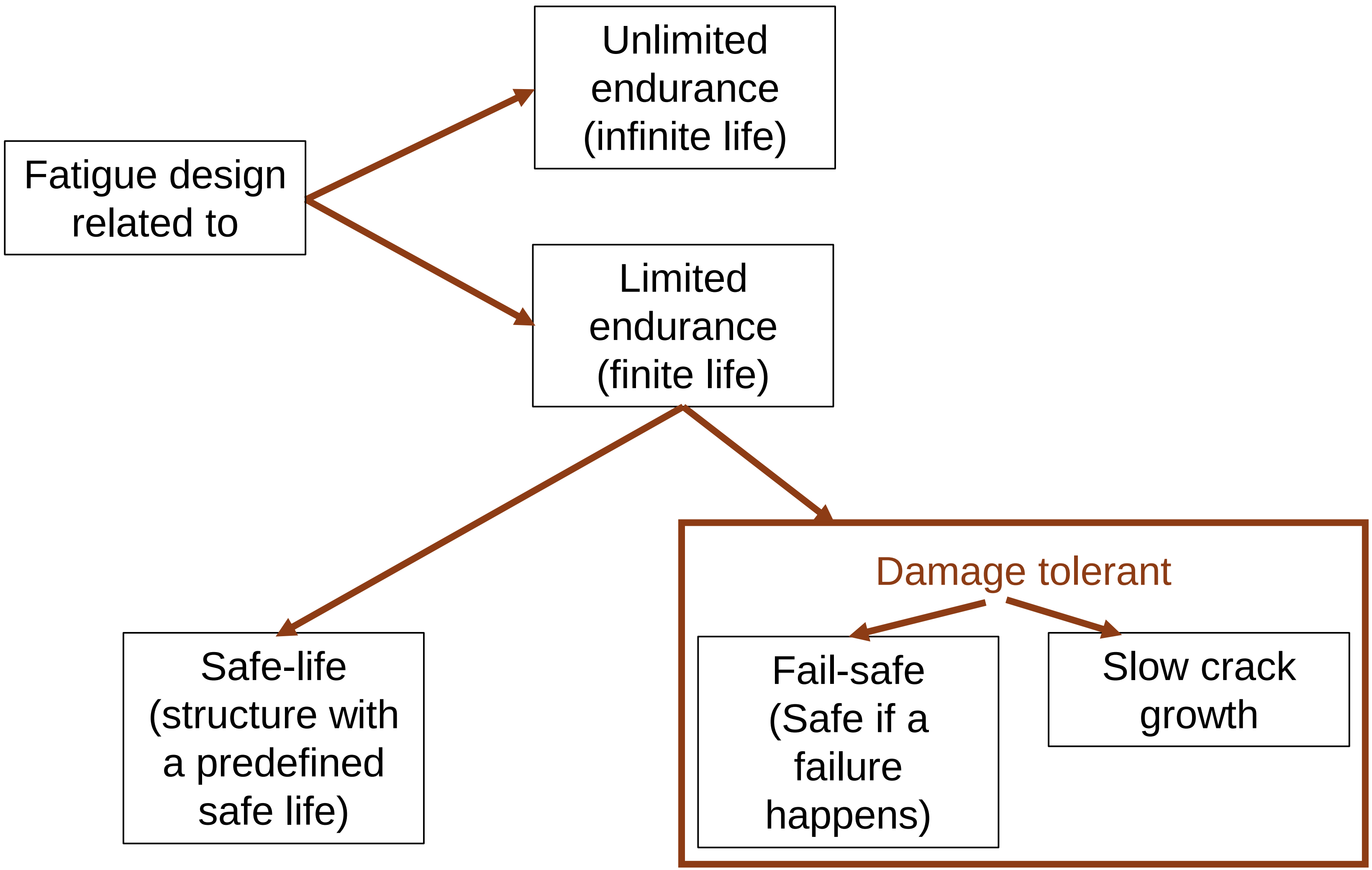
Dynamická únosnost a životnost

Lekce 2
-
Stress-Based Fatigue Analysis, Part #1

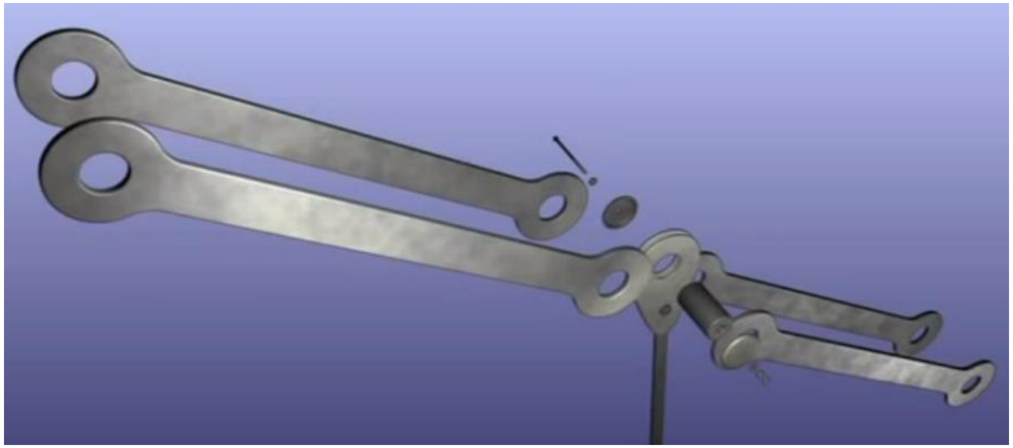
Jan Papuga

Different Attitudes to Fatigue Design



Silver Bridge Collapse - 1967

- Bridge over Ohio river collapsed in less than 1 minute. 46 casualties
- Was not fail-safe
- Corrossion crack in one member lead to complete disintegration
- Right practice here

