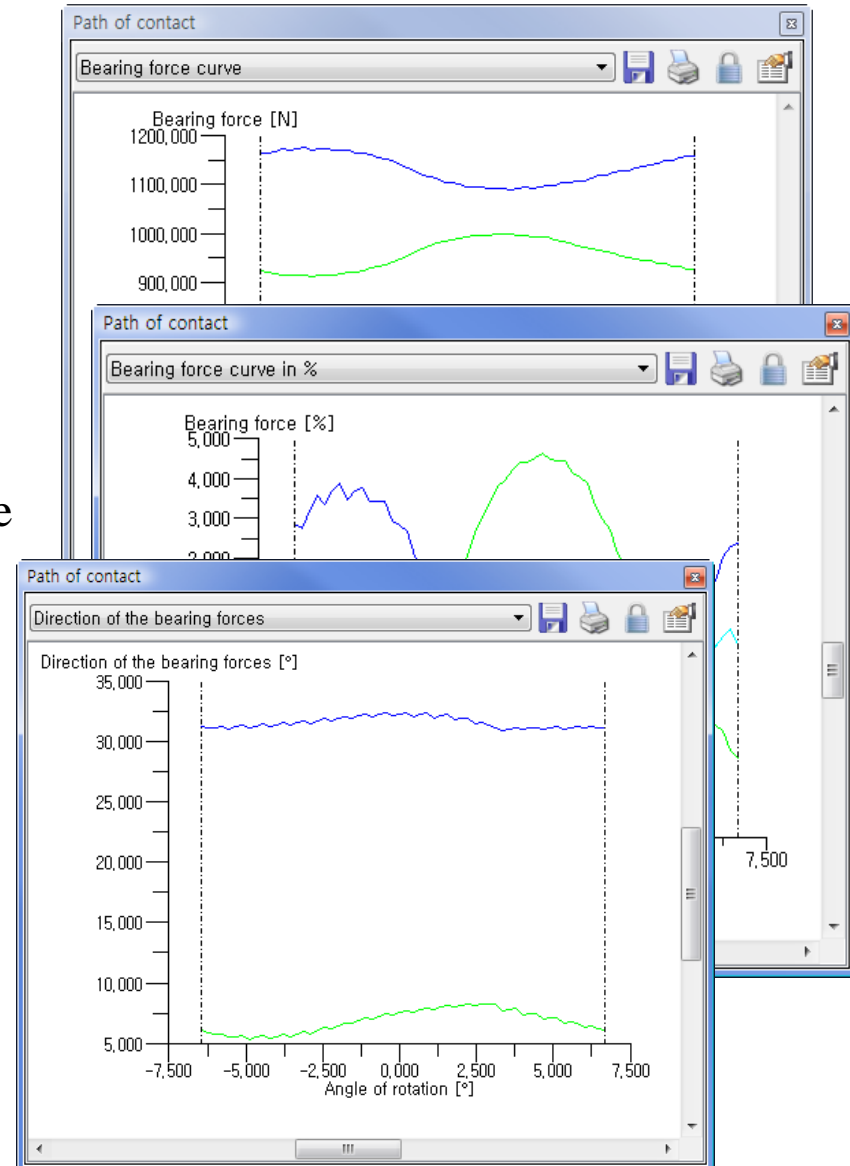


With assumed symmetrical bearing in the distance given for the calculation of face load coefficient the bearing forces are calculated by normal loads, axial and frictional force.

Without friction there will be no variation in bearing forces for involute gears.

Variations in bearing loads will generate vibrations and therefore noise.

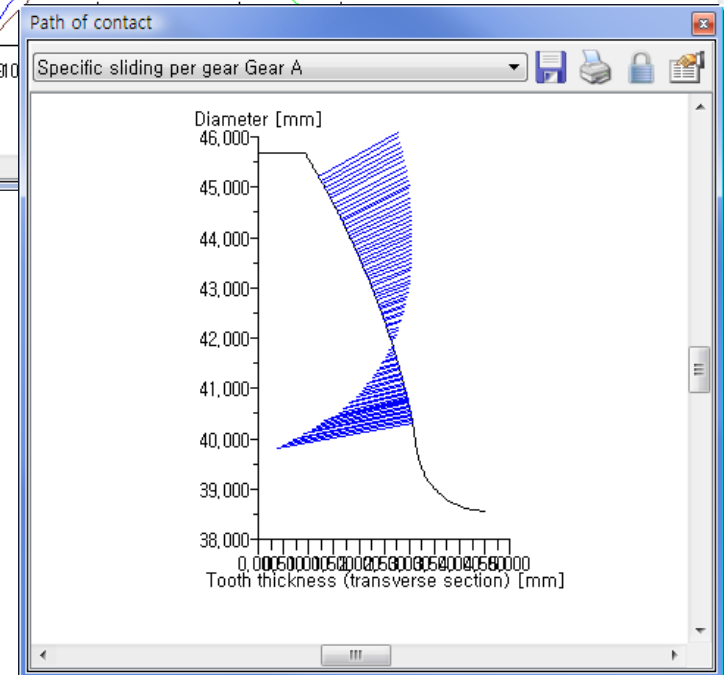
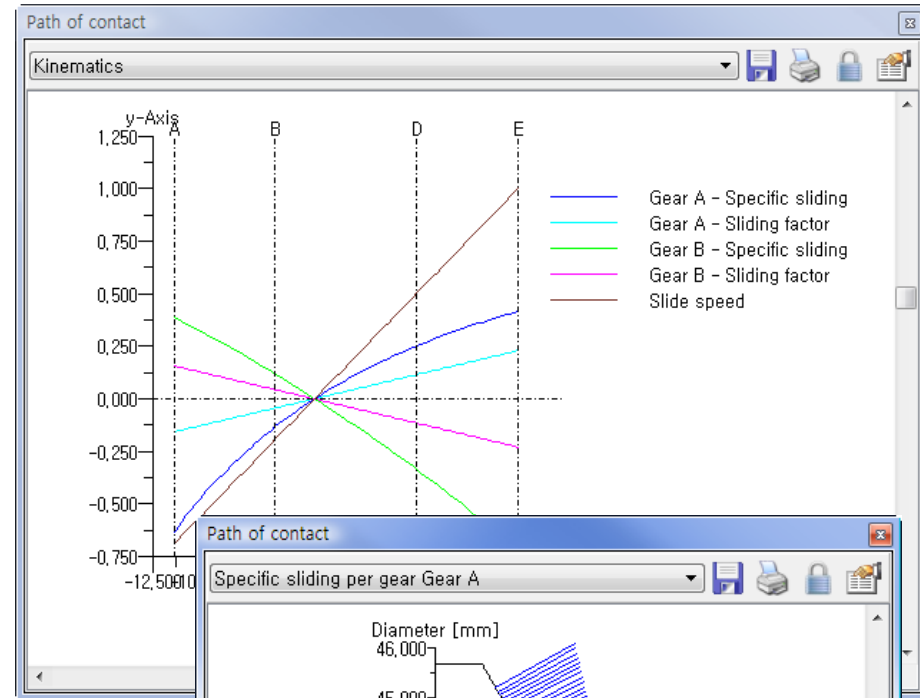
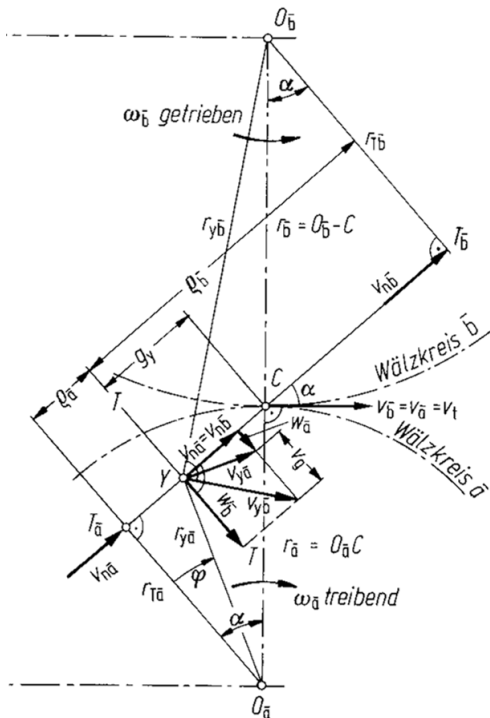
The calculation was performed with a constant torque. In reality inertia will generate variations of torque and therefore normal loads because of the transmission error.



