# Cylindrical Gears – Root Stresses and Root Strength

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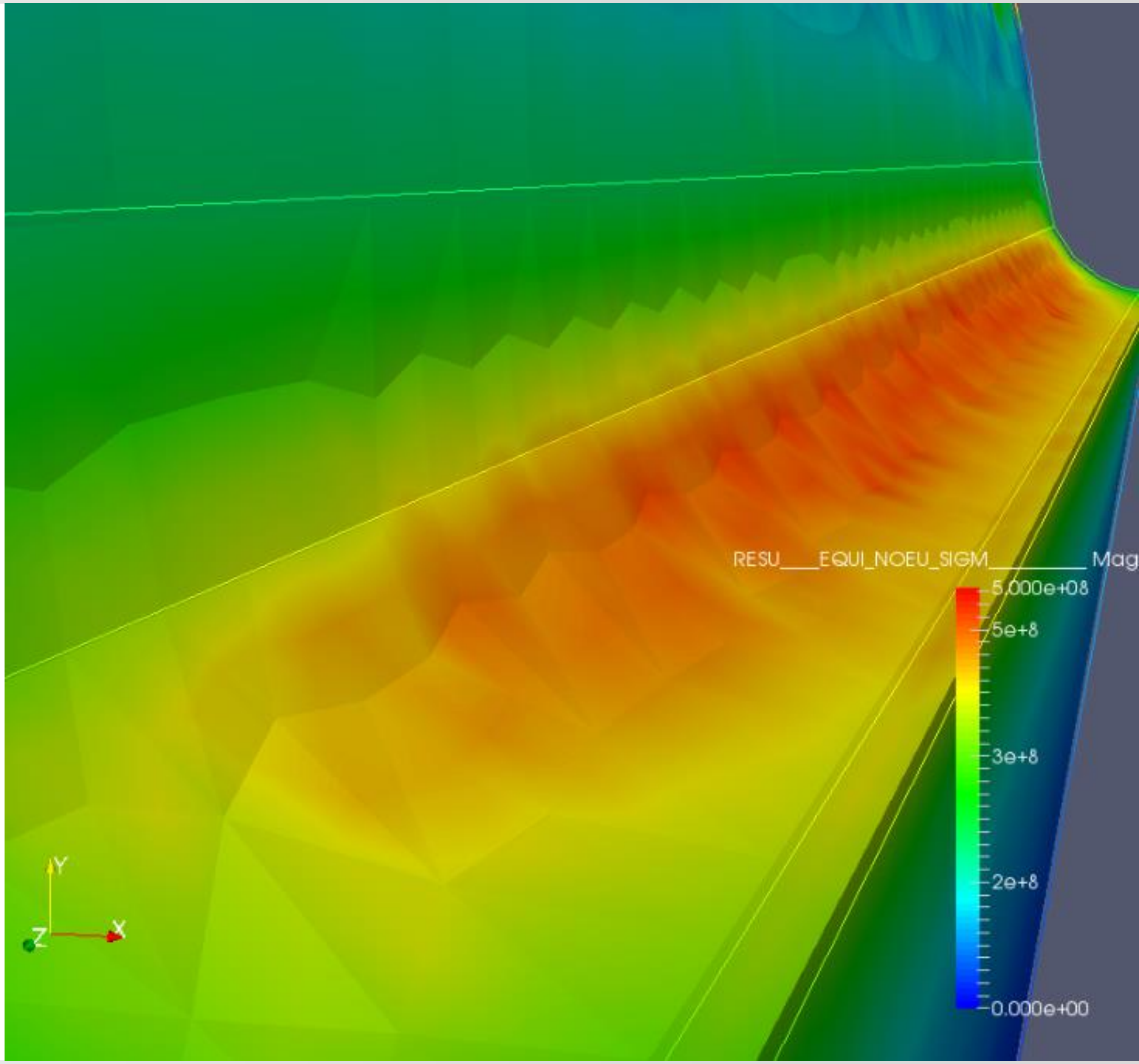
# Content



# Bending failure

- What is Bending Failure?

  Fatigue cracks usually initiated in the root fillets on the loaded flank side caused by high tensile bending stress.

  The crack grows to critical size in a direction perpendicular to the maximum tensile stress / to the surface.

  If the crack propagates intermittently, it may leave a pattern of “beach marks”.

- Causes

  High tensile stress from bending

  Reduced root thickness by undercut

  Final machining notch of low radius of curvature

  Low grade material

  Improper heat treatment (Improper case hardness and depth vs. core hardness)

Figure 43 – Bending fatigue crack

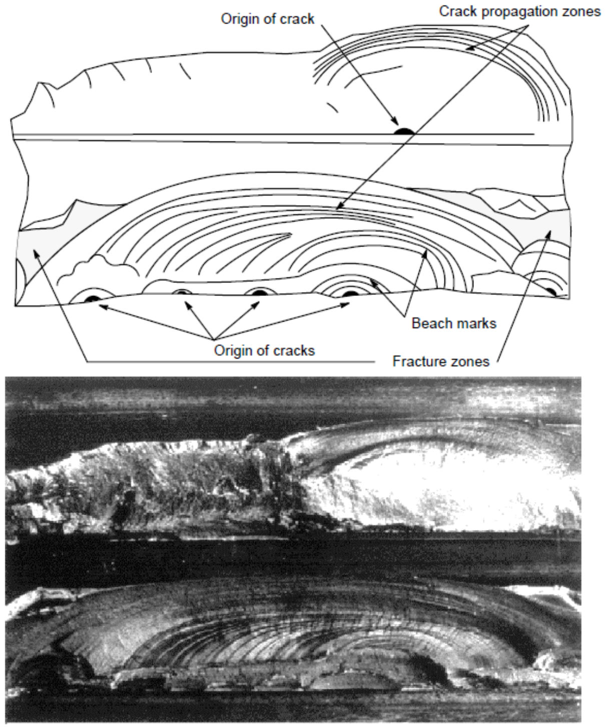


Figure 51 – Fatigue of two spur teeth

ANSI/AGMA 1010-E95,
Appearance of Gear Teeth – Terminology of Wear and Failure

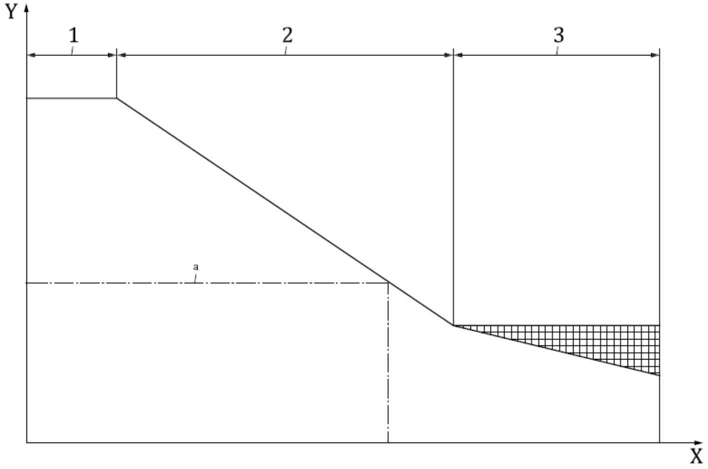
# Strength vs. stress

Target: Permissible bending stress ≥ bending stress

Permissible stress is a function of part strength and minimum (required, user defined) safety factor

$\sigma FP = \sigma FG / SFmin$

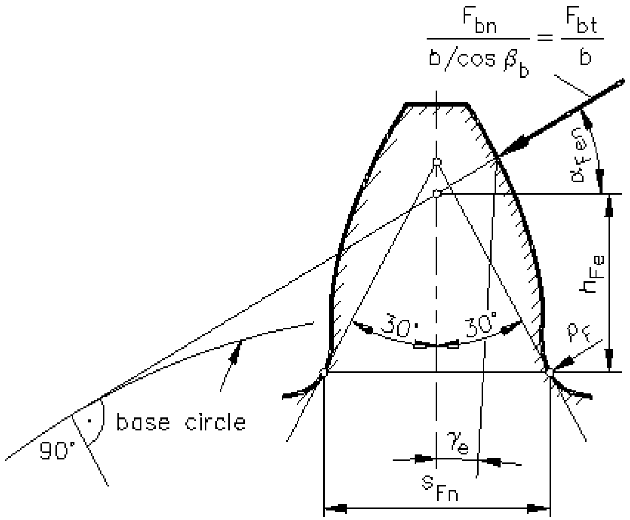
$\sigma FG = \sigma Flim * YST * YNT * Y\delta relT * YRrelT * YX$



Bending stress is a function of part geometry, material properties, external load and load amplification factors

$\sigma F = \sigma F0 * KA * K\gamma * Kv * KF\alpha * KF\beta$

$\sigma F0 = F_t / (b * m_n) * YF * YS * Y\beta * YB * YDT$



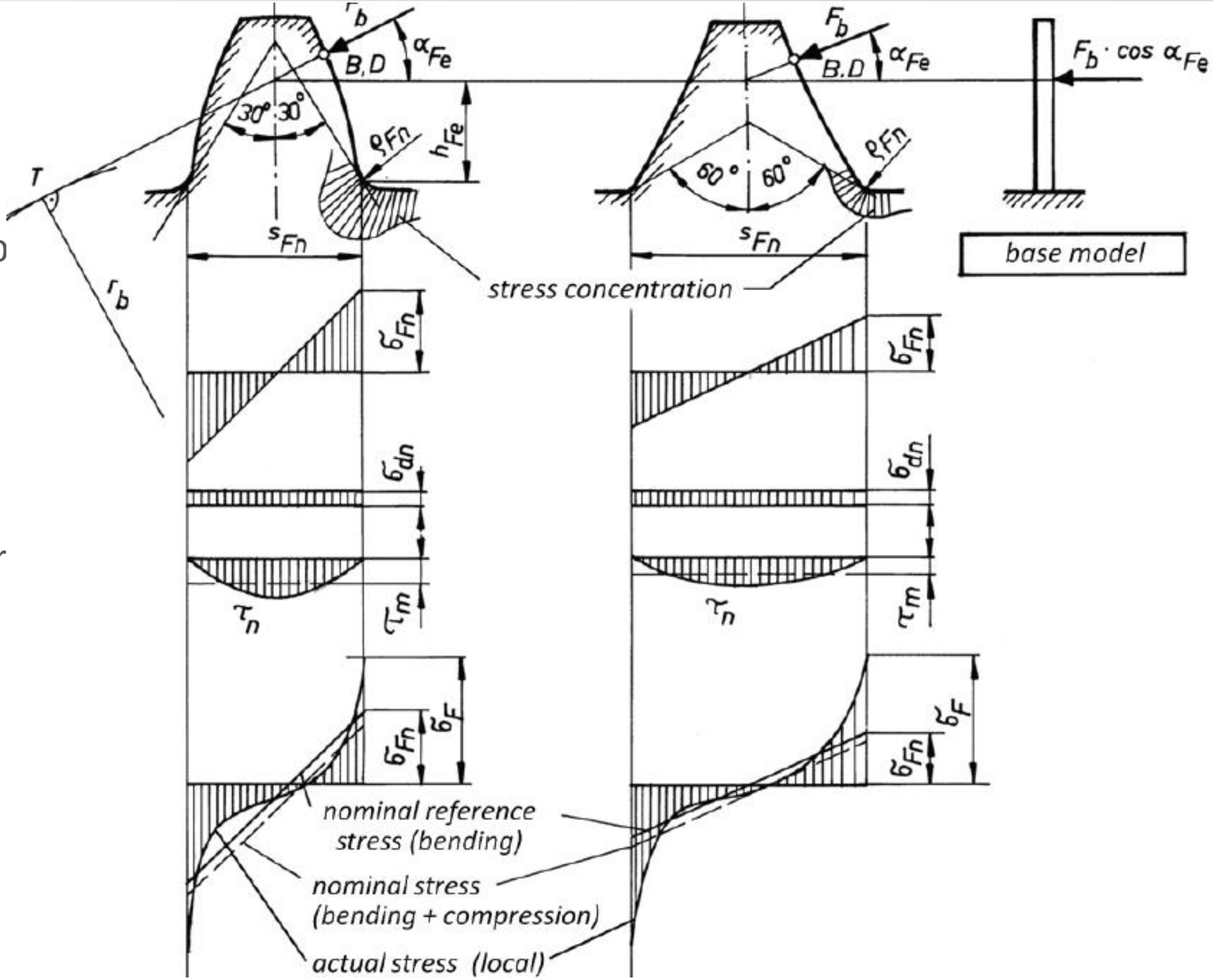
# Stress components

For internal and external gears

Local stress is calculated (stress concentratio is considered)

Local strength is used as reference (stress concentration was present in tests when gear root strength was measured)

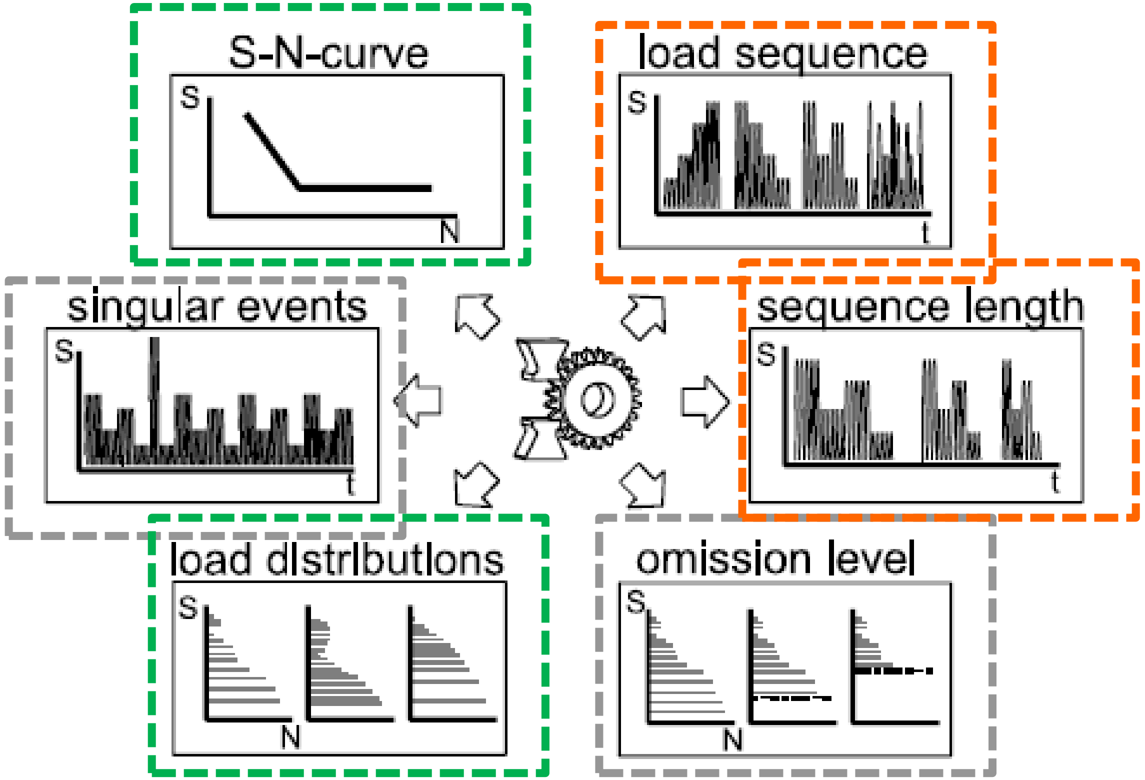
For solid gears, with sufficient amount of material below the root diameter (see additional factor YB below)



# Effects of variable loads

## Root fatigue life under variable load, effects in ISO 6336

| | |
|---|---|
| S-N curve | considered |
| Load sequence | not considered |
| Load sequence length | not considered |
| Singular events | partially considered |
| Torque reversals | partially considered |
| Omission level | partially considered (if modified S-N curve is used) |



→ Duration and sequence of load cycles not considered

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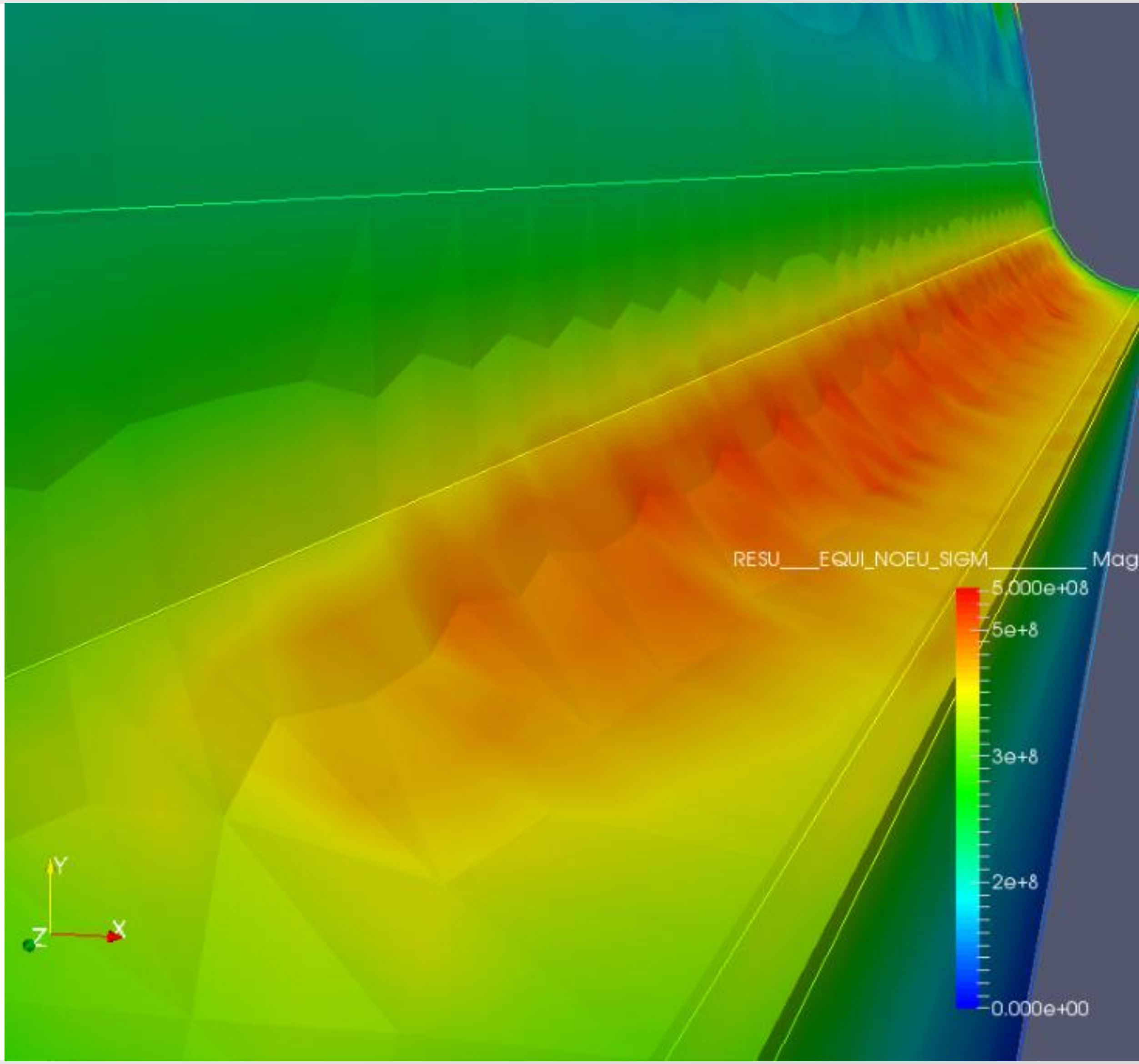


# Tooth form factor YF

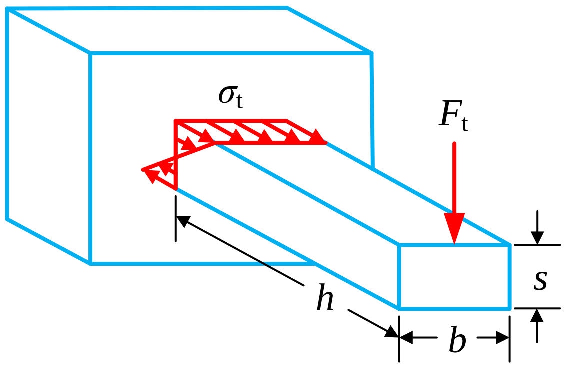
Calculation of YF for different tooth thicknesses

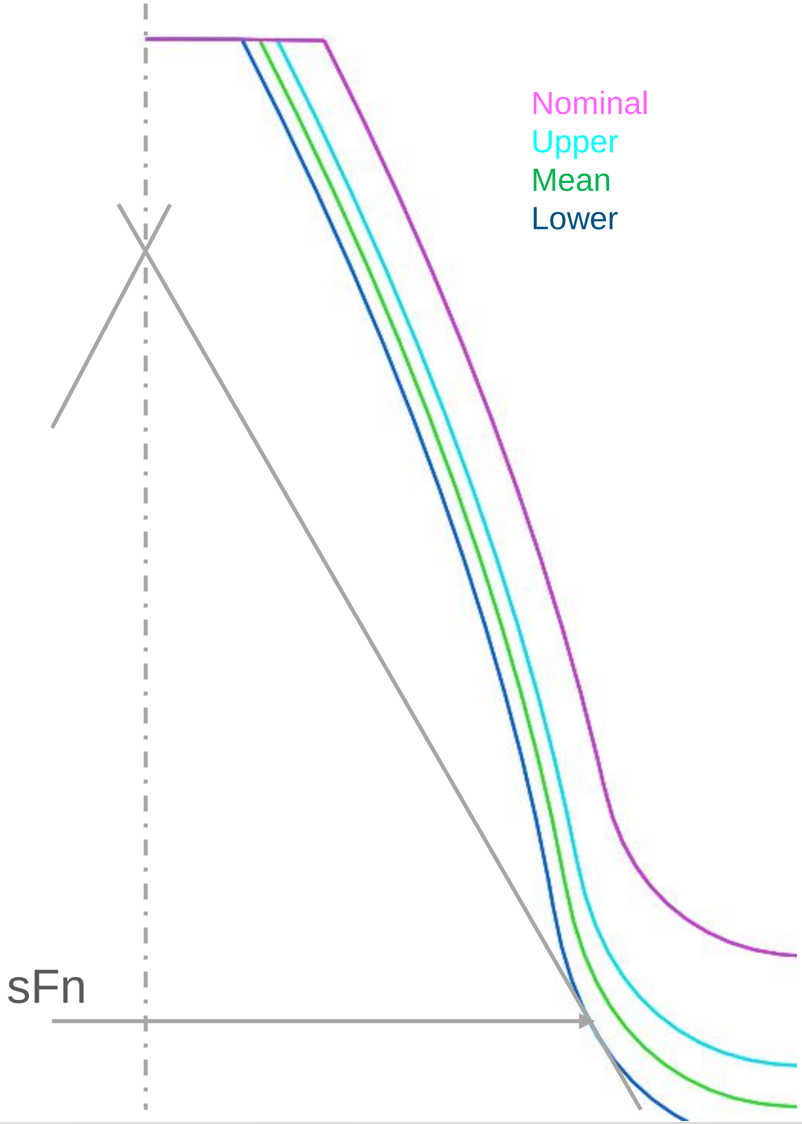
ISO 6336:2019: tooth thickness sFn (corresponding to beam thickness s below) of tooth with lower limit in thickness tolerance range shall be used.