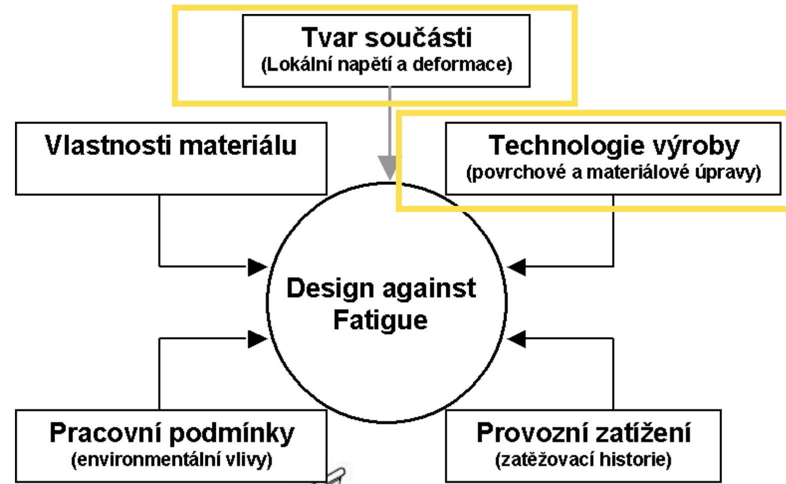


https://www.youtube.com/watch?v=dGQfUWvP0II

General Dynamics F-111 (1969)

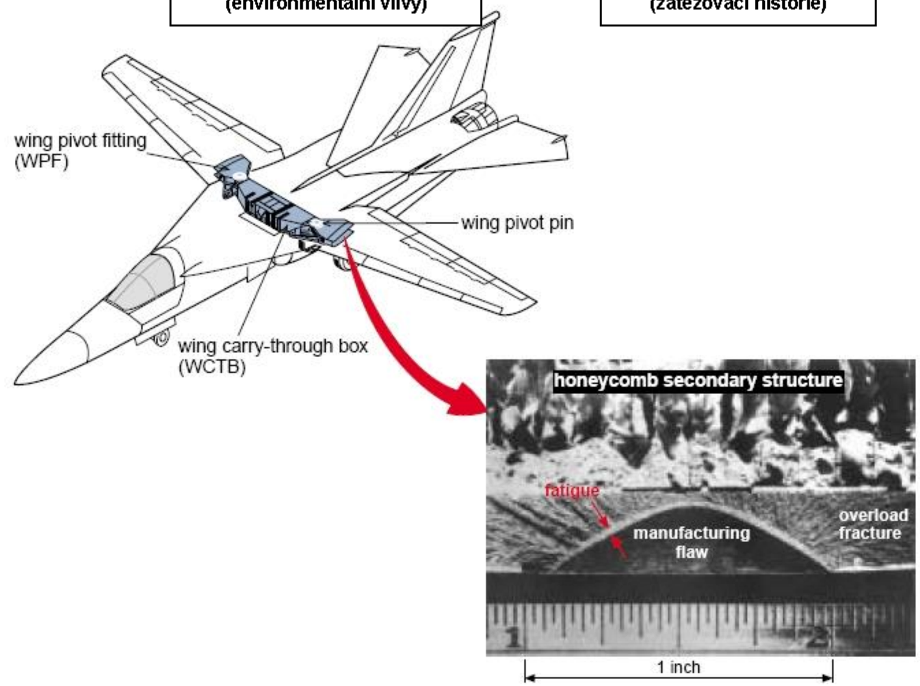
Bomber / fighter with variable wing configuration

F-111#94 lost a wing during a training flight (just one year a service)



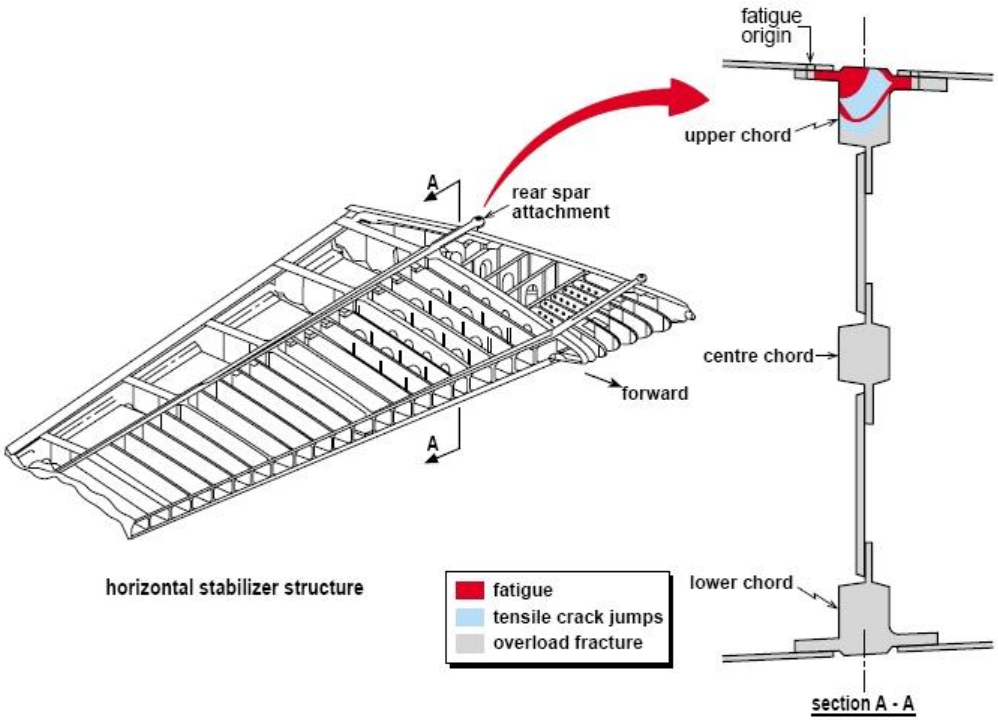
Reason:

- **Large crack in the hinge already from the manufacturing process (23.4x5.9mm!) – only short growth (low fracture toughness of the high-strength steel)**



DAN-Air Boeing 707-300 (1977)

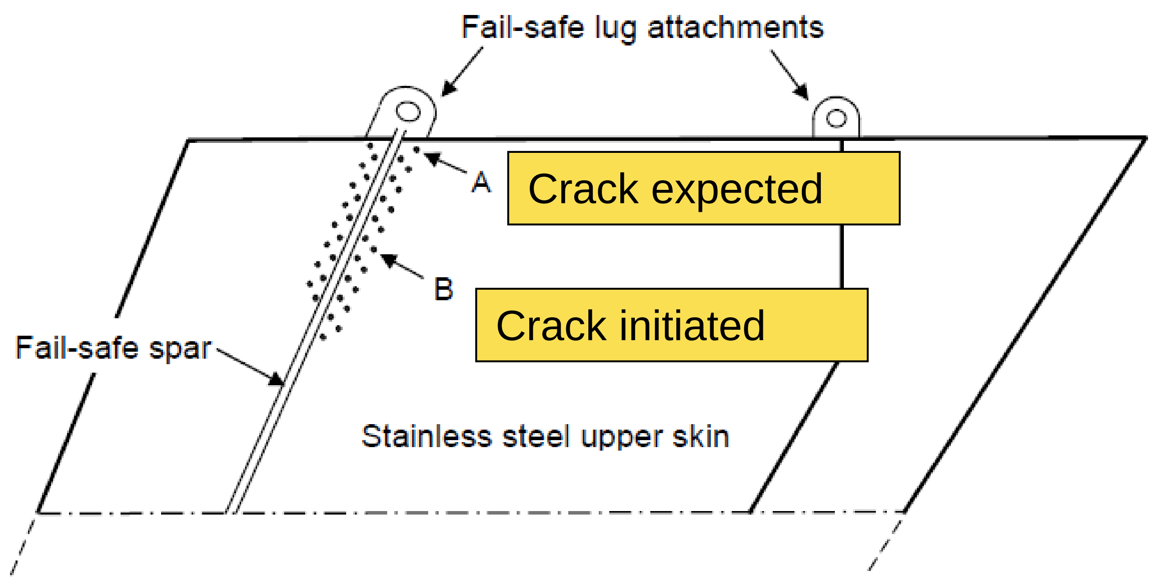
- **Lost whole right stabilizer with elevator**
- **47621 flight hours. 14 year old (60000 FH designed. 20 years)**
- **Reason: Stabilizer (its upper flange) designed as fail-safe but:**
  - Skin material changed from 7075-T6 to steel. higher stiffness – was not supported by a full scale test
  - Only visual inspection prescribed
  - Once the flange broke. there was not enough time till the next inspection
  - Stabilizer was not fail-safe due to the chosen way of inspection



DAN-Air Boeing 707 (1977)

Consequences:

- **Fail-Safe is not guaranteed by design only. but also by the selection of the inspection method**
- **Material exchange can lead to stress redistribution and should be supported by**
  - **FEA**
  - **Experiment**

