AEROSPACE,
DEFENCE & RAILWAYS

Systematic Analysis and Comparison of Stress Minimizing Notch Shapes

- Designing Stress-Concentration free Notches without FEM-Code -

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Altran GmbH & Co. KG

6th SAXSIM, TU Chemnitz

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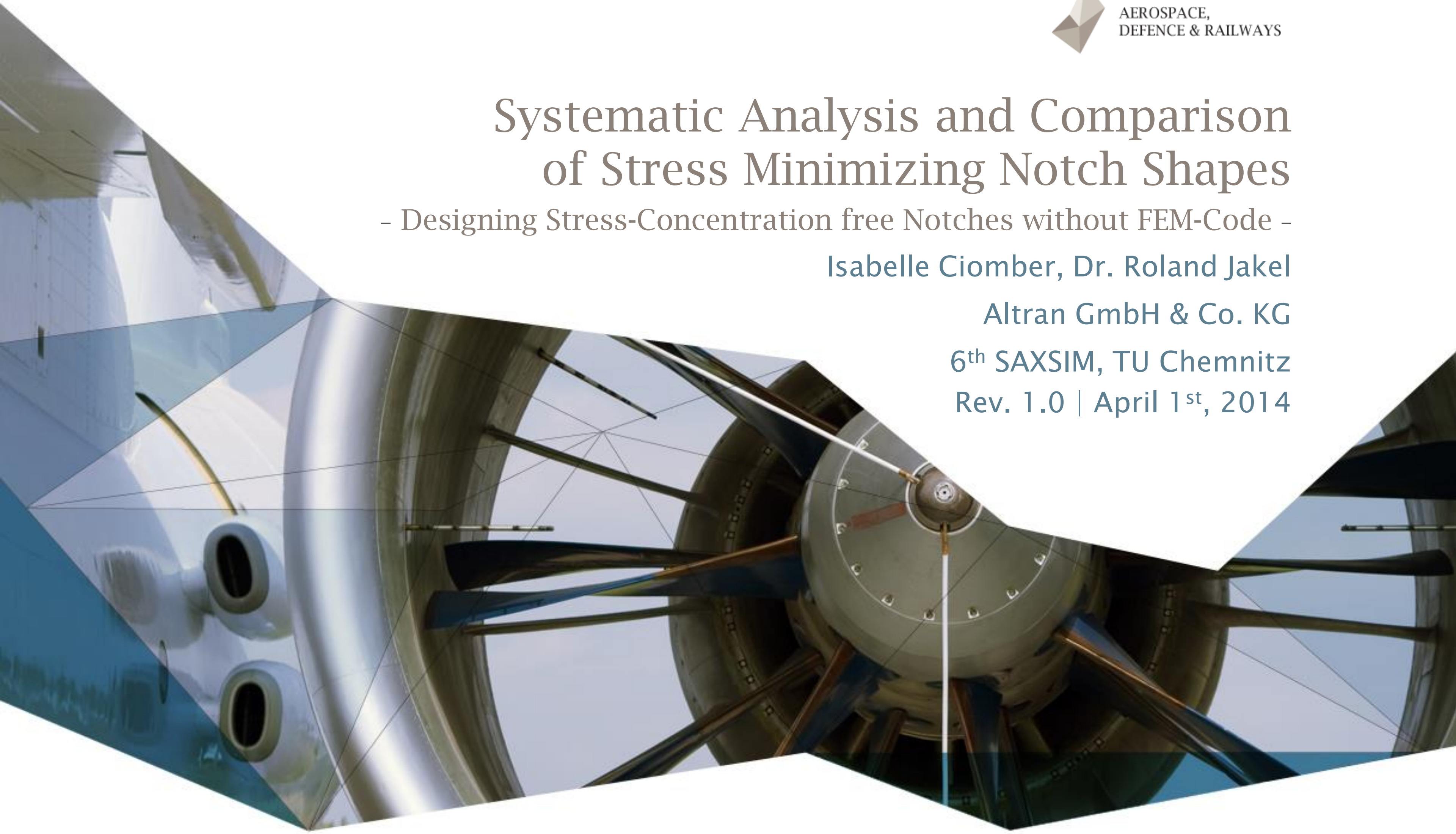

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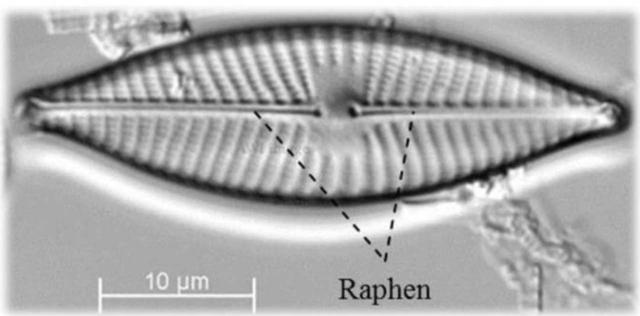
Personal Profile – Isabelle Ciomber

- Bachelor of Science in Biomimetics (Bionik)
- University of Applied Science Bremen

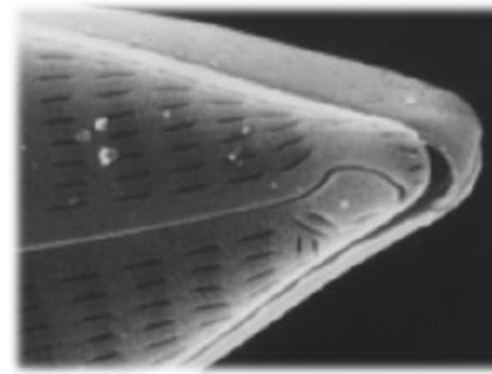
  Bachelor-Thesis:
  Notch Stress Decrease due to Shape Modification inspired by pennate Diatoms

HOCHSCHULE BREMEN
UNIVERSITY OF APPLIED SCIENCES



Adapted from AWI 2013

Light and stable, in spite of notches



Adapted from Round et al. 1990

- Since September 2013:
  Student trainee at Altran
- Since October:
  Master in Production Engineering
  Industrial Management at University Bremen

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Universität Bremen

Personal Profile – Dr. Roland Jakel

Education

- 1985–1990: Mechanical engineering studies at the Technische Universität Clausthal, degree Dipl.-Ing. (TU)
- 1996: Ph.-D. (Dr.-Ing.) in design and analysis of engineering ceramics

Professional Activities

- 1990–1996: Scientific assistant at the Institut für Maschinenwesen, TU Clausthal
- 1996–2001: Development engineer and project leader at Daimler-Benz Aerospace AG, Space Infrastructure (today Airbus Defense & Space); Tasks: Structural simulation of space systems (using NASTRAN and Mechanica) & project (hardware) management for different systems within the Ariane 5 upper stage ESC-A (e.g. stage damping system SARO using friction dampers at cryogenic temperatures)
- 2001 – 2005: Responsible for structural simulation consulting with the PTC products Mechanica, MDX, MDO and BMX at DENC AG (Design ENgineering Consultants, a former technical consultancy for virtual product development)
- 2006 – 2012: After the DENC acquisition, Principal Consultant for the simulation business within the global service organization GSO of PTC for CER (Central Europe: Germany, Austria, Switzerland)
- Since 06/2012: Senior Consultant at TECCON Consulting and Engineering, since 01/2014 Altran GmbH & Co. KG, ASD-R (Aerospace, Defense & Railway)
- Several publications and papers, some are available for download at www.saxsim.de

Who are we?

1,456 M€
REVENUES in 2012

49 %
REVENUES outside France in 2012

20 000
EMPLOYEES in 2012

20+
COUNTRIES in operation

International Group

Altran operates in some twenty countries throughout Europe, Asia and America

Strategic Partner

Altran offers its customers global project support while guaranteeing a consistent level of service

Local Dimension

Altran keeps a local dimension while offering a specific support through geographical management

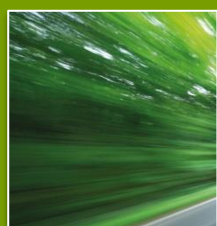
What do we do ?

Altran offers a balanced sector exposure

Altran Group Sales Breakdown by sector (2012)

INDUSTRIES

17%
AUTOMOTIVE, INFRASTRUCTURE & TRANSPORTS



26%
AEROSPACE, DEFENCE & RAILWAY

26%
ENERGY, INDUSTRY & LIFE SCIENCES

12%
FINANCIAL SERVICES & GOVERNMENT

19%
TELECOMS & MEDIA

Global Solutions

Mechanical Engineering
Information Systems
Embedded & Critical Systems
Product Lifecycle Management

Biomimetics: Learning from Nature

Sticking without glue

www.das-auge-denkt-mit.de

Intelligent division of work

Strong

Patek et al. 2006

www.wikipedia.org

Adaption of geometry

Adaptive stress distribution

www.lifeandscience.de

Lightweight design

Perfect flow

Kate Sprogis

Flying

www.wikipedia.org

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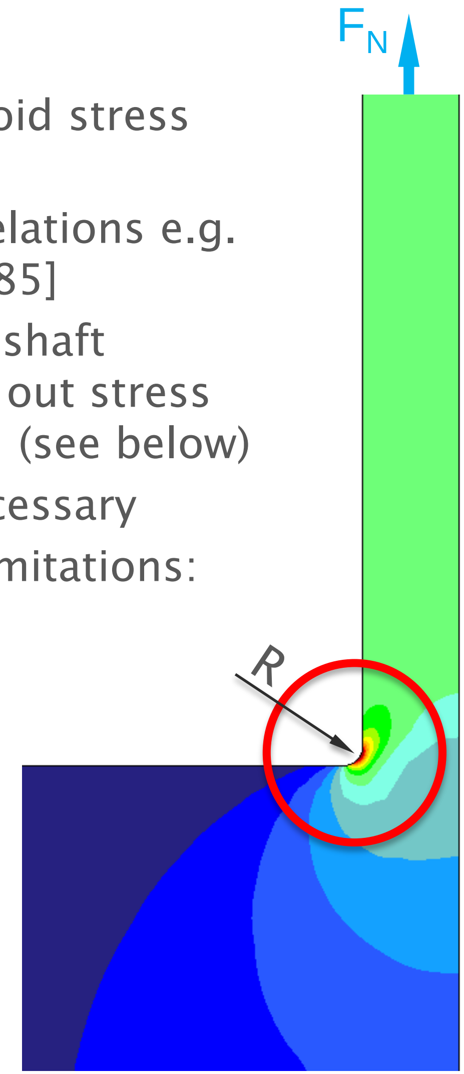
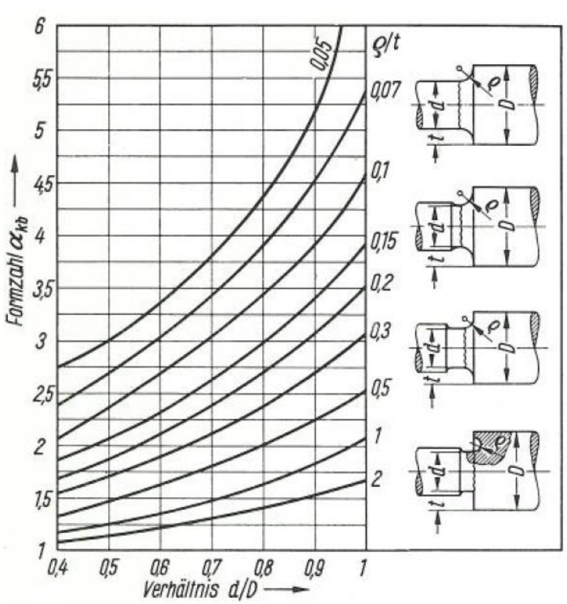
Introduction

1. Motivation



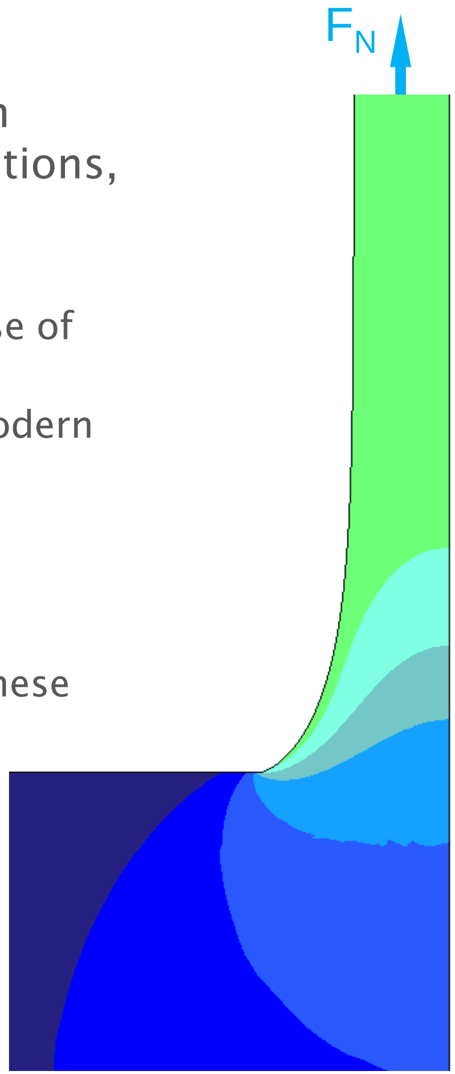
- State-of-the-art is a simple circular radius fillet to avoid stress concentration (typically a quarter circle)
- Classification in literature available for different size relations e.g. at shaft shoulders (diagrams, tables for K_t) [Decker 1985]
- Common design procedure: Determine geometry (e.g. shaft shoulder measures & radius) and load state, then read out stress concentration factor (K_t, in German α_k) from a diagram (see below)
- Advantage: Simple to apply, no tests or FE analysis necessary
- However, the state-of-the-art shows two significant limitations:



1. The efficiency of stress reduction is very limited with a circular notch
2. More efficient non-circular shapes usually have to be individually analyzed by numerical methods (FEM)

2. Intention

- Goal is to provide well characterized non-circular notch shapes for standard load cases and cross section transitions, which
  1. are very efficient for stress reduction;
  2. are easy to analyze without use of a FE analysis, only by use of special normalized diagrams as for the circular notch and
  3. can be easily created by the use of standard features of modern CAD tools
- To reach this goal efficiently:
  - A CAE tool is required that allows to create full-parametric geometry models
  - Subsequent FEM sensitivity and optimization analyses of these shapes based on the parametric model must be supported
- Therefore, Creo Parametric and Creo Simulate in embedded mode were chosen (PTC Inc.)

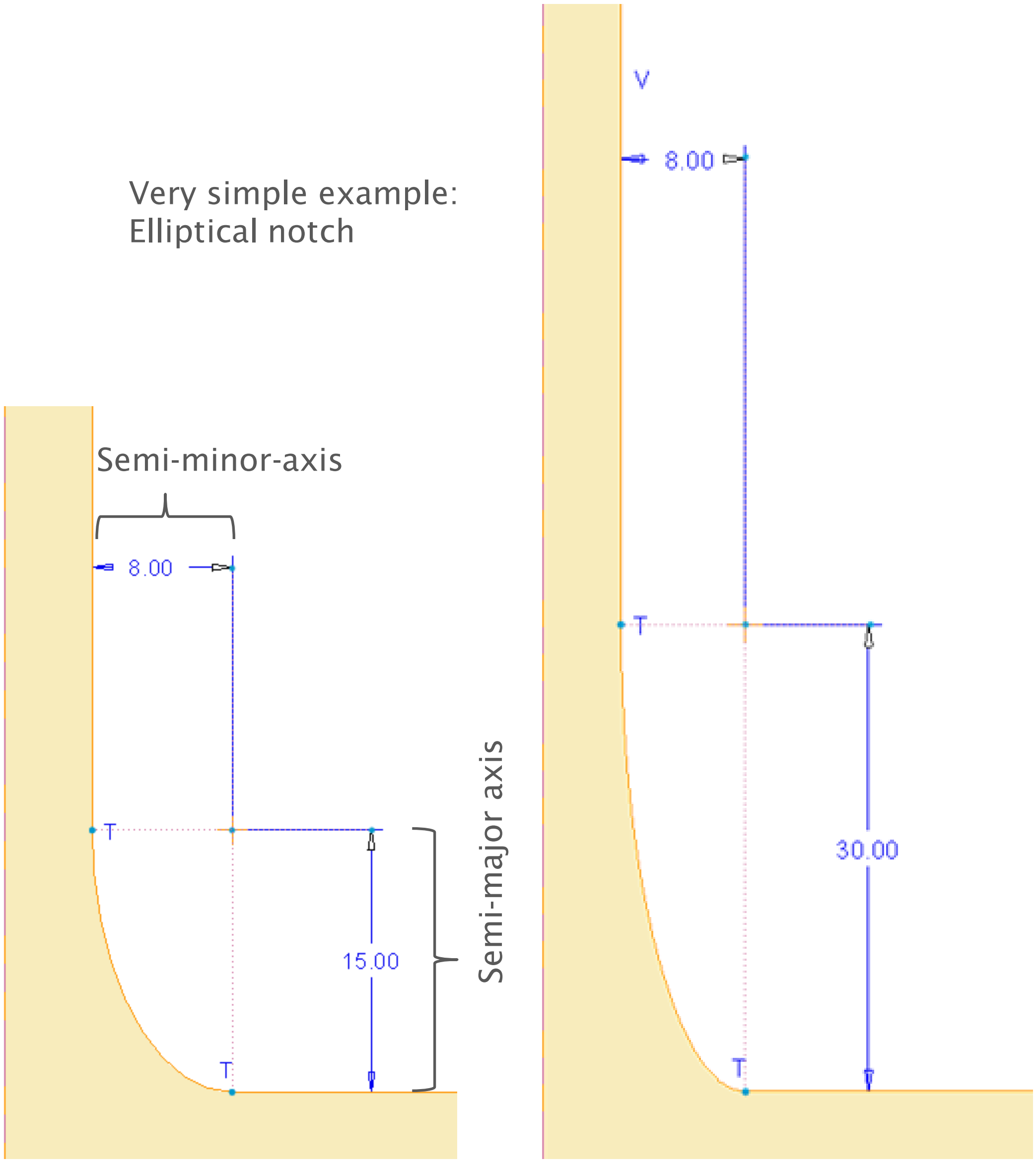


Part A: Used Software Functionality

1. Parametric Modelling

- In a parametric CAD model, all objects/features like points, lines, curves, planes, etc. are described associatively as parameters
- The parameter values can be changed easily; the model just has to be newly regenerated
- Important for this is the clean, logical construction of the CAD model
- The ideal base of a successful notch sensitivity and optimization analysis is a good parameterization of the CAD notch model, since many different sizes/relations shall be automatically examined!

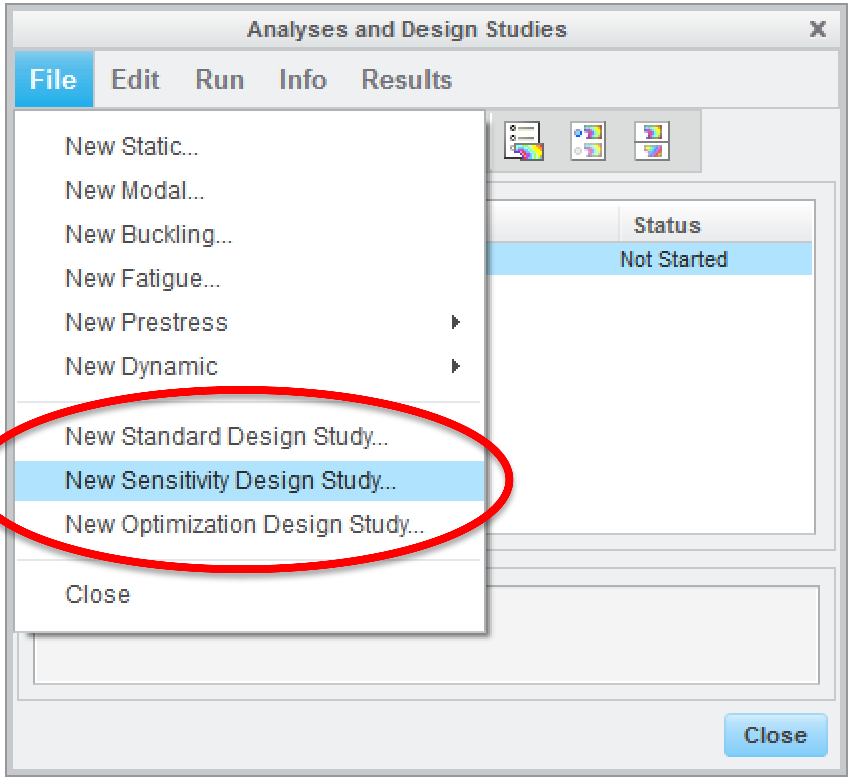
Very simple example:
Elliptical notch



2. Software tools for analyzing parametric geometry

2.1 Introduction

- Creo Simulate offers different study types that allow to examine the influence of different parameters to certain physical quantities (stress, strain, stiffness, mass,…) and optimize the structure for certain goals
- These parameters can be feature dimensions (e.g. wall thickness, radii), like used for this project, or certain physical properties (E-modulus, spring stiffness,…)
- Since the CAD model is set up fully parametric, a wide range of parameter values (measures of the notches) can be studied and the results can be very efficiently displayed in sensitivity diagrams
- Note: Simulate just owns a parameter optimizer, no shape optimizer! Smart model (dimension) parameters therefore have to be created!