$F_t / (b * m_n) * YF$ corresponds to $\sigma t$ below

Cantilever beam model

$$\sigma_t = \frac{6F_t \cdot h}{b \cdot s^2}$$



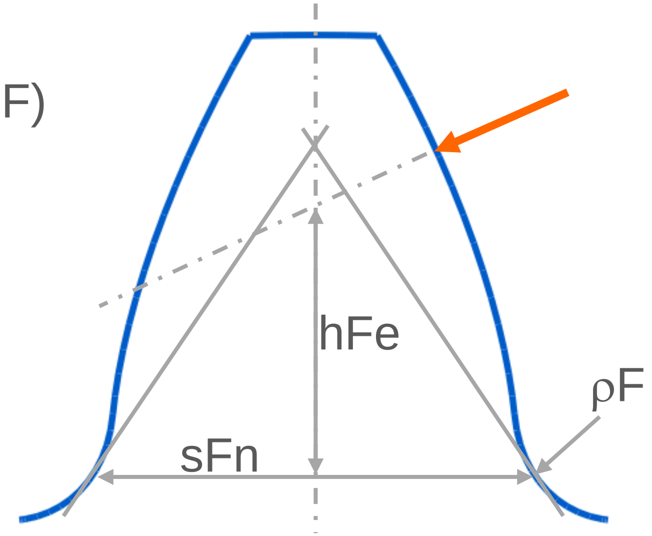


# Stress correction factor YS

Accounts for local stress increase due root rounding

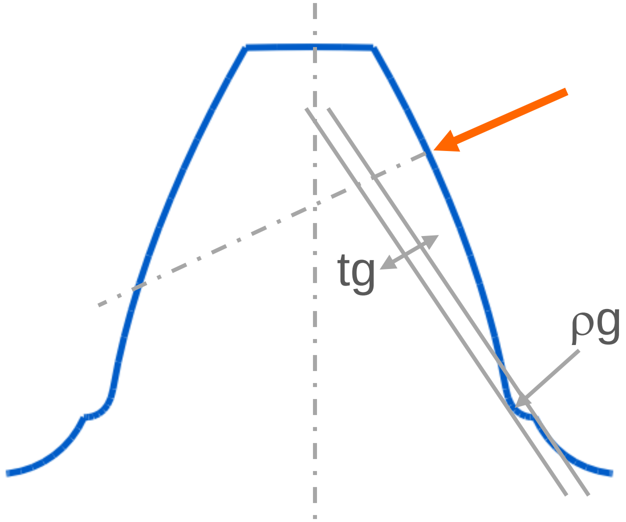
For root shapes without final machining notch

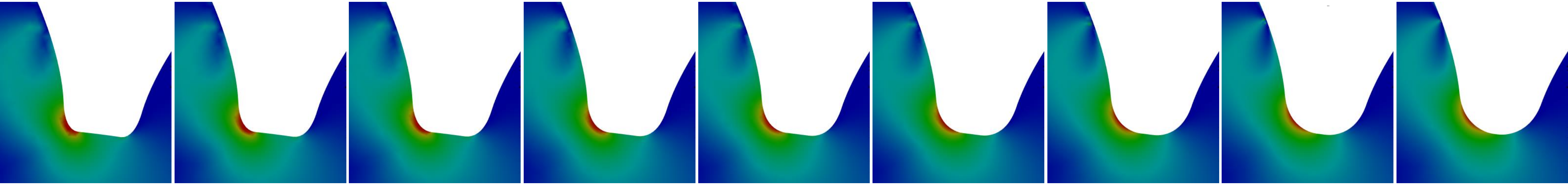
$YS = f(hFe, sFn, \rho F)$



If final machining or grinding notch is present

$Ysg = f(YS, tg, \rho g)$





# Root stresses in internal gears

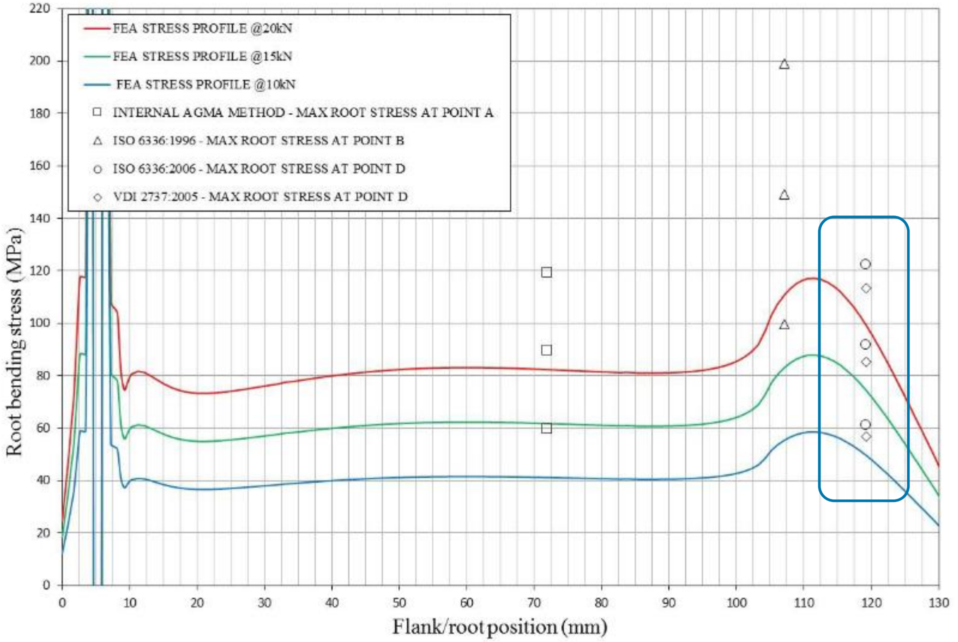
YF along ISO 6336:1996, ISO 6336:2006, ISO 6336:2019, VDI 2737:2016 (thick rim), AGMA 2000 series [39]

ISO 6336-3:1996 root stress at 30deg, root shape calculated using virtual rack type tool

ISO 6336-3:2006 root stress at 60deg, root shape calculated using virtual rack type tool

VDI 2737:2016, ISO 6336-3:2019 root stress at 60deg, root shape calculated using shaping cutter

| **gear x*** | **pinion cutter x0** | $\rho_{fP}$ | $\rho_{fPv}$ | $\rho_F$ **ISO 6336-3:2006 / 2007-02** | $\rho_F$ **2007-04** | $\rho_F$ **measured** | $\rho_F$ **VDI 2736:2016** | $\rho_F$ **ISO 6336-3:2019** | **Deviation % (2007/2019)** |
|---|---|---|---|---|---|---|---|---|---|
| -0.75 | 0.1 | 0.2 | 0.32 | 0.201 | 0.426 | 0.233 | 0.233 | **0.233** | 45% |
| -0.75 | 0.0 | 0.2 | 0.296 | 0.175 | 0.403 | 0.220 | 0.220 | **0.220** | 45% |
| 0.00 | 0.1 | 0.2 | 0.332 | 0.298 | 0.364 | 0.284 | 0.286 | **0.286** | 21% |
| 0.00 | 0.0 | 0.2 | 0.310 | 0.274 | 0.343 | 0.265 | 0.264 | **0.264** | 23% |

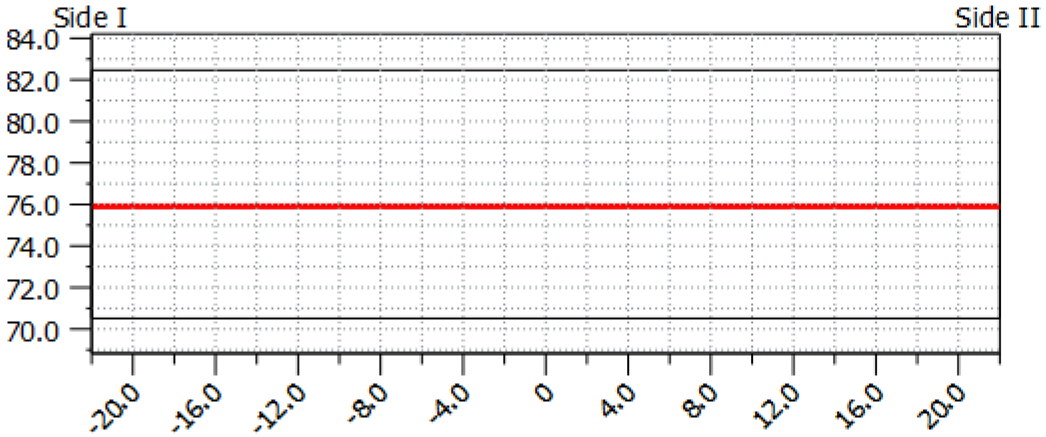


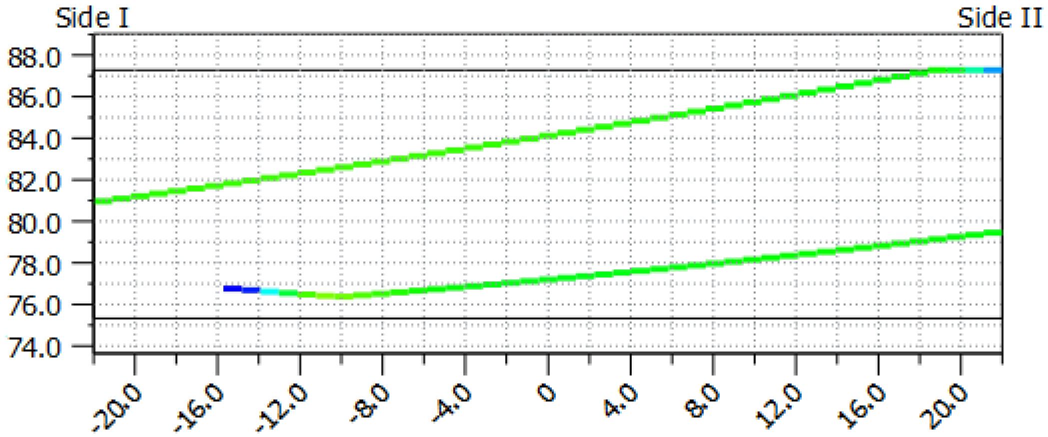
# Helix angle factor Yβ

## Virtual spur gear vs. actual helical gear

Conversion of tooth root stress of virtual spur to that of the actual helical gear.

Oblique orientation of contact lines is considered (lower tooth root stress).





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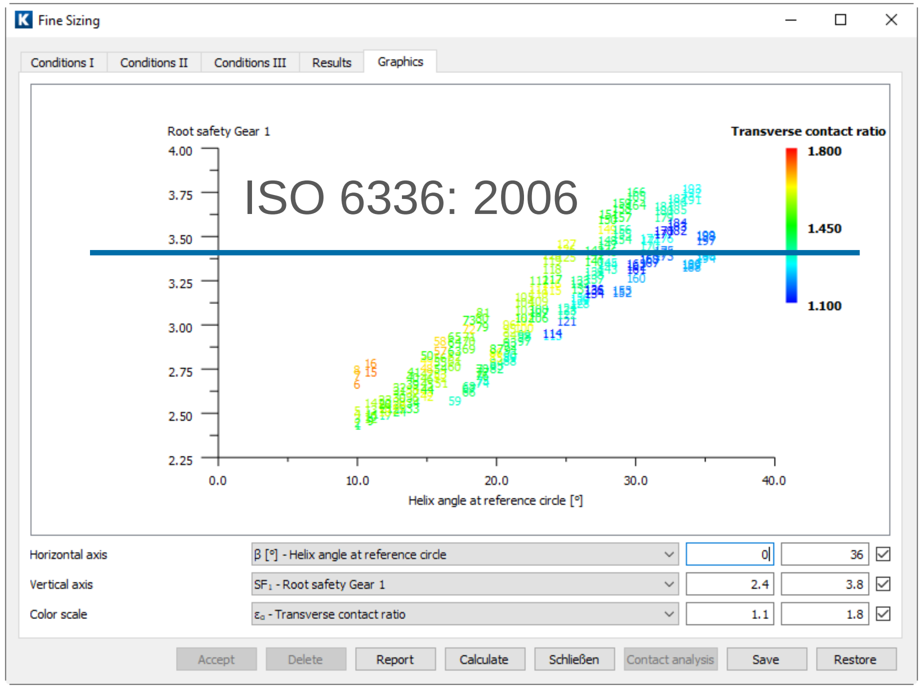
# Helix angle factor Yβ

Changes in ISO 6336-3, from 2006 edition to 2019 edition

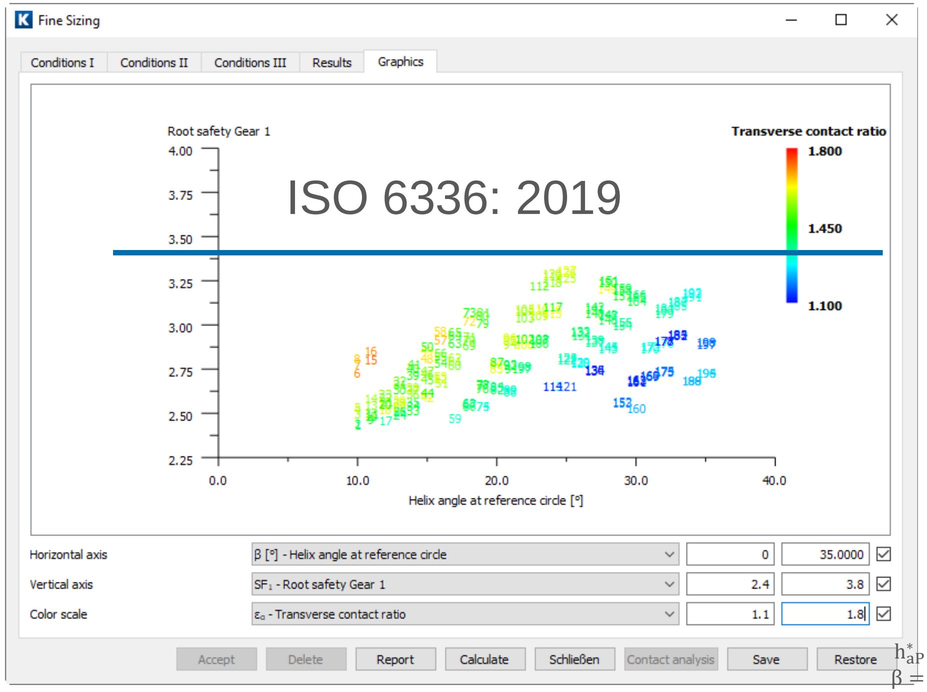
Effect of Yβ has been changed considerably



Horizontal axis: β [°] - Helix angle at reference circle, 0, 36 ☑

Vertical axis: SF₁ - Root safety Gear 1, 2.4, 3.8 ☑

Color scale: εα - Transverse contact ratio, 1.1, 1.8 ☑



Horizontal axis: β [°] - Helix angle at reference circle, 0, 35.0000 ☑

Vertical axis: SF₁ - Root safety Gear 1, 2.4, 3.8 ☑

Color scale: εα - Transverse contact ratio, 1.1, 1.8 ☑

$h^*_{aP} = 1.0 \quad h^*_{fP} = h^*_{aP} + 0.25 \quad \rho^*_{fP} = 0.25$

$\beta = [10°, 11°, \ldots, 35°]$, helix angle is varied

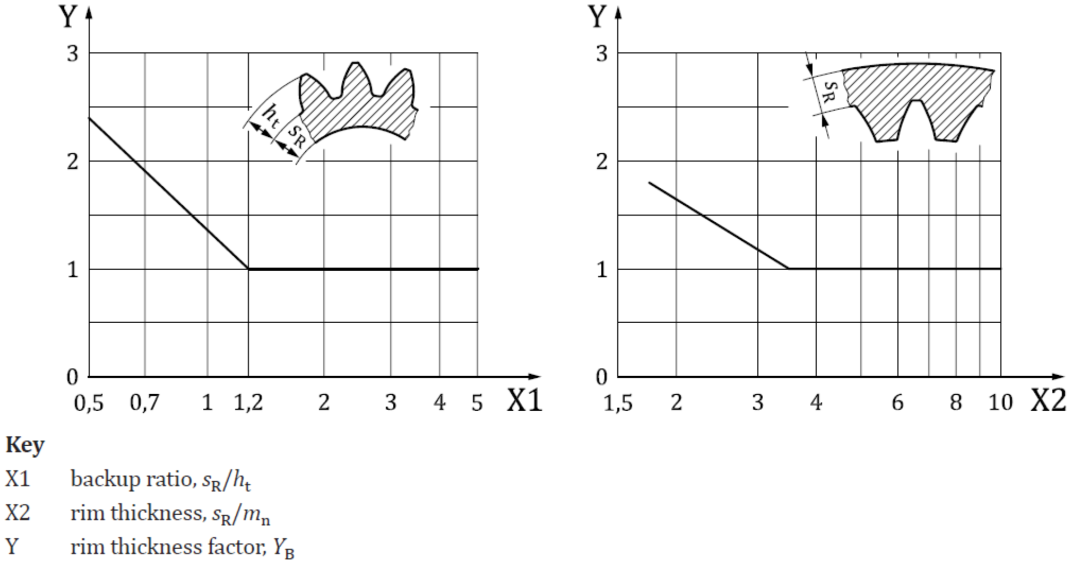
$a = 303\ \mathrm{mm}$

$m_n = 6\ \mathrm{mm}$

# Rim thickness factor YB

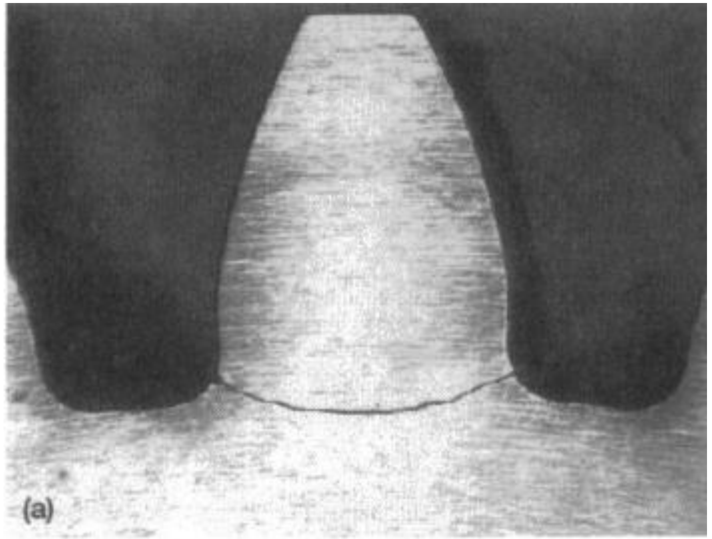
Backup ratio = rim thickness / tooth height

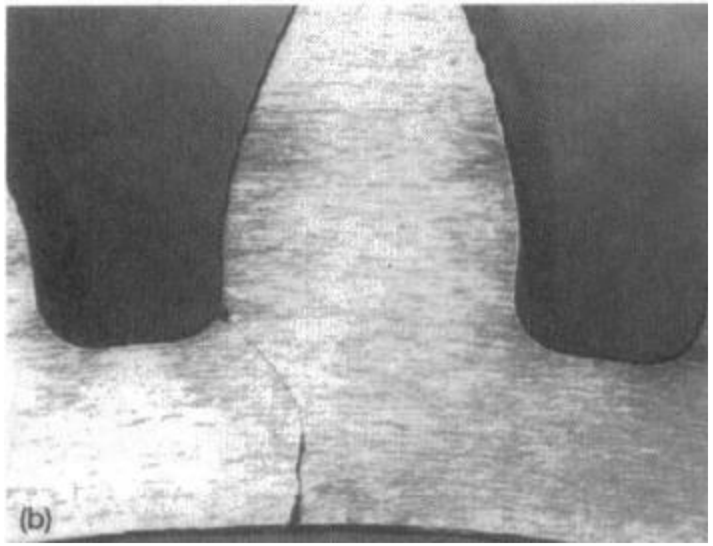
Where the rim thickness is not sufficient to provide full support for the root, the location of bending fatigue failure may be through the gear rim, rather than at the root fillet. YB is a simplified factor to rate thin rimmed gears, when no detailed calculation of stresses in both tension and compression fillets are available. [38]



**Key**

| | |
|---|---|
| X1 | backup ratio, $s_R/h_t$ |
| X2 | rim thickness, $s_R/m_n$ |
| Y | rim thickness factor, $Y_B$ |