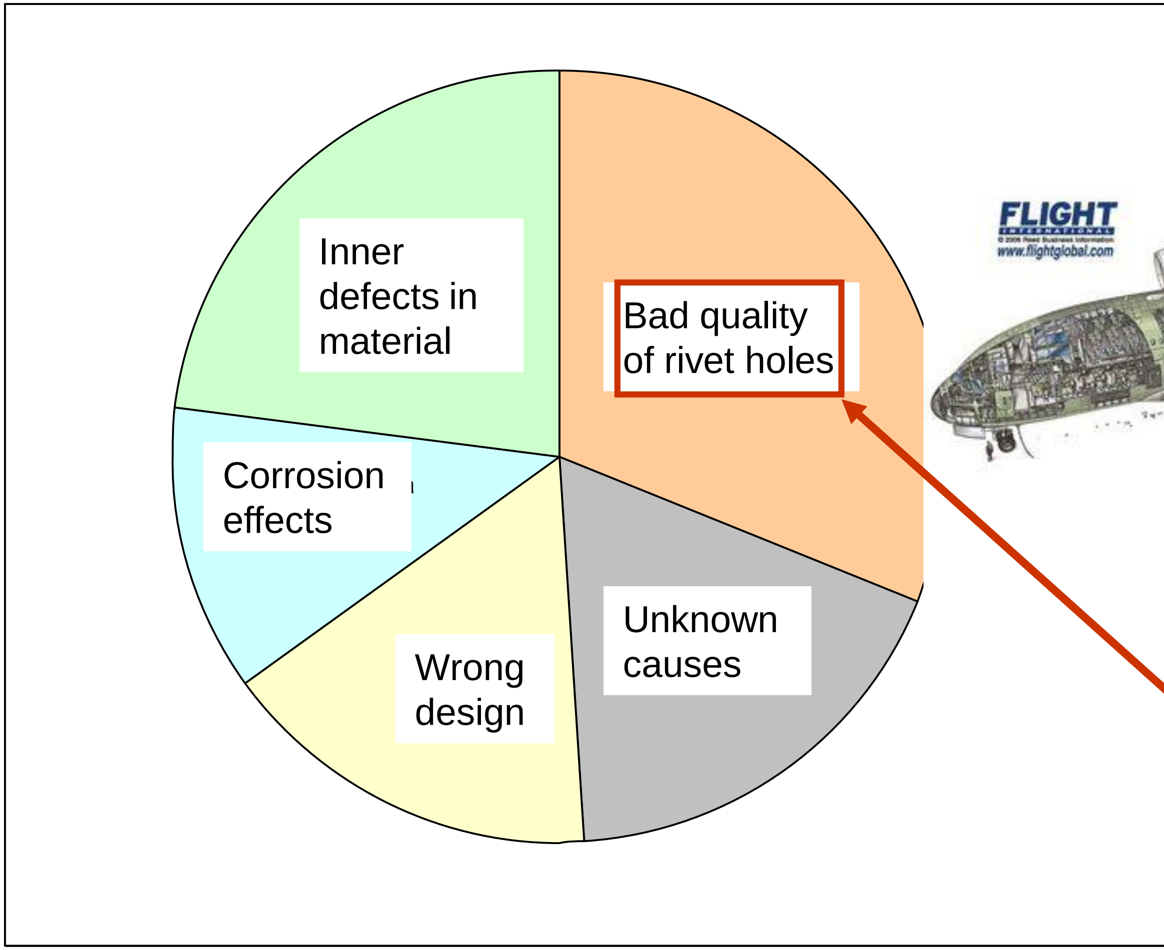


Schijve. J.: Fatigue Damage in Aircraft Structures. Not Wanted. But Tolerated?

Boeing 737 (1988)

- **Aircraft lost 5.5 m of the pressurized cabin**
- **35496 FH. 89680 landings. 19 years in service. short flights. humid sea air**
- **Reason:**
  - **Quick joining of multiple small cracks in a row – Multiple Site Damage (MSD)**

Causes of fatigue issues



See NASA experiments on retired Boeing

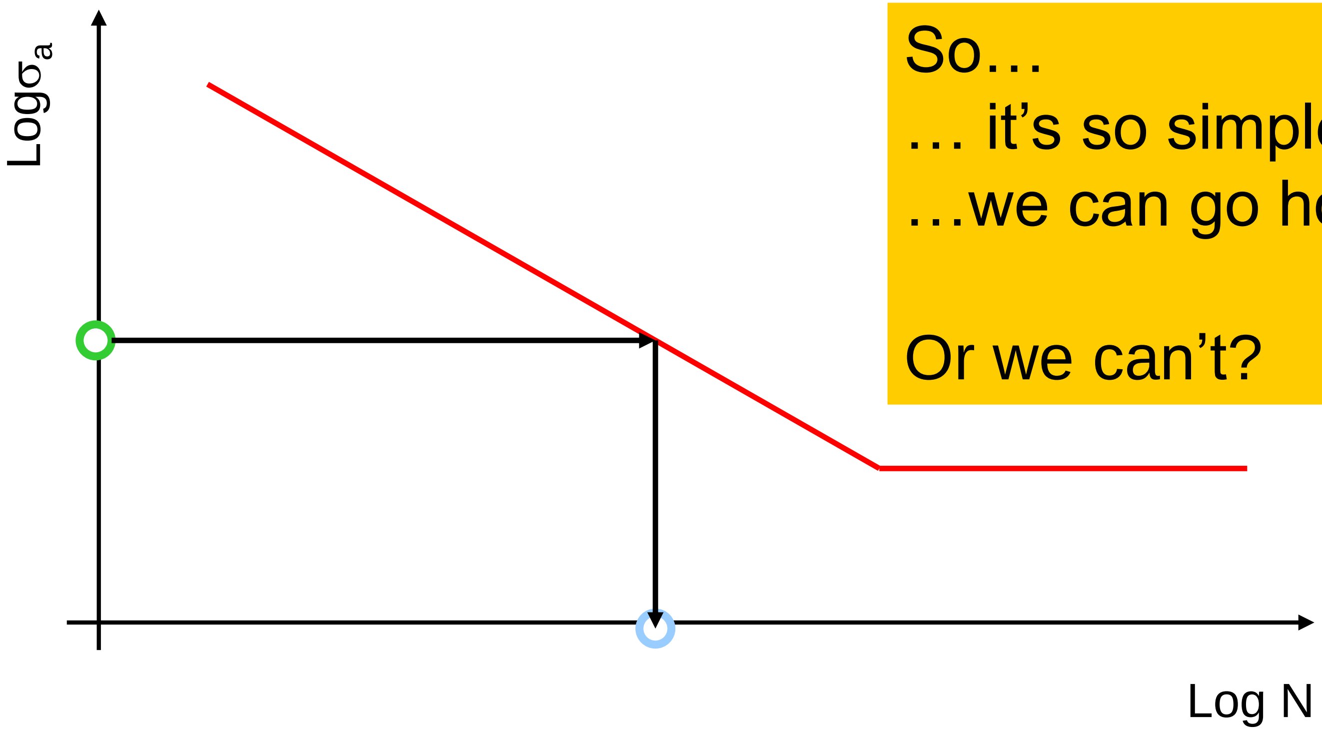
Sampath. S.G. and Simpson. D.: Airframe Inspection Reliability Under Field/Depot Conditions. Terms of Reference of AGARD Structures and Materials Panel Proposed Activity SC.77. October 1995.