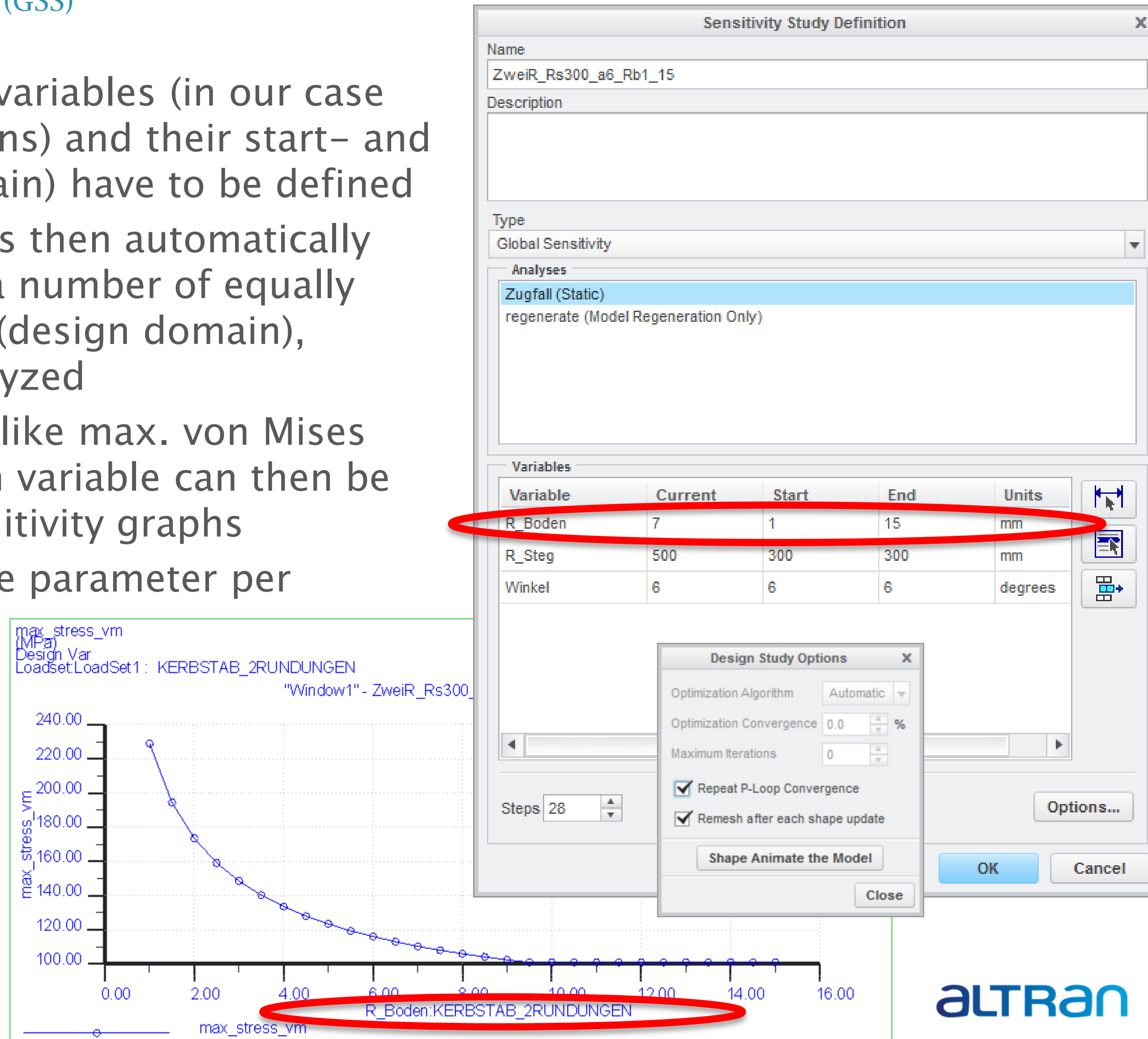


2. Software tools for analyzing parametric geometry

2.2 Global Sensitivity Study (GSS)

- In a GSS, design variables (in our case feature dimensions) and their start- and end values (domain) have to be defined
- The CAD model is then automatically regenerated for a number of equally spaced intervals (design domain), meshed and analyzed
- Measure results (like max. von Mises stress) vs. design variable can then be displayed in sensitivity graphs

Hint: Just change one parameter per analysis to better understand its influence!

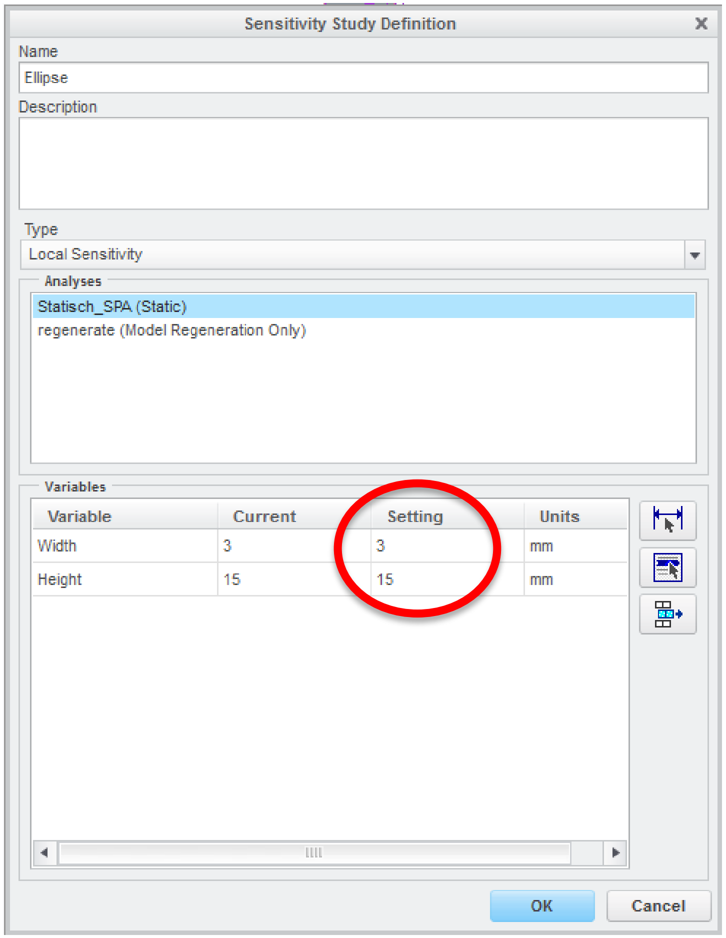


2. Software tools for analyzing parametric geometry

2.3 Local Sensitivity Study (LSS)

- The LSS was not used directly in the project; it is described here just for completeness
- This study allows to determine the variable affecting the results the most – but just for one “operating point” per analysis!
- It is thought to preliminary examine the model to exclude design variables from an optimization study that have no or just a small influence only
- Creo Simulate varies (±1%) the setting value individually for each variable and then just computes the *slope* for it at the actual operating point

Hint: Unlike in the GSS, in the LSS you may therefore vary all design variables simultaneously!

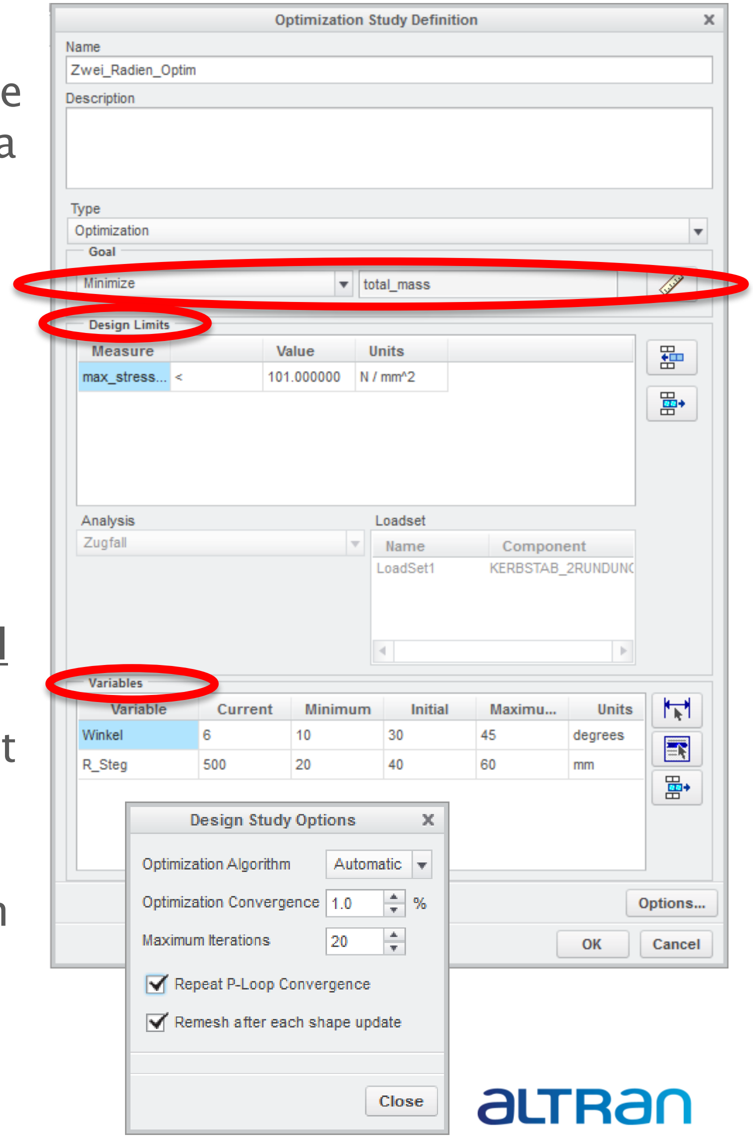


2. Software tools for analyzing parametric geometry

2.4 Optimization Study (OS)

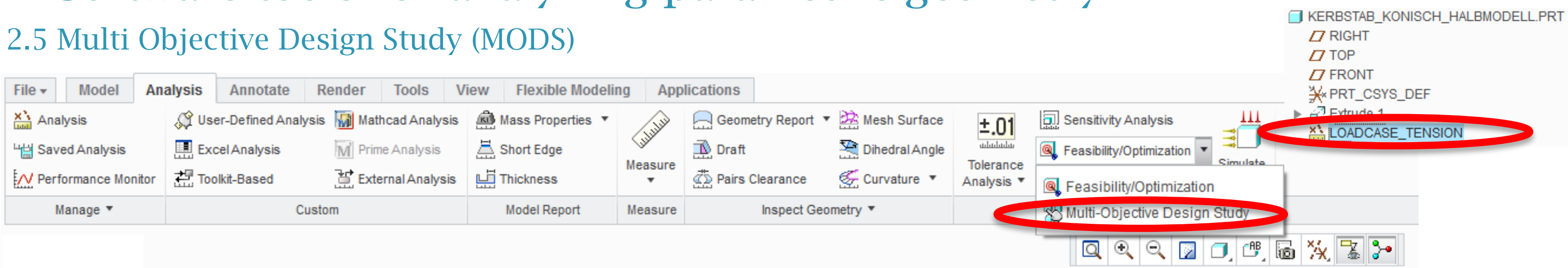
- In this study, the defined design variables are driven within their allowed domain to reach a certain goal while still defined design limits have to be satisfied (e.g. stress < 100 MPa)
- An optimization study without a goal (e.g. “minimize total mass”) is a feasibility study!
- Two algorithms are available:
  SQP – Sequential Quadratic Programming
  GDP – Gradient Projection
- External optimizers may be embedded
- This study allows to search for the next <u>local</u> minimum or maximum, which is not necessarily the global one: The starting point may be important therefore!

<u>Hint</u>: Application experience shows that with more than two design variables, this study often does not work very efficient!

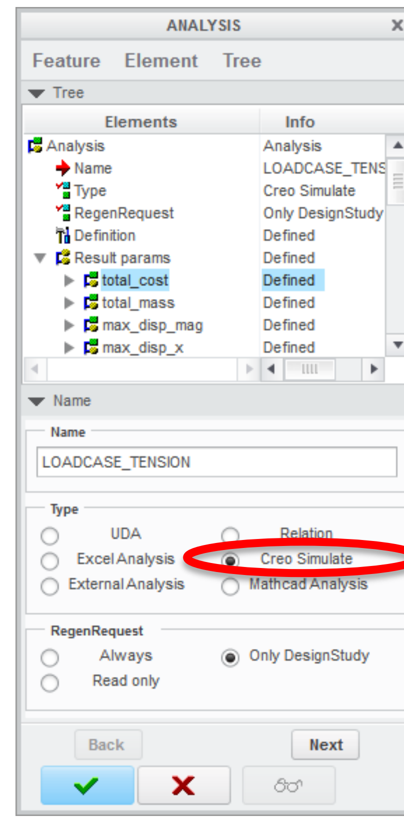


2. Software tools for analyzing parametric geometry

2.5 Multi Objective Design Study (MODS)



- Probably less known to Simulate users is the MODS, since this is part of Creo Parametric BMX functionality (Behavioral Modeling EXtension)
- This study allows to examine a complete design space with many design variables systematically
- To obtain the link from BMX to Creo Simulate, an analysis feature has to be created in the model tree that refers to a Simulate FEM analysis
- This analysis feature defines which result parameters (=measures from the simulate analysis) have to be computed by Simulate and recorded by the BMX MODS
- The obtained parameter results for the complete design space can then be evaluated by various table and graph display functions to better understand the model behavior



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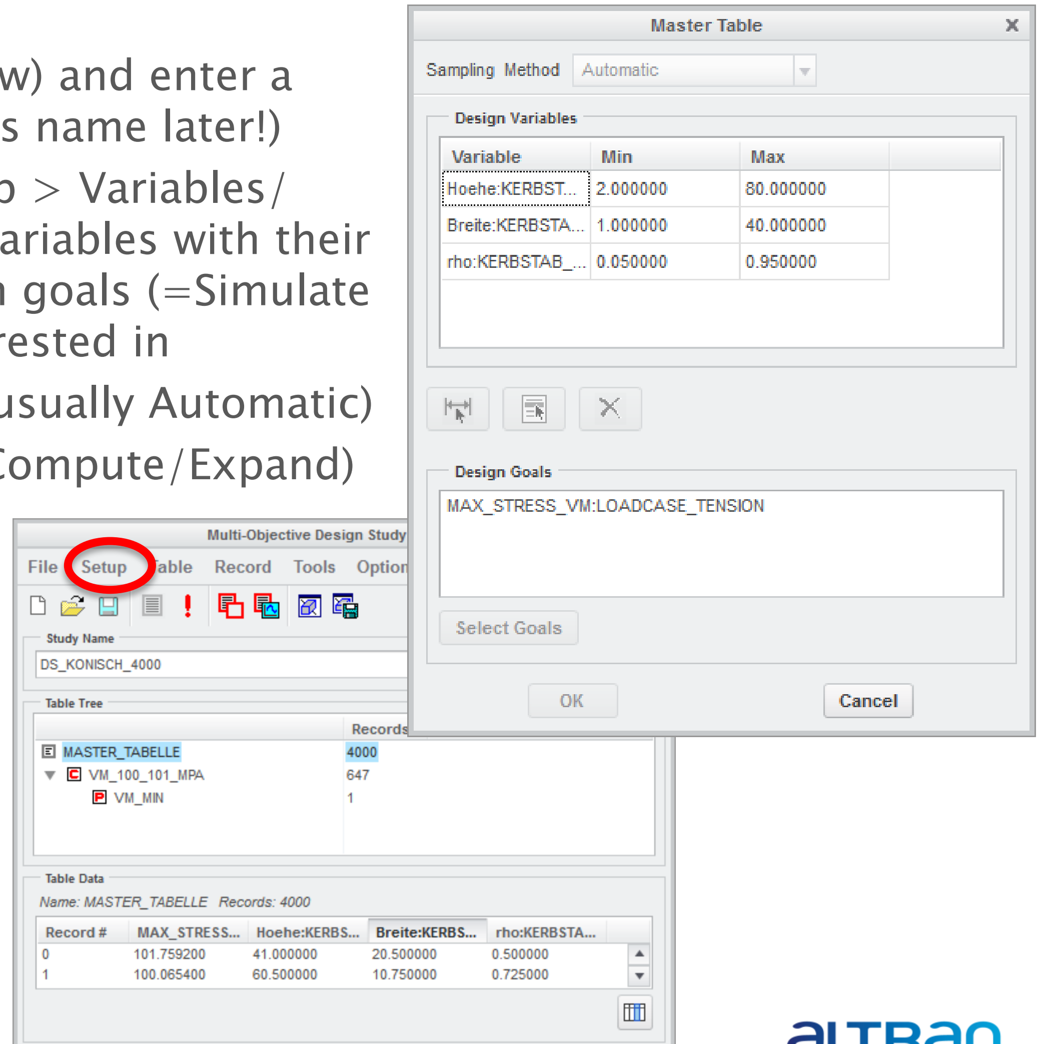
2. Software tools for analyzing parametric geometry

2.5 Multi Objective Design Study (MODS)

Setting up a MODS:

1. Create a new study (File > New) and enter a study name (never change this name later!)
2. Create the master table (Setup > Variables/ Goals) and define all design variables with their domains as well as the design goals (=Simulate measure results) you are interested in
3. Select the Sampling Method (usually Automatic)
4. Compute the table (Setup > Compute/Expand) by entering the number of experiments to generate (often more than 1000 may be required!)
5. Wait (e.g. 4000 notch samples need 8 hours to complete)



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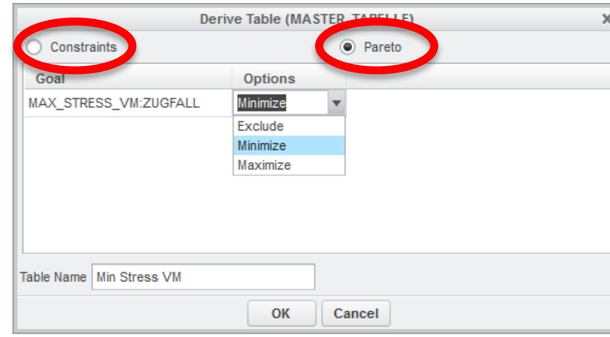
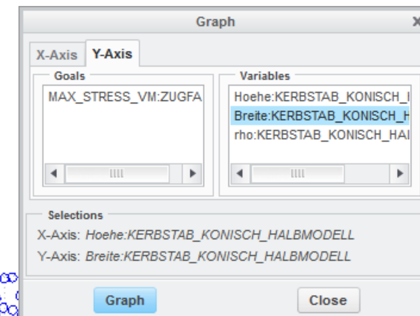
2. Software tools for analyzing parametric geometry

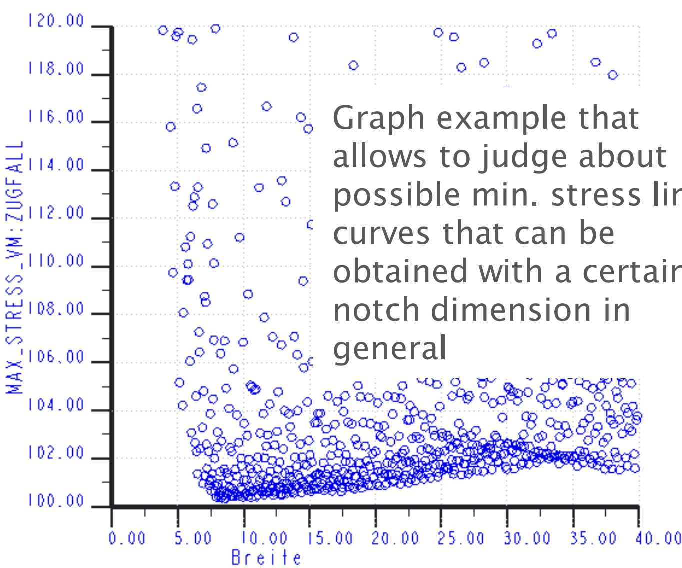
2.5 Multi Objective Design Study (MODS)

Evaluating a MODS analysis result:

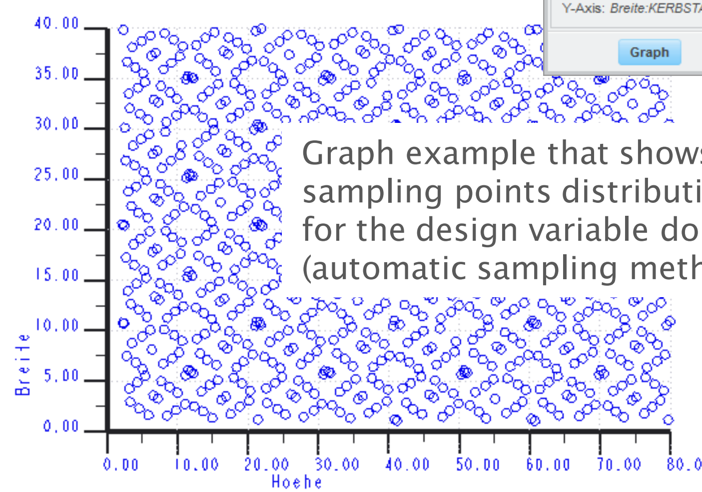
- Study the table result data with all the single experiment data (Table > Show Data)
- Derive new subtable(s) using constraints → define min/max values of the goal(s)
- Derive new subtable(s) using Pareto method → minimize, maximize or exclude goal(s)
- Derive 2D-Graphs (Tools > Graph Study) (unfortunately, no 3D graphs are supported)







Graph example that allows to judge about possible min. stress limit curves that can be obtained with a certain notch dimension in general



Graph example that shows sampling points distribution for the design variable domain (automatic sampling method)

3. Software tools for analysis speed and accuracy control

3.1 p-FEM Mesh: Auto-GEM controls

Goal:

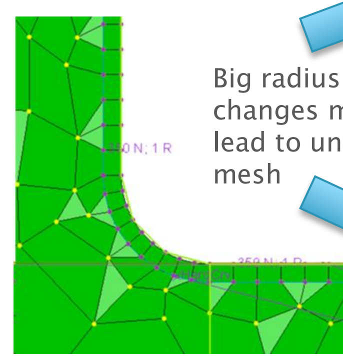
- Stress results have to be very accurate for each notch size to draw smooth sensitivity graphs for the complete domain

First Approach:

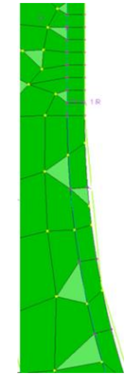
- Use surface areas and edge distribution AutoGEM Controls to obtain undistorted quad elements
- It turned out that if the geometry changes heavily during the sensitivity analysis, elements will distort and a new, often unwanted mesh appears

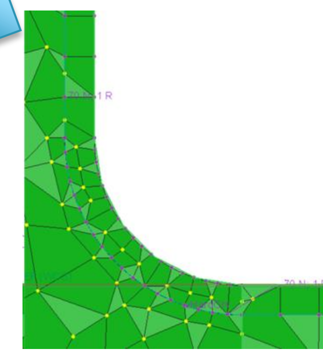
Chosen Approach:

- Mesh the most interesting part of the geometry just with smaller elements, no attention to triangles or quad elements
- In addition, use advanced SPA controls like described next slide



Big radius size changes may lead to unwanted mesh





Edge distribution AutoGEM control cannot be parameterized!