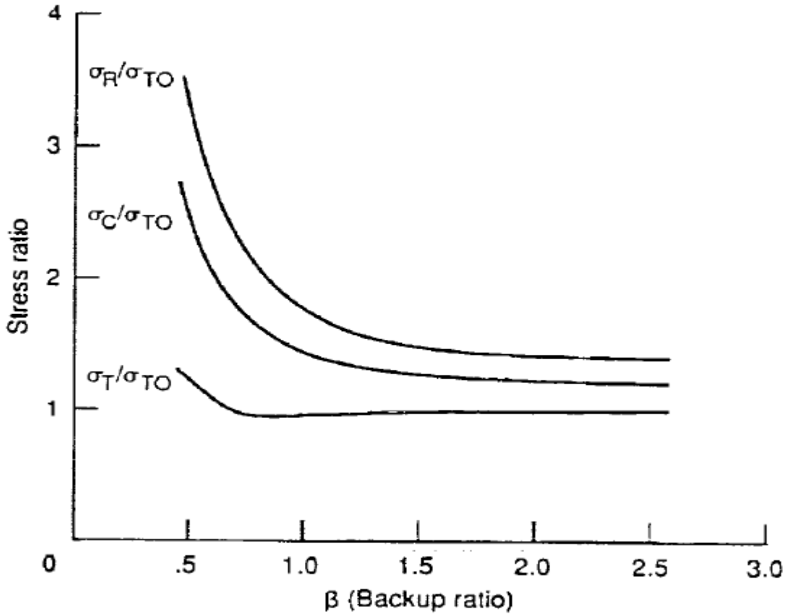




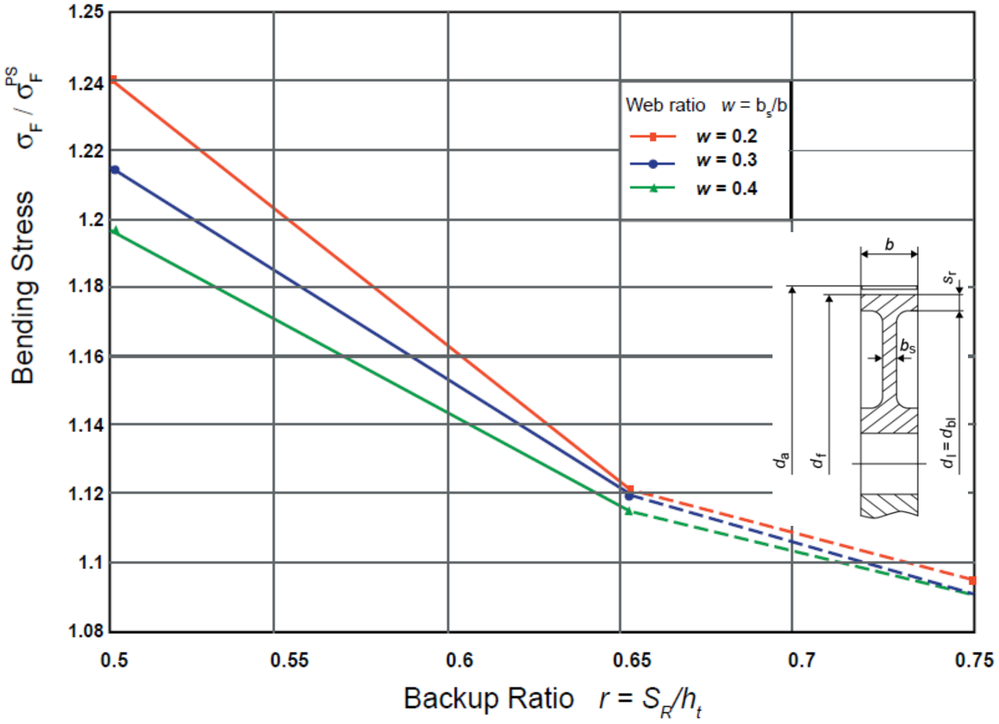
# Rim thickness factor YB

Experimental and analytical studies

If sr/h > 1.3, effect may be neglected [36]



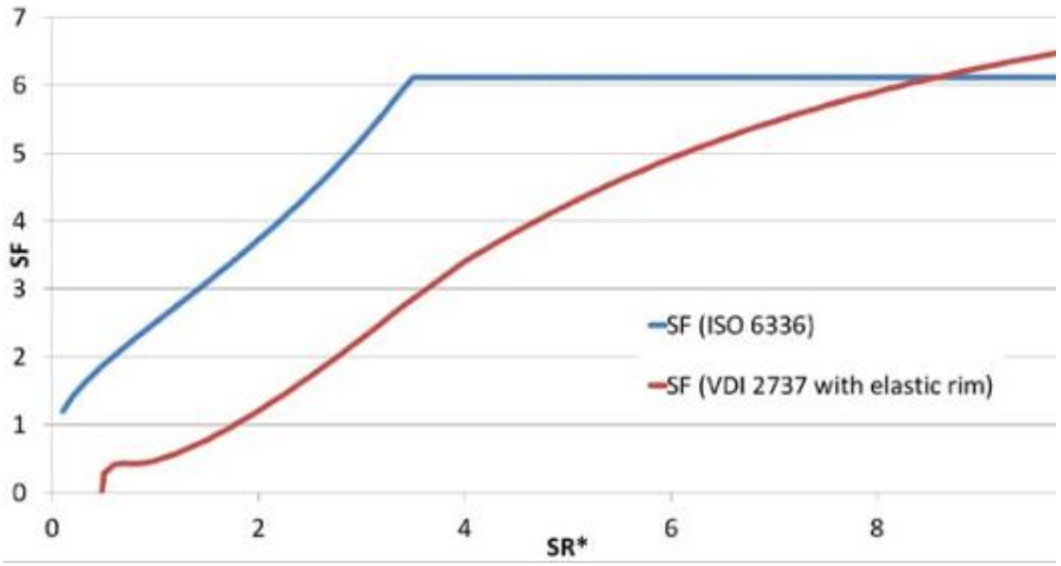
Effect of web thickness may be superimposed [38]



# Rim thickness factor YB

## VDI 2737

For internal gears, considers gear body bending stresses. However, now ring gear bores or web, use FEM for complex shapes, [37], [40].



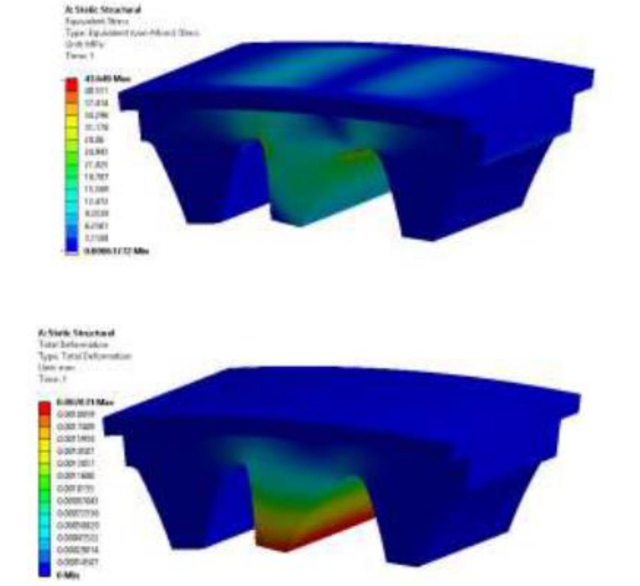


Figure 19. Effect of rim shape A on (a) bending stress and (b) displacement

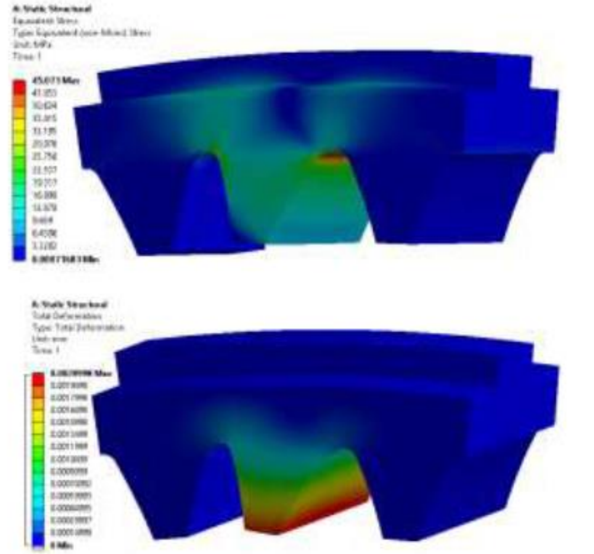
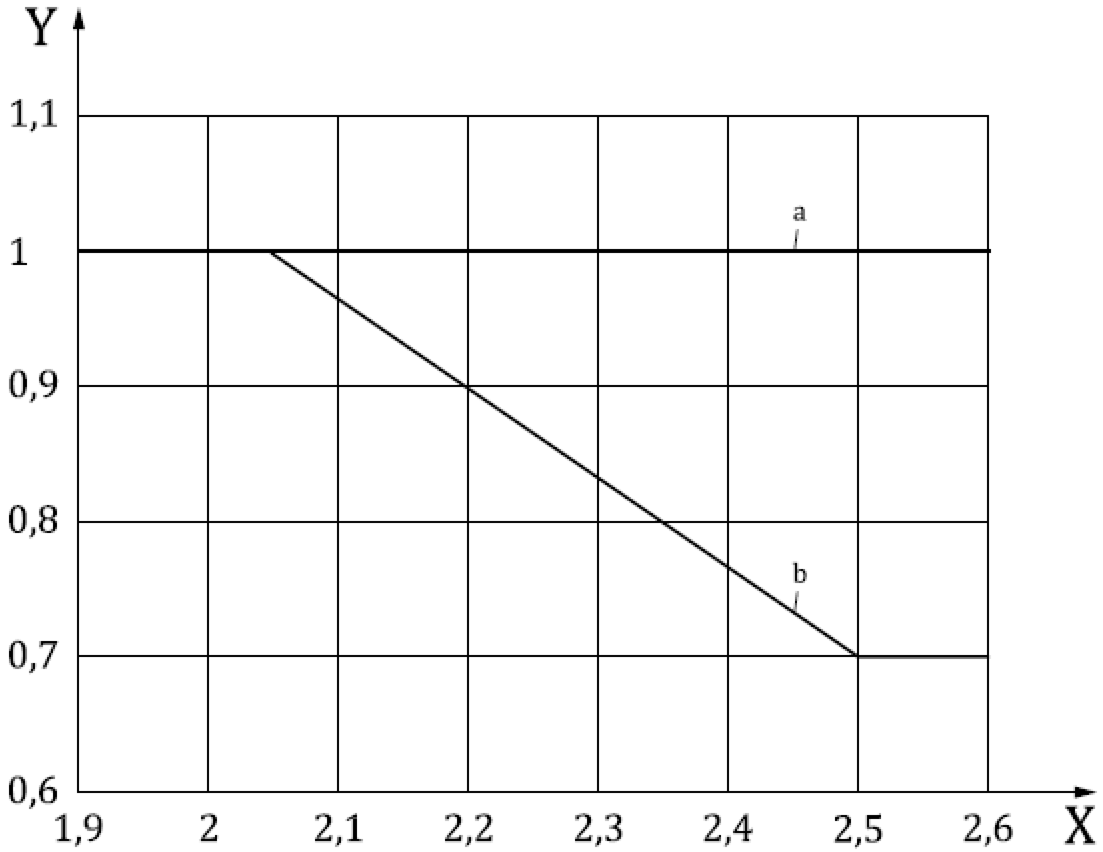


Figure 21. Effect of rim shape B on (a) bending stress and (b) displacement

# Deep tooth factor YDT

## Load sharing of precise, deep tooth gears

ISO Tolerance Class ≤ 4 and 2.00 ≤ εαn < 2.50 and suitable profile modification → nominal tooth root stress, σF0, is adjusted by the deep tooth factor, YDT



**Key**

| | |
|---|---|
| X | virtual contact ratio, $\varepsilon_{\alpha n}$ |
| Y | deep tooth factor, $Y_{DT}$ |
| a | ISO Tolerance Class > 4. |
| b | ISO Tolerance Class ≤ 4. |

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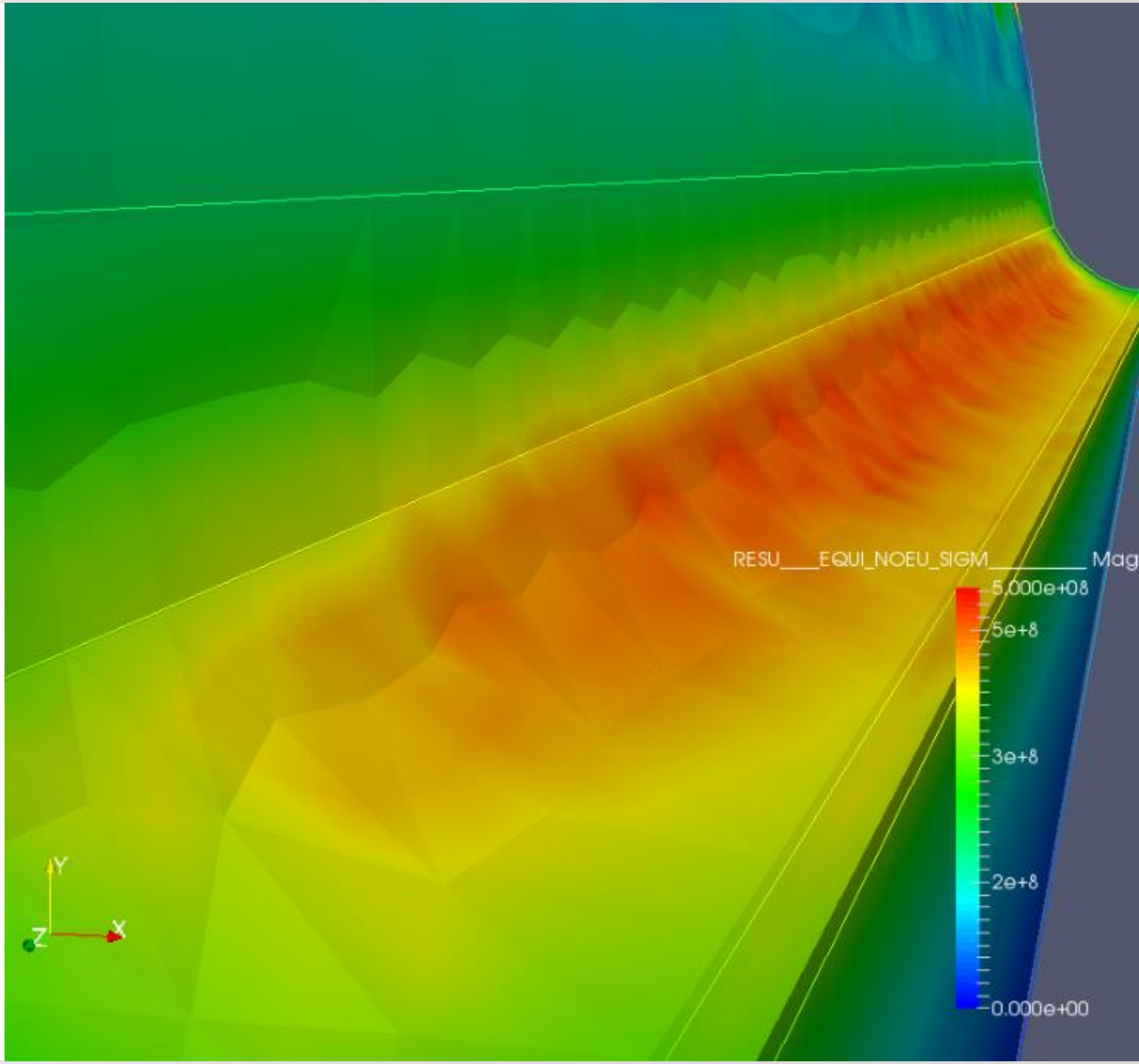
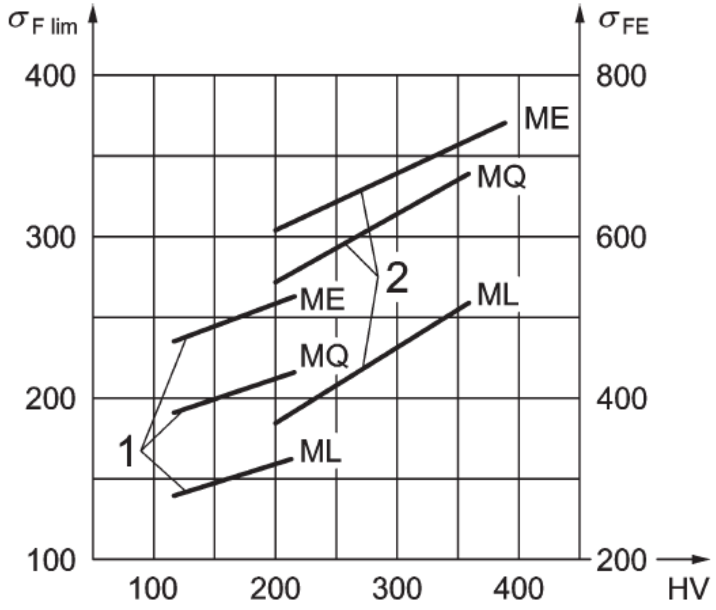


# Gear Root Strength

Standards for gear rating: no specific materials but material classes

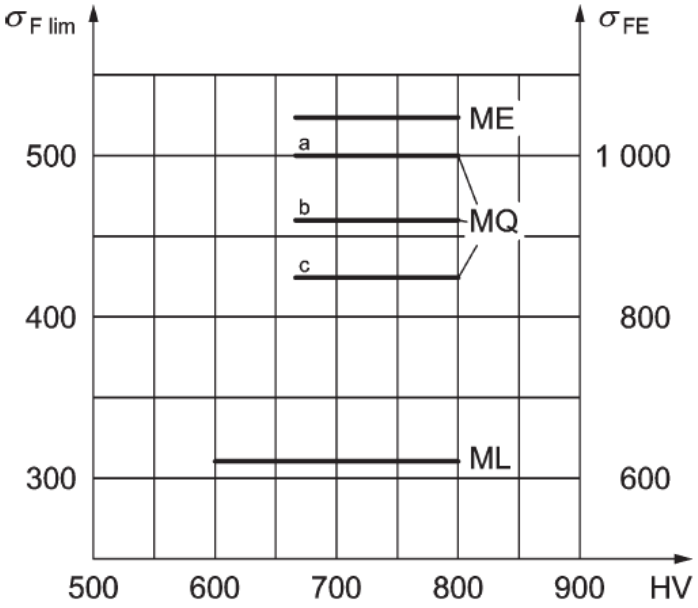
For through hardened wrought steels, fatigue limit of root is a linear function of the hardness

$\Delta \sigma$ Flim / $\Delta$HV = constant



For case hardened wrought steels, fatigue limit of root is a constant function of the case hardness