Intuitively

For inputs:

- Material with a known S-N curve
- Load amplitude -> FEA -> stress amplitude σ_a

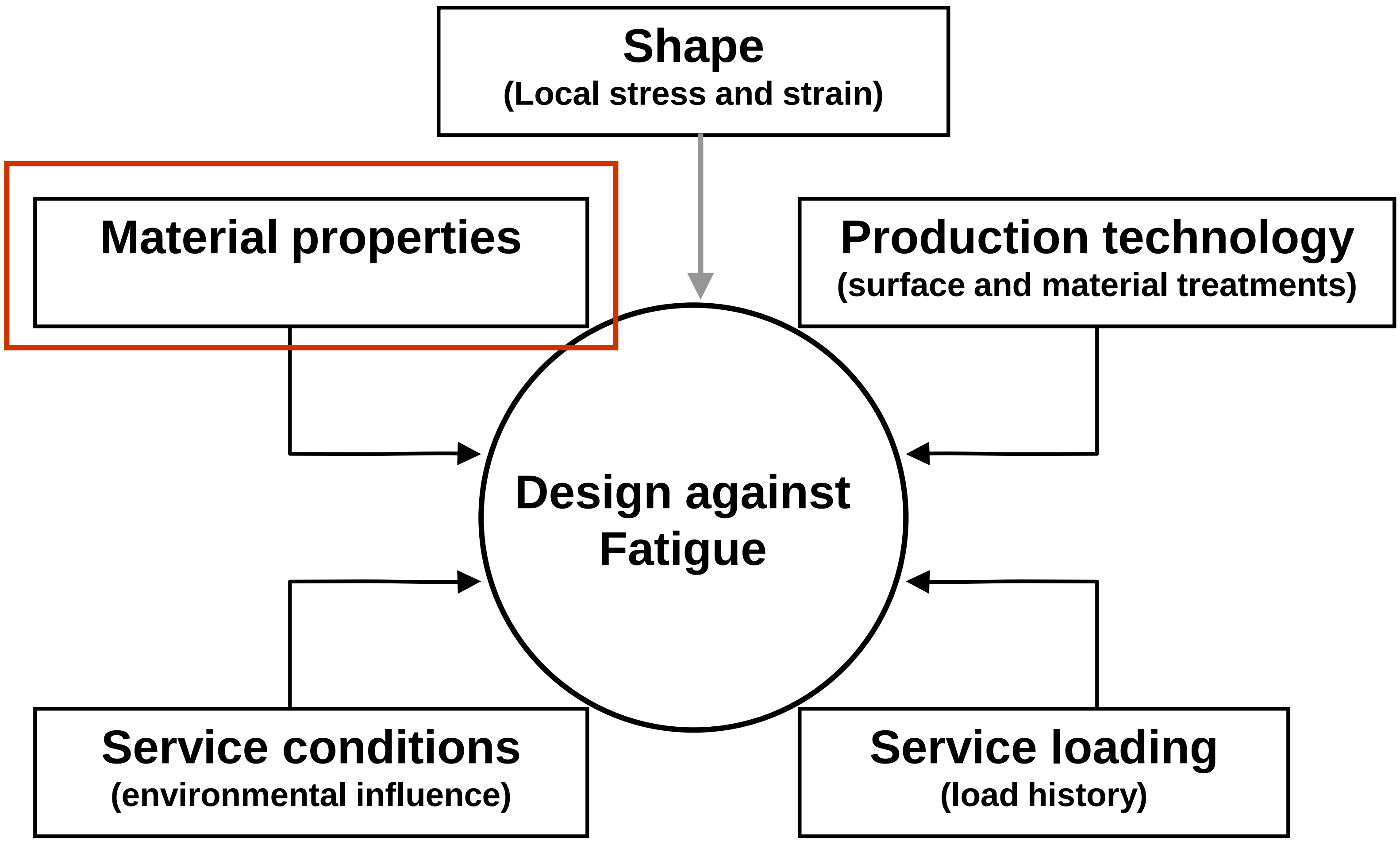
We can get immediately the final lifetime



So…
… it's so simple…
…we can go home

Or we can't?

Aspects influencing the fatigue life



Questions – Part I

1. How fatigue design and static design of structures differ?
2. Typical attributes of low cycle fatigue and of high cycle fatigue?
3. Draw a hysteresis loop and describe on it elastic and plastic part of strain.
4. Specify phases of damage and fatigue progress in metals.
5. What is the main difference between safe-life and fail-safe design philosophy?
6. Which are main attributes of the damage tolerant design philosophy?
7. Define the fatigue limit of a material.
8. Which type of fatigue curve describes high-cycle fatigue primarily? Draw this curve.
9. Which type of fatigue curve describes low-cycle fatigue? Draw this curve.
10. Could be the fatigue limit higher than yield strength?
11. How can you estimate the fatigue limit of carbon steel from tensile strength?