The specific sliding is calculated from the local sliding velocity and the tangential speed in the contact point:

$$\zeta_a = v_g/w_a = 1-w_b/w_a$$

$$\zeta_b = v_g/w_b = 1-w_a/w_b$$

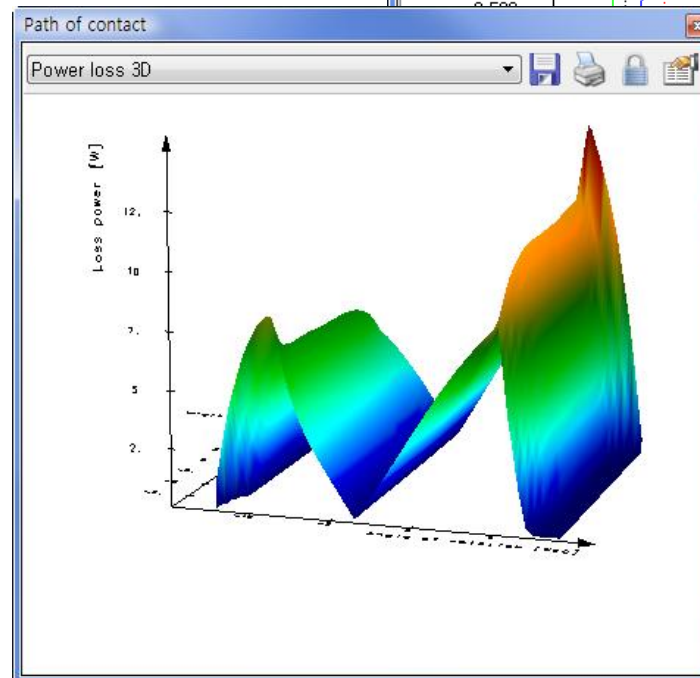
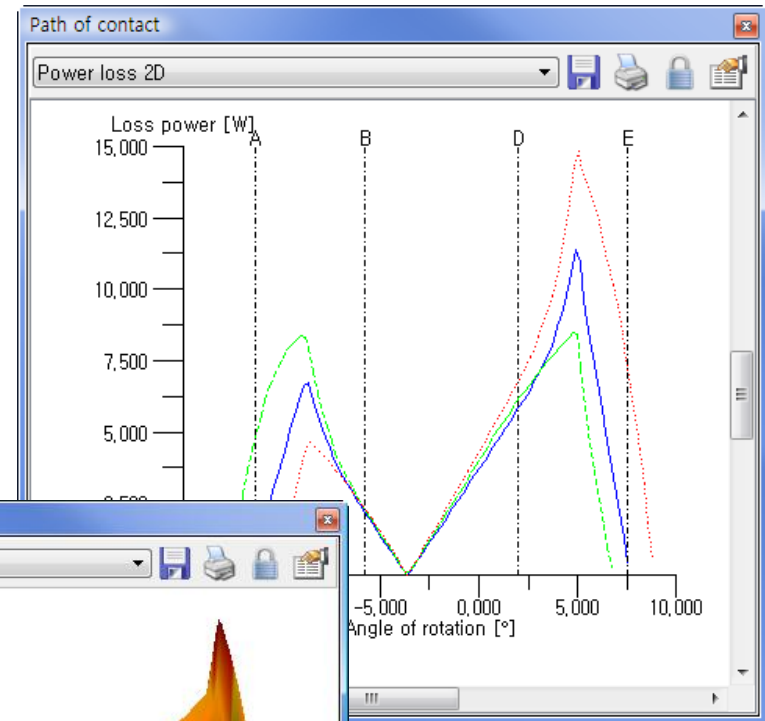


Calculation of power loss

The power loss on one flank is calculated as $P_{loss} = v_g \cdot \mu \cdot F_n$

The local heat development is the power loss per tangential speed:
 $Heat = P_{loss}/w$

Both values are given for the total face width.