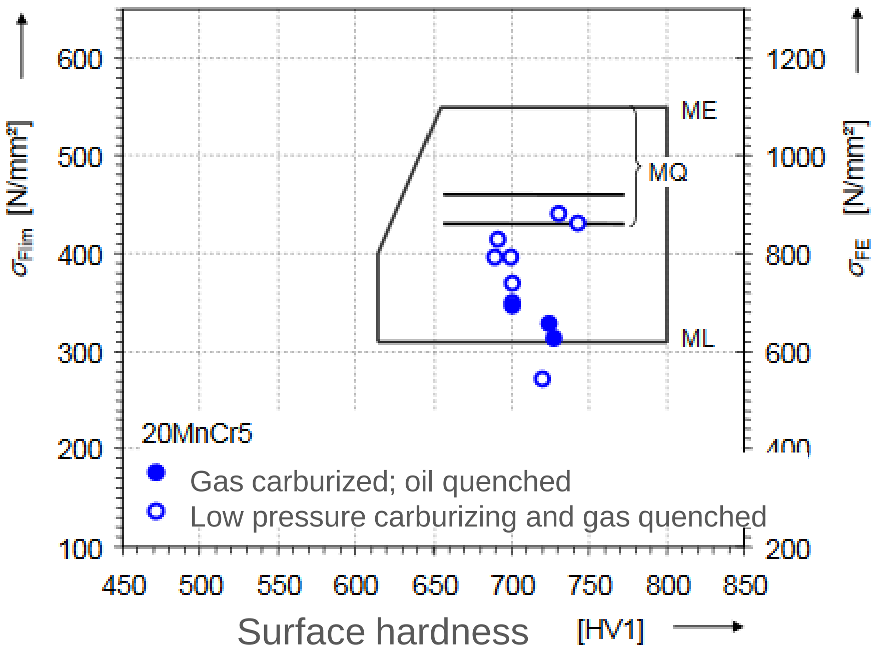
$\sigma$ Flim = constant



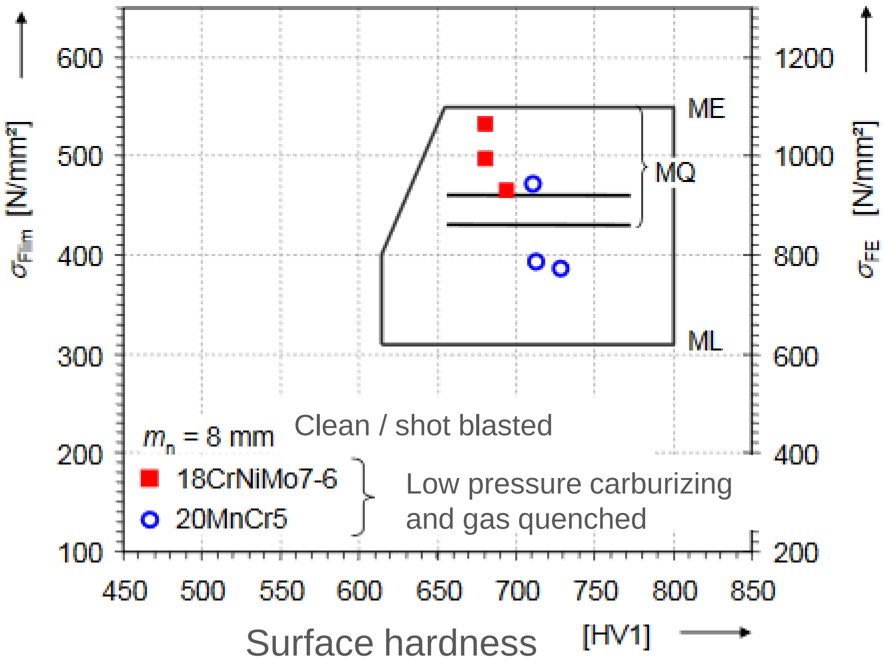
# Gear Root Strength

Material and heat treatment however have an influence

Root strength, same material, two heat treatments



Root strength, same heat treatment, two materials

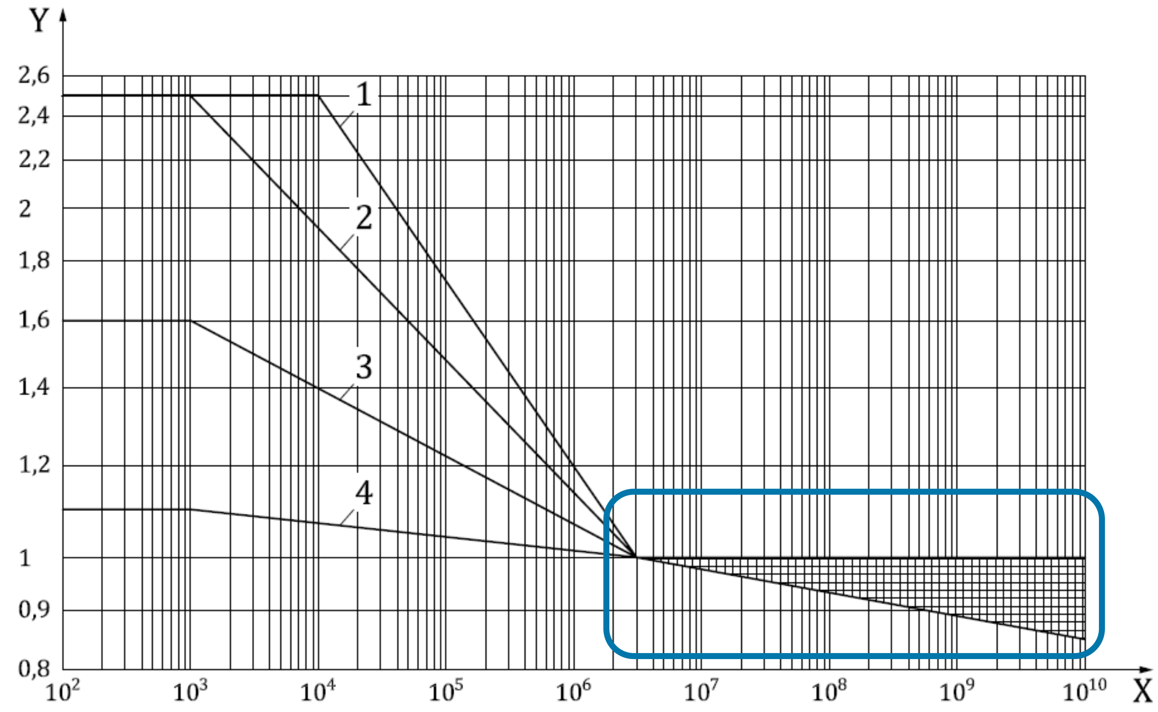


# Life factor YN

S-N curve is defined through life factor

The life factor accounts for the higher tooth root stress, which may be tolerable for a limited life (number of load cycles), as compared with the allowable stress at 3 × 10e6 cycles

The allowable stress numbers are established for 3 × 10e6 tooth load cycles at 99 % reliability



**Key**

| | |
|---|---|
| X | number of load cycles, $N_L$ |
| Y | life factor, $Y_{NT}$ |
| 1 | GTS (perl.), St, V, GGG (perl. bai.) |
| 2 | Eh, IF (root) |
| 3 | NT, NV (nitr.), GGG (ferr.), GG |
| 4 | NV (nitrocar.) |

# Gear Root Strength

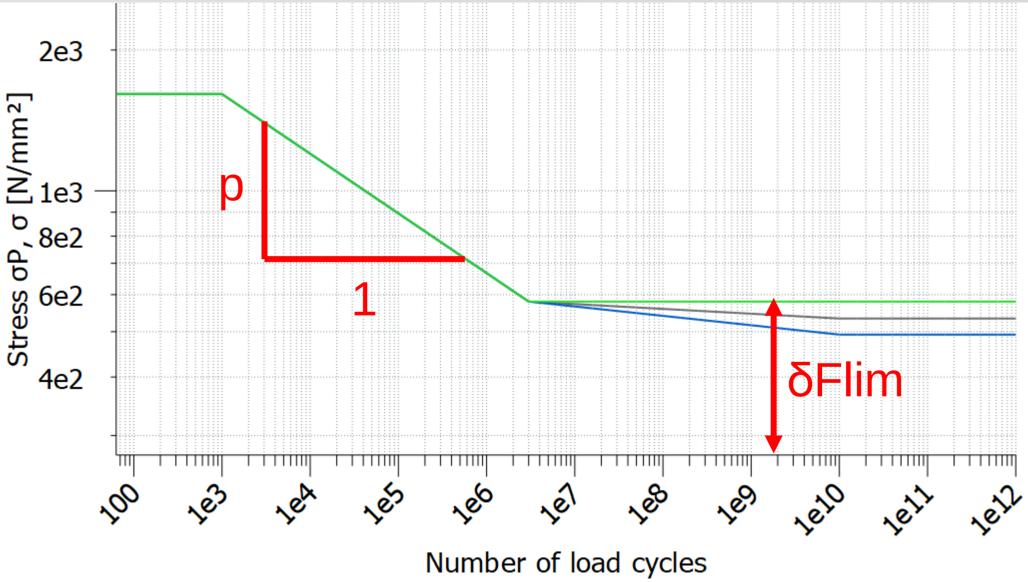
## S-N curves

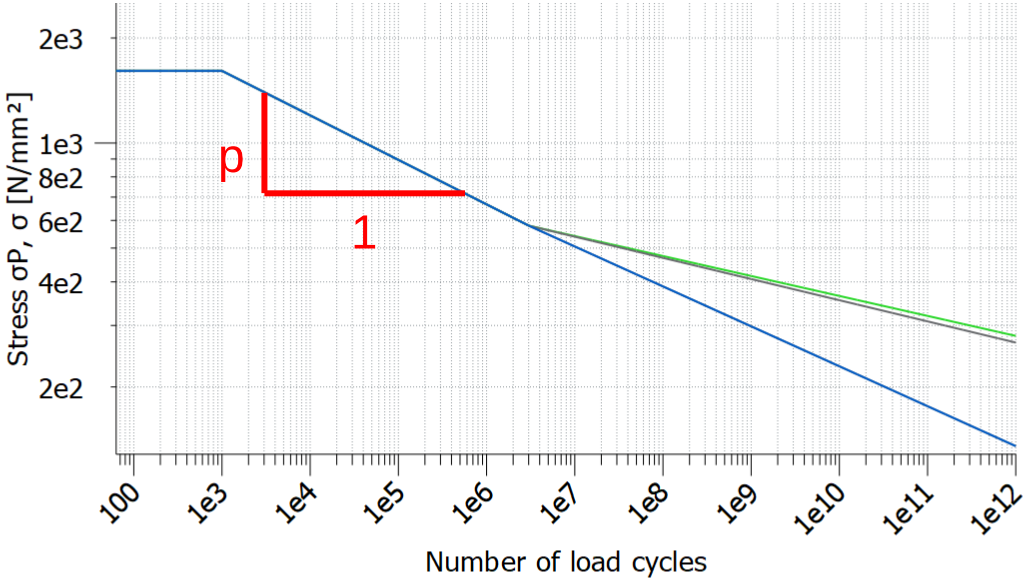
ISO 6336: endurance limit in range of 0.85…1.00 of σFlim (ISO 6336-3:2019, Table 3, footnote a)

ISO 9085: endurance limit for ML material grade = 0.85 * σFlim, for MQ material grade  = 0.92 * σFlim, for ME material grade = 1.00 * σFlim (ISO 9085:2002, Table 6)

Corten / Dolan: constant slope p over whole fatigue domain

Haibach: high cycle fatigue domain $p_h$ = 2 * p or $p_h$ = (2 * p)-1





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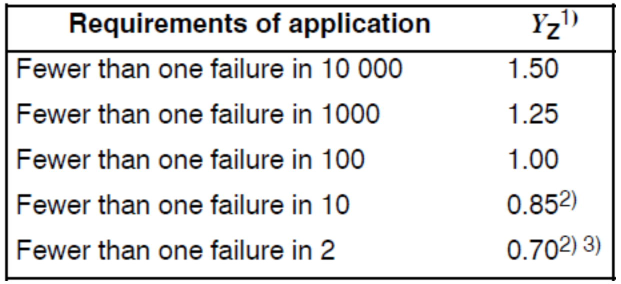
# Gear Root Strength

## Probabilistic approach

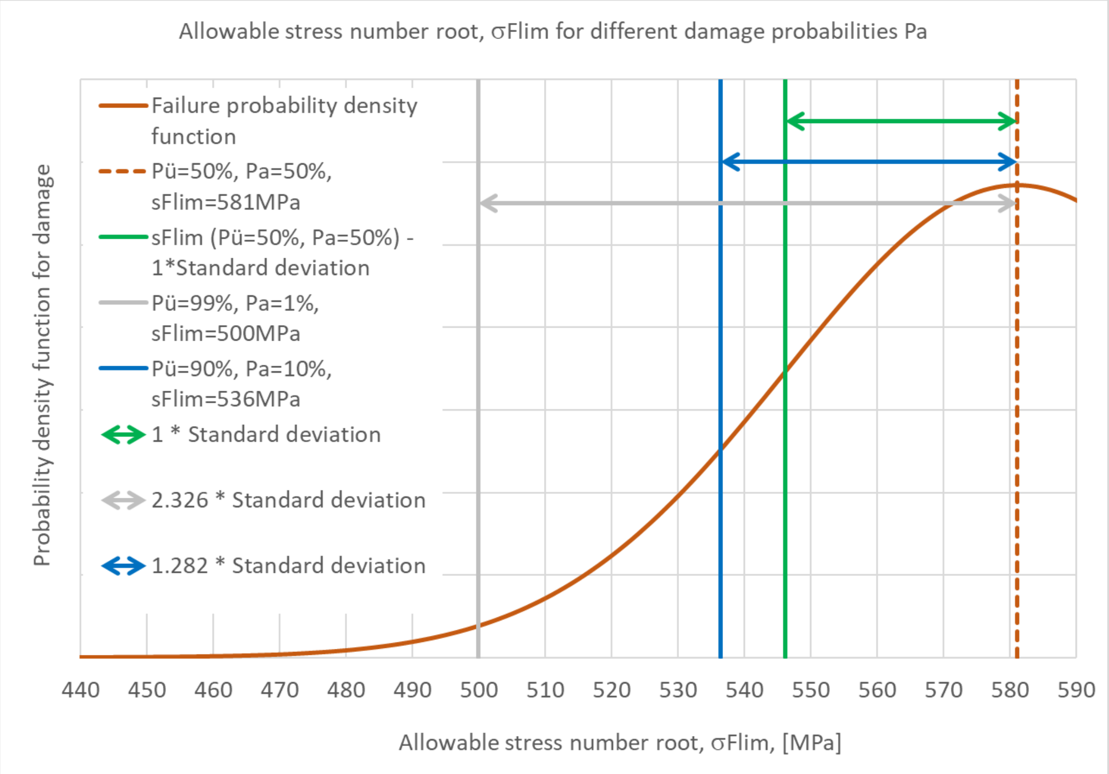
S-N curve as per ISO 6336 is based on 99% reliability or 1% probability of damage and 0.15 * $m_n$…0.20 * $m_n$ case depth.

For other reliability level than R = 0.99

- ANSI/AGMA 2101: uses reliability factor YZ
- ISO 6336: currently no details given, Hein proposes the introduction of a reliability level factor YZ [35]
- σFlim may be determined from standard deviation of strength measurement data [4]



| Requirements of application | $Y_Z$1) |
|---|---|
| Fewer than one failure in 10 000 | 1.50 |
| Fewer than one failure in 1000 | 1.25 |
| Fewer than one failure in 100 | 1.00 |
| Fewer than one failure in 10 | 0.85 2) |
| Fewer than one failure in 2 | 0.70 2) 3) |



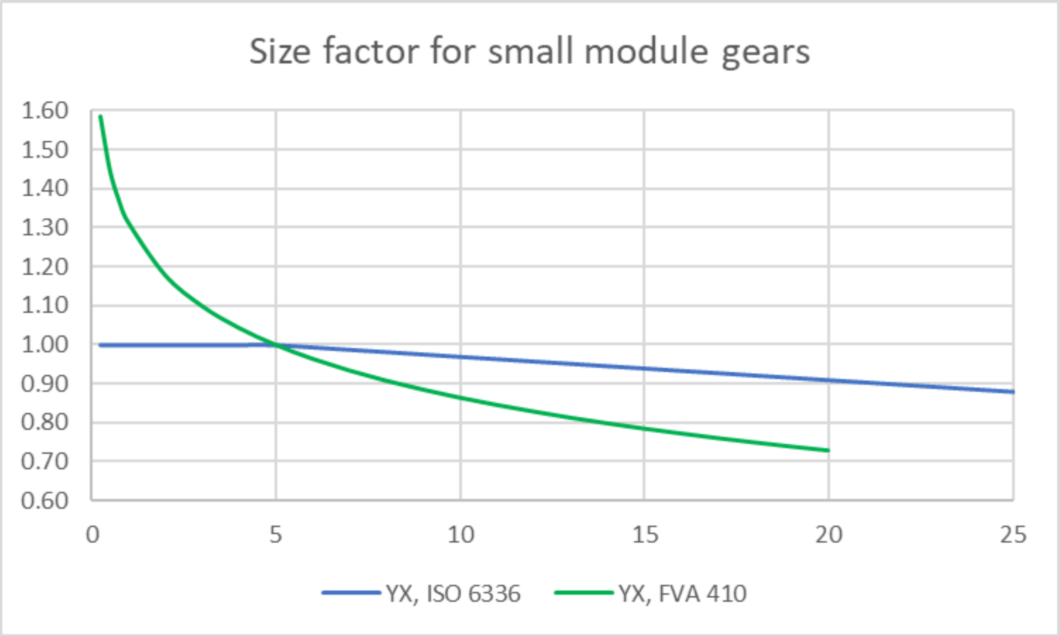
Allowable stress number root, σFlim for different damage probabilities Pa

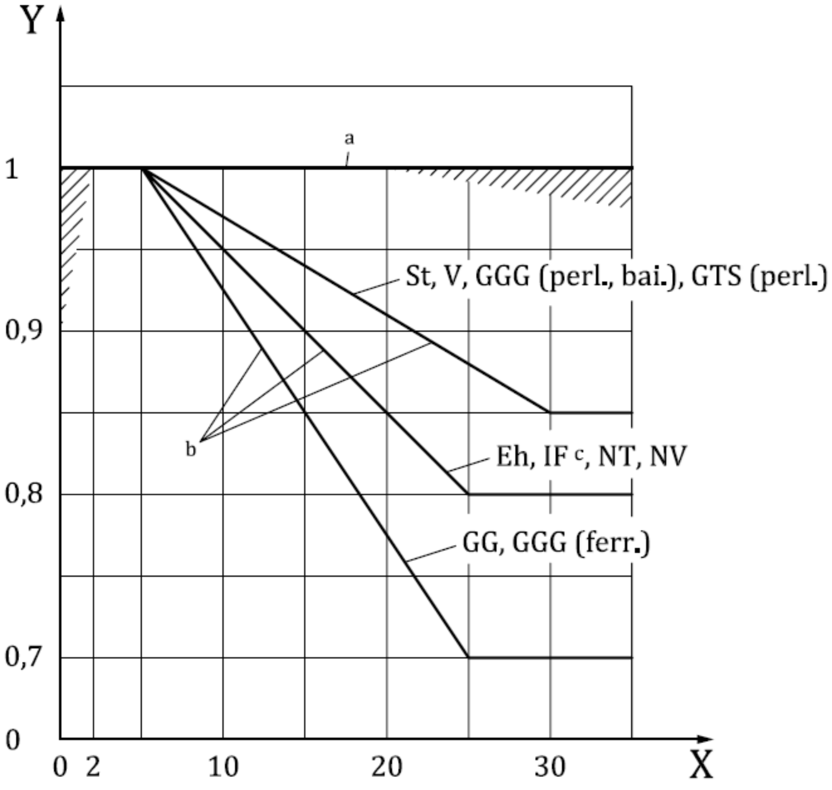
# Size factor YX for small gears

## Influence of size on specific strength

Material flaws increase with increasing size

Rating standards are known to be conservative for small gears [41]





**Key**

| | |
|---|---|
| X | normal module, $m_n$, mm |
| Y | size factor, $Y_X$ |
| a | Static stress (all materials). |
| b | Reference stress. |
| c | (root). |

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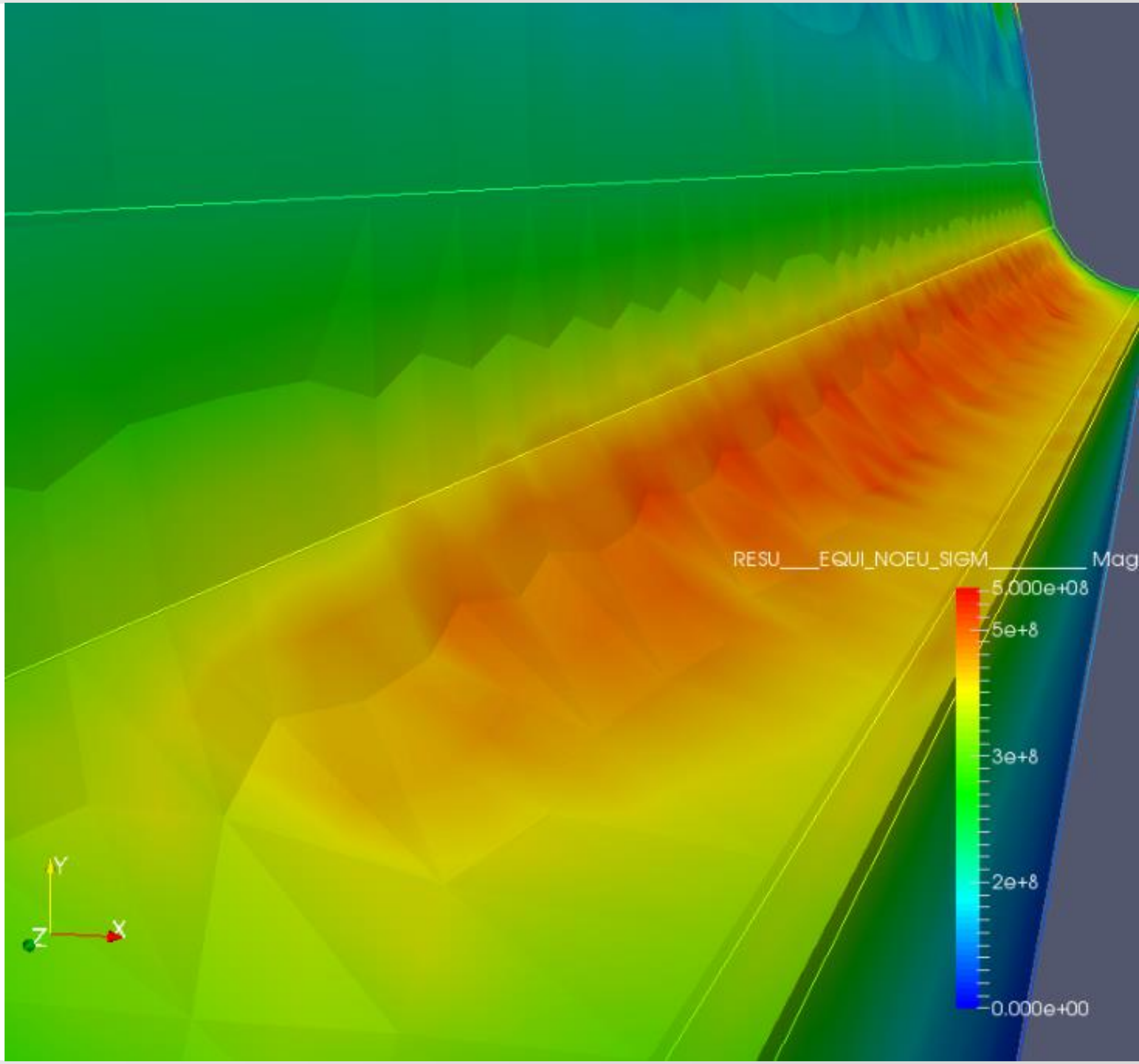

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# Influence of shot peening

## Effects of shot peening [36]

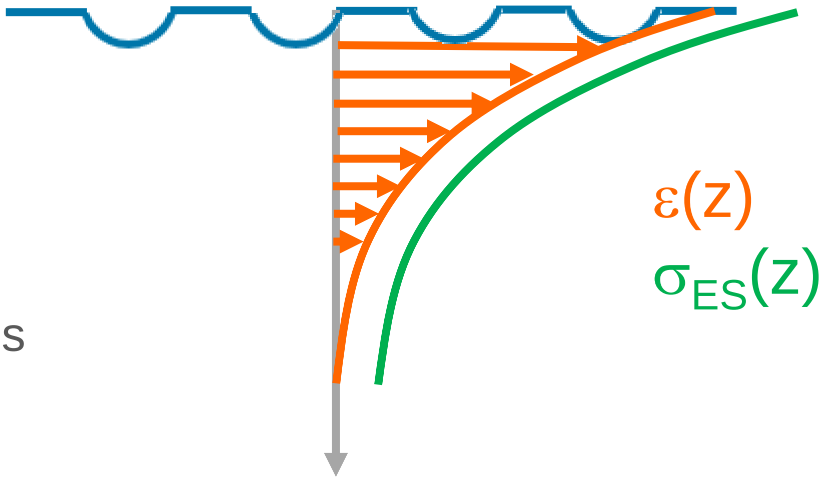
Near surface residual compressive stresses ↑