Just smaller elements and reduced allowed element edge angle domain and aspect ratios (1:3) are better to reach the goal!

3. Software tools for analysis speed and accuracy control

3.2 Convergence settings for sensitivity analysis

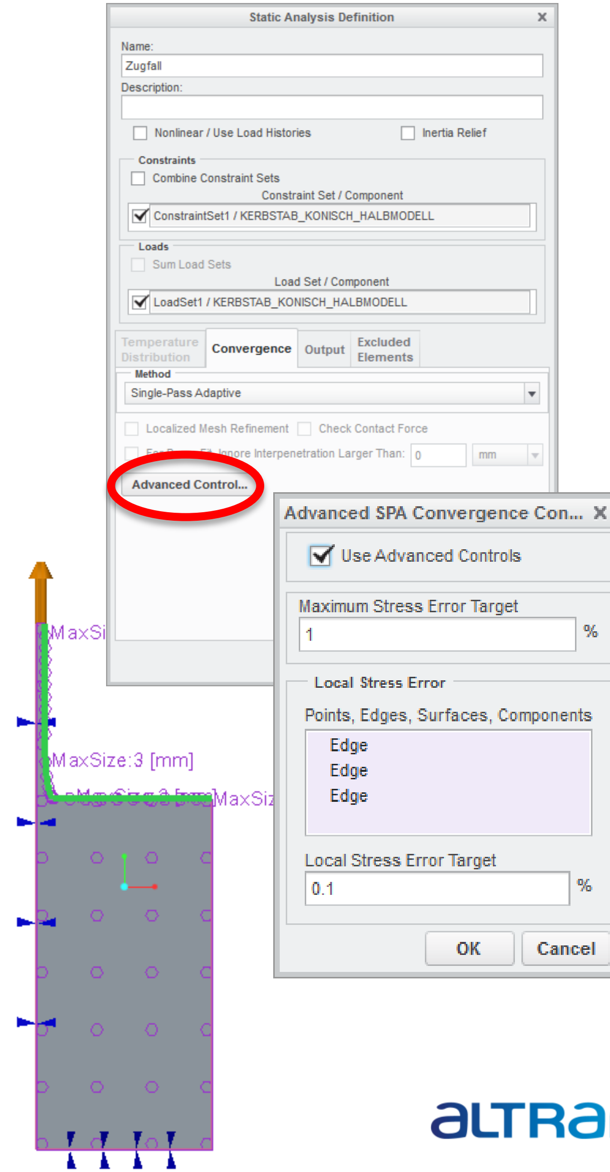
Goals:

- Stress results have to be very accurate
- Analysis speed has to be fast even though, since a huge amount of notch sensitivity studies have to be performed in limited time

Approaches to reach the goals:

- Use 2D models where ever possible (plane stress, plane strain, axial symmetric)
- Do not use the accurate, but relatively slow multi-pass adaptive (MPA), instead use single pass adaptive convergence method (SPA)
- Use advanced SPA controls to decrease the local RMS stress error from default 8 % to values <<1 % near the notch surface

<u>Remark:</u>
The RMS stress error is the max. difference between direct (raw) element stress and smoothened (superconverged) stress [Zienkiewics, Zhu 1987]



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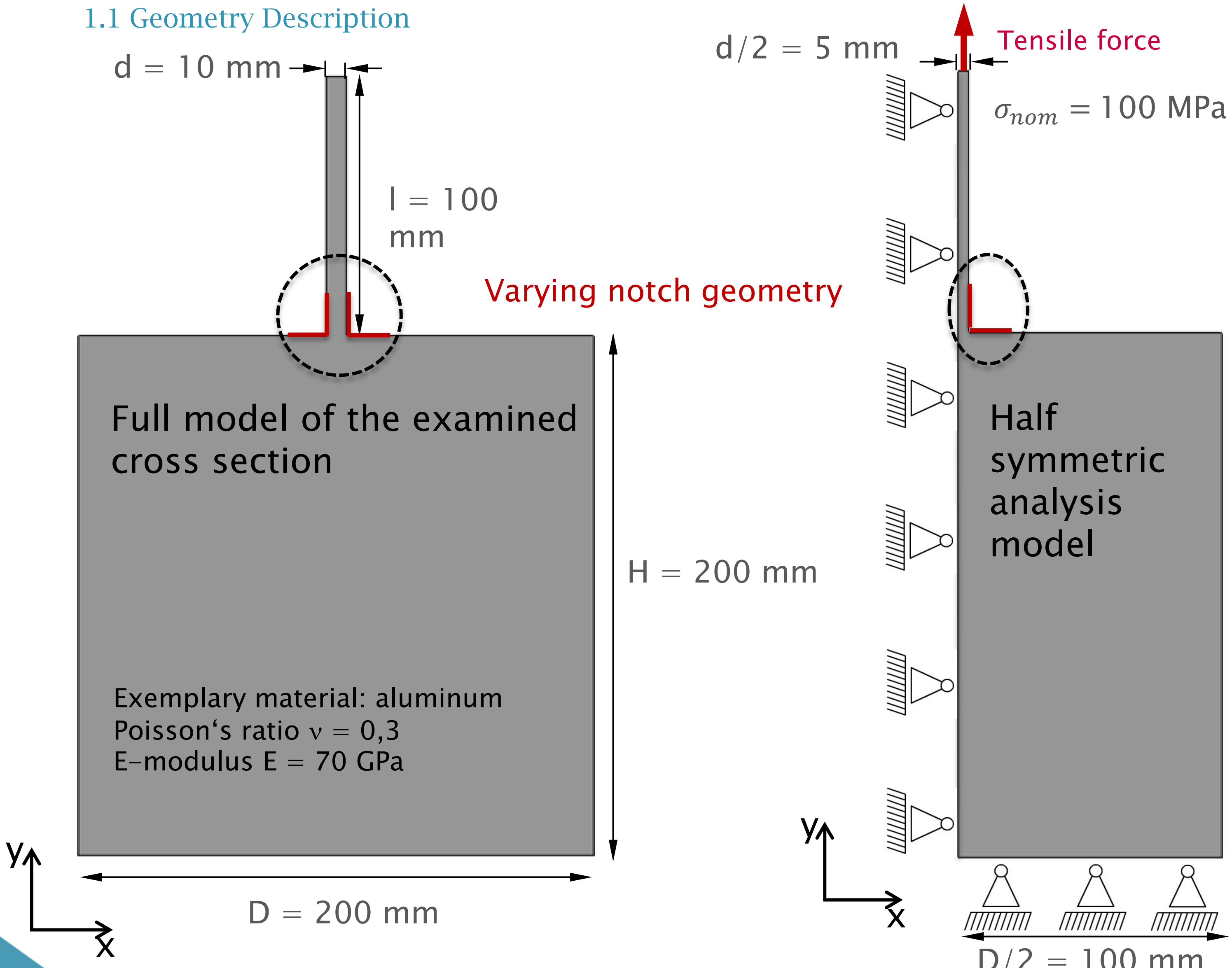
Part B: Application
Notch Stress Decrease due to the Variation of Geometry

1. Model of the Cross Section Change used for Notch Examination

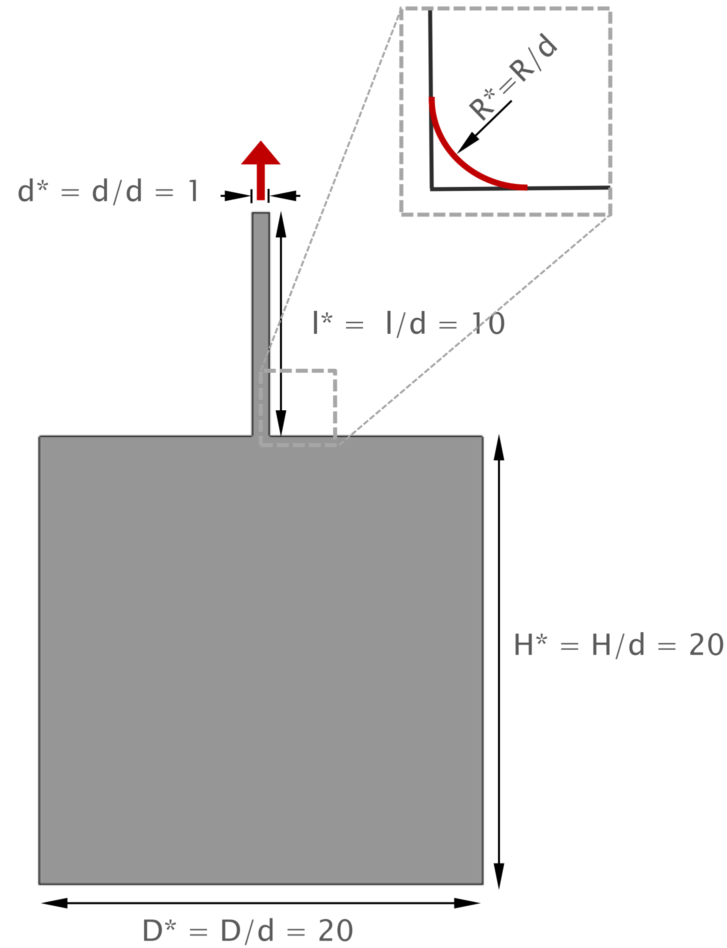
1.1 Geometry Description



1. Model of the Cross Section Change used for Notch Examination

1.2 Normalization of Dimensions

- In order to create dimensionless diagrams, the model dimensions have been normalized
- All measures are related to the web thickness d=10 mm
- It will be shown later that D* and H* are large enough to have worst-case notch stresses. Smaller values, especially for D*, usually lead to a notch stress decrease
- l* was arbitrarily chosen to be 10: It just has to be long enough so that the load introduction point is far enough away from the notch; therefore it is subsequently not further taken into account
- For non-circular notches, the normalized notch height is named h* = h/d and the width b* = b/d

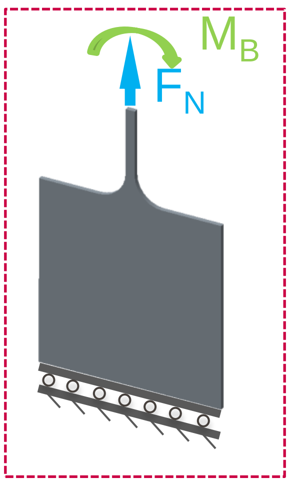


2. Range of Validity

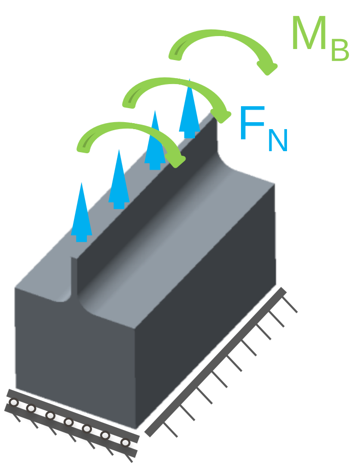
2.1 Load Cases and Geometry Models

Analyzed was just the tension load for 2D plane stress, since it turned out this is the worst-case condition for the examined cross-section:

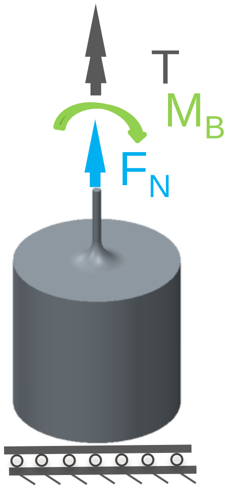
- Tension creates higher notch stress than bending: Bending is less critical
- Notch stress for the 2D plane stress condition is always higher than in the state of 2D plane strain or rotational symmetric case
- Tension creates higher notch stress than torsion in the rotational symmetry-model: Torsion is less critical



Plane stress model



Plane strain model

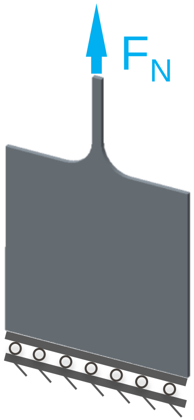


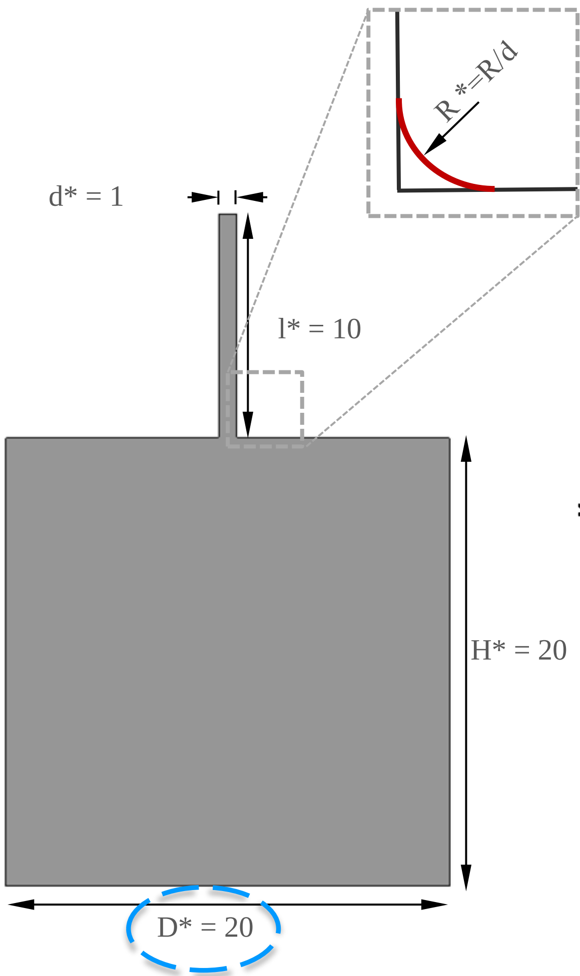
Rotational symmetric model

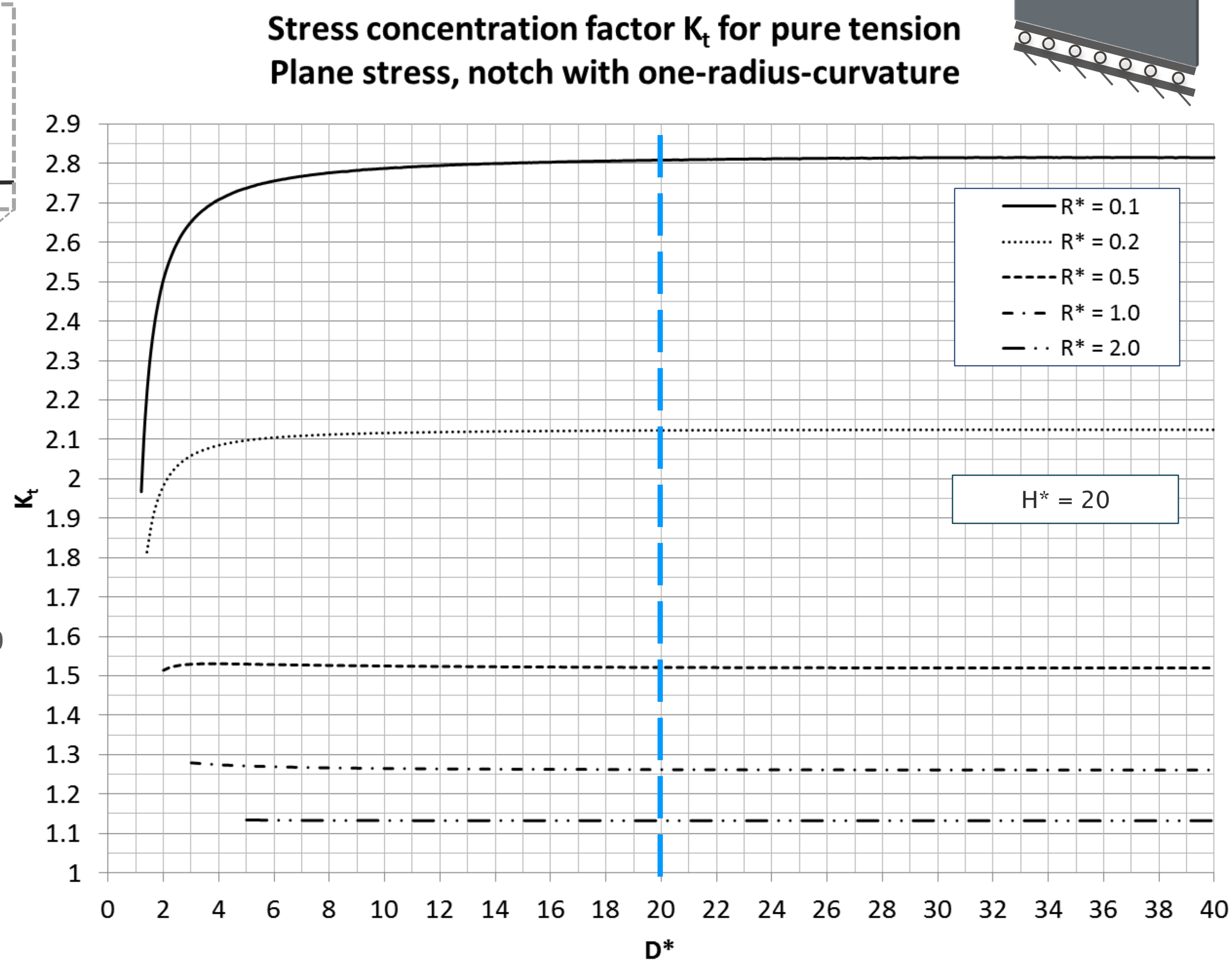
2. Range of Validity

2.2 Width Sensitivity

Width D* = 20 chosen sufficiently large
→ Almost no further influence on notch stress



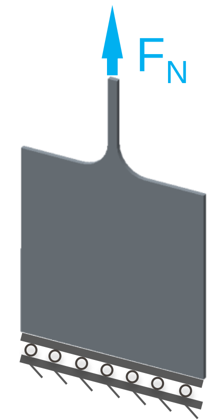


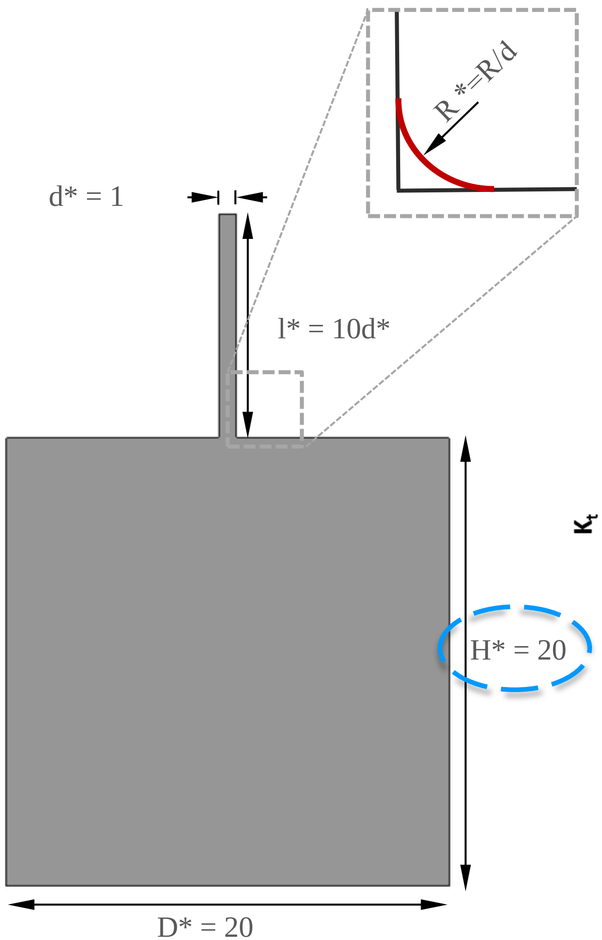


2. Range of Validity

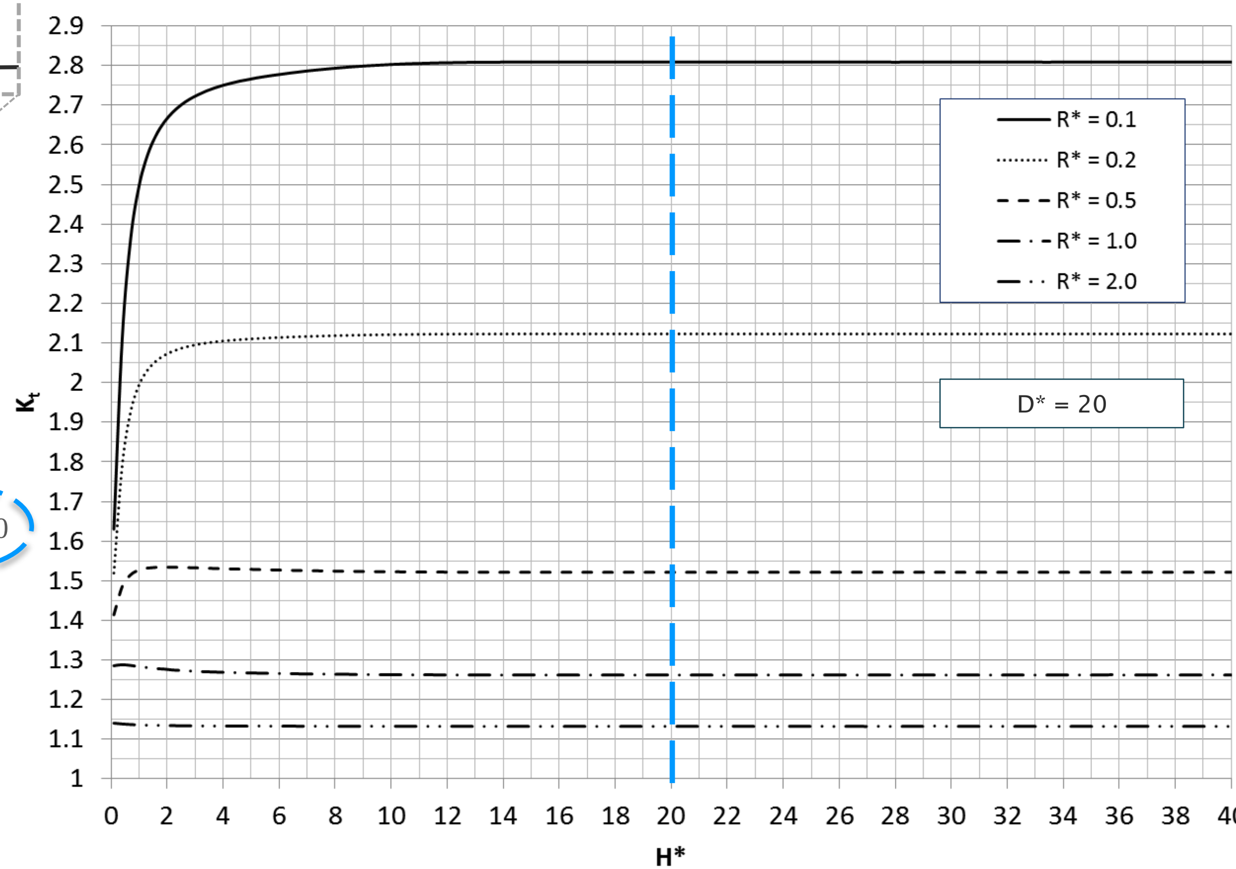
2.3 Height Sensitivity

Height H* = 20 chosen sufficiently large
→ Almost no further influence on notch stress





Stress concentration factor K_t for pure tension
Plane stress, notch with one-radius-curvature



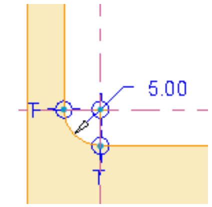
3. Examined Notches

3.1 Overview

Six different notch shapes have been examined:

1. One-radius-fillet as “state-of-the-art”
2. Two-radii-fillet
3. Baud-fillet (R.V. Baud)
4. Method of tensile triangles (C. Mattheck)
5. Standard elliptical fillet
6. Conical round as generalized elliptical fillet

1.