Calculation of heat development

Calculation of path of contact

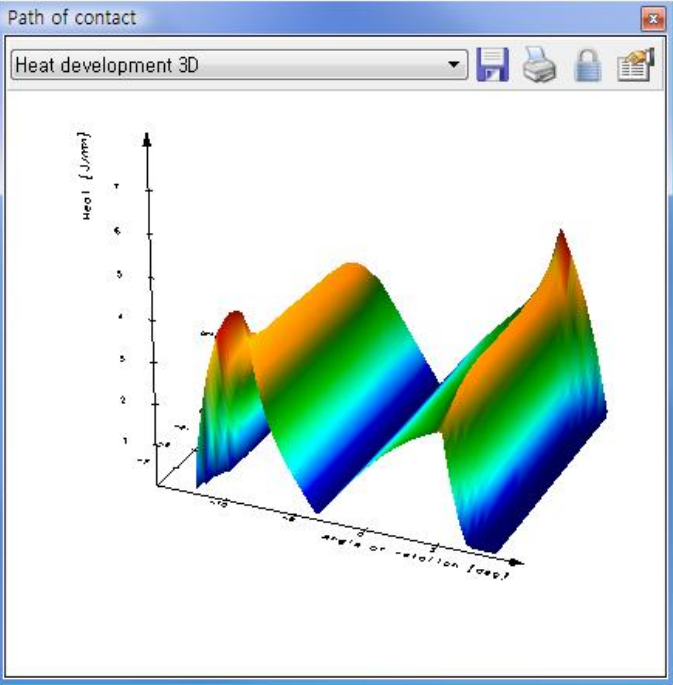
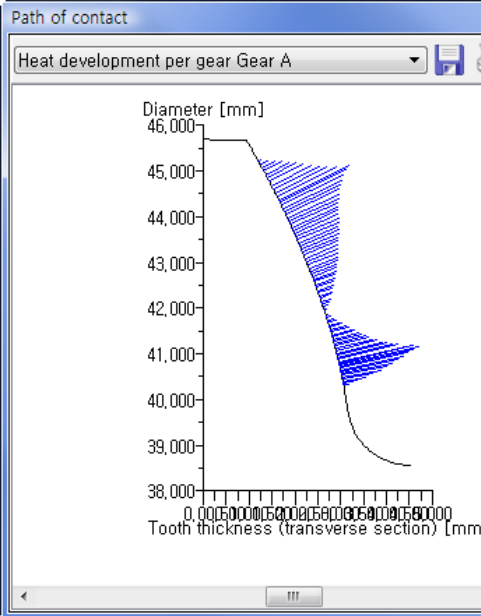
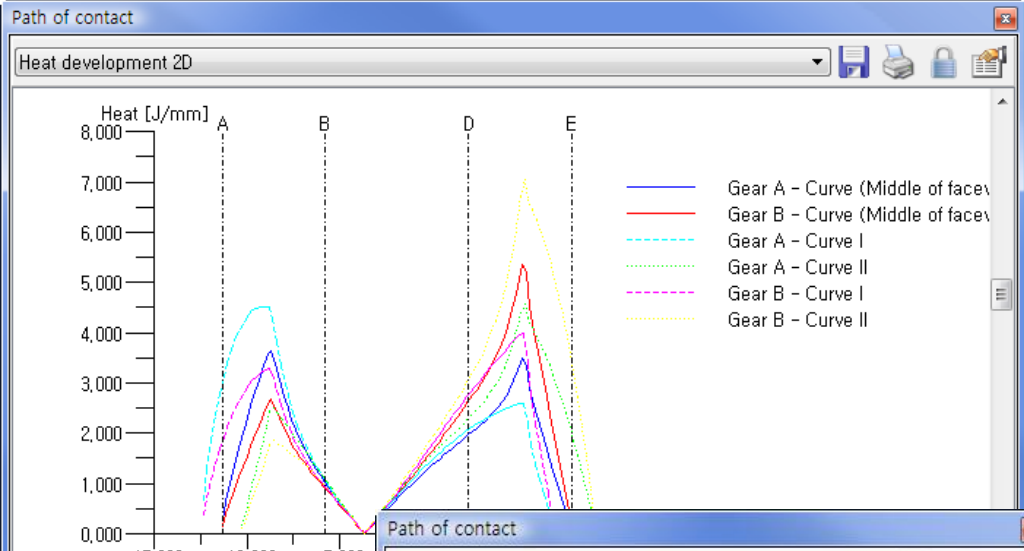
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$$Heat = P_{loss}/w$$