Retained austenite content ↓

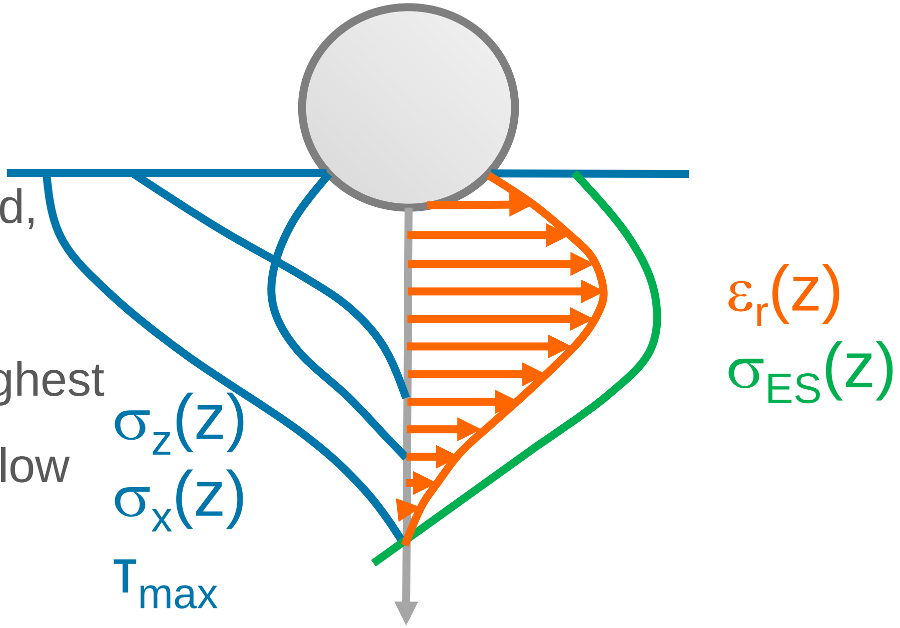
Surface roughness ↑

Work strengthening / structure dislocation ↑

After shot peening, surface plastified, highest residual stress at surface



Under contact load, elastic stresses superimposed, highest residual stress below surface



# Influence of shot peening

Values for technology factor YT

ISO 6336-5 allows for strength increase of

- 0% in case of ML material quality grade
- 10% in case of MQ material quality grade
- 5% in case of ME material quality grade

Lloyd's Register of Shipping allows for strength increase of

- 20%

Higher values are reported in literature, using highly controlled processes, for "automotive" size, case carburized gears, [13]

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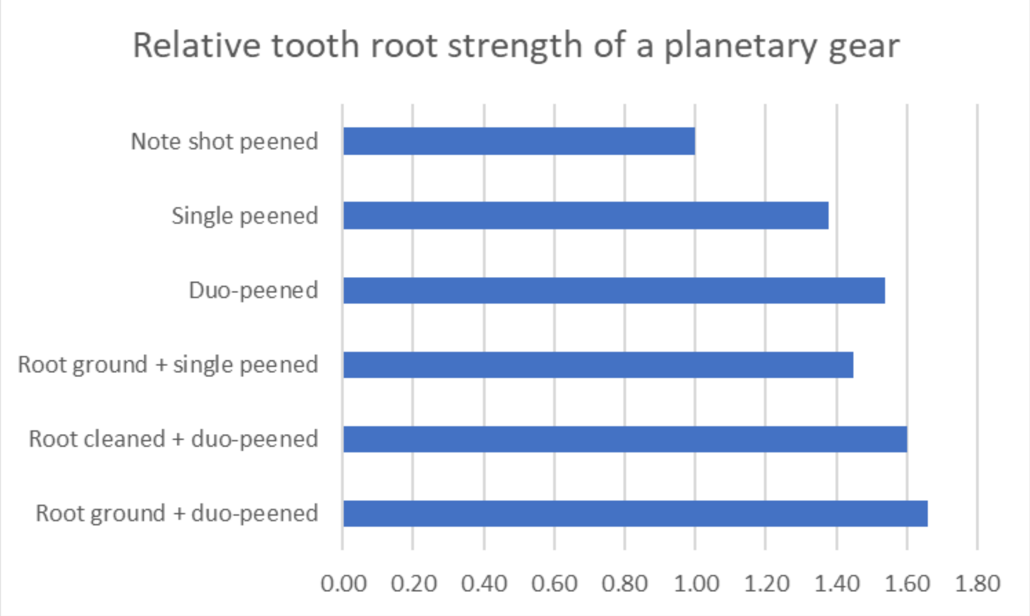

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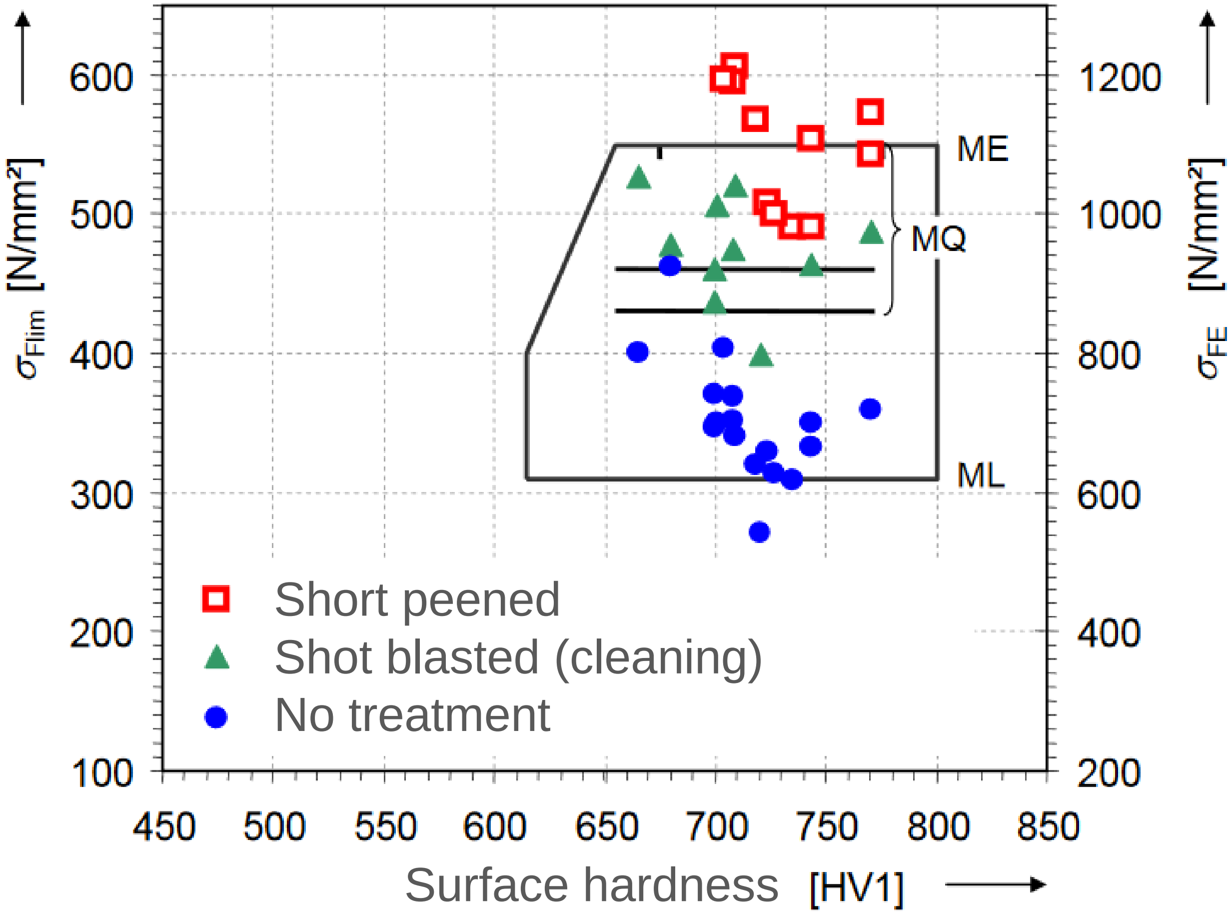
# Influence of shot peening

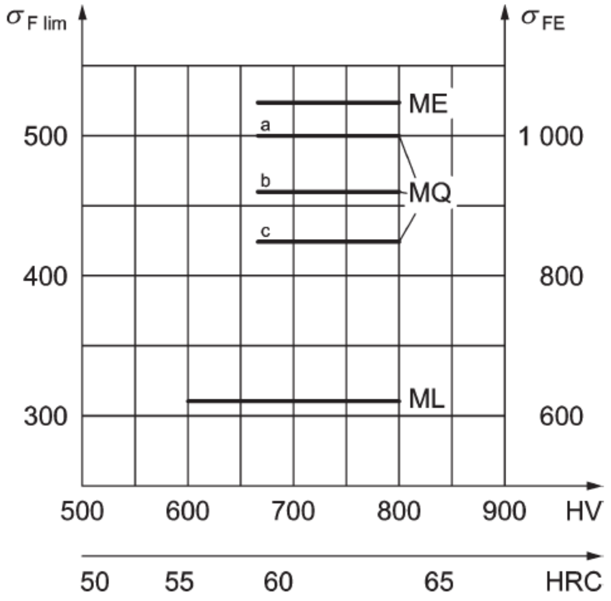
## Issues and questions

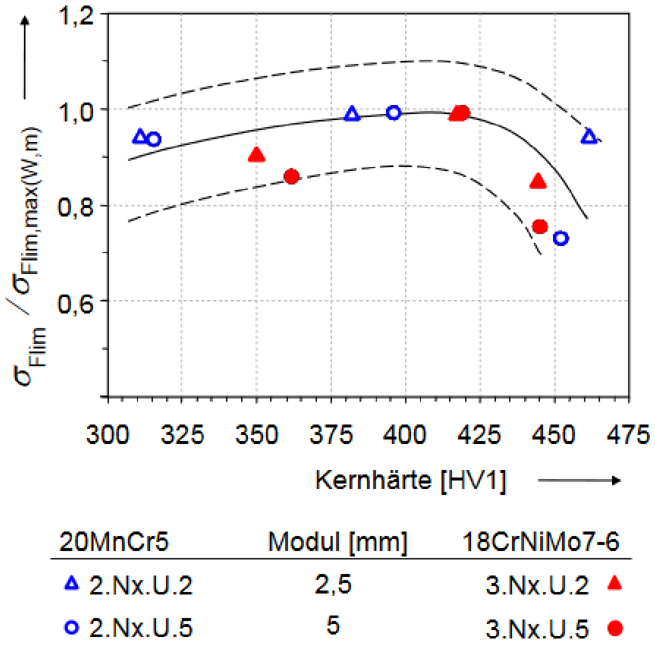
- Effectiveness for large gears?
- Effectiveness for gears under alternating bending, lower effectiveness is reported, [9]
- Introduction of a shot peening factor ZS is proposed [41]
- Strength values as per ISO 6336-5 require a mechanical cleaning of gears by a shot blasting (not a shot peening) process [11]

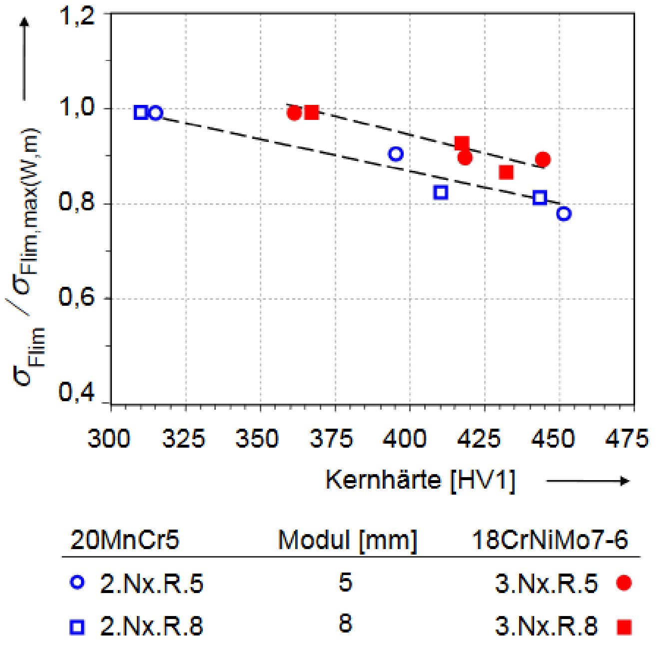


# Influence of material and heat treatment

## Core hardness







- Left: Core hardness ≥ 30 HRC (300 HV) gives highest root strength
- Middle: Core hardness ~ 400 HV gives highest root strength, not cleaned gear [11]
- Right: Core hardness > 30 HRC (300 HV), shot / clean blasted gear, root strength drops [11]

# Mean stress influence factor YM

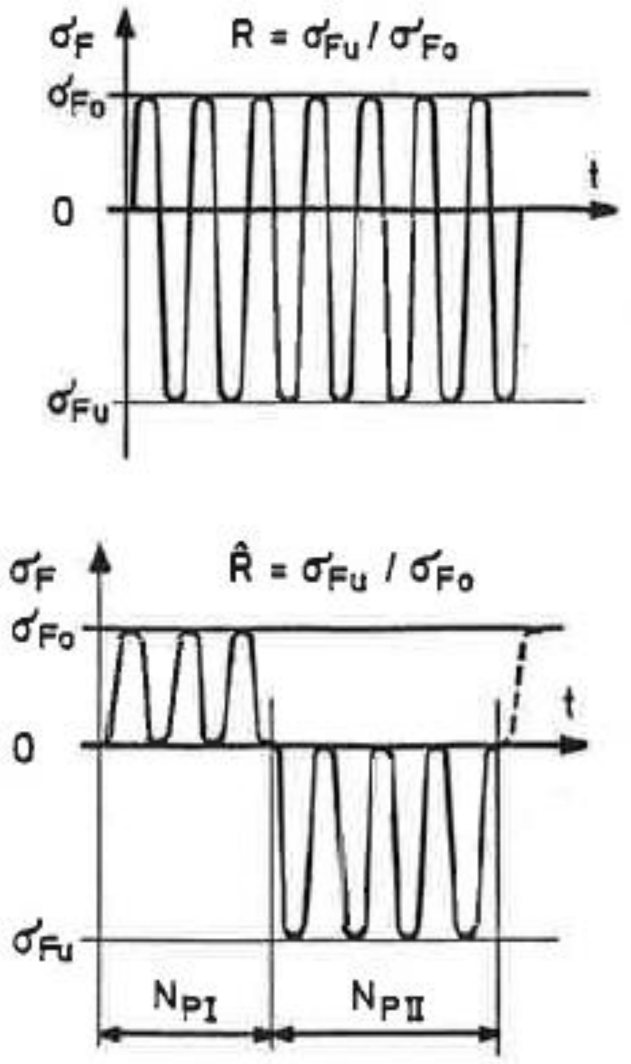
## Strength reduction in case of mean stress or load reversal

ISO 6336-3 Annex B (informative), mean stress influence factor YM

$$R = -1{,}2\,\frac{\text{load per unit facewidth of the lower loaded flank}}{\text{load per unit facewidth of the higher loaded flank}} \qquad Y_{\mathrm{M}} = \frac{1}{1 - R\,\dfrac{1-M}{1+M}}$$

M from below table

| | **Endurance limit** | **Static strength** |
|---|---|---|
| Case hardened | $0{,}8 - 0{,}15\,Y_{\mathrm{S}}$ | 0,7 |
| Case hardened and shot peened | 0,4 | 0,6 |
| Nitrided | 0,3 | 0,3 |
| Induction or flame hardened | 0,4 | 0,6 |
| Not surface hardened steels | 0,3 | 0,5 |
| Cast steels | 0,4 | 0,6 |



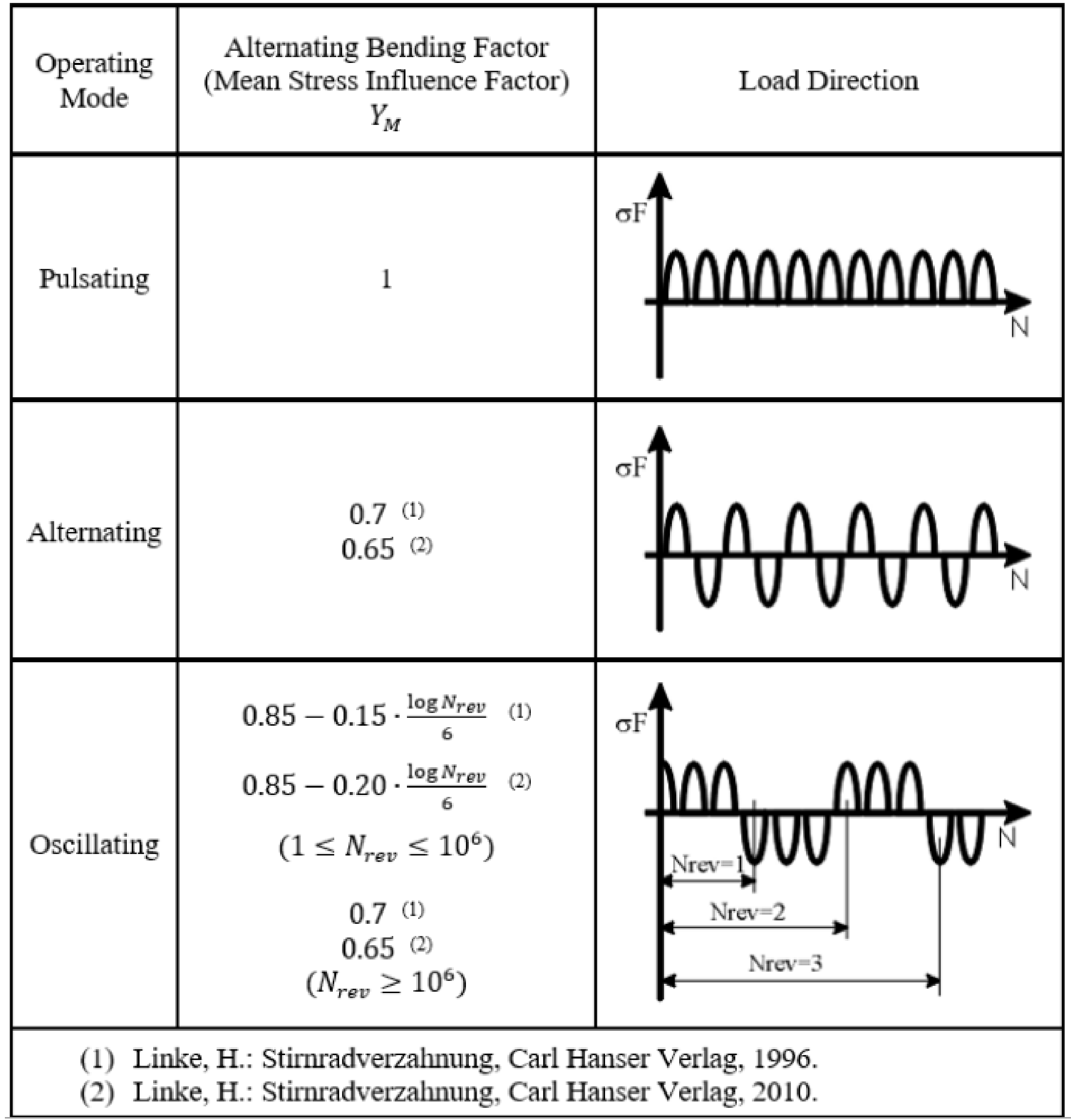
YM as per ISO 6336-3, for periodical load reversal → assume R = 1

# Mean stress influence factor YM for periodic load reversal

In case of periodical change of loading direction

Calculate YM as a function of the number of load reversals [35], [42]

For case carburized gears, a 30% strength loss is quickly achieved in case of load reversals.



| Operating Mode | Alternating Bending Factor (Mean Stress Influence Factor) $Y_M$ | Load Direction |
|---|---|---|
| Pulsating | 1 | |
| Alternating | 0.7 (1)<br>0.65 (2) | |
| Oscillating | $0.85 - 0.15 \cdot \frac{\log N_{rev}}{6}$ (1)<br>$0.85 - 0.20 \cdot \frac{\log N_{rev}}{6}$ (2)<br>$(1 \leq N_{rev} \leq 10^6)$<br>0.7 (1)<br>0.65 (2)<br>$(N_{rev} \geq 10^6)$ | |

(1) Linke, H.: Stirnradverzahnung, Carl Hanser Verlag, 1996.
(2) Linke, H.: Stirnradverzahnung, Carl Hanser Verlag, 2010.