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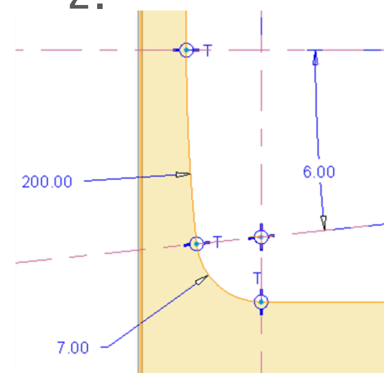

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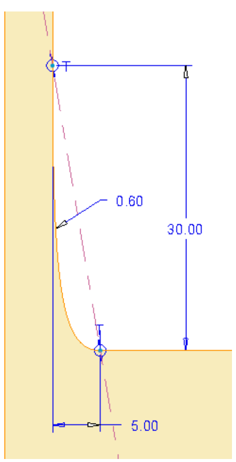

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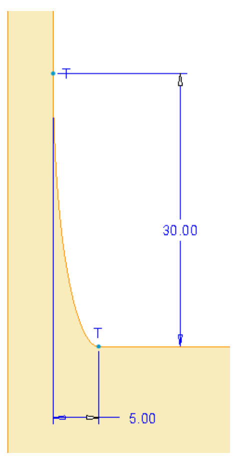

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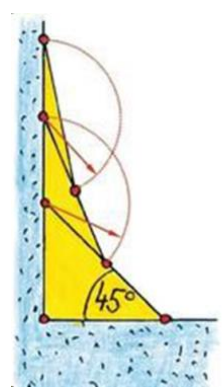

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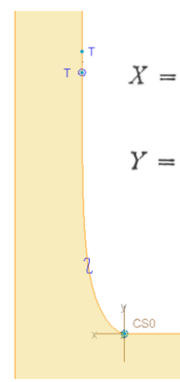
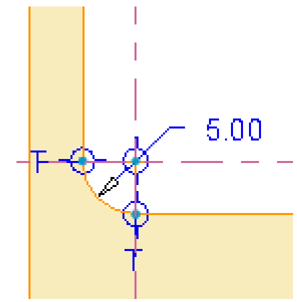


$$X = 2\,C \sin^2 \frac{\theta}{2}$$

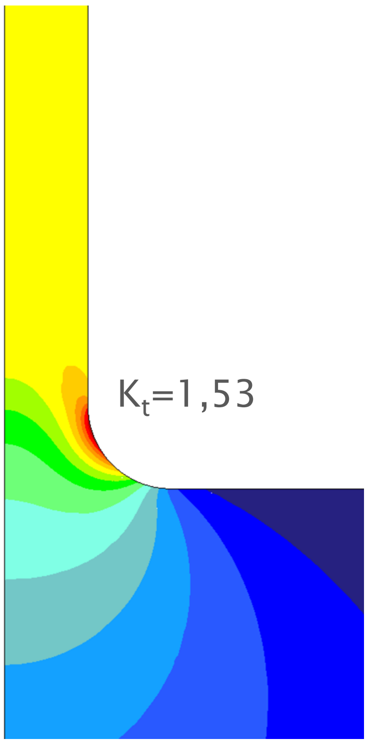
$$Y = C\left[\log \tan\left(\frac{\theta}{2} + \frac{\pi}{4}\right) - \sin\theta\right]$$

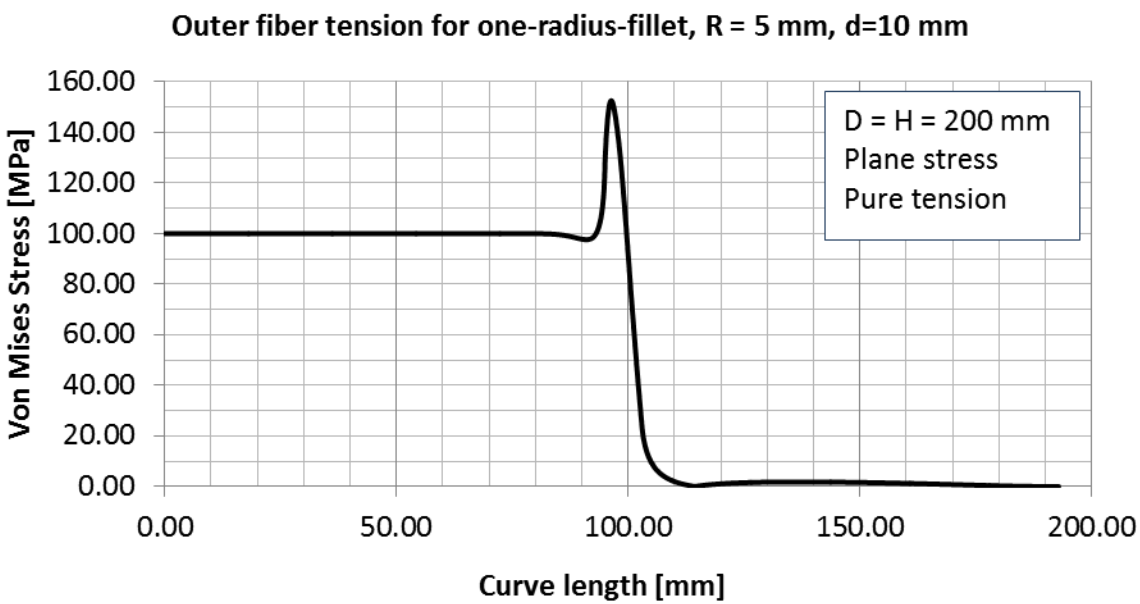
3. Examined Notches

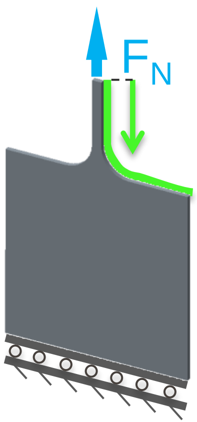
3.2 One-Radius Fillet



- For one–radius–fillets still notch stresses remain
- E.g. radius 5 mm, normalized to d = 10 mm → R* = 0.5:



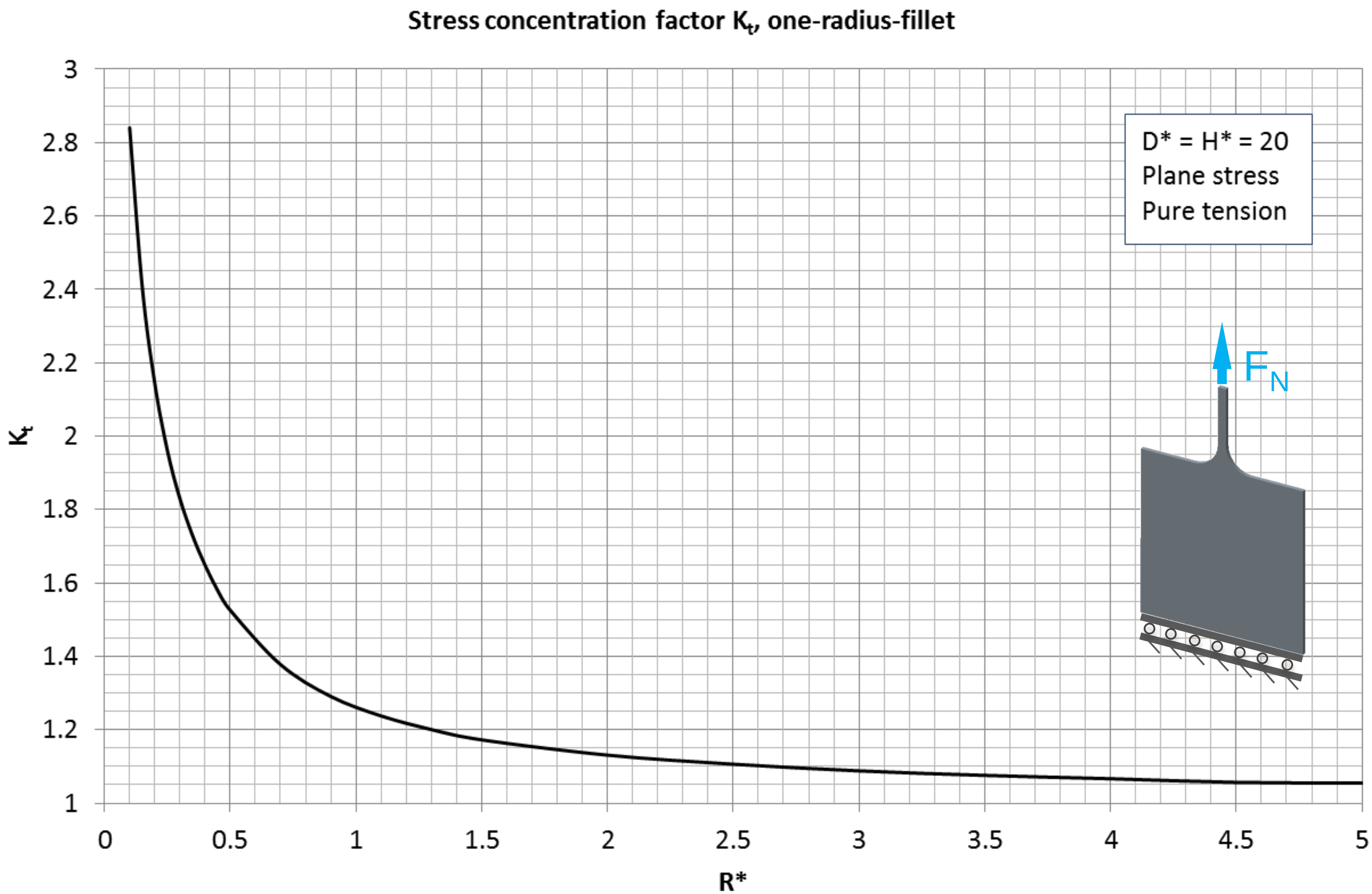




- Exmined radius domain: 0.1 ≤ R* ≤ 5
- Minimal K_t = 1.05 for R* = 5

3. Examined Notches

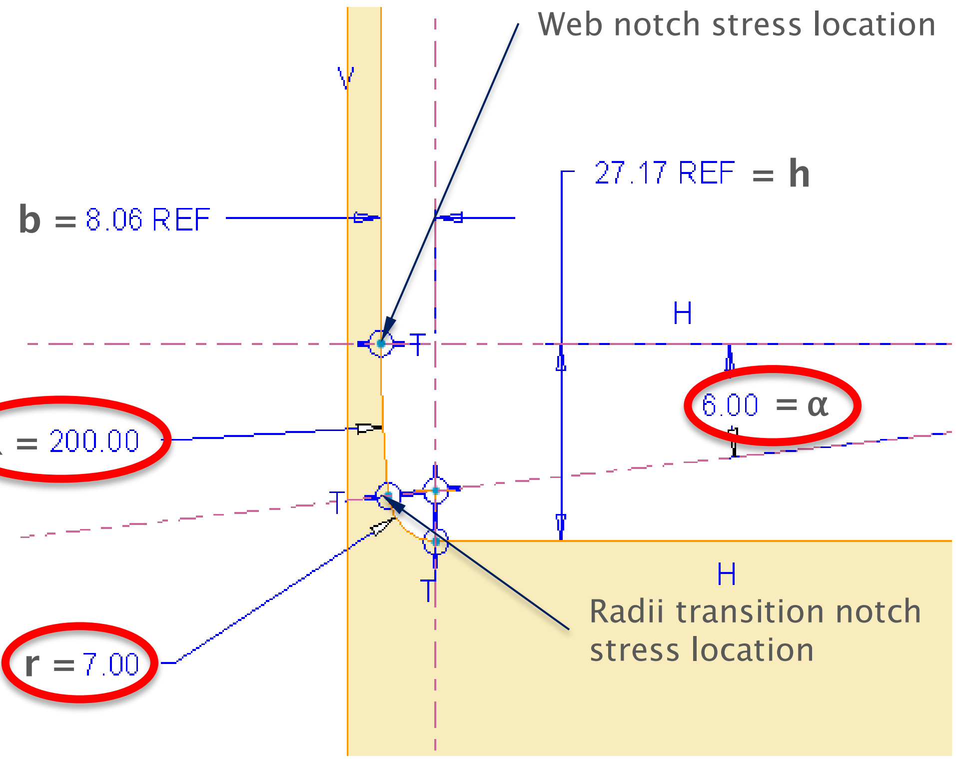
3.2 One-Radius Fillet



3. Examined Notches

3.3 Two-Radii Fillet

- This notch type uses a large radius R at the web joined to a small radius r at the base with tangent transitions, respectively
- R has the dominant influence on the web notch stress → choose R as large as possible
- For size reasons, R only prevails for a small angle, e.g. $3° \leq \alpha \leq 10°$ (horizontal line as starting point)
- r and α are adjusted in order that radii transition notch stress does not increase over the web notch stress and the used design space is as small as possible

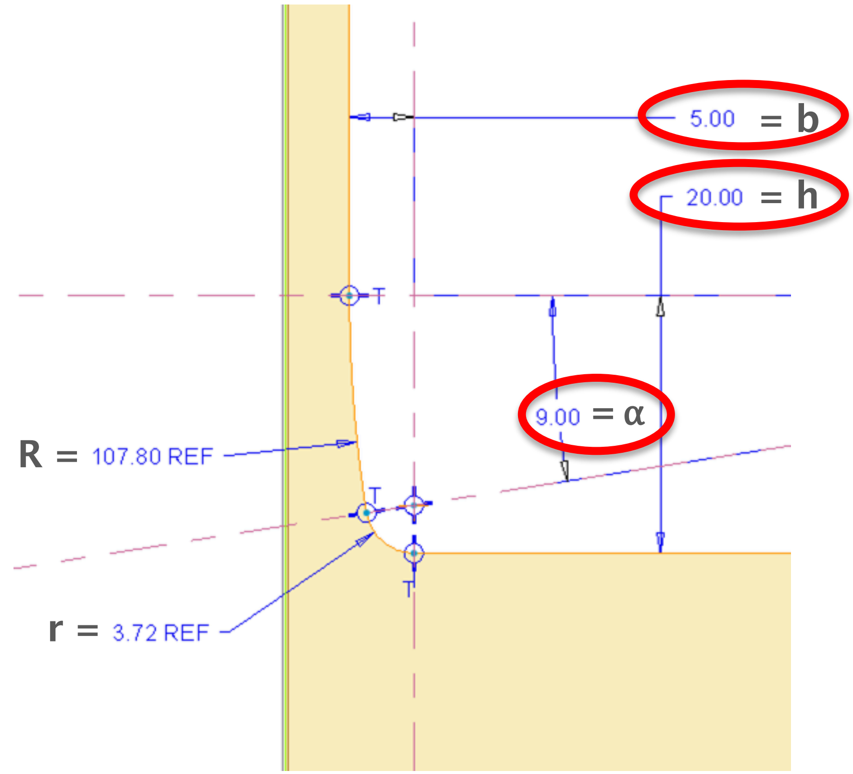


Radii R, r and angle α are varied (height h and width b of the notch geometry only as reference dimension!)

3. Examined Notches

3.3 Two-Radii Fillet

- Alternatively, sensitivity studies with a different sketch set-up have been performed: Height h, width b and angle α of the notch geometry are varied; radii R & r only as reference dimensions
- This method of approach is advantageous if the focus is on the absolute notch size, not on the radii to be manufactured
- Subsequently, we will first show some results for this set-up



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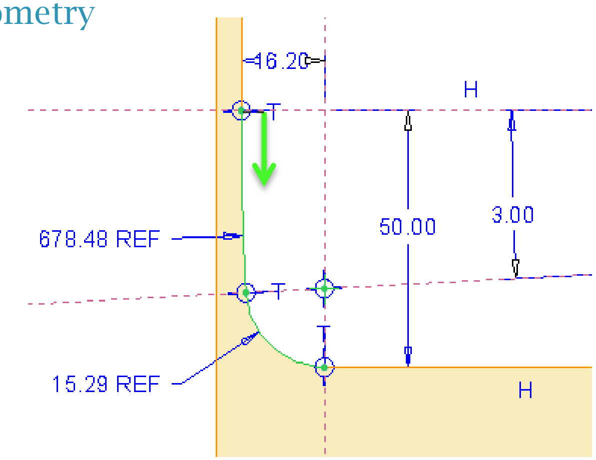
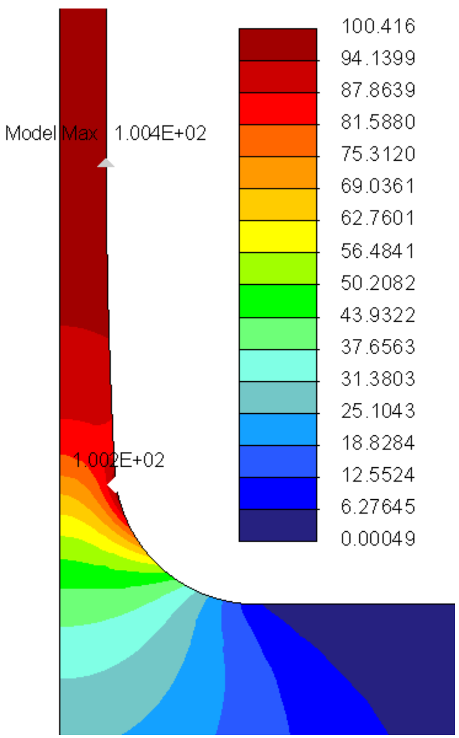
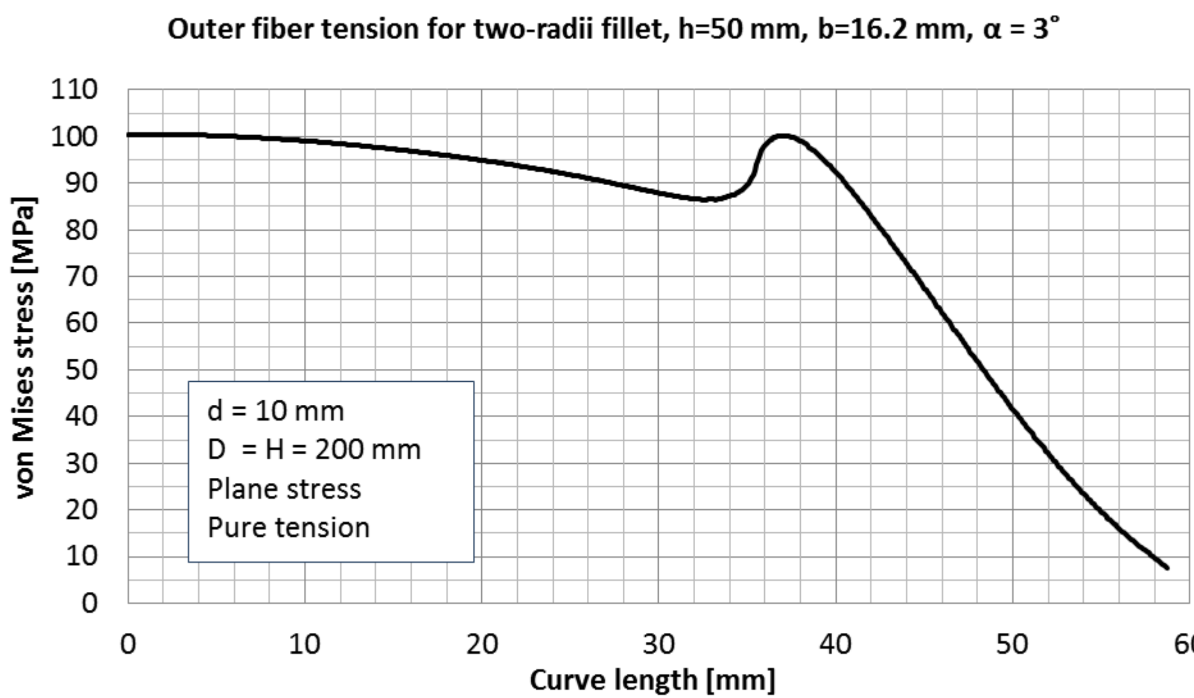
3. Examined Notches

3.3 Two-Radii Fillet

- For a two-radii fillet, in relation to the needed design space a much better stress reduction can be obtained compared to the one-radius fillet
- A very good example with $K_t=1.0042$ is depicted here:



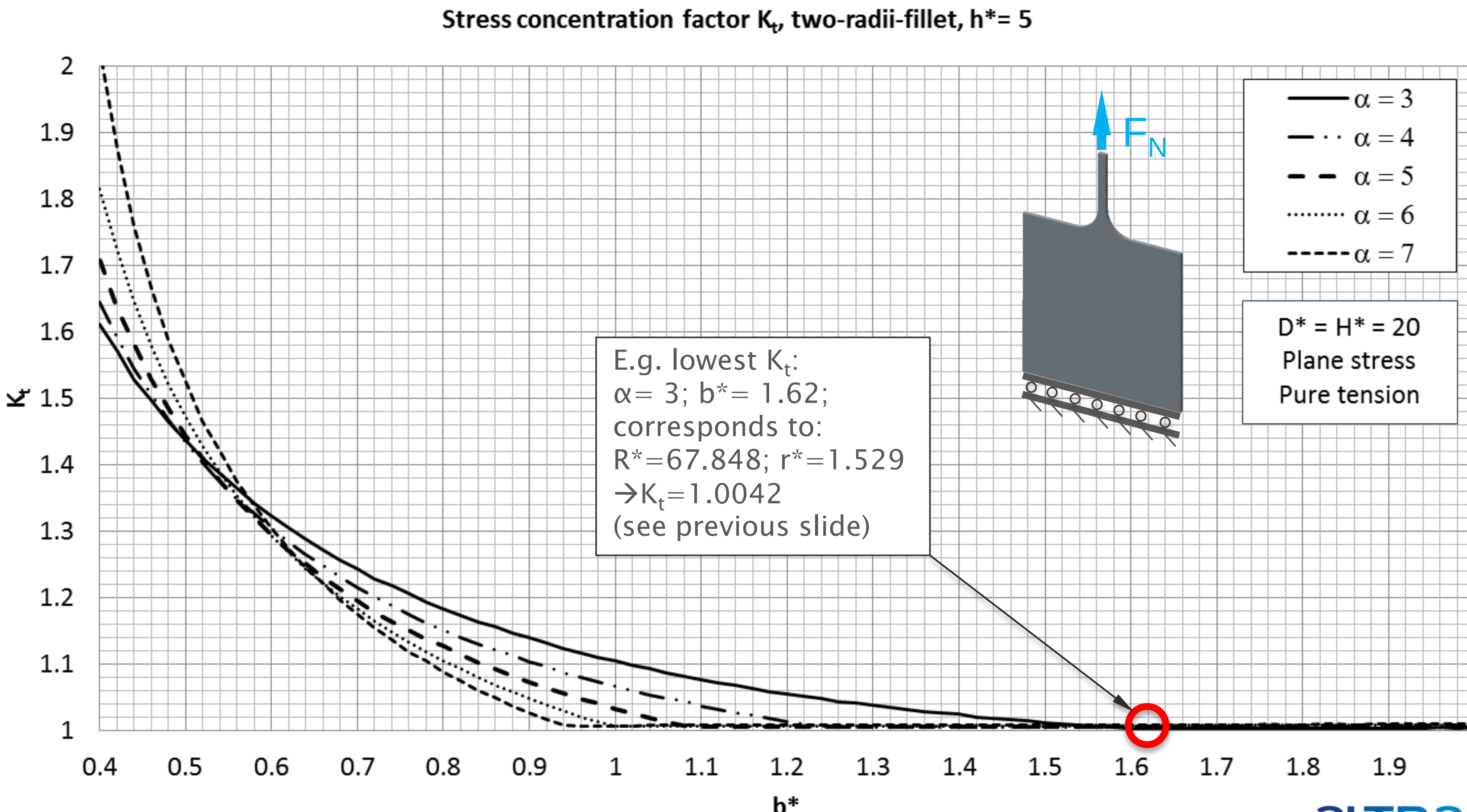




3. Examined Notches

3.3 Two-Radii Fillet

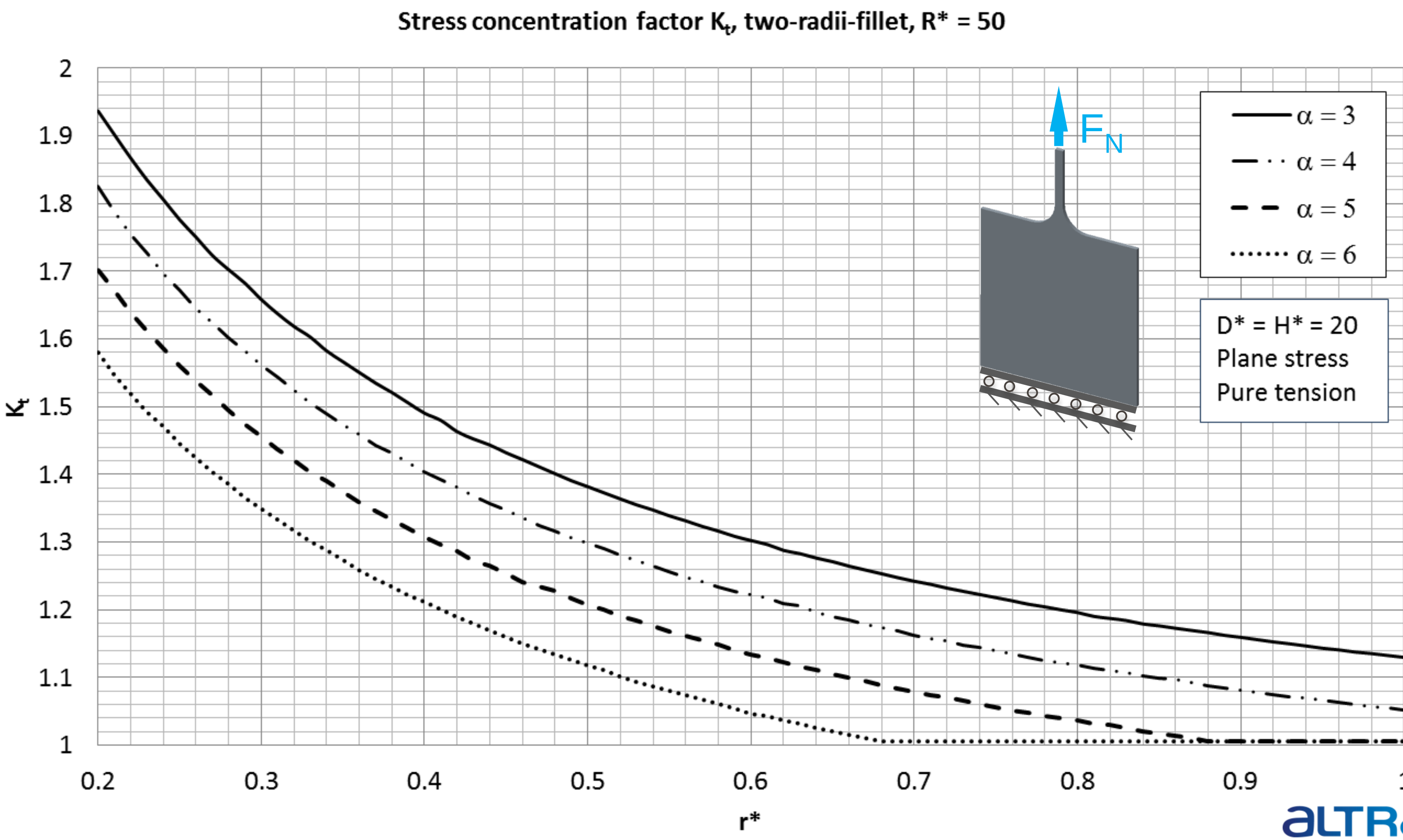
- The following diagram exemplifies the stress concentration factor K_t for a notch height of h*=5 (variable are b* and α)



3. Examined Notches

3.3 Two-Radii Fillet

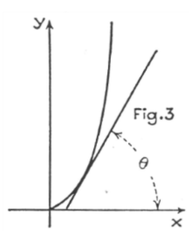
- The following diagram exemplifies the stress concentration factor K_t for a notch radius $R^*=50$ (variable: r^* and α)



3. Examined Notches

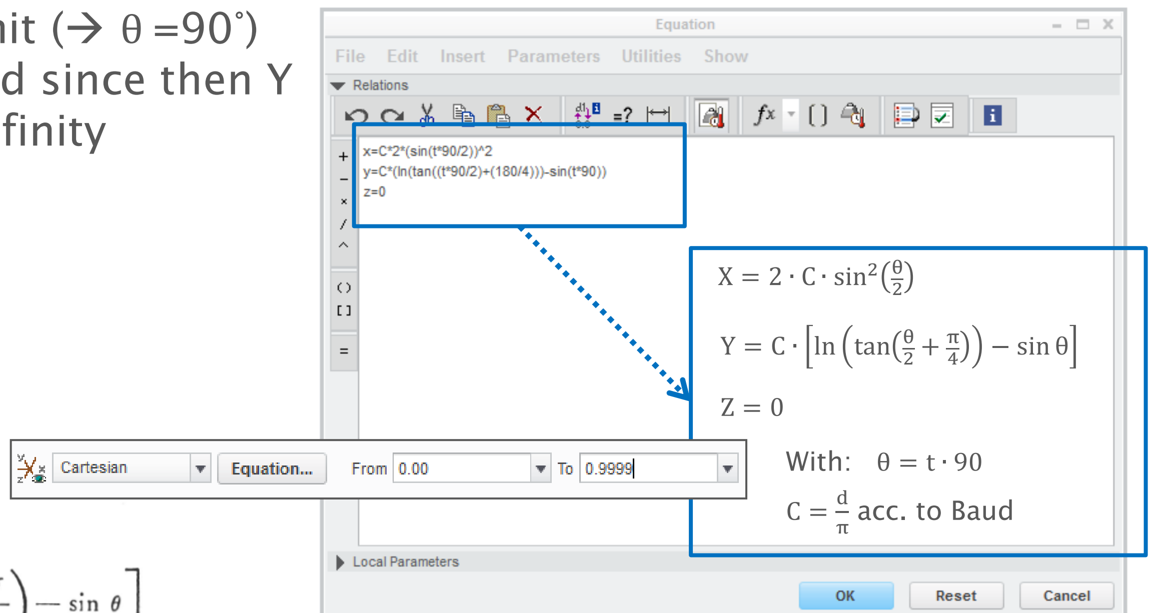
3.4 Baud Fillet

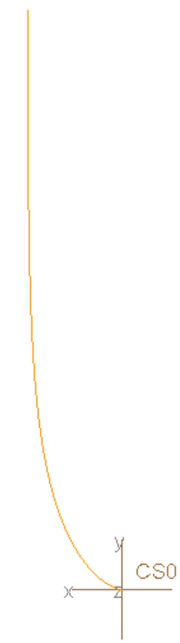
- [Baud, R.V, 1934] recommended to use the shape of a free jet of water with the equation given below, proposing C=d/π (note log = ln)
- This is a curve described in parametric representation with the control variable θ as angle of the curve to the X-axis
- This can be easily coded in Creo Parametric, which expects a parametric representation in the curve equation editor
- Here, t is used as control variable $0 \le t \le 0.9999$ to generate the curve
- 1 as upper limit (→ θ =90°) cannot be used since then Y approaches infinity



$$X = 2\,C \sin^2 \frac{\theta}{2}$$

$$Y = C\left[\log \tan\left(\frac{\theta}{2} + \frac{\pi}{4}\right) - \sin\theta\right]$$





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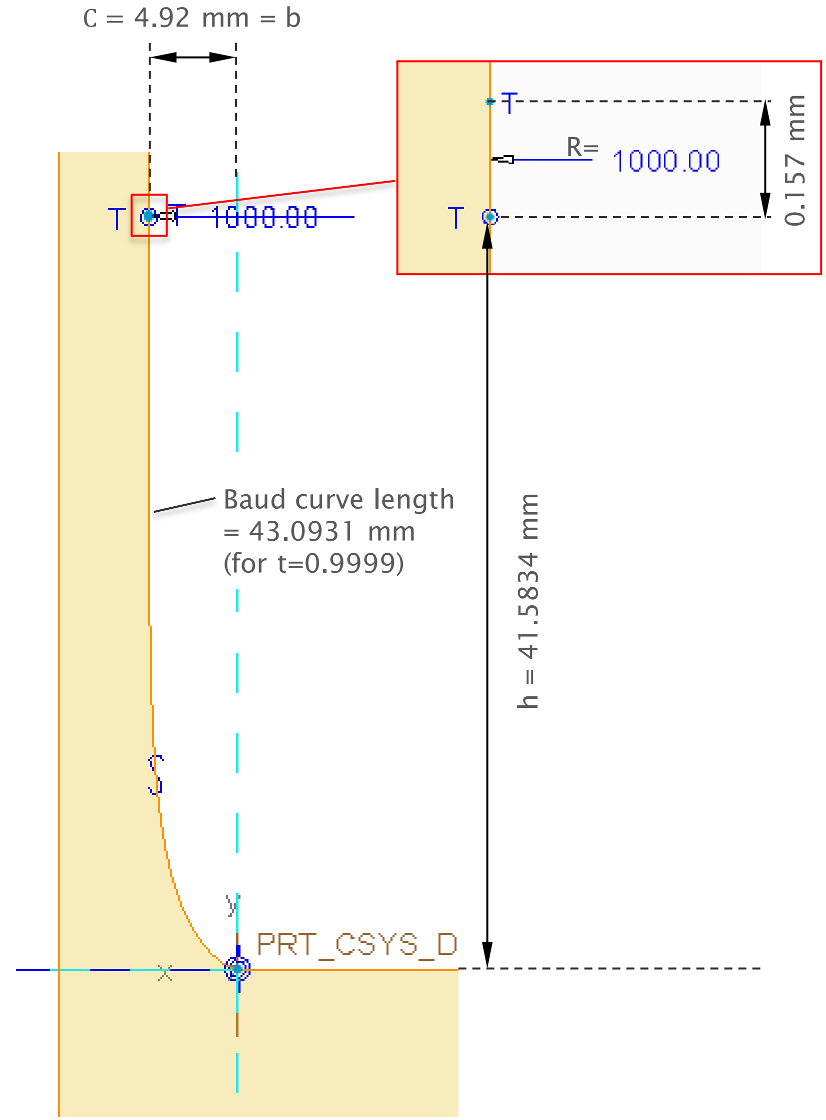
3. Examined Notches

3.4 Baud Fillet

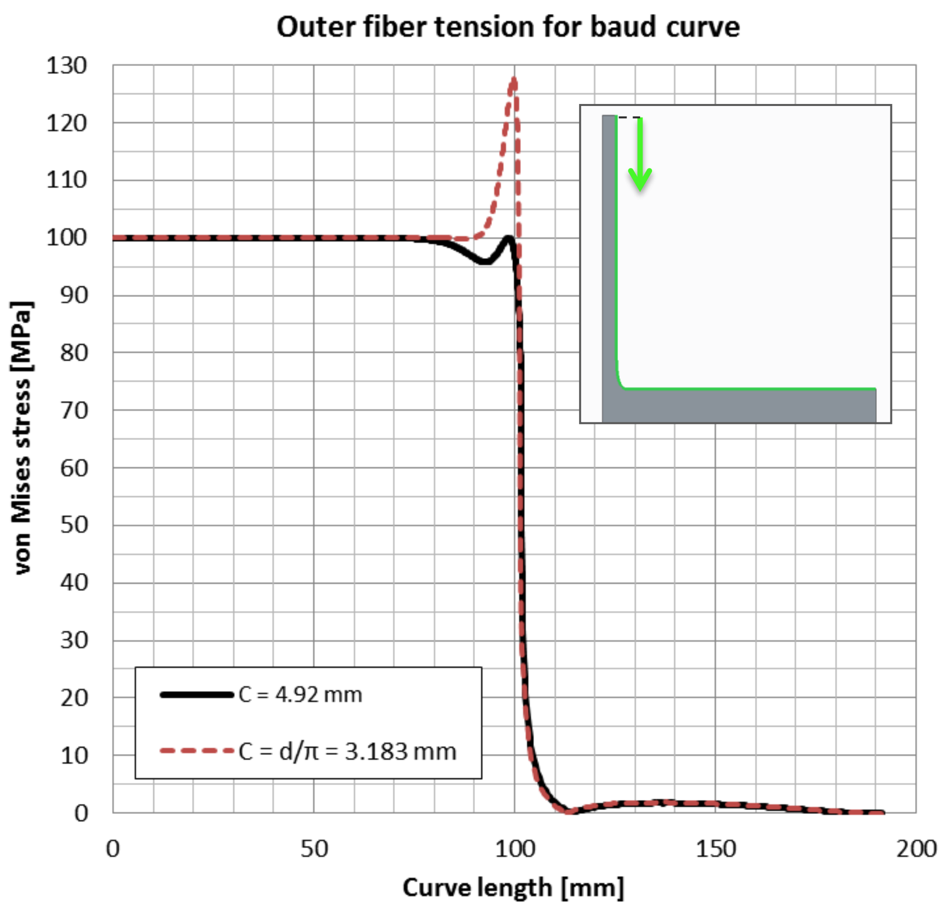
- To obtain a smooth transition to the web after t approaches 0.9999, a tangent constant radius of R=1000 mm was added
- C reflects the width b of the notch. It was therefore regarded as design variable and was varied between 1 ≤ C ≤ 6
- After normalization, we obtain b*=C*=C/d=0.1…0.6
- For the Baud recommendation we have b*=1/π=0.3183
- The notch dimensions shown right reflect the ideal dimensions found in this project (b*=0.492): Smallest notch size with K_T=1 (exact value 1.00046)



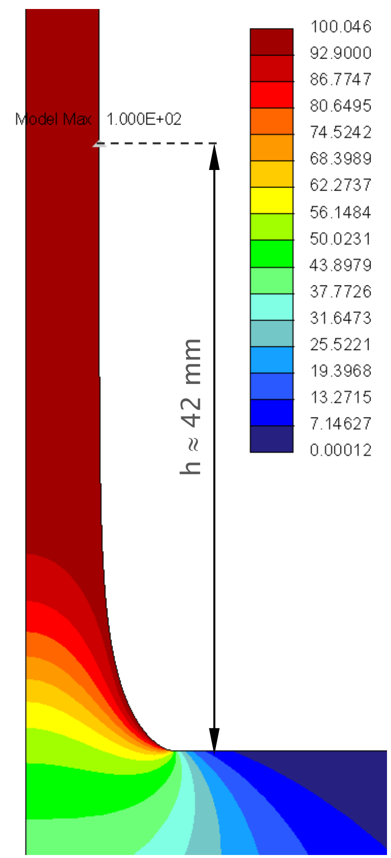
3. Examined Notches

3.4 Baud Fillet

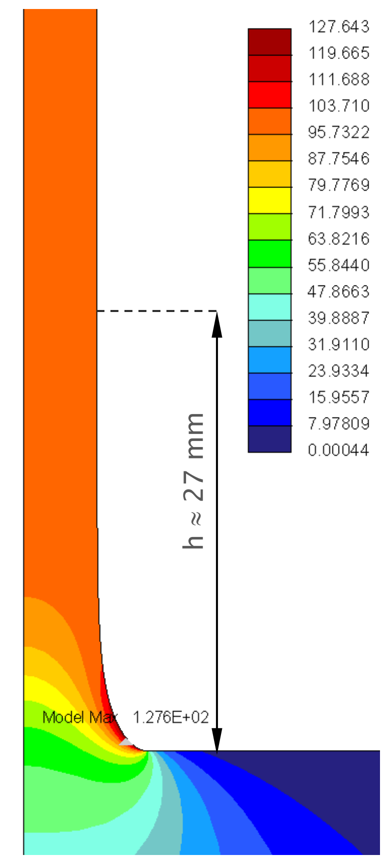
- Notch analysis results



Von Mises Stress
b = C = 4.92 mm
Best solution found



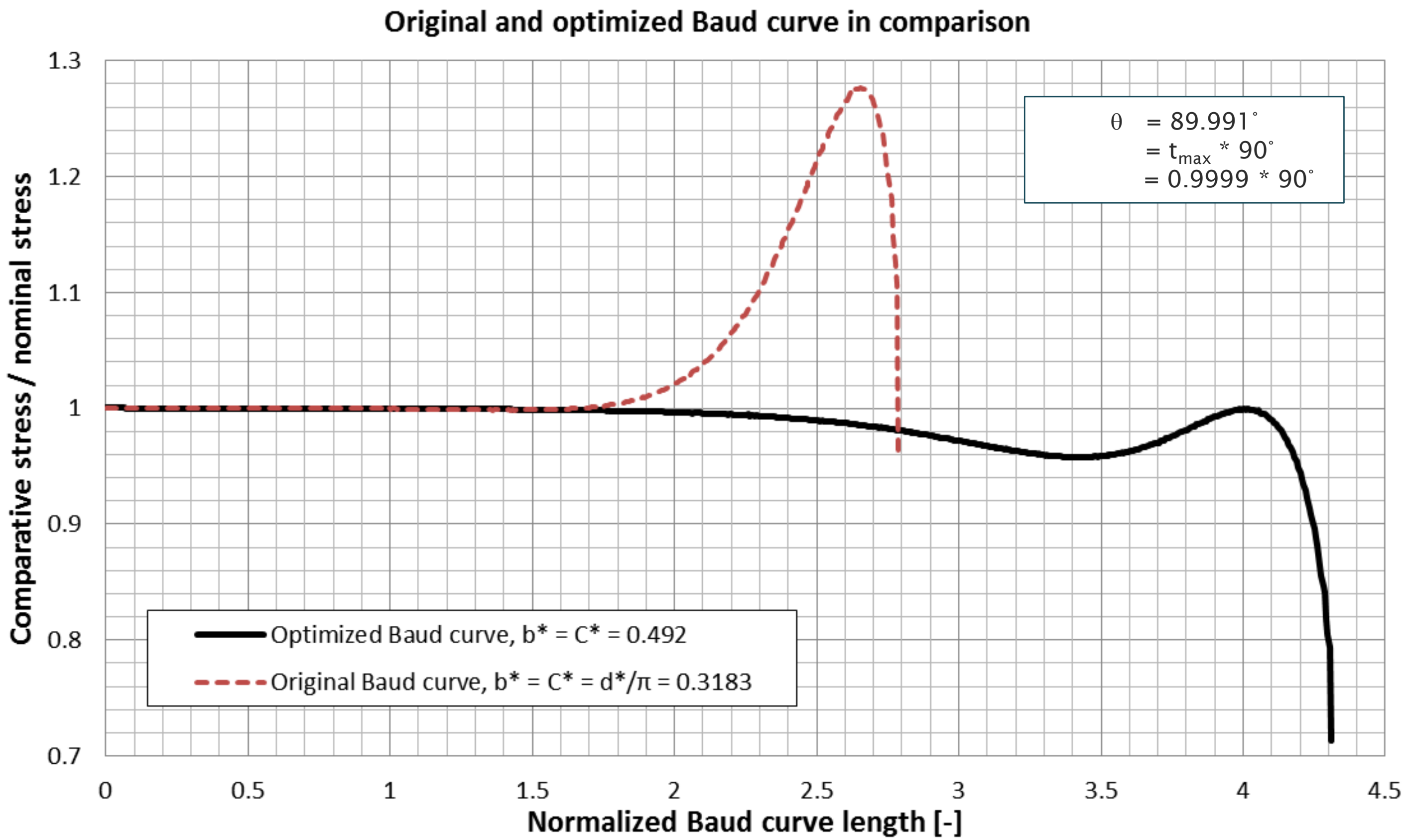
Von Mises Stress
b = C = d/π = 3.183 mm
Baud recommendation