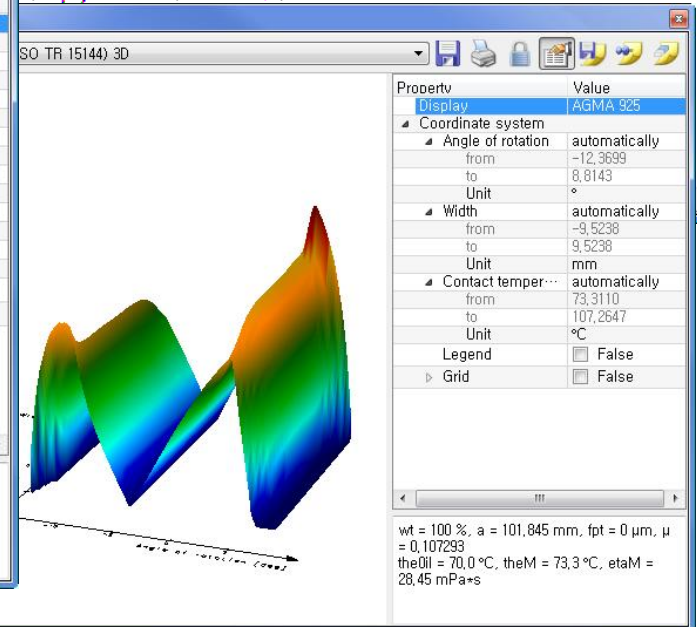
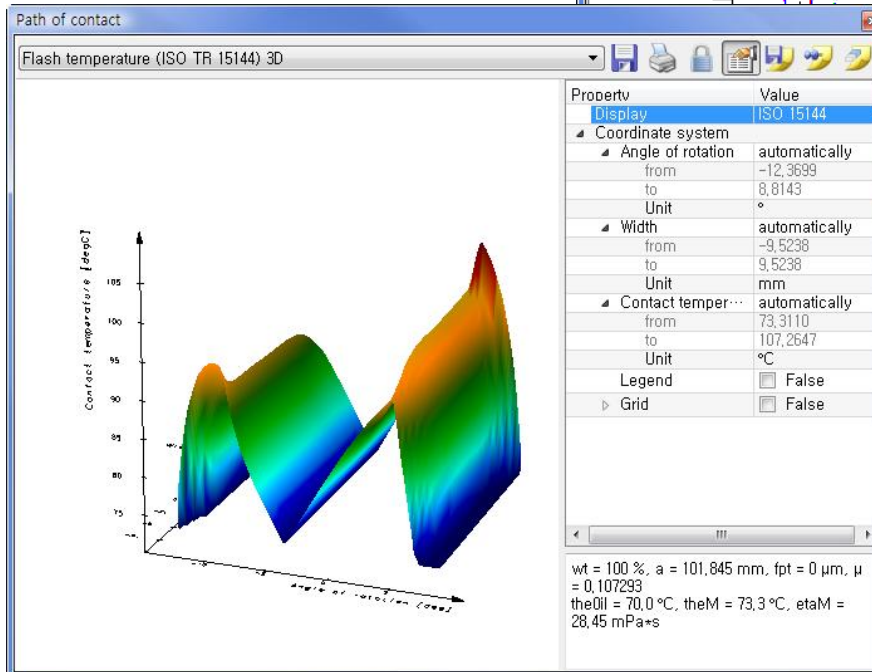
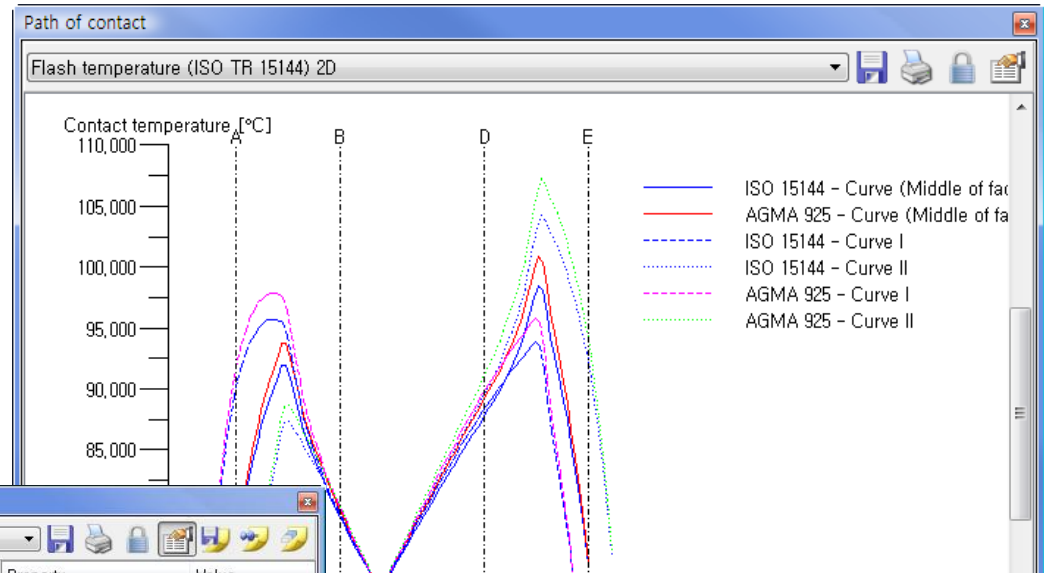
Both values are given for the total face width.



Calculation of flash temperature

The flash temperature is calculated according to ISO TR 15144 and AGMA 925.

Both values are given for the face width.

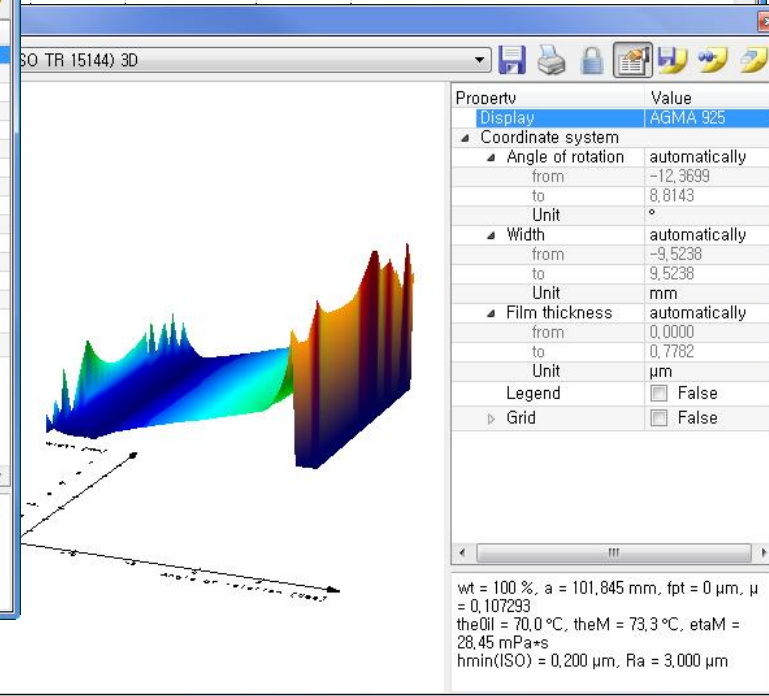
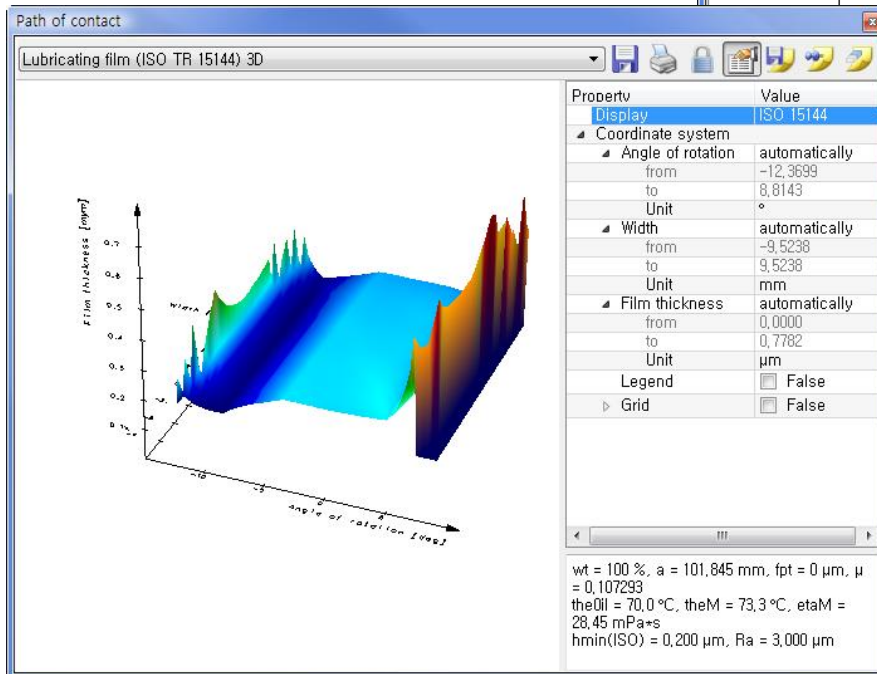
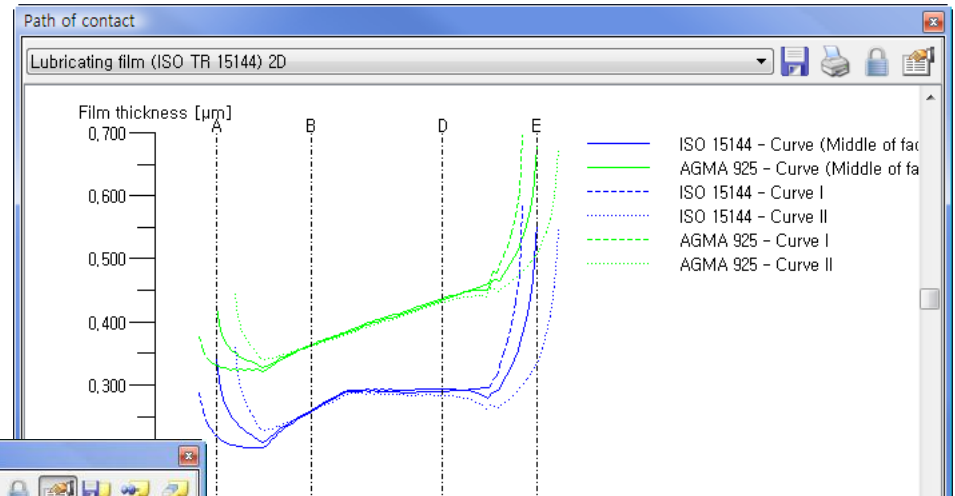