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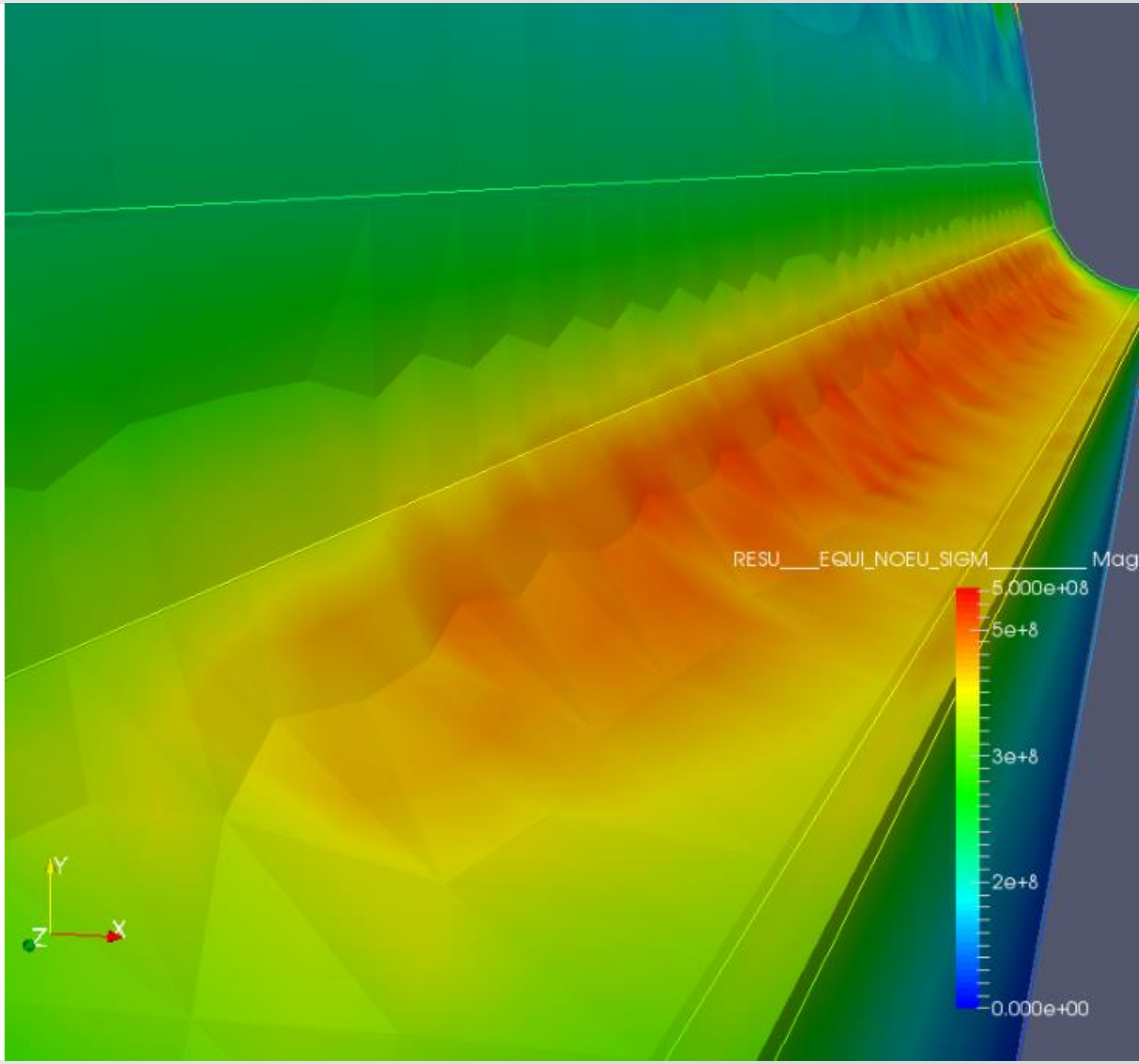
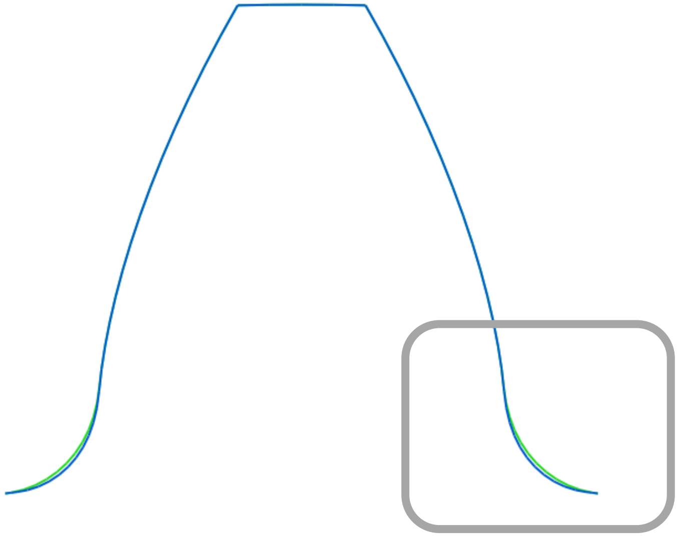


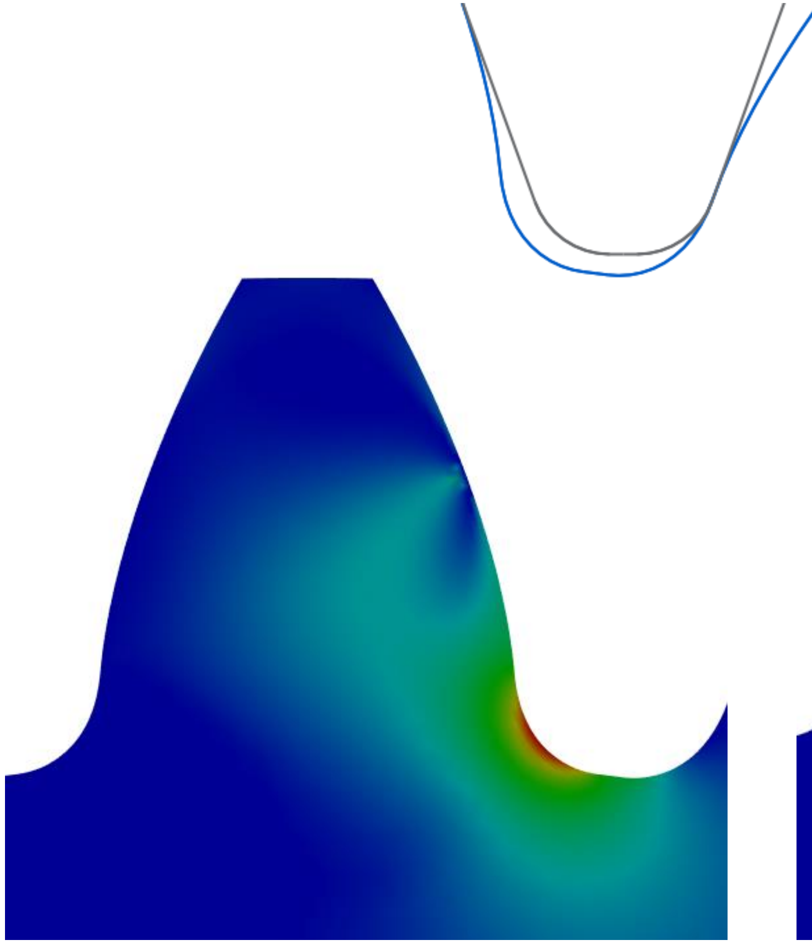
# Root shape and root stress

## Effect of tool tip radius

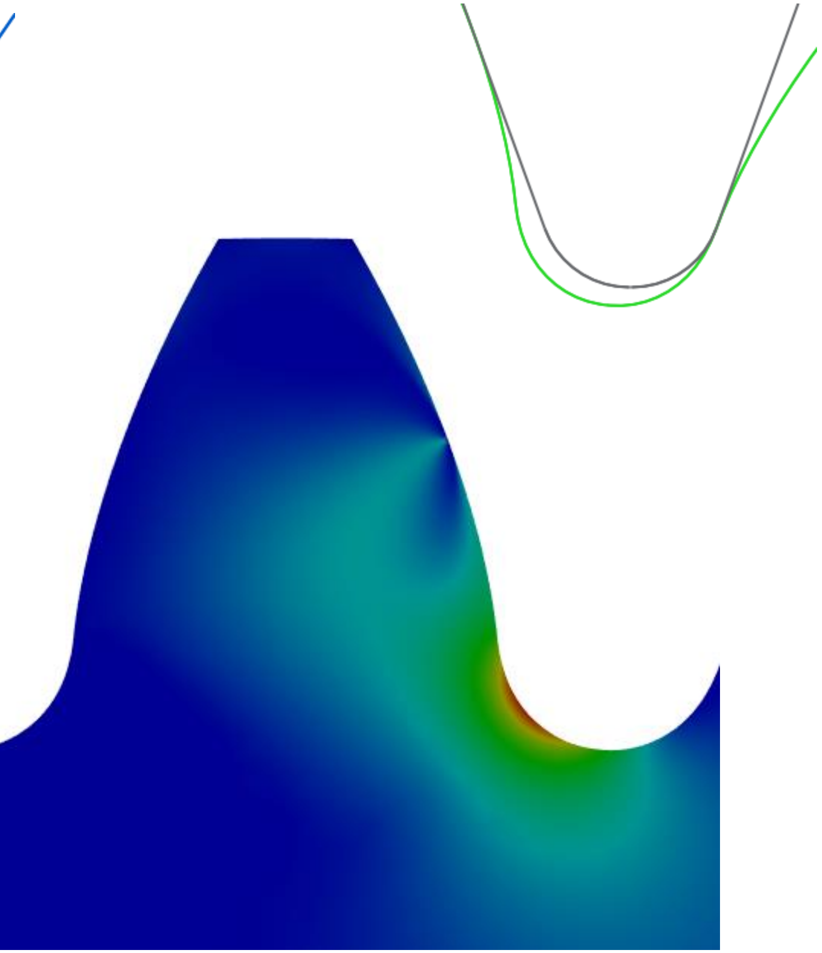
z = 25, mn = 6.00 mm, β = 0° ρaP0* = 0.38

Cut with tool standard tip radius or by maximized tool tip radius, resulting in different root shapes:





Hob → 207 MPa



Hob, ρaP0* = 0.46 → 194 MPa

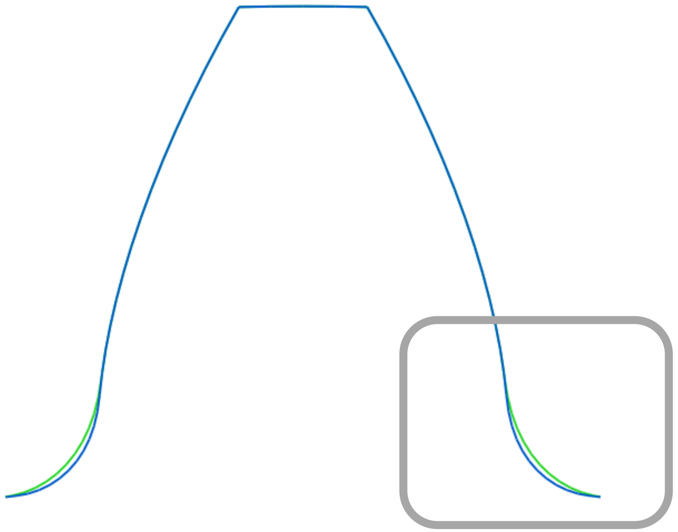
Images not to scale

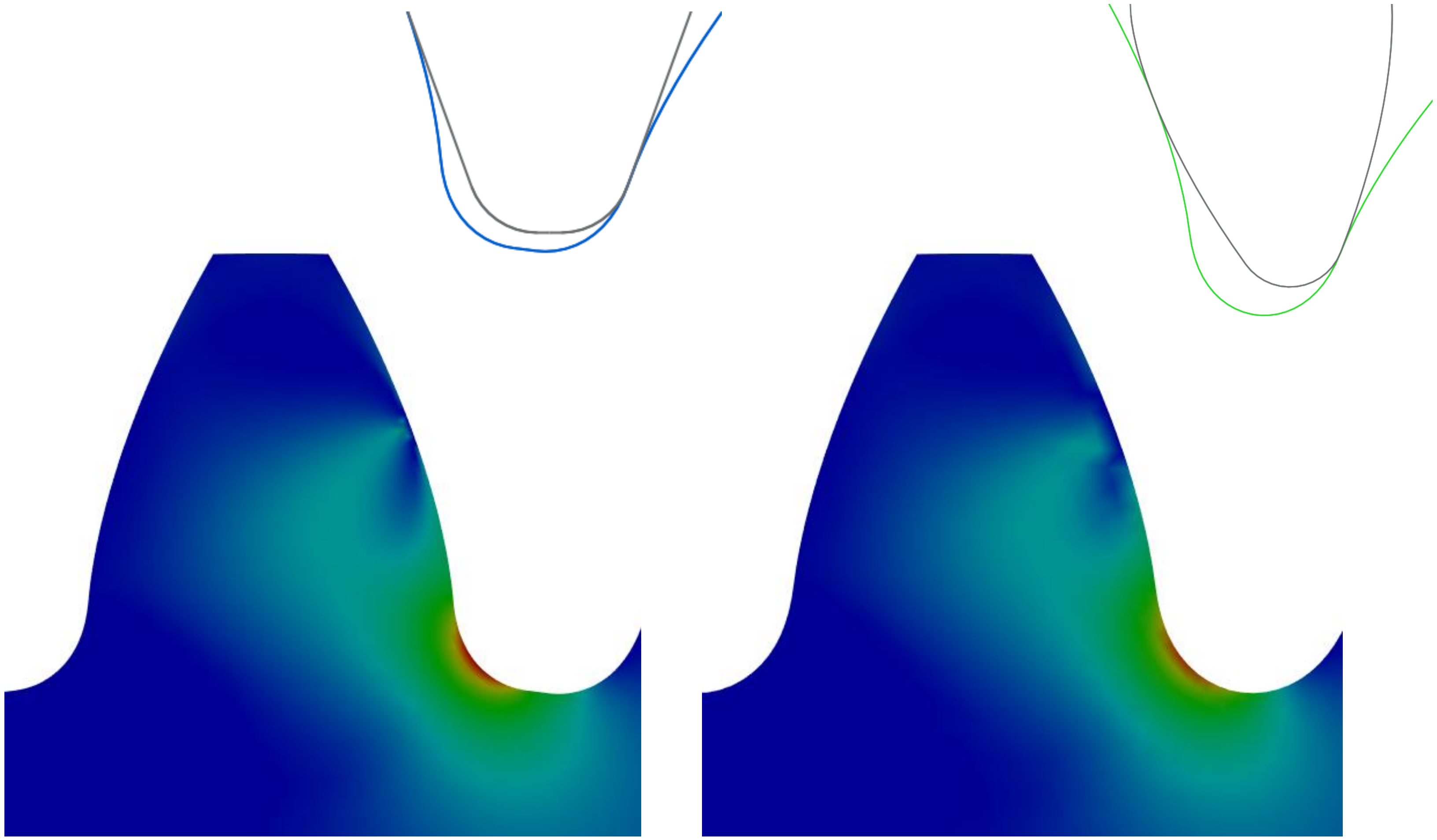
# Root shape and root stress

## Effect of tool type

z = 25, mn = 6.00 mm, β = 0°   ρaP0* = 0.38

Cut by rack type tool or by shaping cutter type tool, resulting in different root shapes:





Hob → 207 MPa

Shaping cutter → 188 MPa

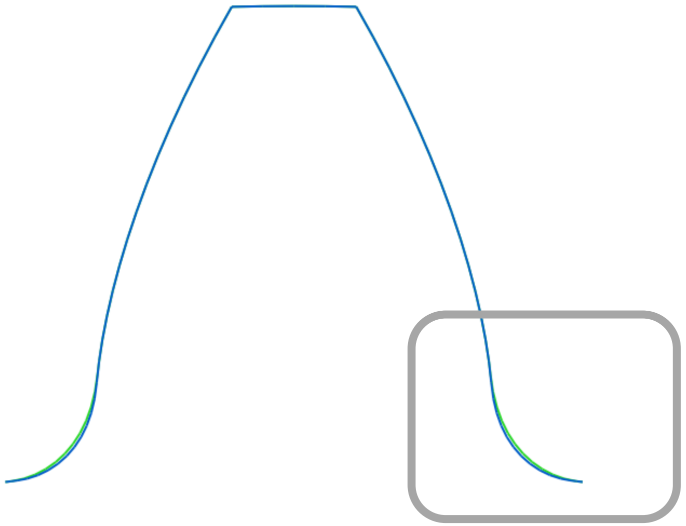
Images not to scale

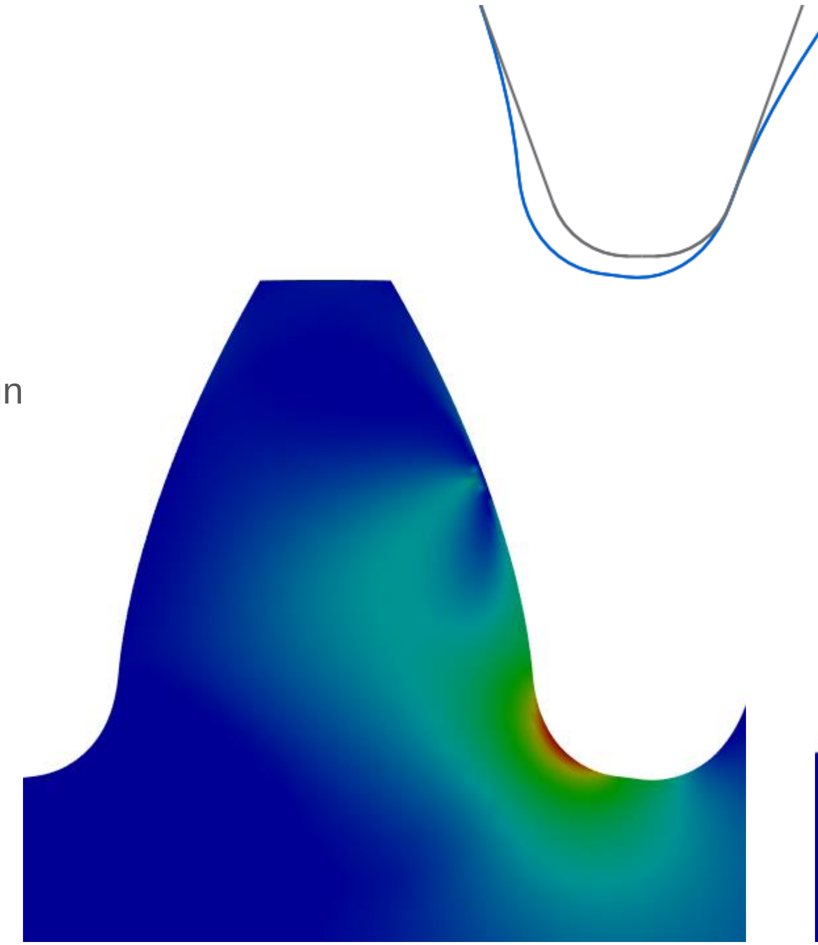
# Root shape and root stress

## Effect of rack type tool module

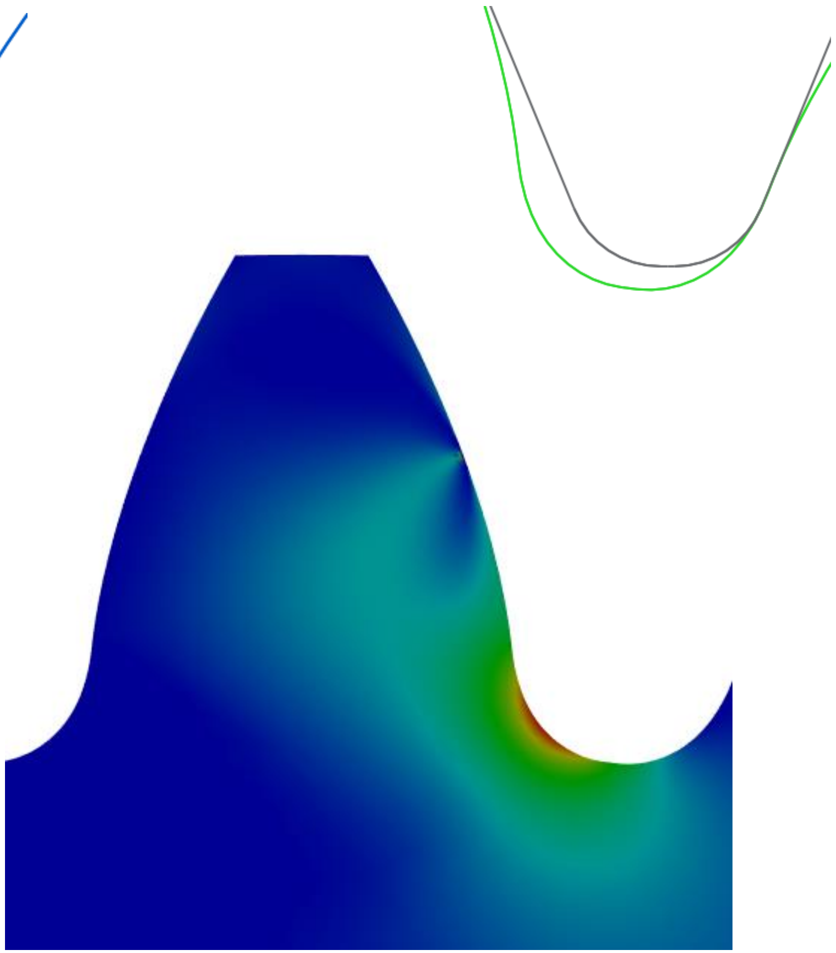
z = 25, mn = 6.00 mm, β = 0°   ρaP0* = 0.38

Cut by rack type tool with same module or by rack type tool with different module, resulting in different root shapes:





Tool module 6.00 mn → 207 MPa



Tool module 6.10 mn → 196 MPa

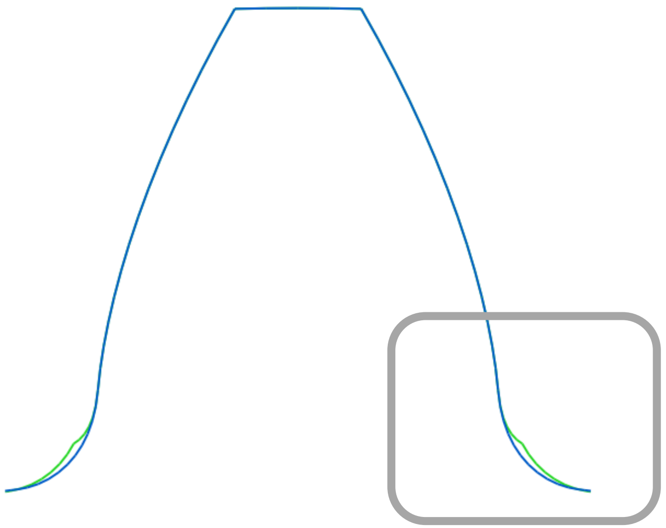
Images not to scale

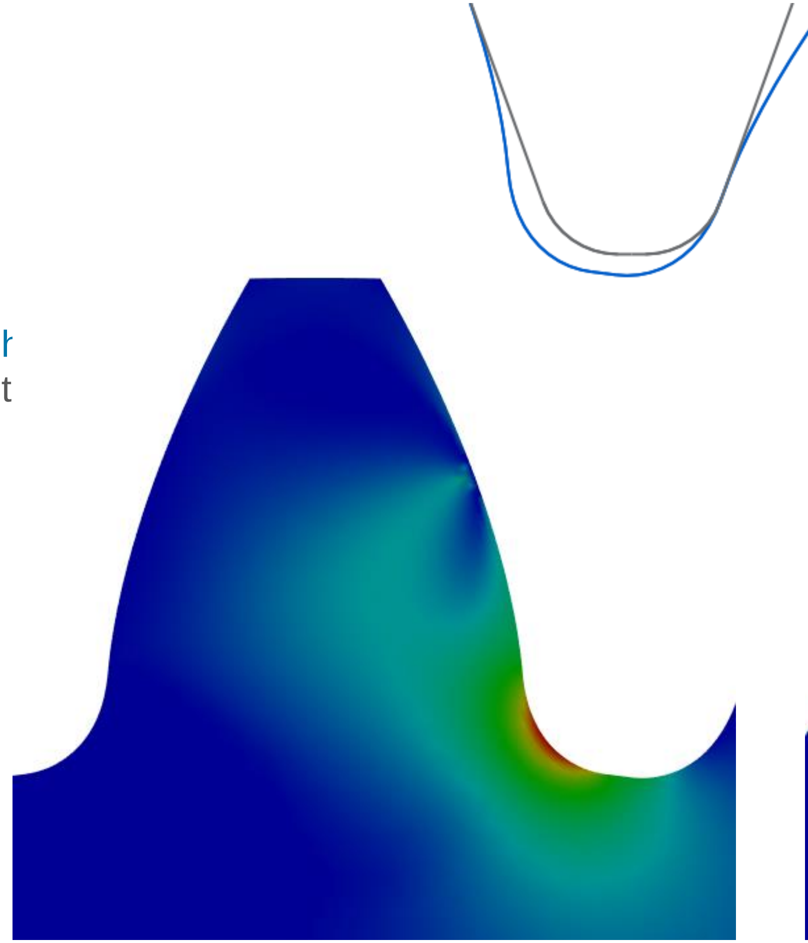
# Root shape and root stress

## Effect of grinding notch

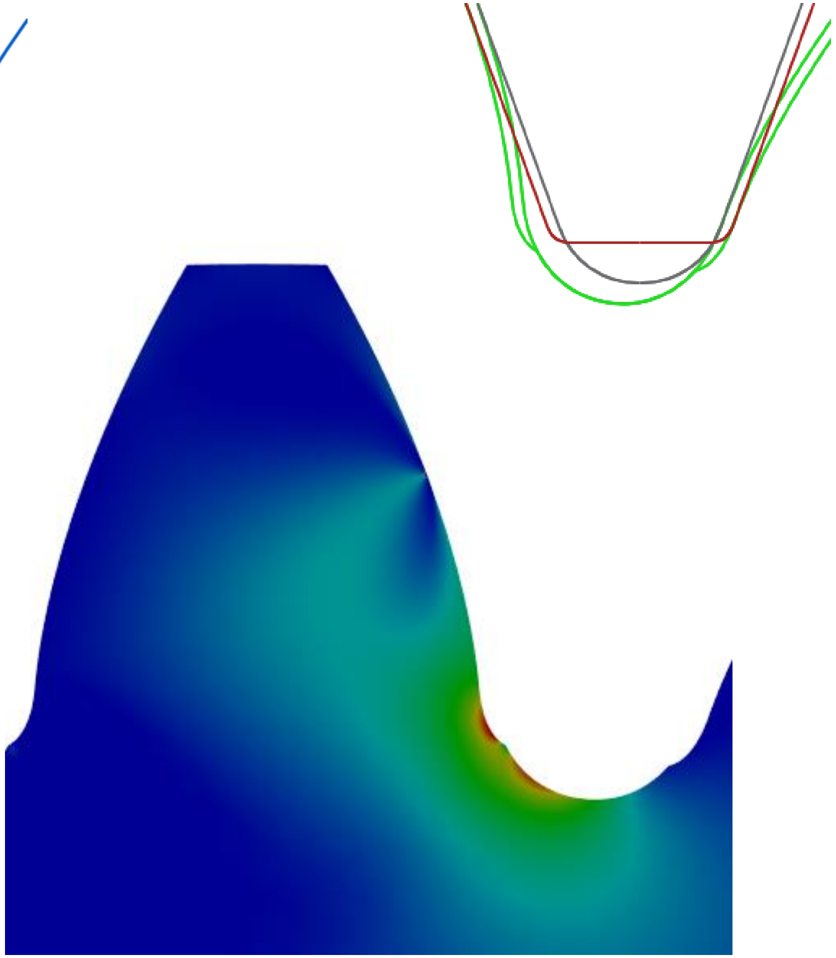
z = 25, mn = 6.00 mm, β = 0°   ρaP0* = 0.38

Root shape trochoidal, no final machining notch or with grinding notch, resulting in different root shapes:





Trochoidal root shape → 207 MPa



Grinding notch → 217 MPa

Images not to scale

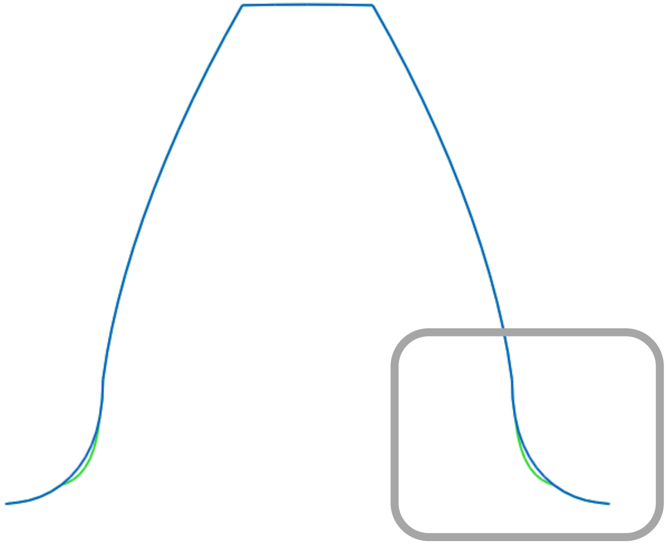
KISSsoft

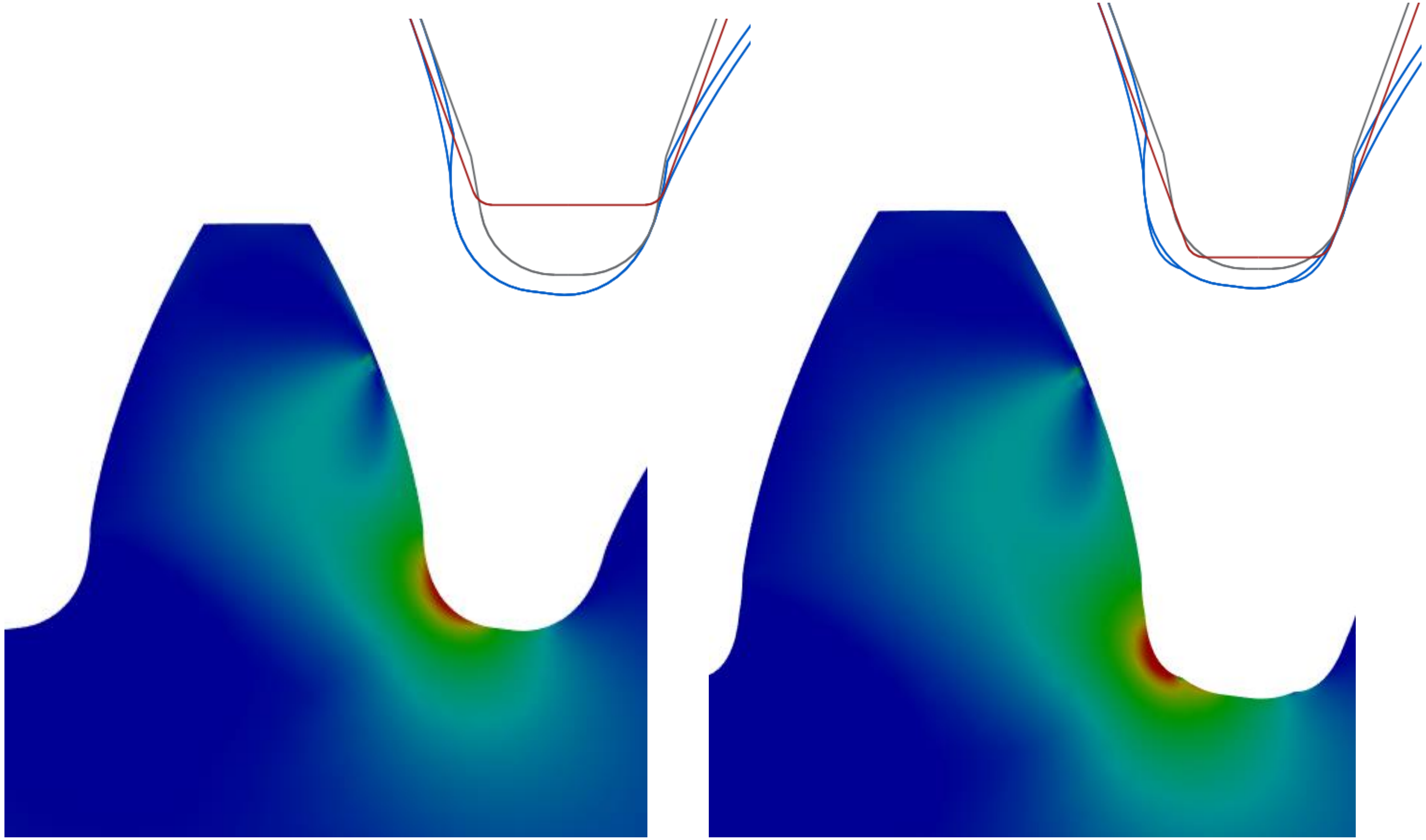
# Root shape and root stress

## Effect of root grinding error

z = 25, mn = 6.00 mm, β = 0°   ρaP0* = 0.38

Root shape trochoidal, with protuberance or with protuberance and grinding error, resulting in different root shapes:





Protuberance only → 222 MPa

Grinding error → 279 MPa

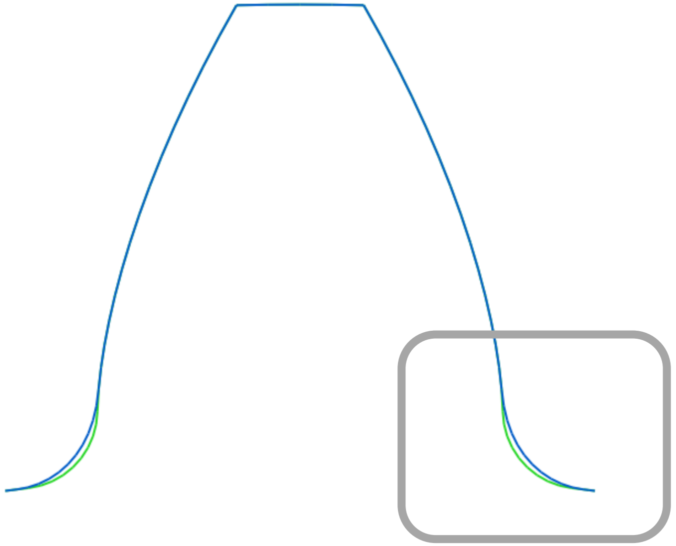
Images not to scale

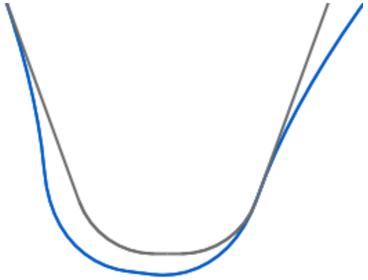
# Root shape and root stress

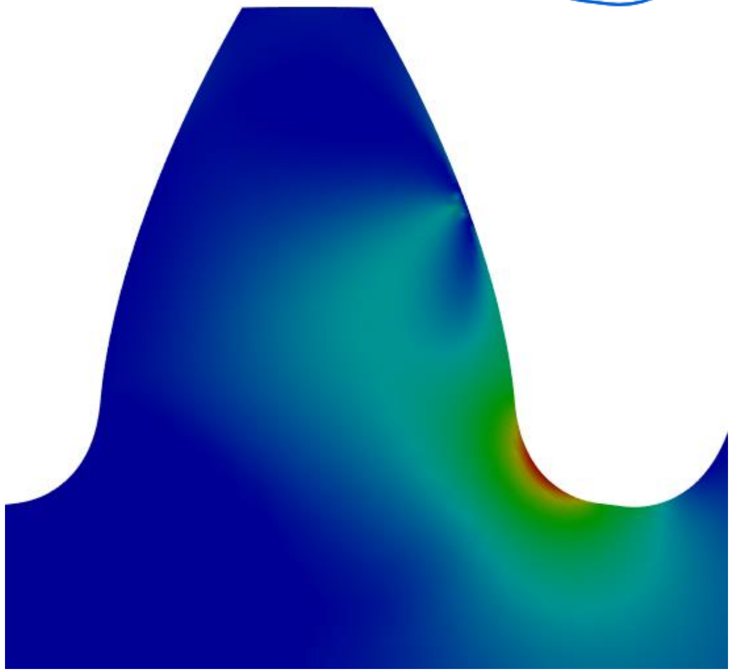
## Effect of circular root shape

z = 25, mn = 6.00 mm, β = 0°   ρaP0* = 0.38

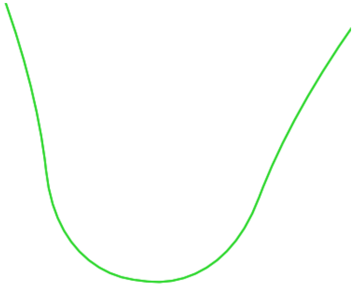
Root shape trochoidal, from rack type or by circular, resulting in different root shapes:

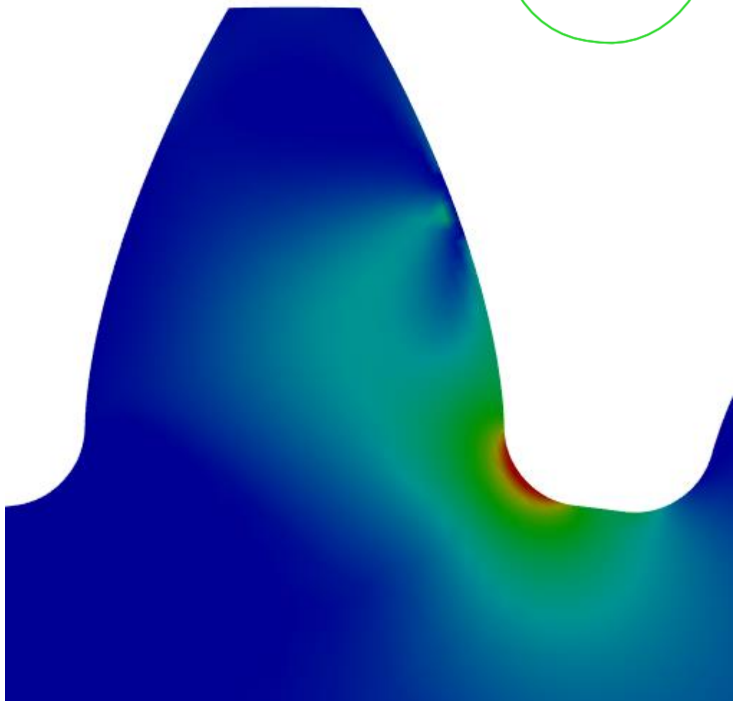






Trochoidal root shape → 207 MPa





Circular arc root shape → 233 MPa

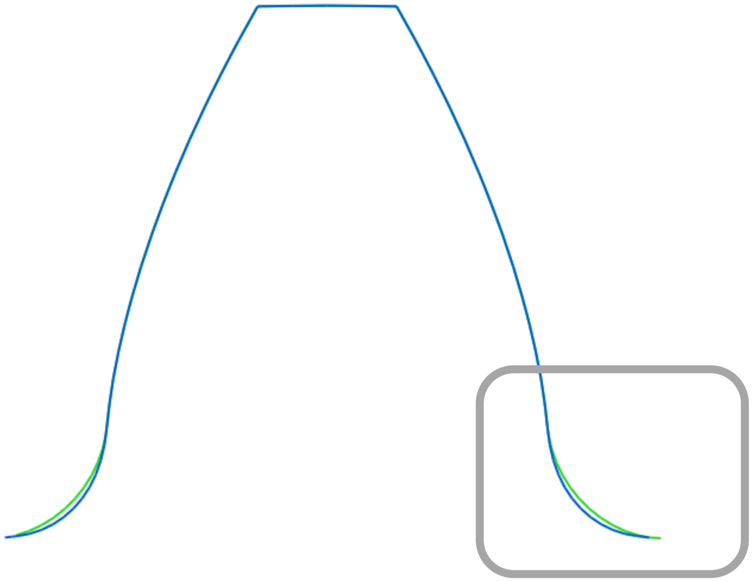
Images not to scale

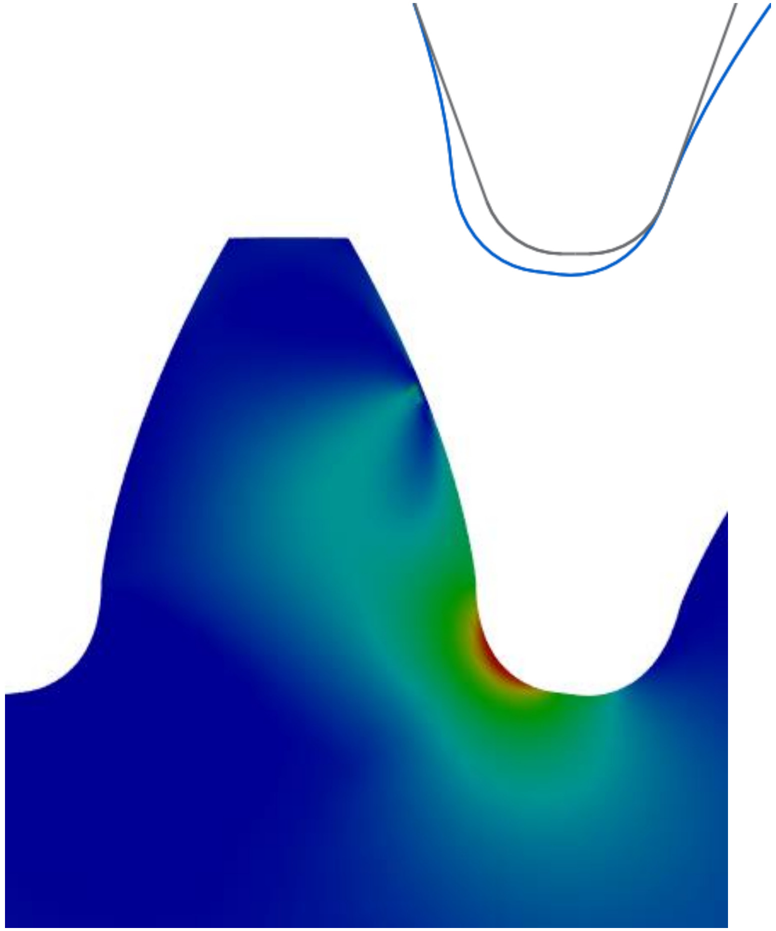
# Root shape and root stress

## Effect of elliptical root shape

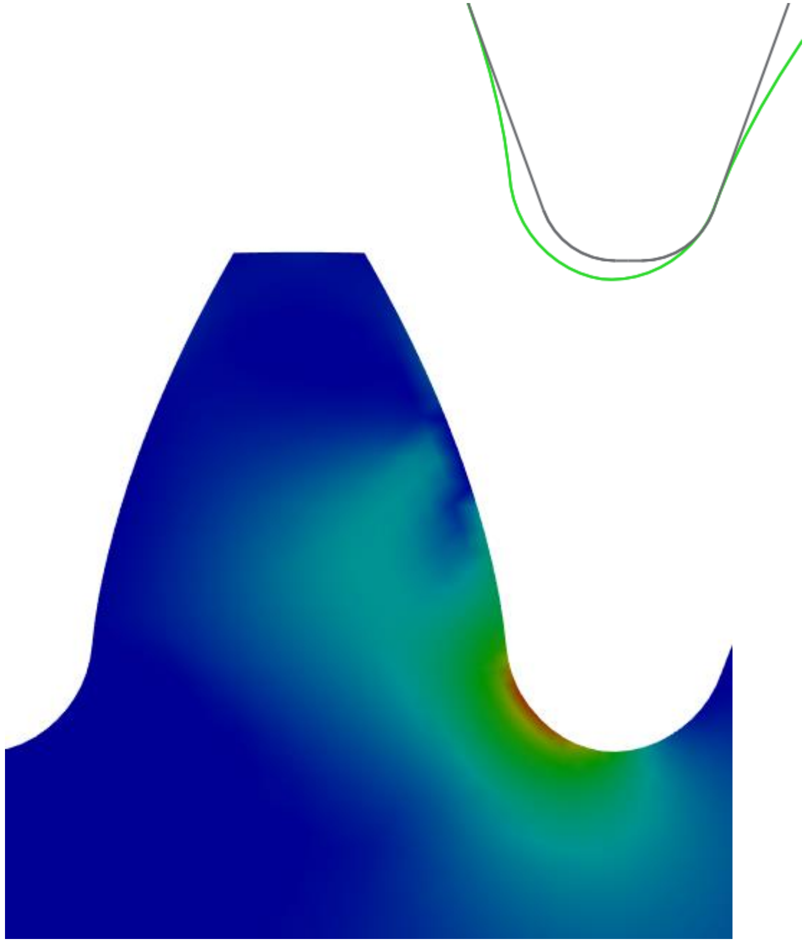
z = 25, mn = 6.00 mm, β = 0° ρaP0* = 0.38