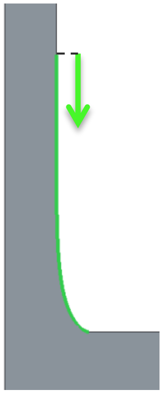


3. Examined Notches

3.4 Baud Fillet

- It can be observed that no other notch shape keeps the outer fiber stress so close to the nominal stress along the complete notch length like the optimized baud curve:

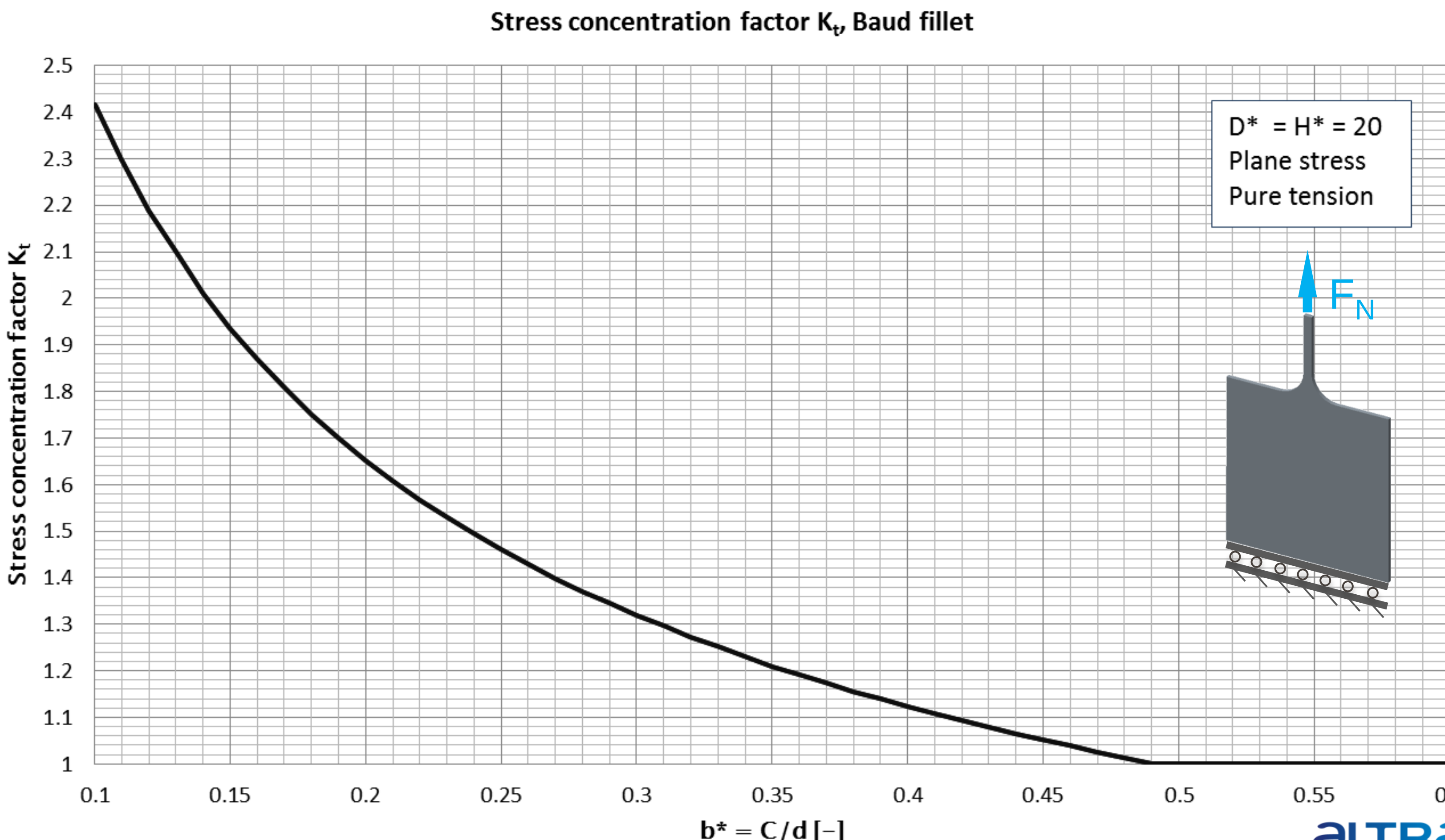




3. Examined Notches

3.4 Baud Fillet

- Stress concentration factor K_t as function of the normalized Baud curve width b*(=C*) for dimensioning

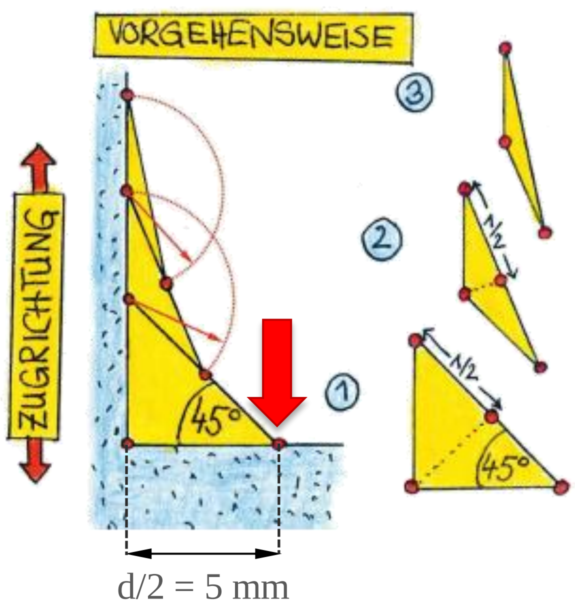


3. Examined Notches

3.5 Method of Tensile Triangles

Mattheck recommends

- Three/ Four isosceles triangles
- First triangle: 45°
- For the following triangles the angle is bisected (22.5°; 11.25°; etc.) and the starting point is the middle point of the hypotenuse from before
- To avoid high singular stress at triangle transition points:
  - Manually put a spline over it → tangential transition in tensile direction
  - Or use one-radius fillets as large as possible → tangential transition in tensile direction



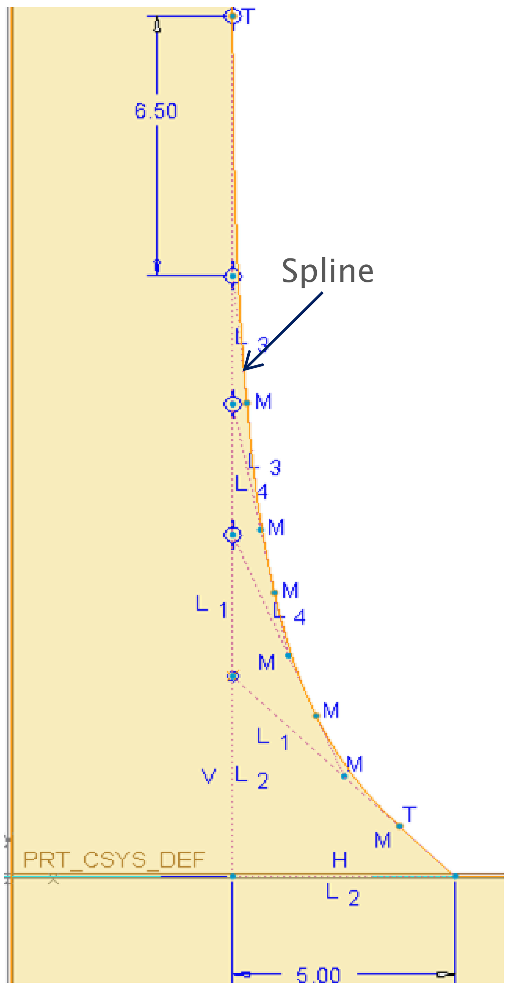
Adverse:

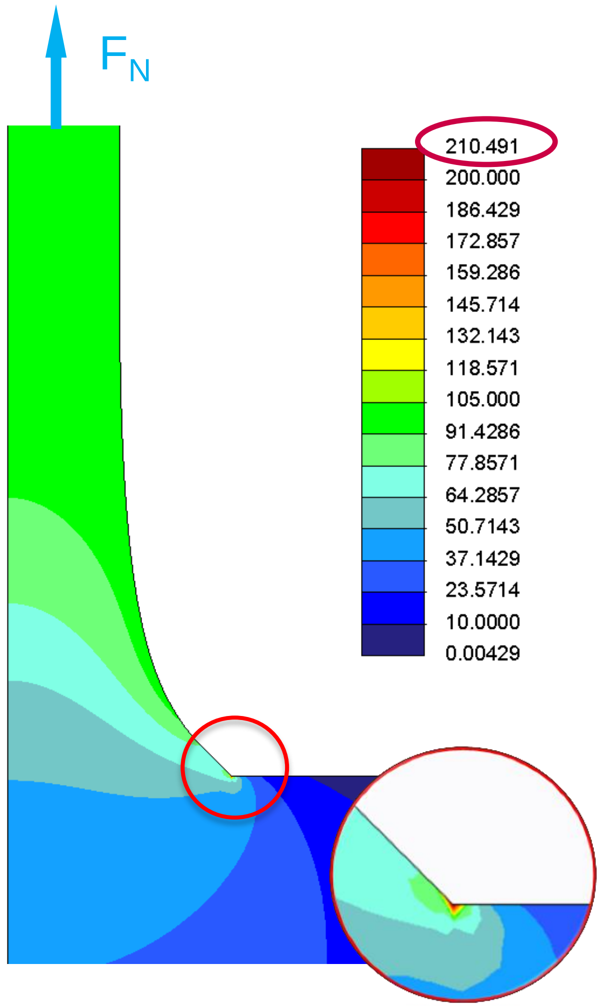
- Construction in CAD program is problematical
  - Spline is individual and unhandy
  - Several one-radius fillets result in an irregular contour
- 45° corner at the bottom: Singularity!
- According to Mattheck: "This location is not critical."

Sketch varied from [Mattheck, C., 2006]

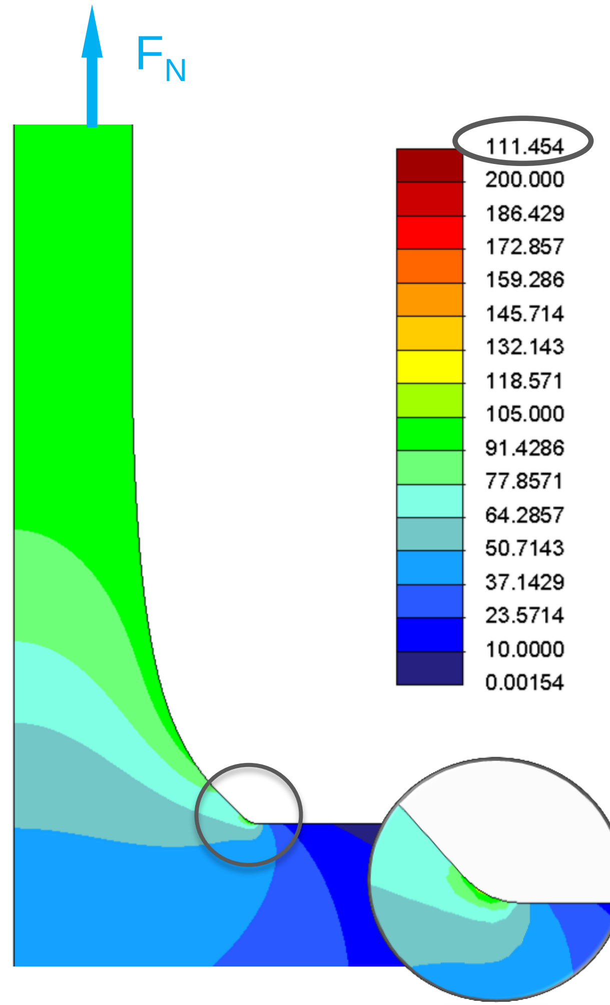
3. Examined Notches

3.5 Method of Tensile Triangles





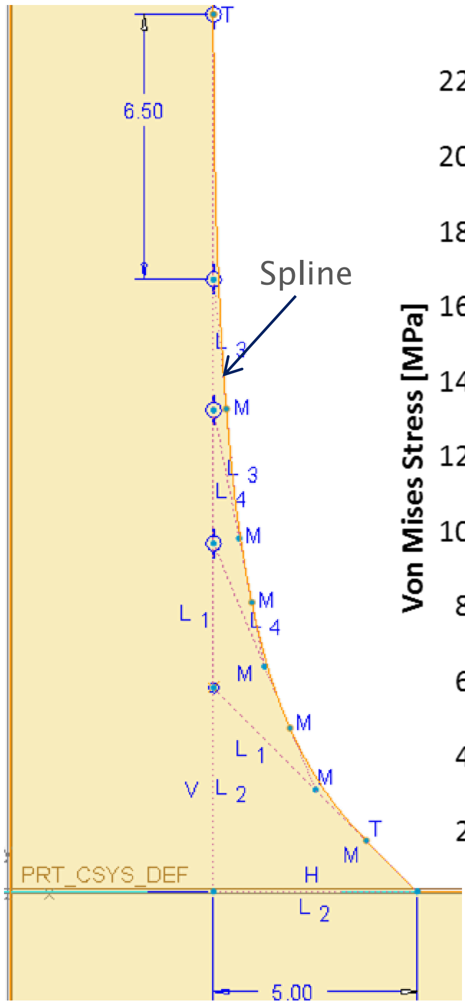
With 45° corner at bottom:
Singular peak stress!

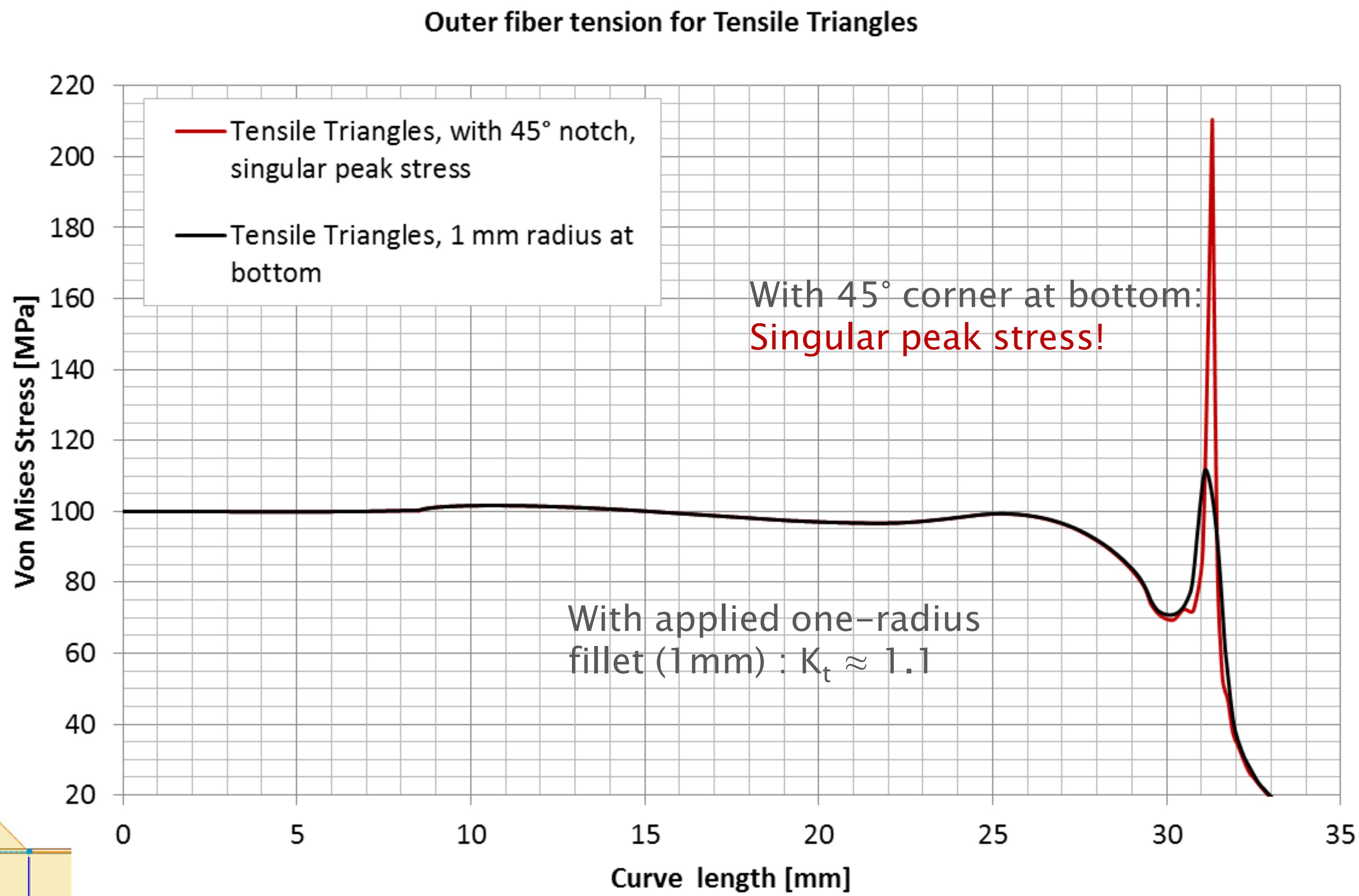


With applied one-radius fillet (1mm) : $K_t \approx 1.1$

3. Examined Notches

3.5 Method of Tensile Triangles

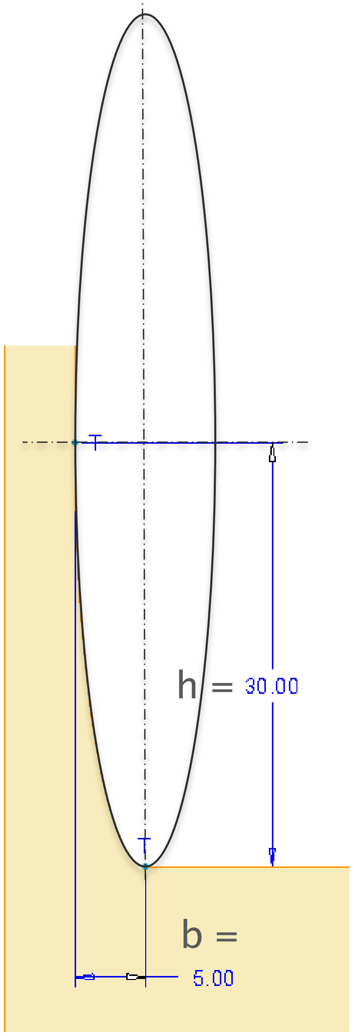




3. Examined Notches

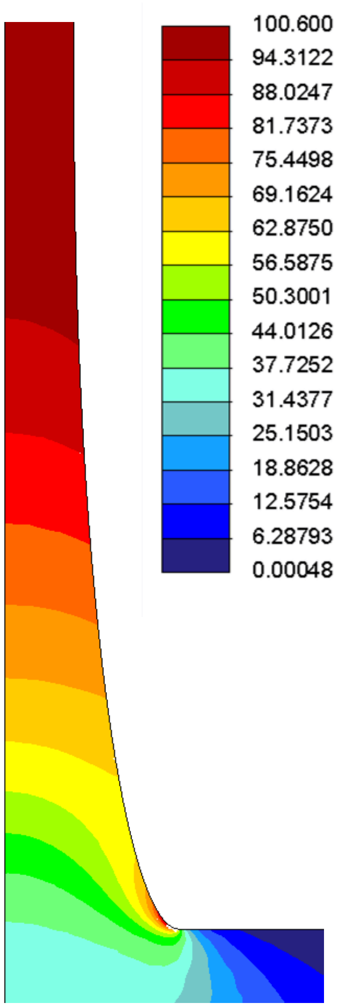
3.6 Standard Elliptical Fillet

- For the standard elliptical fillet, the two semi axes are always parallel with the web/base, respectively
- So here, the notch height h and notch width b always reflect the semi-major and semi-minor axis of the ellipse:

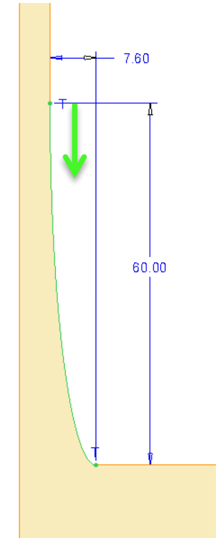


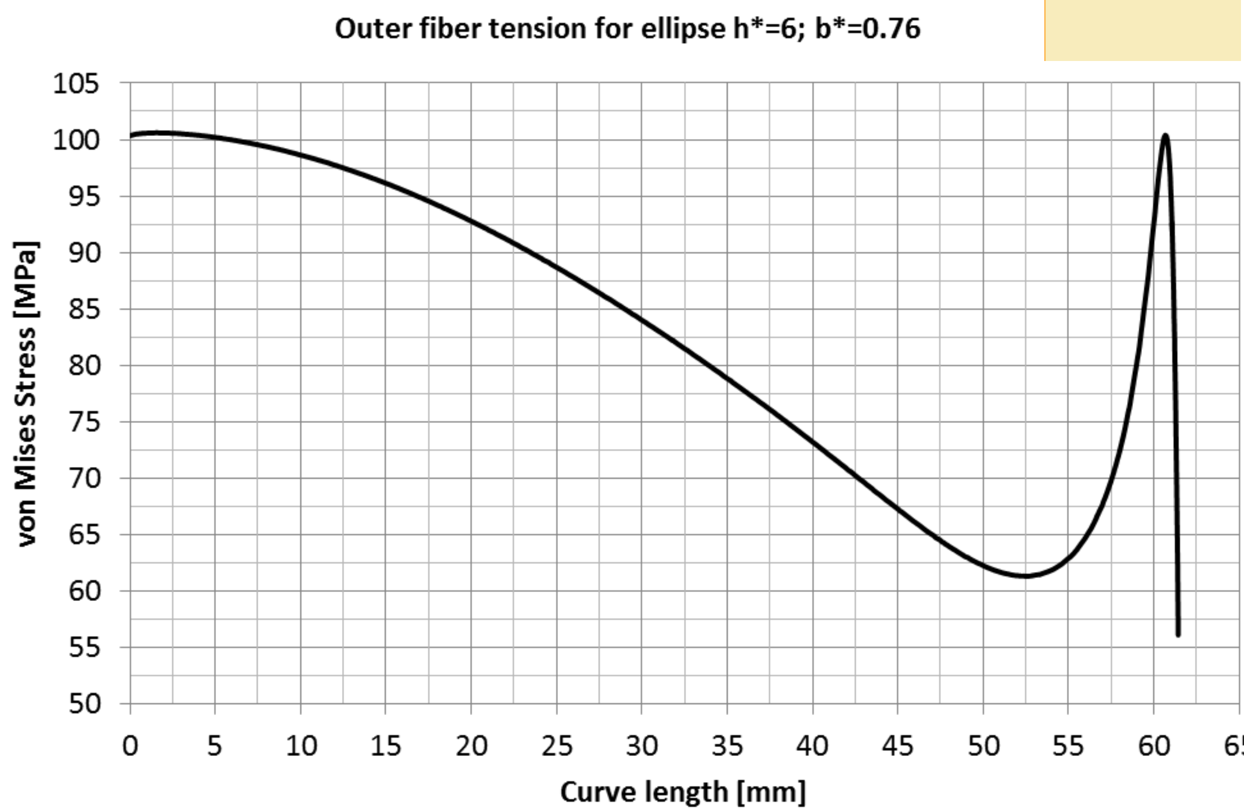
3. Examined Notches

3.6 Standard Elliptical Fillet



- Also with the elliptical fillet type low K_t can be obtained (here e.g. $K_t = 1.006$), but the utilization of material is by far not as good as with the Baud fillet:

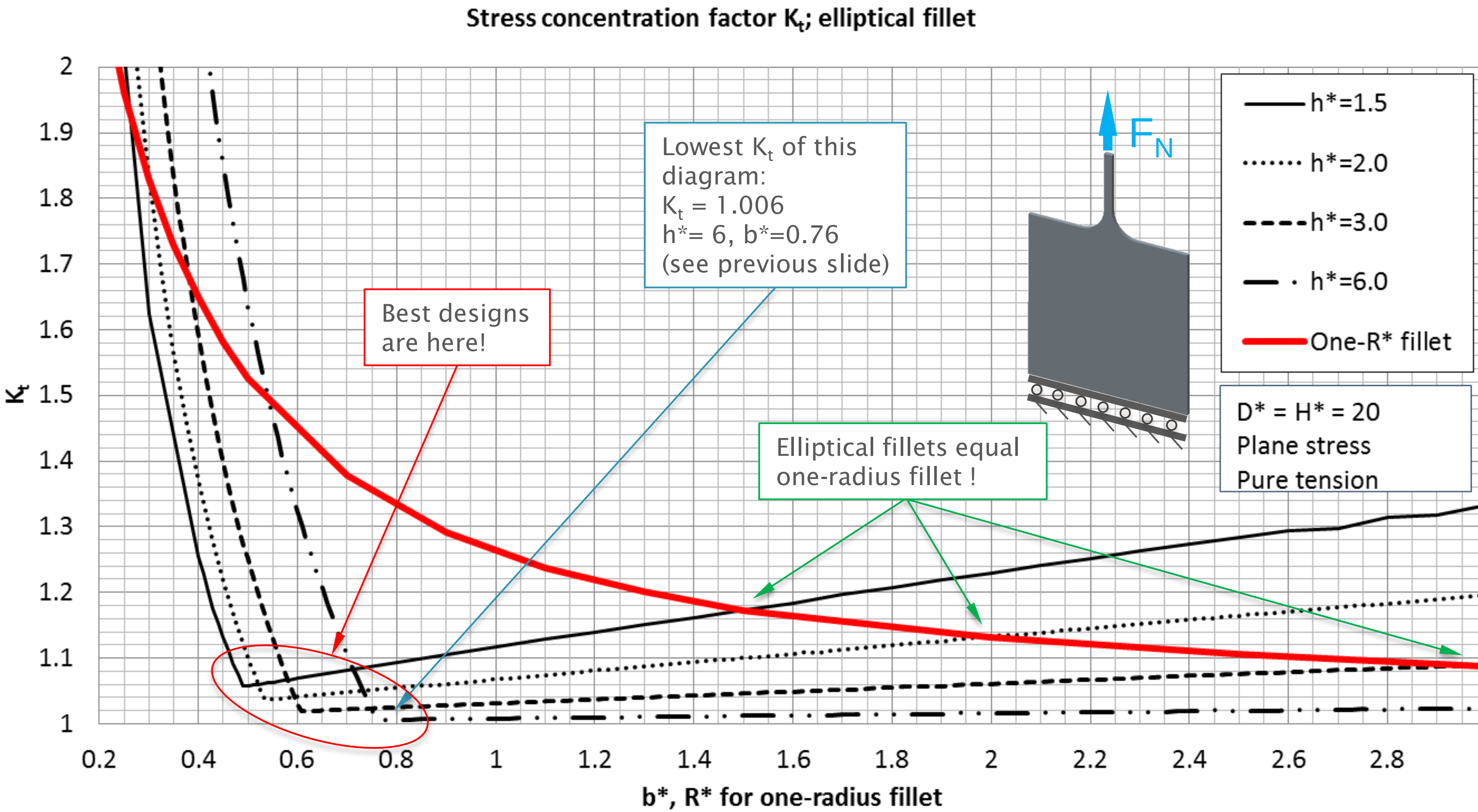




3. Examined Notches

3.6 Standard Elliptical Fillet

- The following diagram exemplifies the stress concentration factor K_t for a couple of normalized notch heights h* for dimensioning



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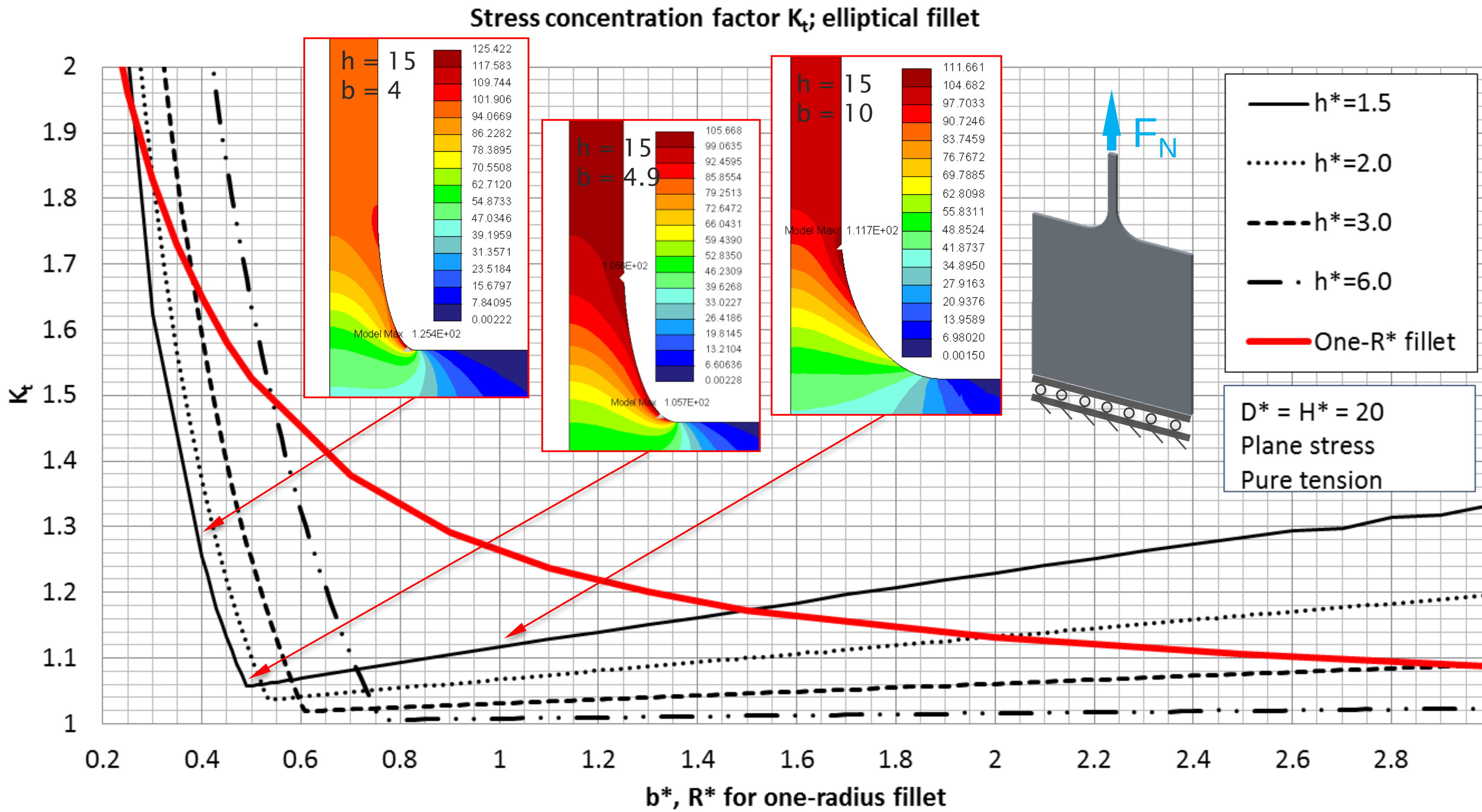
3. Examined Notches

3.6 Standard Elliptical Fillet

- Why do the curves have a sharp bend?



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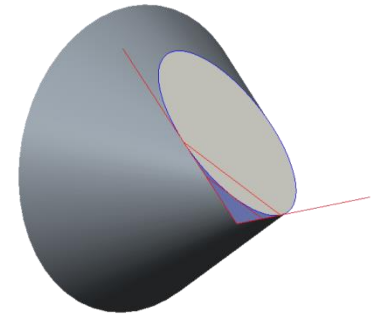
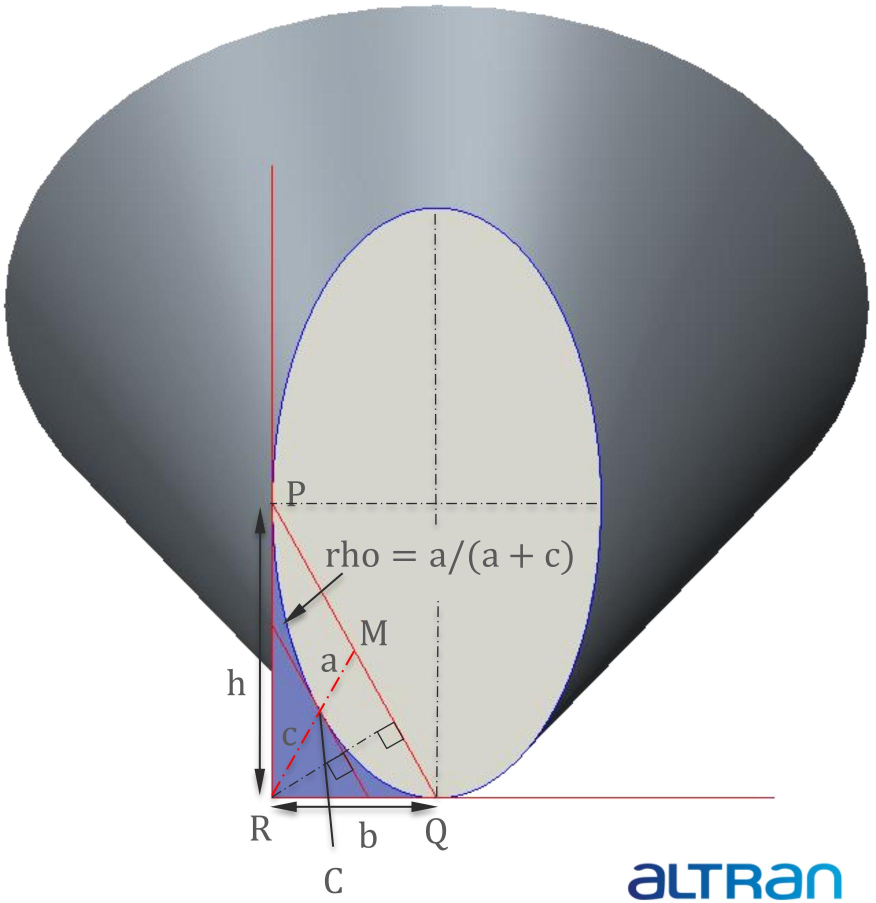
3. Examined Notches

3.7 Conical Rounds as Generalized Elliptical Fillets



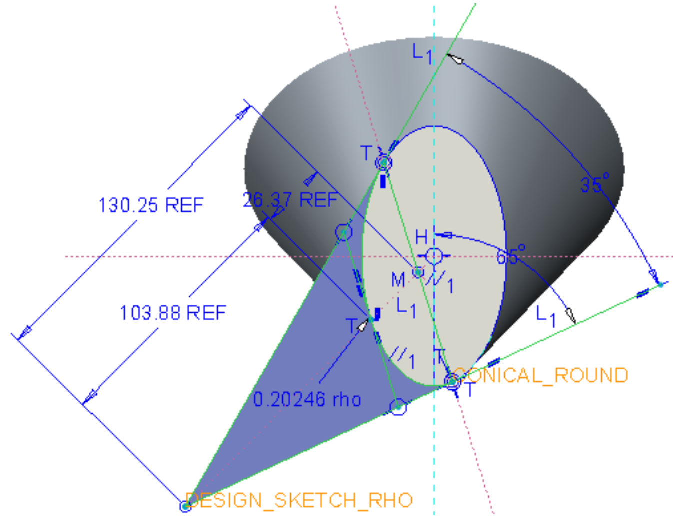
- The conical round is a standard feature in Creo Parametric
- It is defined as follows:
  - Line segments PR and QR are tangential to the ellipse at points P and Q
  - The line segment RM = a+c touches chord PQ at its midpoint M
  - Point C is the crossing point of the parallel line tangential to the ellipse with RM
  - Rho=a/(a+c) determines for the conical bow segment PQ the shape of the conic section
  - In our special case depicted right, if RP and RQ are perpendicular to each other AND exactly reflect the major-semi axis h and the minor-semi axis b of the ellipse, we have rho $= \sqrt{2} - 1 =$ 0.41421…
  - For other values of rho see next slide



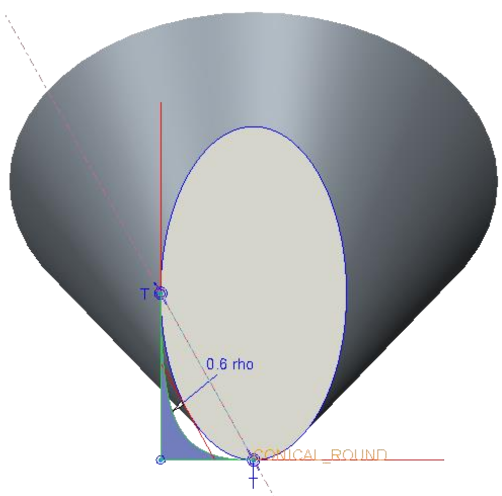
3. Examined Notches

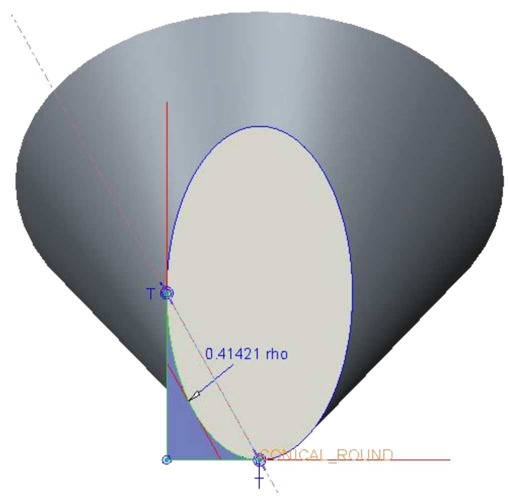
3.7 Conical Rounds as Generalized Elliptical Fillets

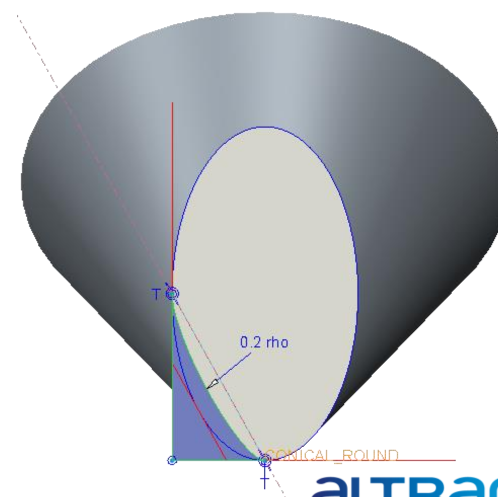
- For our special notch case with PR = h and QR= b we conclude below:
  - rho > 0.41421: “slim” fillet
  - rho < 0.41421: “fat” fillet
  - The parameter rho therefore seems to be an ideal additional control to further optimize our standard elliptical notch (having rho = $\sqrt{2} - 1$ fixed)
  - For better understanding, a more general case example is shown right



Remark:
In Creo Parametric, rho values between 0.05 and 0.95 are supported







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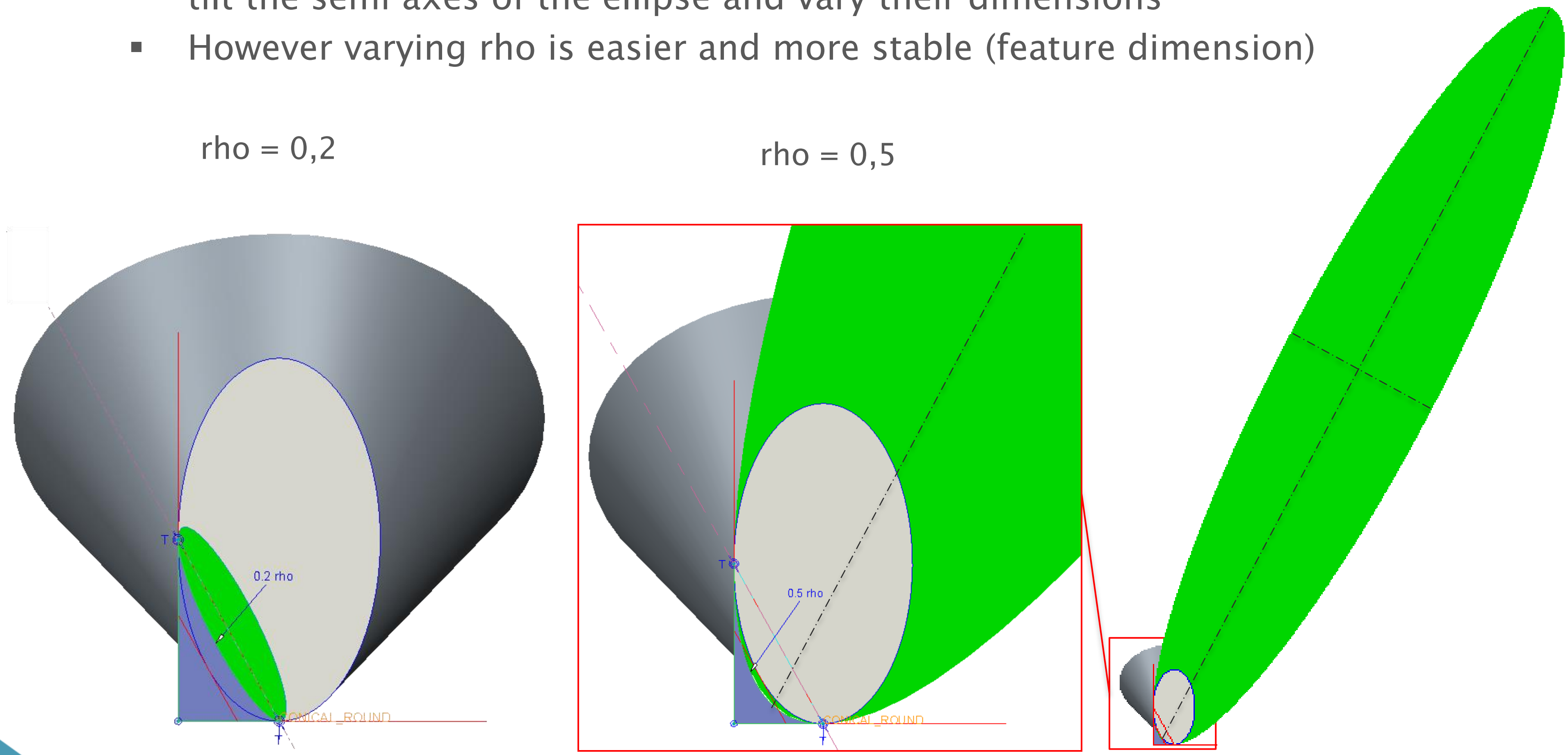
3. Examined Notches

3.7 Conical Rounds as Generalized Elliptical Fillets

- Alternatively, instead of changing rho of the conical round we could tilt the semi axes of the ellipse and vary their dimensions
- However varying rho is easier and more stable (feature dimension)