Calculation of lubricating film

Calculation of path of contact

The lubricating film is calculated according to ISO TR 15144 and AGMA 925.

Both values are given for the face width.

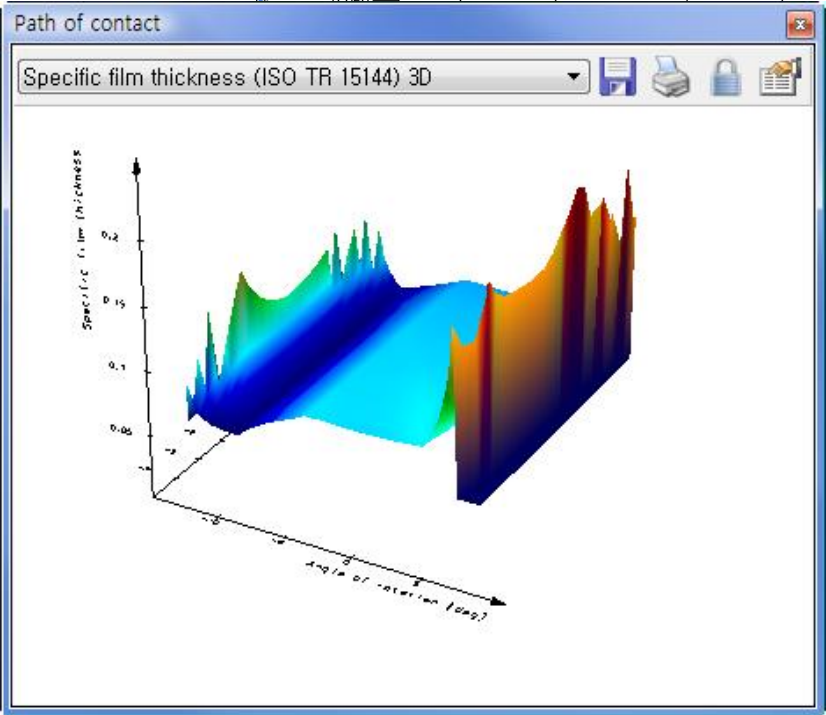
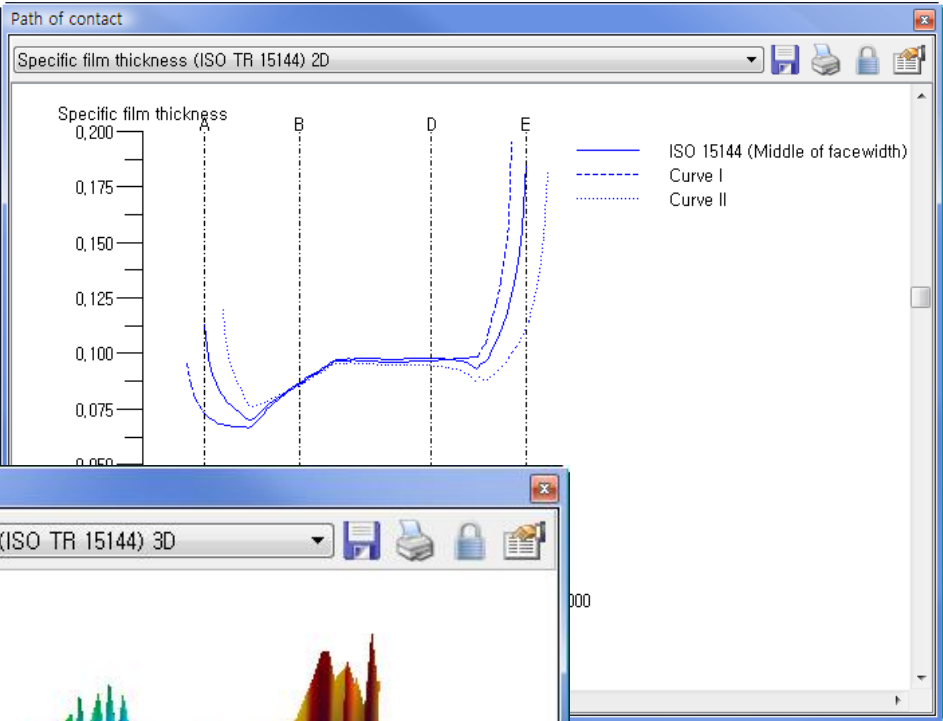