Root shape trochoidal or elliptic, resulting in different root shapes:





Trochoidal root shape → 207 MPa



Elliptical root shape → 189 MPa

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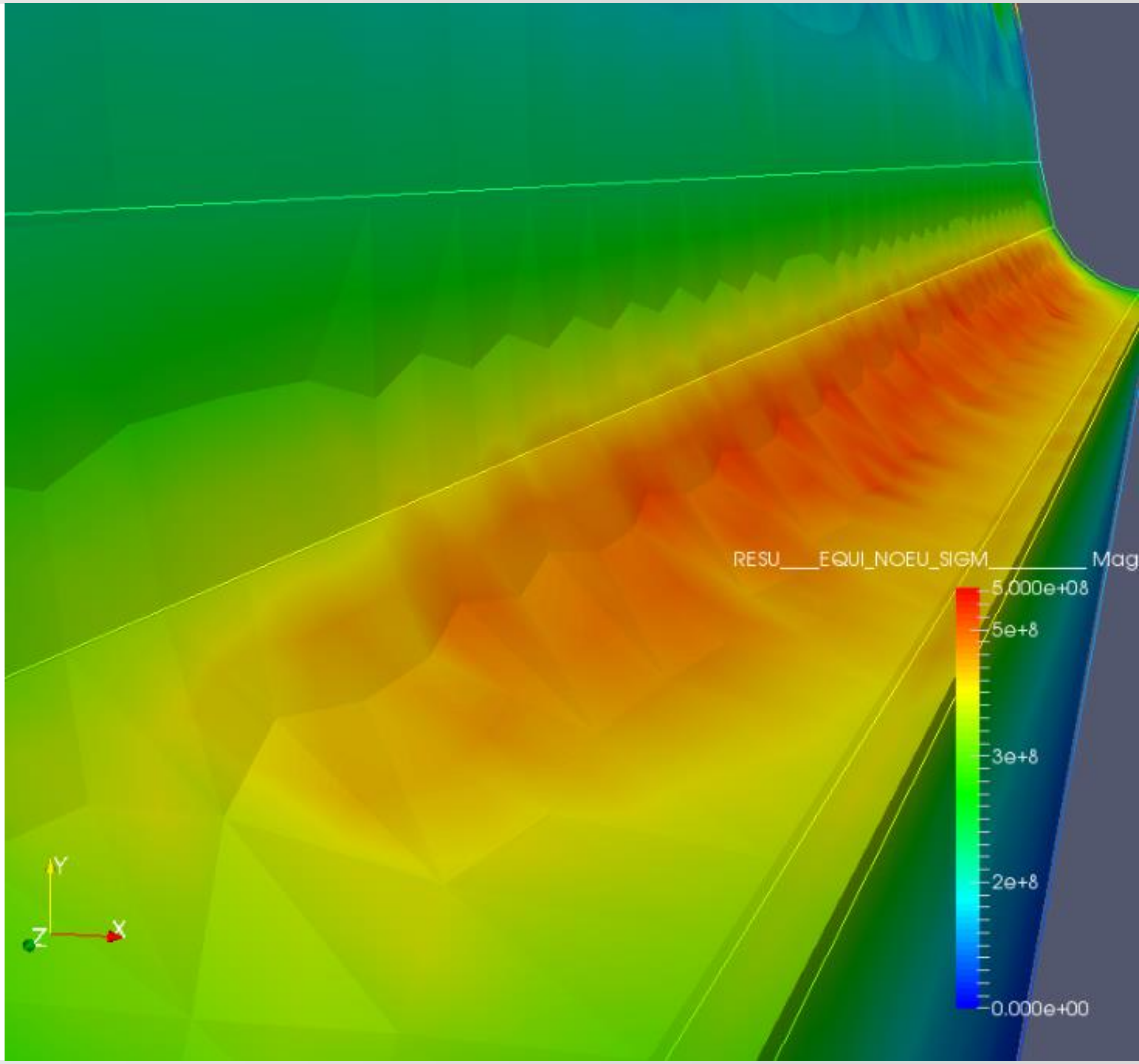


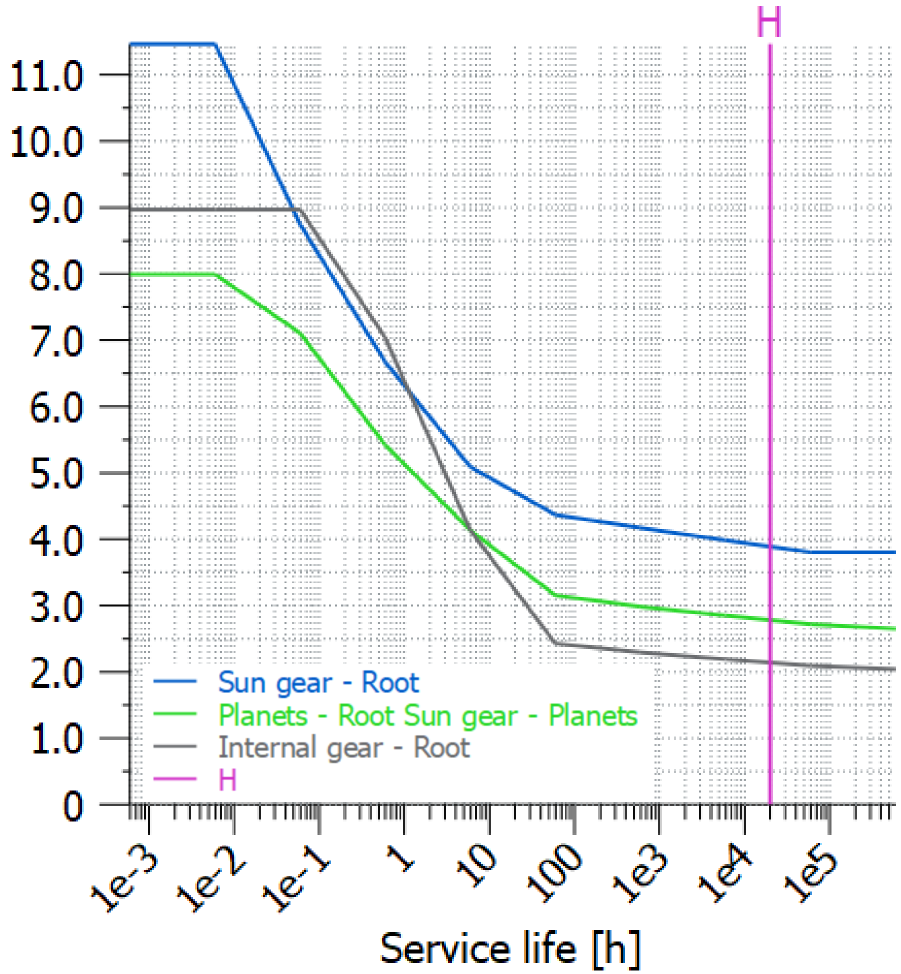
# Conclusion

It’s not so simple

Stress calculation is well defined, understood and calculations match measurements very well.

Part strength depends on less well-known factors including

- Material, material quality, surface and core hardness
- Heat treatment, surface treatment, machining process
- Load sequence
- Calculation methods e.g. in standards to derive permissible stress values do not cover these aspects in full



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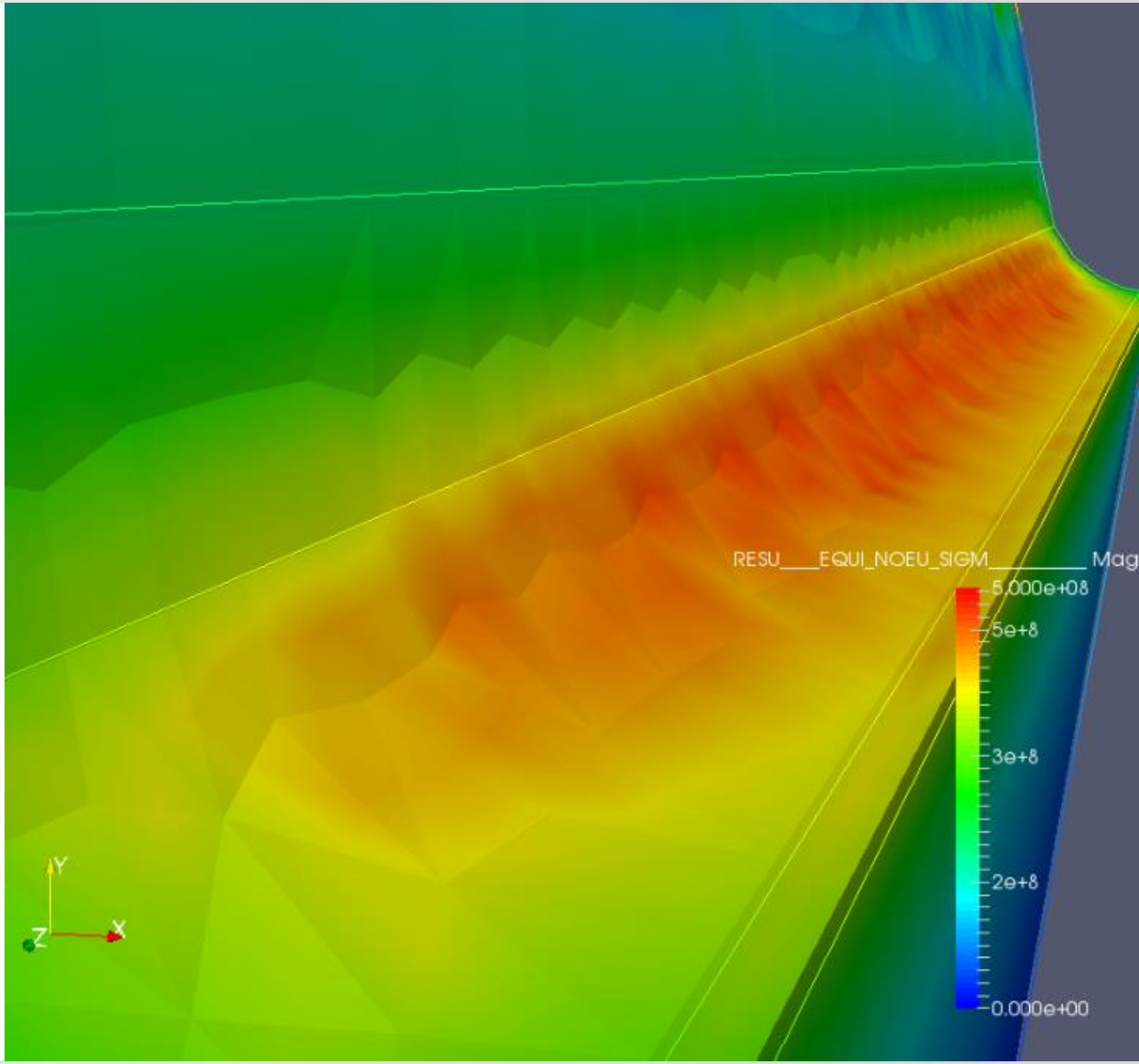

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This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4.

*By Dr. A. Pogacnik and Dr. S. Beermann*

The number of plastic gears used in different applications is increasing every year, mostly due to their cost effectiveness for large series production and lubrication-free running. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required. Unfortunately for gear designers, "gear fatigue data" (S-N curves) are rarely measured for new materials.

If the decision is made to measure gear-fatigue data, it is worth doing it correctly so that the measured data can be used. This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4, which was published in 2016. The procedure to measure temperature and other challenges that arise are discussed. Furthermore, the statistical procedure to evaluate test results is explained.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2. To help test engineers, a software tool was developed. The measured data is introduced, then the software performs the evaluation of the test results automatically and calculates the permissible root/flank stress data for different temperatures.

In the conclusion, some remaining problems are discussed. For the calculation of achievable lifetime or for loads based on duty cycles, the S-N curves (Wöhler lines) should be defined up to $10^{30}$ cycles, which is far beyond the obtainable test results. Therefore, an extrapolation of the measured data to higher cycle numbers should be considered.

### INTRODUCTION

Due to the well-known advantages of plastic gears, their use is increasing every year, especially in the automotive industry where lubrication-free running, low noise, and high serial production are required [1,2]. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required [3,4]. But this is a problem for gear engineers as gear data (permissible stresses) are not being measured at the same rate as new plastics are developed. In fact, there are just a few new materials measured every year, for which permissible root/flank stresses are publicly available. The initiative is mainly coming from the major plastic material companies, which want to promote their materials also for gear applications.

As an engineer, how can you design plastic gears if no reliable fatigue data is available for the material that is used in the application? It is possible to calculate root/flank safety factors with fatigue data from a material, which has gear data available. But due to uncertainties between the two materials in question, the safety factors should be increased/decreased. For high temperatures, there is very little gear data available. Often, extrapolation is used to project permissible stresses at high temperatures, but this can be very inaccurate. At the end, all these uncertainties lead to gear designs that are not optimized in terms of strength and can be far from an optimal solution in terms of price.

An alternative to the above-mentioned procedure is to generate gear data for a material that is being used in the application. Generating root/flank fatigue data on actual gears (on a gear test rig) is a huge effort in terms of time and money spent. Generating permissible root stresses for three different temperatures and for five different numbers of cycles (between $0.1 \cdot 10^6$ and $5 \cdot 10^6$) can easily take between 3-4 months and can cost up to 50,000 euros.

If decision is taken to measure gear data, it is worth doing it correctly so that the measured data can be used at the end. In this paper, an overview of the testing procedure for plastic gears according to the VDI 2736-4 [5], which was published in 2016, is provided. The procedure to measure temperature and other challenges that arise are discussed. The statistical procedure to evaluate test results is also explained. A comparison of the permissible tooth root strength, measured on different test gears, will be discussed, indicating a large scatter between the measured permissible stresses.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2 [6]. The calculation of tooth root and flank safety factors requires S-N curves, which are temperature dependent. The wear calculation requires wear factors.

To help test engineers, a tool was developed based on the VDI 2736-4. The measured data is introduced, then the tool performs the evaluation of the test results automatically and calculates the permissible root/flank stresses. This information is documented in a text file, which can directly be used by a calculation tool for the calculation of plastic gears according to the VDI 2736-2.

For the safety factor calculation with duty cycles, the S-N curves should be defined up to the $10^{30}$ cycles, which is far beyond the obtainable test results. Therefore, an

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## Measuring Root/Flank Stress in Plastic Gears

This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4.

*By Dr. A. Pogacnik and Dr. S. Beermann*

The number of plastic gears used in different applications is increasing every year, mostly due to their cost effectiveness for large series production and lubrication-free running. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required. Unfortunately for gear designers, "gear fatigue data" (S-N curves) are rarely measured for new materials.

If the decision is made to measure gear-fatigue data, it is worth doing it correctly so that the measured data can be used. This paper provides an overview of the testing procedure for plastic gears according to the VDI 2736-4, which was published in 2016. The procedure to measure temperature and other challenges that arise are discussed. Furthermore, the statistical procedure to evaluate test results is explained.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2. To help test engineers, a software tool was developed. The measured data is introduced, then the software performs the evaluation of the test results automatically and calculates the permissible root/flank stress data for different temperatures.

In the conclusion, some remaining problems are discussed. For the calculation of achievable lifetime or for loads based on duty cycles, the S-N curves (Wöhler lines) should be defined up to $10^{30}$ cycles, which is far beyond the obtainable test results. Therefore, an extrapolation of the measured data to higher cycle numbers should be considered.

### INTRODUCTION

Due to the well-known advantages of plastic gears, their use is increasing every year, especially in the automotive industry where lubrication-free running, low noise, and high serial production are required [1,2]. With a rapid development of plastic materials in terms of strength and allowable temperature range, plastic gears are finding their way also into more demanding applications, where high transmittable torque at elevated temperatures is required [3,4]. But this is a problem for gear engineers as gear data (permissible stresses) are not being measured at the same rate as new plastics are developed. In fact, there are just a few new materials measured every year, for which permissible root/flank stresses are publicly available. The initiative is mainly coming from the major plastic material companies, which want to promote their materials also for gear applications.

As an engineer, how can you design plastic gears if no reliable fatigue data is available for the material that is used in the application? It is possible to calculate root/flank safety factors with fatigue data from a material, which has gear data available. But due to uncertainties between the two materials in question, the safety factors should be increased/decreased. For high temperatures, there is very little gear data available. Often, extrapolation is used to project permissible stresses at high temperatures, but this can be very inaccurate. At the end, all these uncertainties lead to gear designs that are not optimized in terms of strength and can be far from an optimal solution in terms of price.

An alternative to the above-mentioned procedure is to generate gear data for a material that is being used in the application. Generating root/flank fatigue data on actual gears (on a gear test rig) is a huge effort in terms of time and money spent. Generating permissible root stresses for three different temperatures and for five different numbers of cycles (between $0.1 \cdot 10^6$ and $5 \cdot 10^6$) can easily take between 3-4 months and can cost up to 50,000 euros.

If decision is taken to measure gear data, it is worth doing it correctly so that the measured data can be used at the end. In this paper, an overview of the testing procedure for plastic gears according to the VDI 2736-4 [5], which was published in 2016, is provided. The procedure to measure temperature and other challenges that arise are discussed. The statistical procedure to evaluate test results is also explained. A comparison of the permissible tooth root strength, measured on different test gears, will be discussed, indicating a large scatter between the measured permissible stresses.

Measured data according to the VDI 2736-4 must be converted correctly to become usable for strength calculation according to the VDI 2736-2 [6]. The calculation of tooth root and flank safety factors requires S-N curves, which are temperature dependent. The wear calculation requires wear factors.

To help test engineers, a tool was developed based on the VDI 2736-4. The measured data is introduced, then the tool performs the evaluation of the test results automatically and calculates the permissible root/flank stresses. This information is documented in a text file, which can directly be used by a calculation tool for the calculation of plastic gears according to the VDI 2736-2.

For the safety factor calculation with duty cycles, the S-N curves should be defined up to the $10^{30}$ cycles, which is far beyond the obtainable test results. Therefore, an

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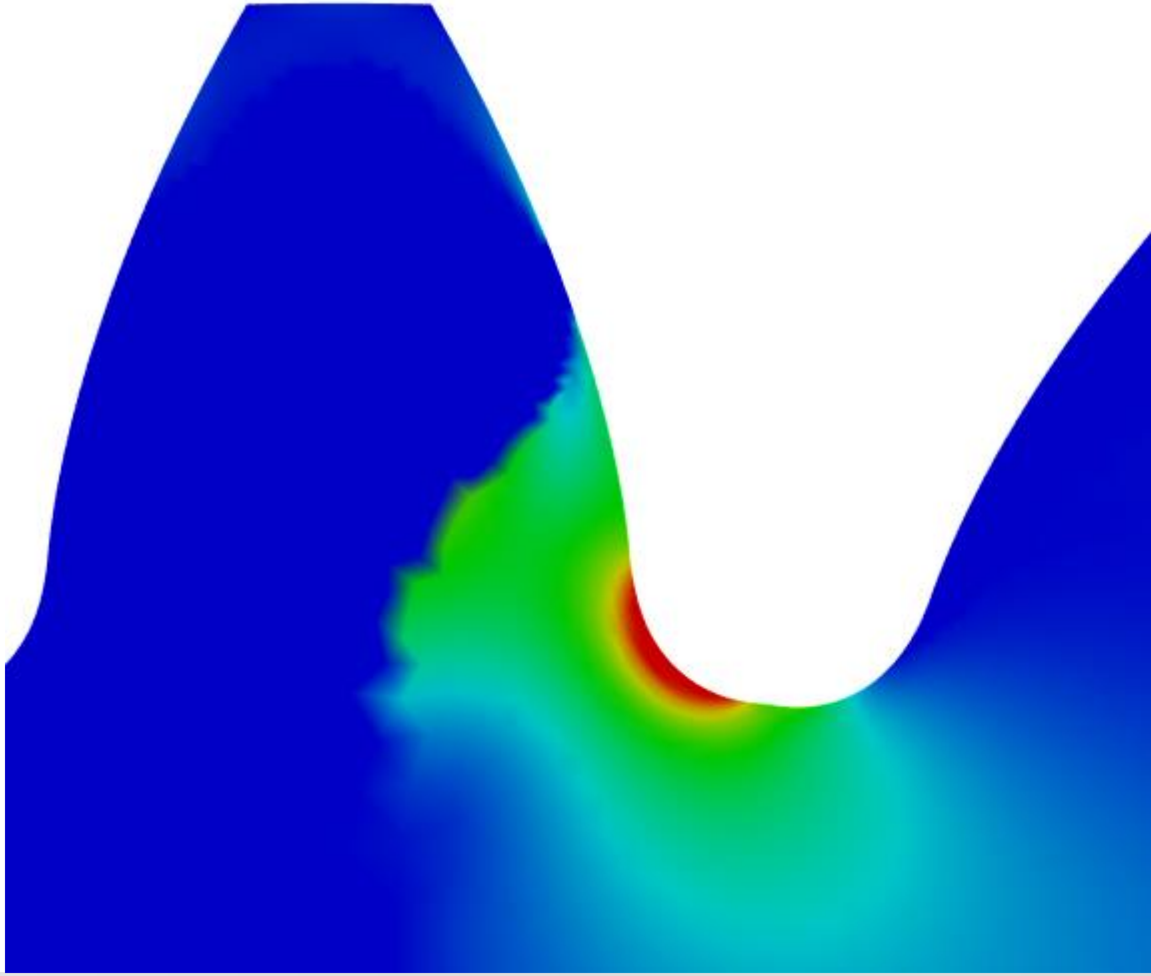

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