Questions – Part II

1. Example 1: Approximation of a stress amplitude is $\sigma_a = K' \cdot \varepsilon_{ap}^{n'}$. Derive equation for the total strain amplitude of a hysteresis loop $\varepsilon_a = \varepsilon_{ae} + \varepsilon_{ap} = ?$
2. Example 2: Approximation of the fatigue curve is $\sigma_a^w \cdot N = C$ or $\sigma_a = \sigma'_f \left(2N\right)^b$. Derive relations between parameters $C, \quad \left(\sigma'_f\right), \quad b, \quad w.$
3. Example 3: There are 6 material fatigue parameters $K', \quad n', \quad \sigma'_f, \quad b, \quad \varepsilon'_f, \quad c$, only are 4 independent. Derive relations between these parameters.
4. Example 4: There is special number of cycles (N_t) in the Strain-life curve, where $\varepsilon_{ae} = \varepsilon_{ap}$. Derive equation to calculate this number N_t .

Intuitively

For inputs:

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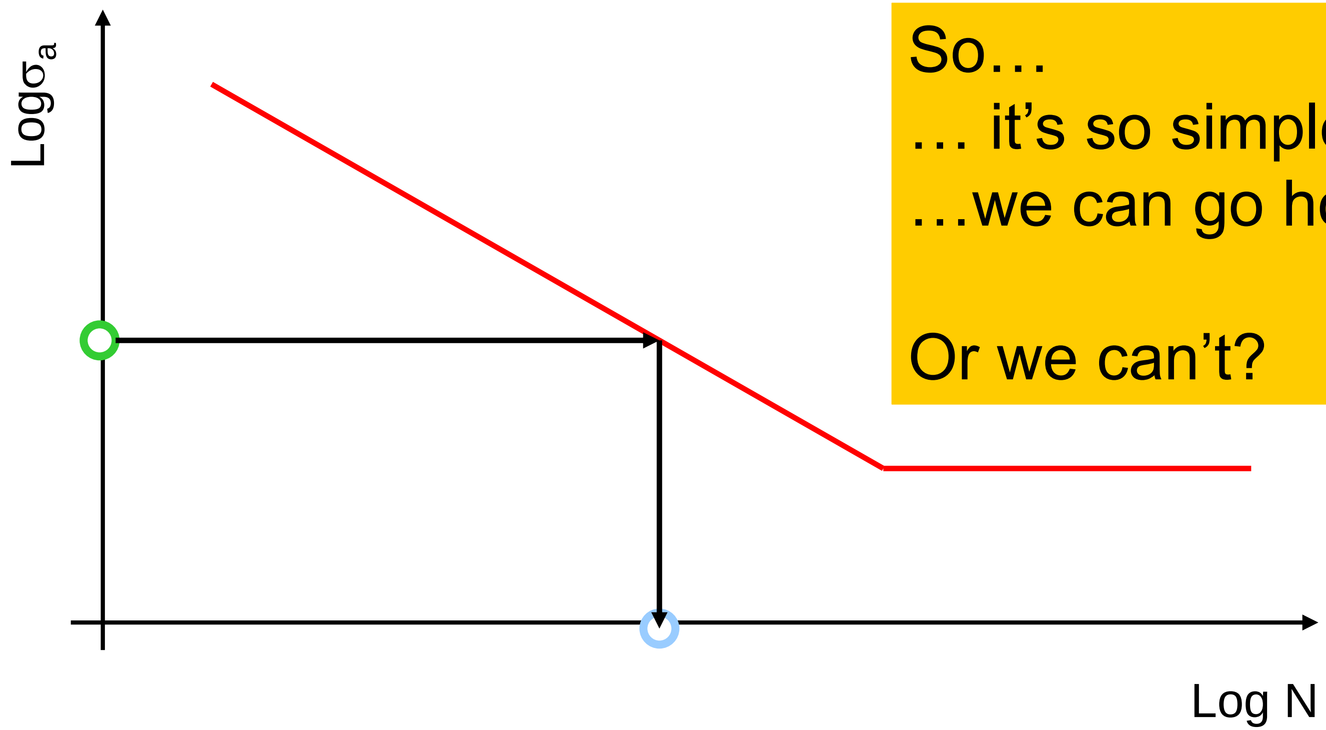

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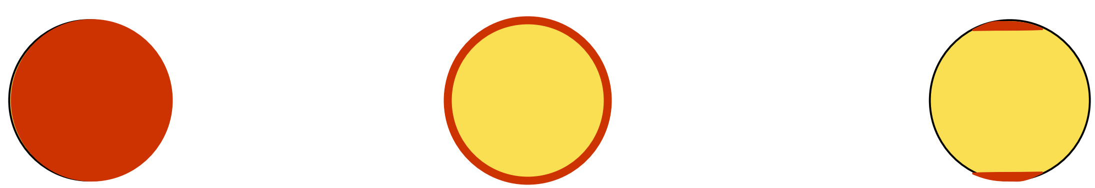
Maximum stress area

- **Fatigue ~ weak link mechanism**
- **If one link is damaged, the complete chain is broken**
- **The bigger area with a big stress, the bigger probability of damage**

Fatigue limit: Push-pull < Rotating bending < Plane bending



Size effect vs. Stress distribution effect

Fatigue limit modifying factors

- Loading factor
- Size factor
- Surface quality factor
- Notch factor
- Mean stress effect
- Effect of thermo-mechanical treatment
- Temperature effect
- Multiaxiality factor
- Oxidation factor
- Hydrogen embrittlement factor
- Irradiation factor
- Load frequency factor
- …