Calculation of specific film thickness

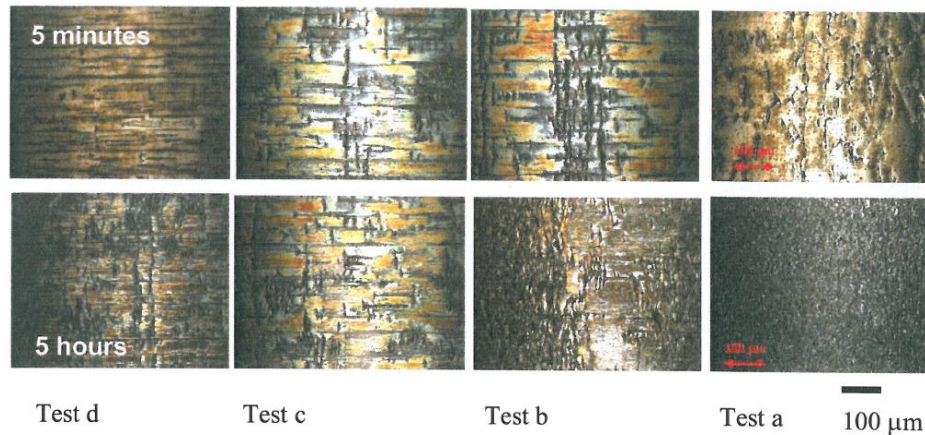
Calculation of path of contact

The specific film thickness is calculated according to ISO TR 15144.

The values are given for the face width.



Micropitting is the generation of numerous surface cracks which are small relative to the size of the contact zone, typically of the order 10 - 20 μm deep by about 25 - 100 μm.



“Roughness and Lubricant Chemistry Effects in Micropitting”
A.V.Oliver et. al., 2007

The safety against micropitting is calculated according to ISO TR 15144.

$$S_{\lambda} = \frac{\lambda_{GF, \min}}{\lambda_{GFP}} \geq S_{\lambda, \min}$$

$$\lambda_{GF, \min} = \min(\lambda_{GF, Y})$$

is the minimum specific film thickness in the contact area;

$$\lambda_{GF, Y}$$

is the local specific film thickness (see 5.3);

$$\lambda_{GFP}$$

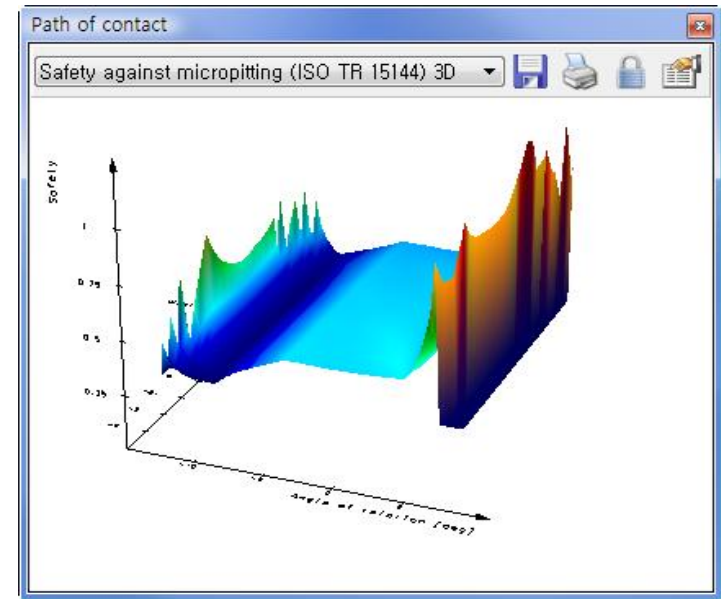
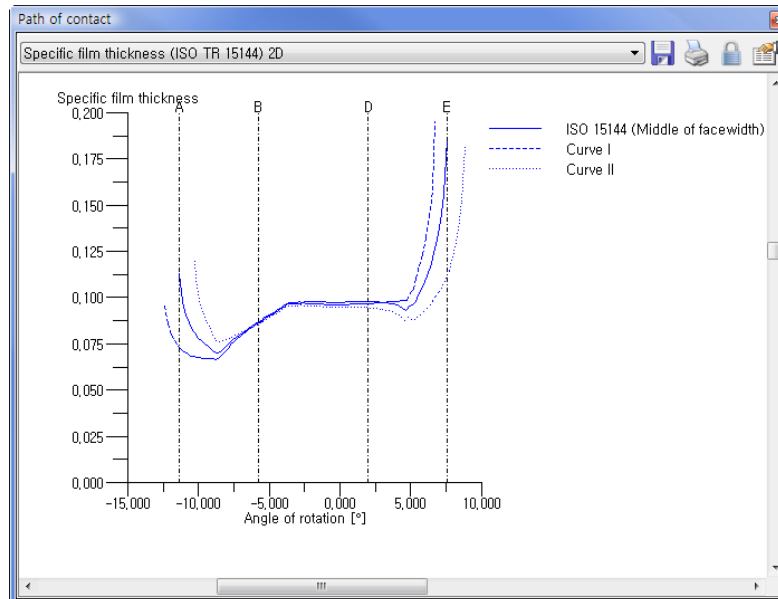
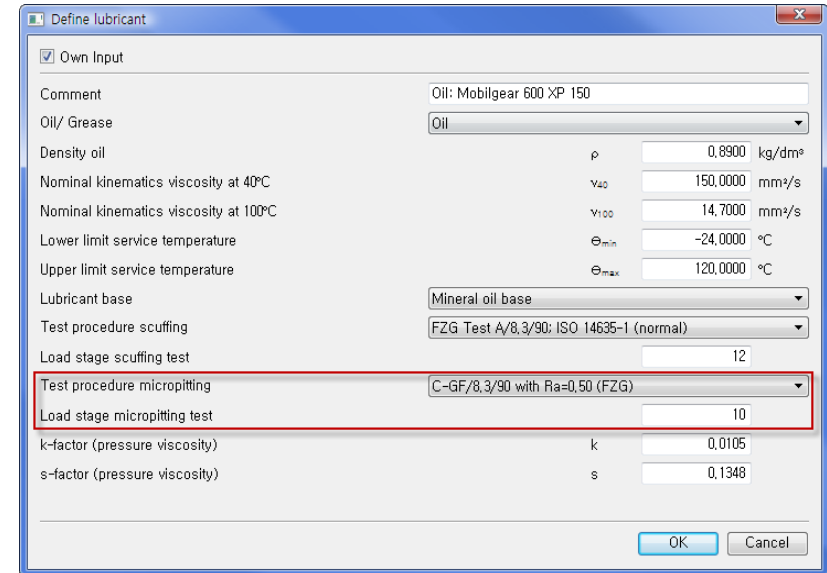
is the permissible specific film thickness (see 5.4);

$$S_{\lambda, \min}$$

is the minimum required safety factor (see 5.5).