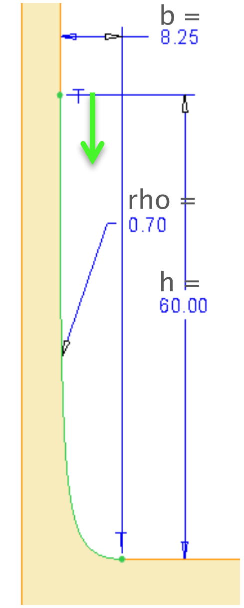
rho = 0,2

rho = 0,5

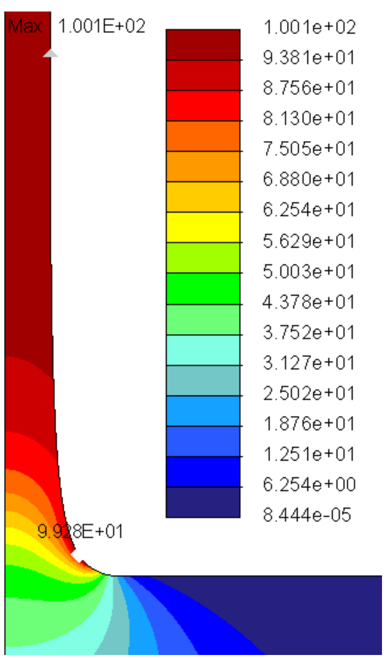


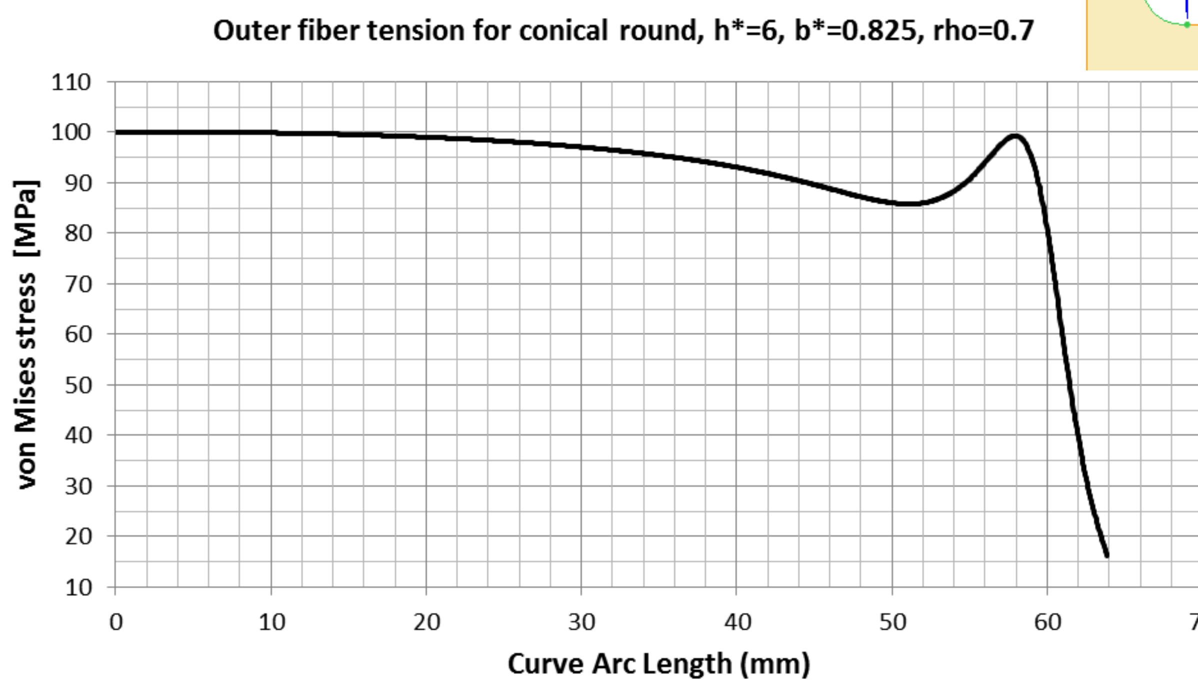
3. Examined Notches

3.7 Conical Rounds as Generalized Elliptical Fillets



- The conical round allows very low K_t (here e.g. $K_t = 1.0006$) for bigger h* and utilizes the material much better than the standard elliptical fillet

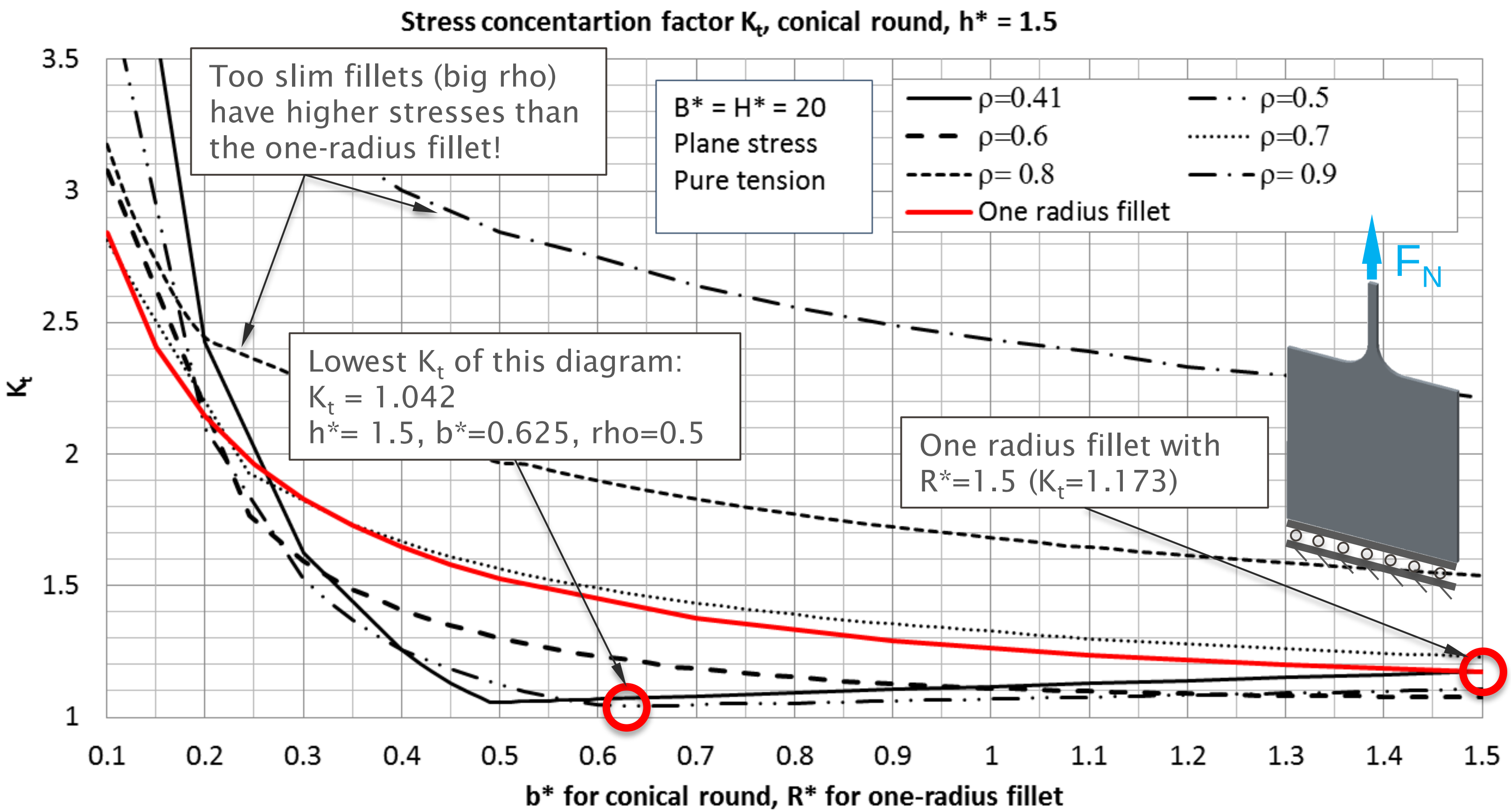




3. Examined Notches

3.7 Conical Rounds as Generalized Elliptical Fillets

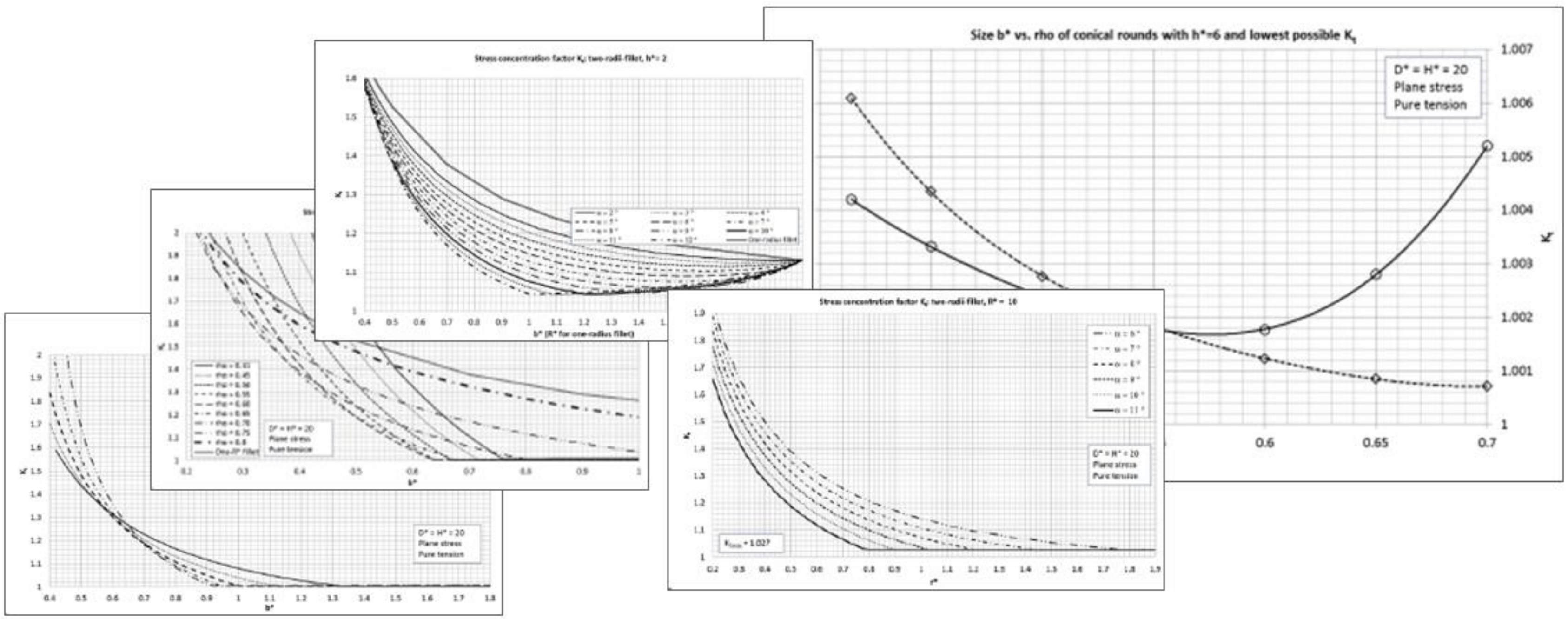
- The following diagram exemplifies the stress concentration factor K_t for a very small normalized notch height of h*=1.5 for dimensioning, for comparison with the one-radius fillet



3. Examined Notches

3.8 Notch layout library

- Further considerations show that it is possible to draw condensed diagrams summarizing the ideal notch dimensions for each notch type to reach a minimum possible K_t (approaching 1)
- The resulting notch layout library allows to quickly design the best notch for the actual K_t value needed without additional, time-consuming and expensive FEM-analysis, just by reading out the necessary dimensions b*, h*, rho, R*, r* or α from the suitable diagram



4. Result Comparison

Notch Shapes and their Efficiency

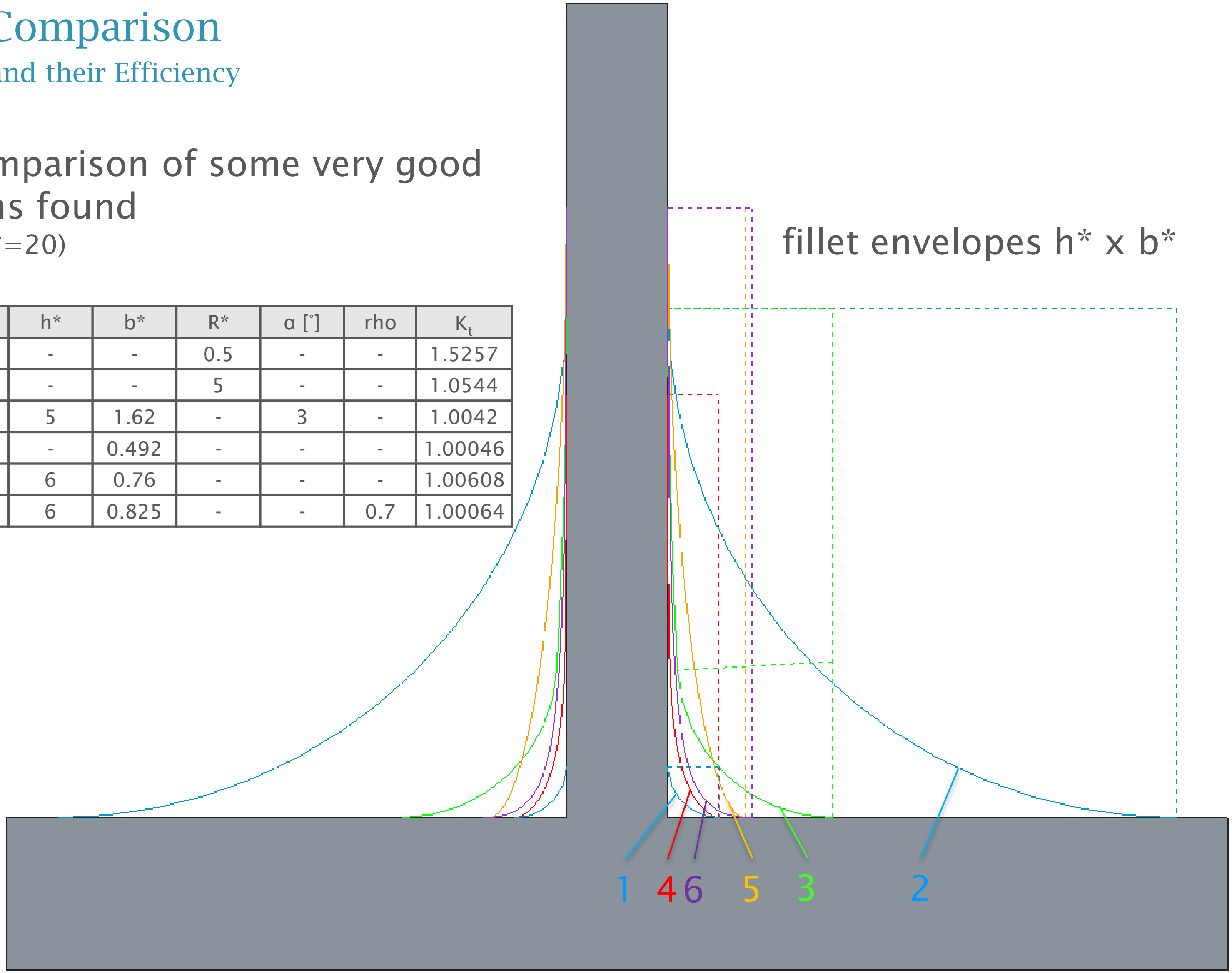
- The standard circular fillet is very inefficient: Very huge R* are necessary to obtain K_t approaching one
- The Baud curve is clearly the best shape from the efficiency point of view, closely followed by conical rounds with rho>$\sqrt{2}-1$ and then standard elliptical fillets
- Surprisingly, with the exception of the Baud curve probably, there seems to be no „ideal“ solution for the notch shape (e.g. a certain “ideal” semi axes relation for the ellipse or a preferred rho for the conical round): The best solution depends on the priority (small h* and/or b* and/or K_t)
- For the Baud curve itself, the theoretical difficulty is that the curve has to be “cut” at its end (e.g. after θ=89,991° like in this presentation) and rounded to prevent a singular location (=edge<180°). This also effects its length along the web and slightly the local stress there
- In general, if $K_t \rightarrow 1$, then all the notches become very size sensitive: A further small decrease in K_t often means a significant increase in fillet size; furthermore, the FEM results appear to be more sensitive to small numerical disturbances

4. Result Comparison

Notch Shapes and their Efficiency

- Size comparison of some very good solutions found
  (for H*=B*=20)

Notch type	No.	h*	b*	R*	α [°]	rho	K_t
One radius fillet	1	-	-	0.5	-	-	1.5257
One radius fillet	2	-	-	5	-	-	1.0544
Two radii fillet	3	5	1.62	-	3	-	1.0042
Baud fillet	4	-	0.492	-	-	-	1.00046
Standard elliptical fillet	5	6	0.76	-	-	-	1.00608
Conical round	6	6	0.825	-	-	0.7	1.00064



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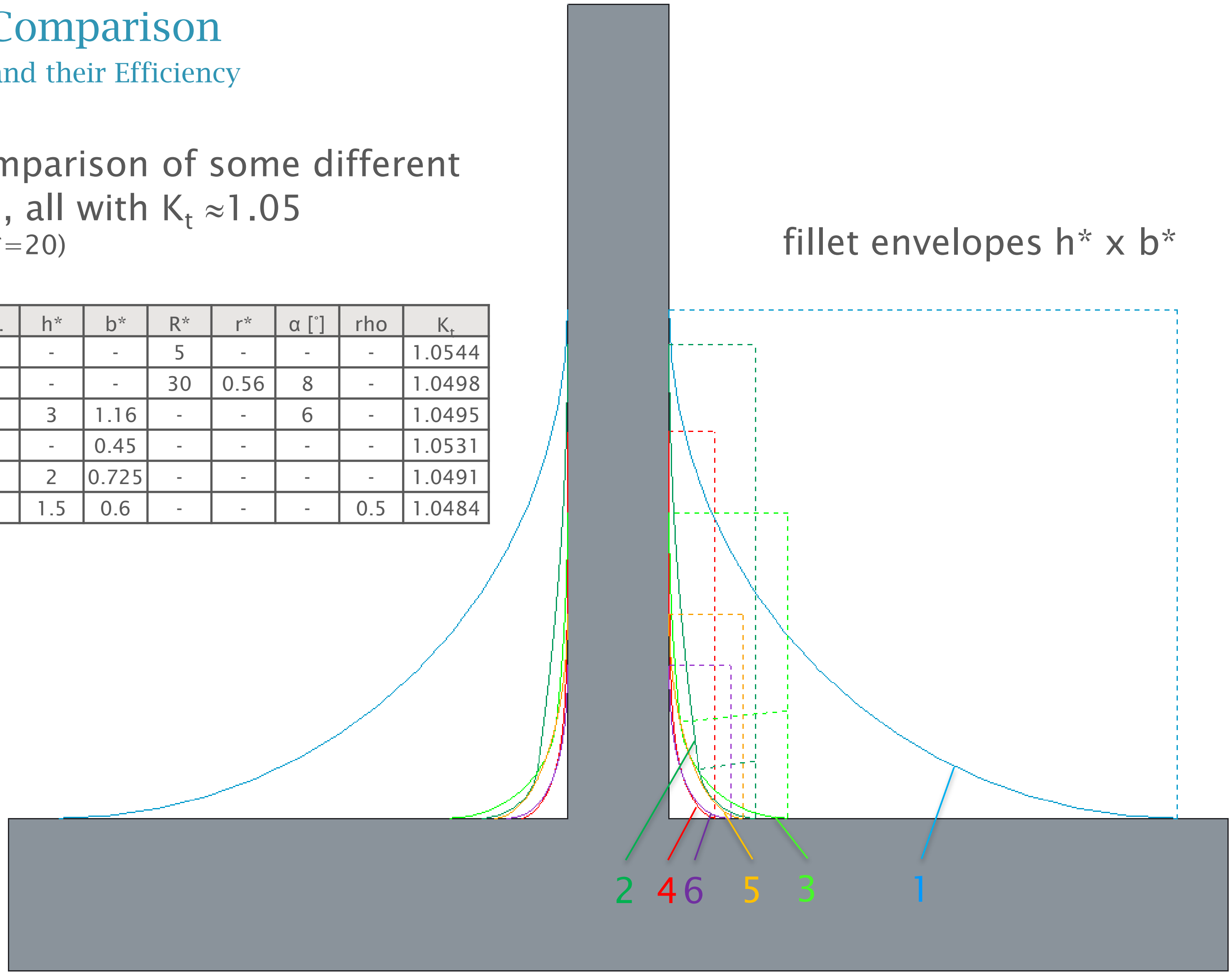
4. Result Comparison

Notch Shapes and their Efficiency

- Size comparison of some different notches, all with $K_t \approx 1.05$
  (for H*=B*=20)

Notch type	No.	h*	b*	R*	r*	α [°]	rho	K_t
One radius fillet	1	-	-	5	-	-	-	1.0544
Two radii fillet	2	-	-	30	0.56	8	-	1.0498
Two radii fillet	3	3	1.16	-	-	6	-	1.0495
Baud fillet	4	-	0.45	-	-	-	-	1.0531
Standard elliptical fillet	5	2	0.725	-	-	-	-	1.0491
Conical round	6	1.5	0.6	-	-	-	0.5	1.0484



INNOVATION MAKERS

Thanks for your attention!

For any questions or services,
please contact the authors under
roland.jakel@altran.com

Part C: Appendix

List of Sources

[Decker, Karl-Heinz, 1985]: Maschinenelemente, Gestaltung und Berechnung, 9. Auflage, ISBN 3-446-14362-9, Hanser Verlag, 1985

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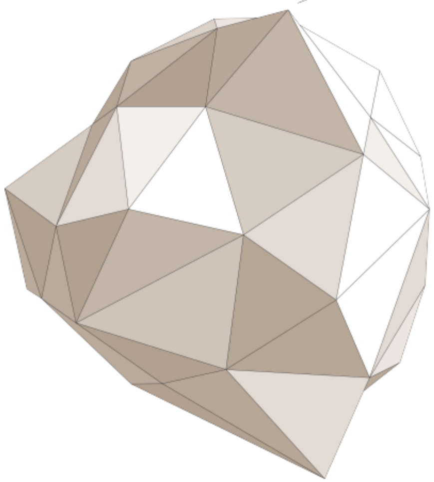

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