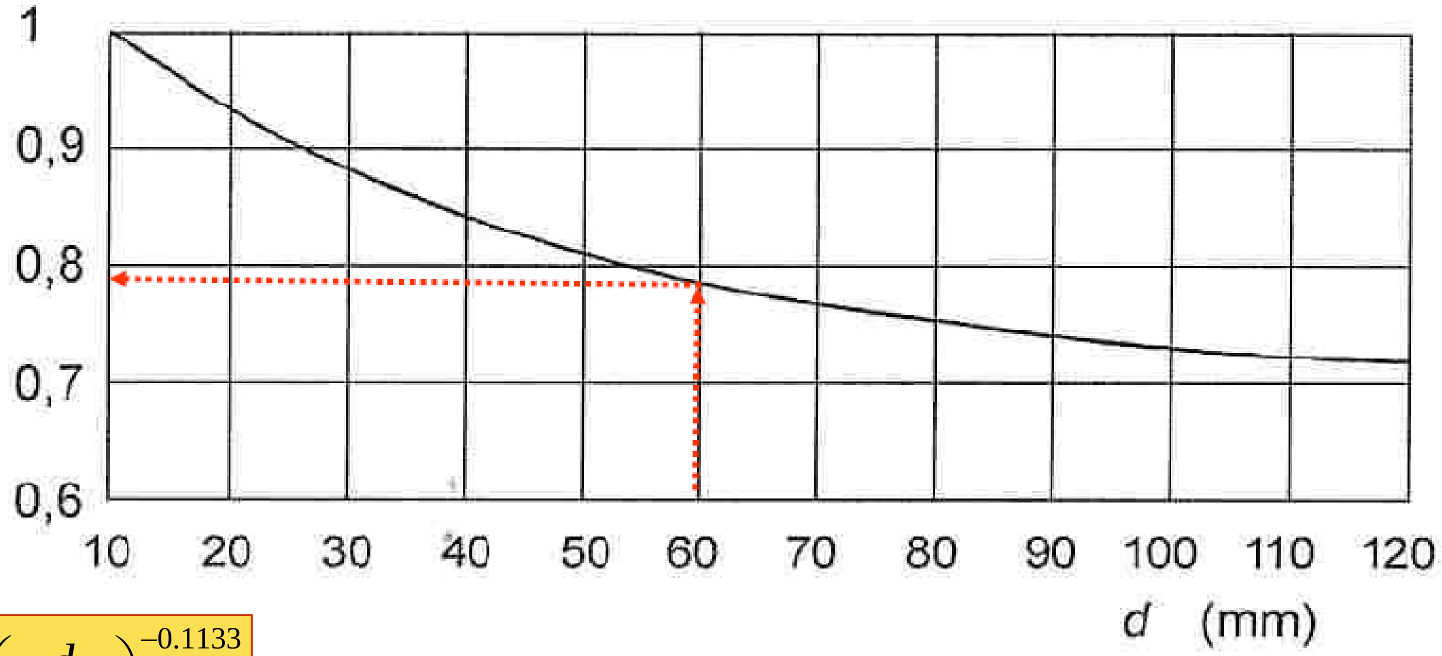
Loading factor, k_L

- Historically, fatigue limits have been determined from simple bending tests with an intrinsic stress gradient in the test specimen.
- A specimen loaded in tension will have a lower fatigue limit than the one loaded in bending.
- An empirical correction factor, called the loading factor, is used to make an allowance for this effect.

Loading Type	k_L
Axial	0.9
Bending	1.0
Torsion	0.57

Size effect

- Experimentally, larger parts have lower fatigue limits than smaller parts
- S-N curves are obtained from small specimens



$$k_S = \left(\frac{d}{7.62}\right)^{-0.1133}$$

Set for *d = 3 - 50 mm*

0.3 inch

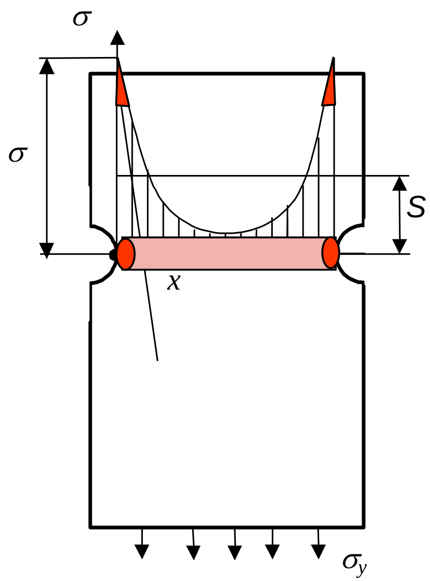
Non-circular cross-section $$d = \sqrt{\frac{A_{0.95}}{0.077}}$$

Size effect (exposed volume)