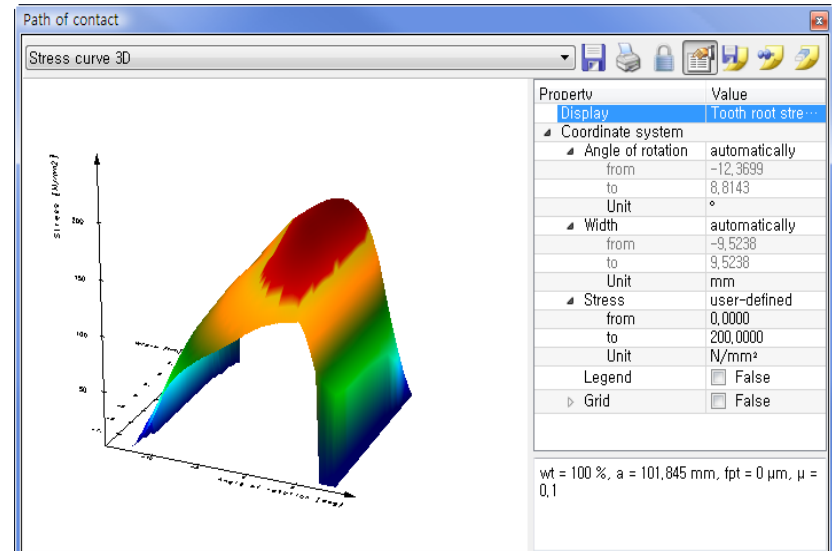
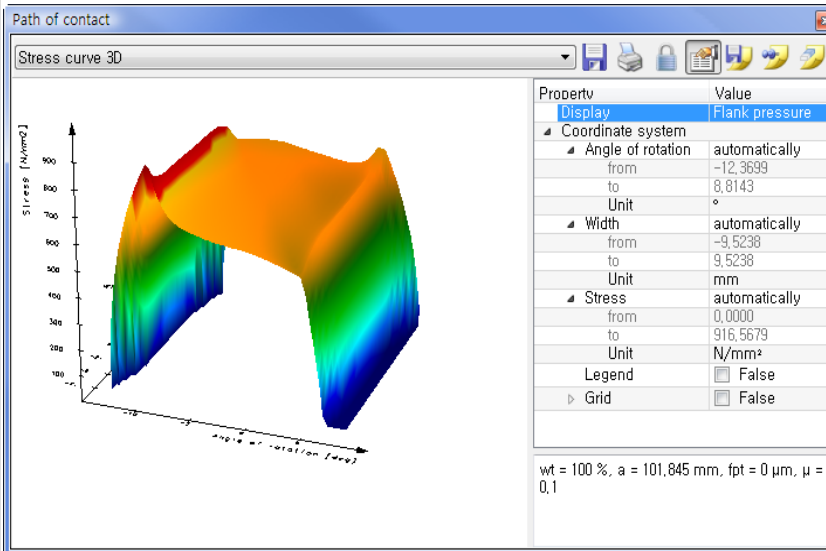
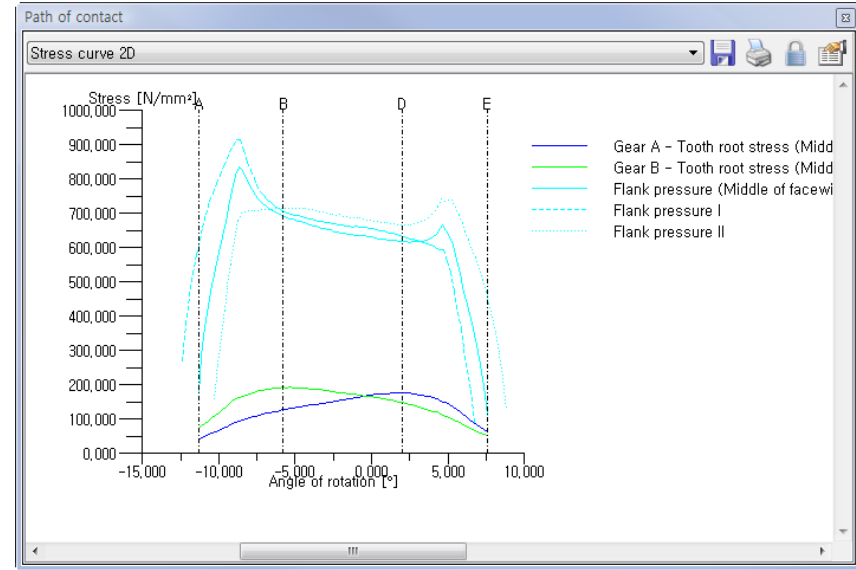
In order to calculate λ_{GFP} , you should define the failure load stage of the lubricant according to FVA-FZG-micropitting test.

Note: This is first implemented in KISSsoft in the world!



Calculation of stress curve

The stresses are calculated acc. to the calculation method but the local normal force and the local curvature of the contact point is used.
The Hertzian stress is mostly slightly higher than DIN/ISO result because no contact ratio factor is considered.
(Factors Z_B , Z_D , Z_ε are not considered)
The root stresses are mostly similar for standard tooth forms.

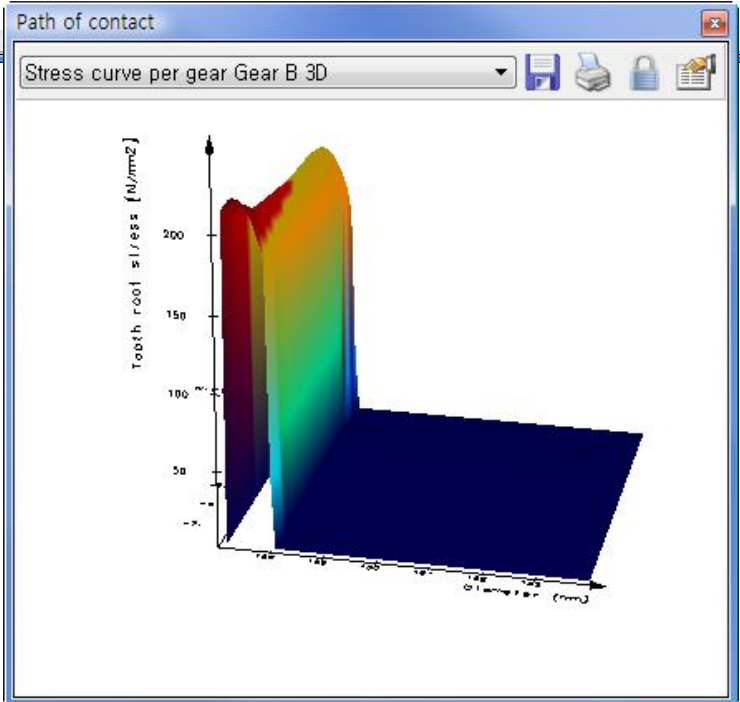
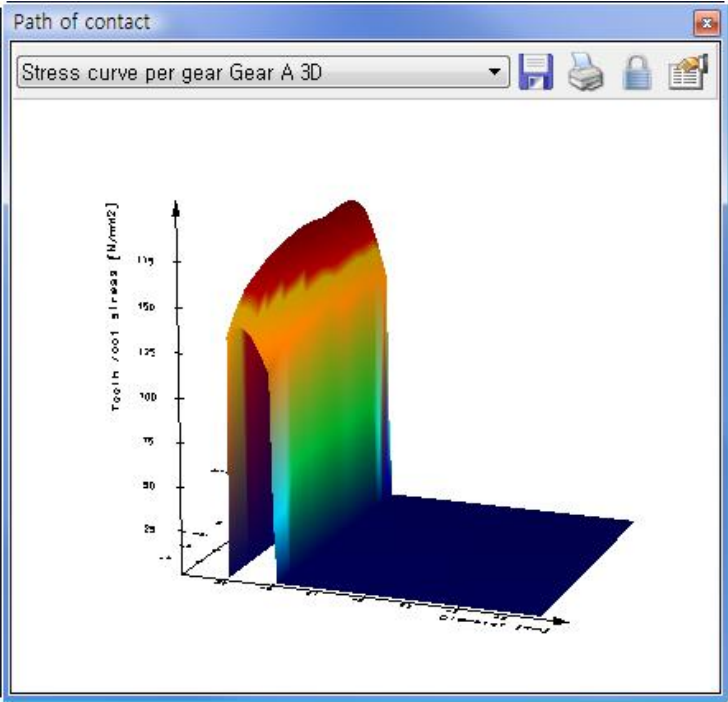
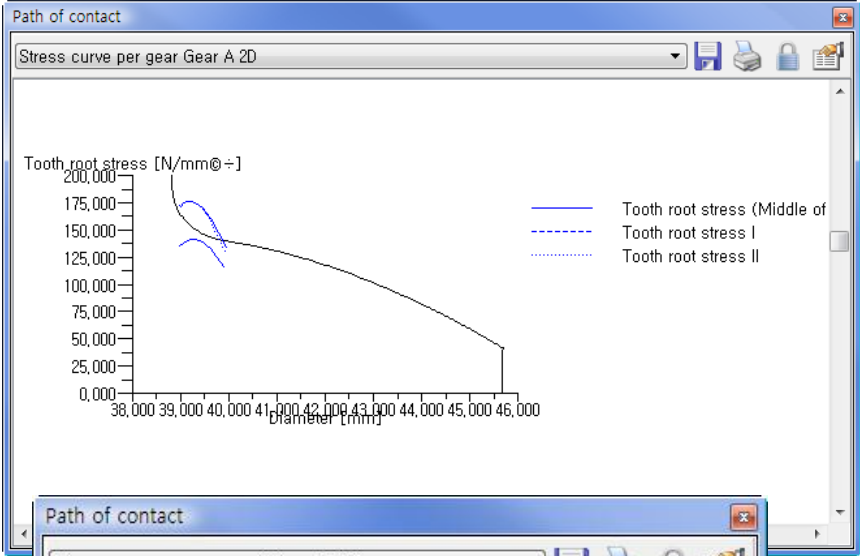


Calculation of stress curve per gear

Calculation of path of contact

The stress curve per gear shows the tooth root stress by FEM.

The values are given for the face width.



The gear pairs with wide face width and/or high helix angle (high overlap ratio) gives bad result. This comes from the limited number of sections for the calculation.

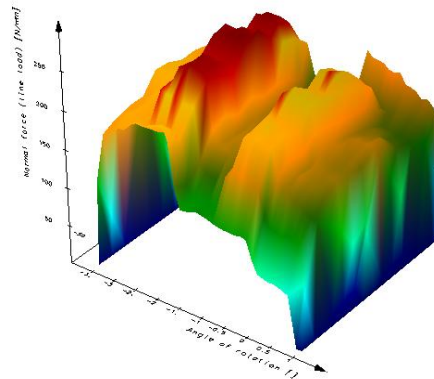
It will be fixed in the next patch to set more number of sections for the calculation.

We will find a solution to give reasonable result without increasing computation time (in the next patch?).

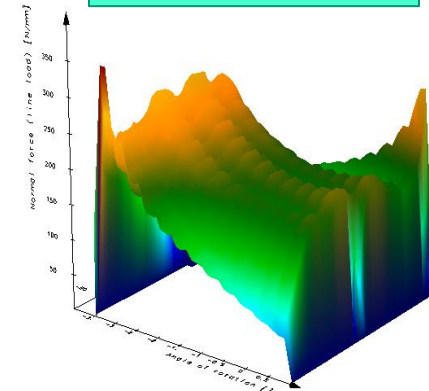
Possible remedy: Set the Accuracy of Calculation to High or Very high.

Module: 8
Face width: 150
Helix angle: 50.6525
Overlap ratio: 2.0

No. of Sections: 21



No. of Sections: 61



Module: 1.5
Face width: 50
Helix angle: 16.424
Overlap ratio: 3.0