Equivalent diameter

- the volume of the component loaded by stress exceeding **95%** of the maximum

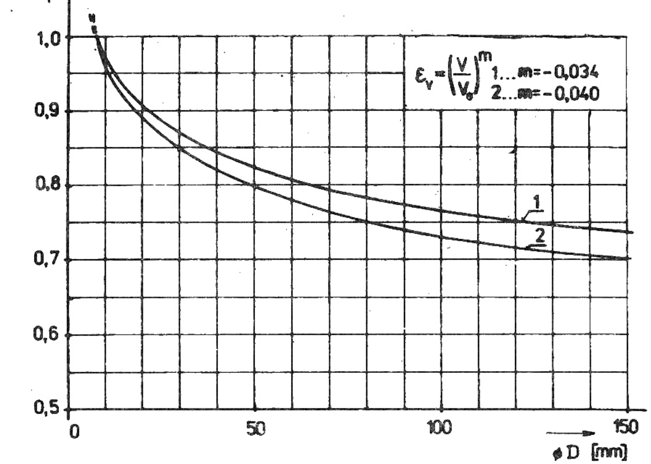
$$k_S = \left(\frac{\sigma_C^D}{\sigma_C^{d=10}} \right) = \left(\frac{V_{\exp}^D}{V_{\exp}^d} \right)^m$$



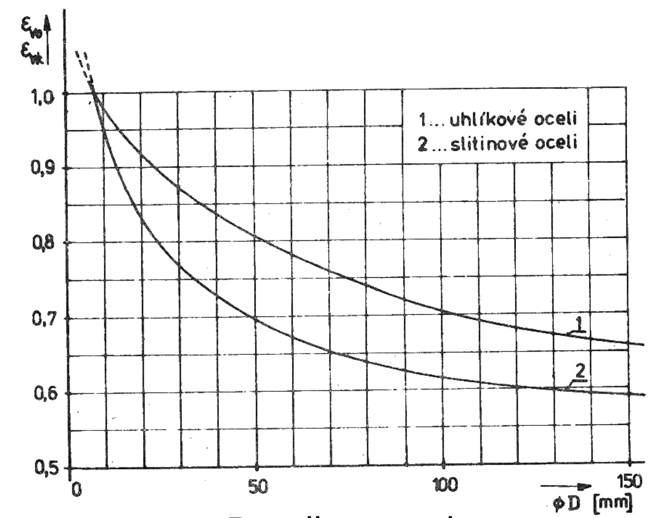
Coefficient m

- Steels: -0,03 ÷ -0,06
- Structural steels: -0,034

1-carbon steel, 2-alloyed steel

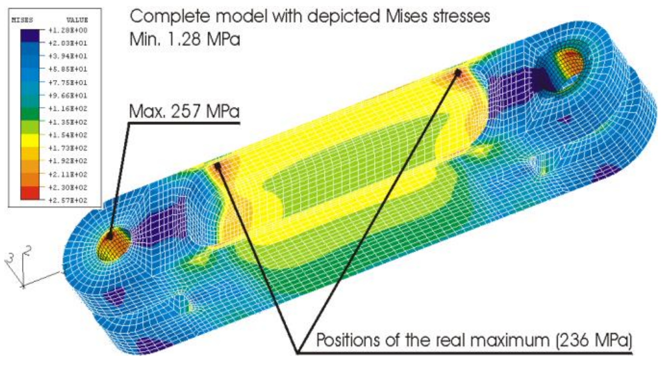


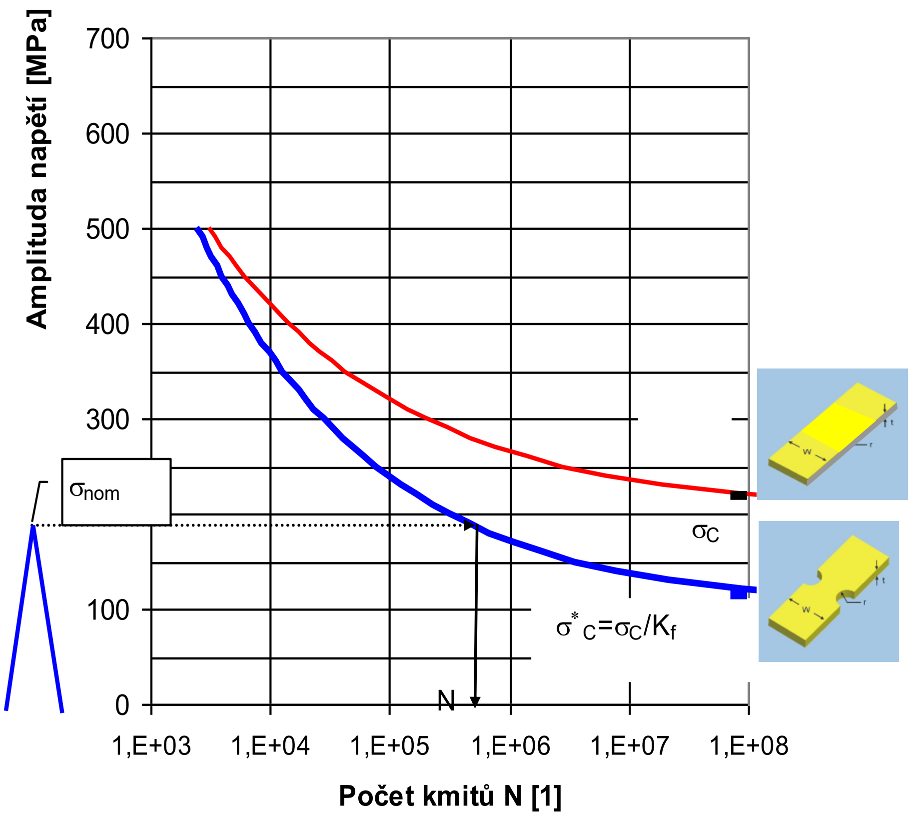
Push-pull



Bending, torsion

Notch effect on fatigue properties





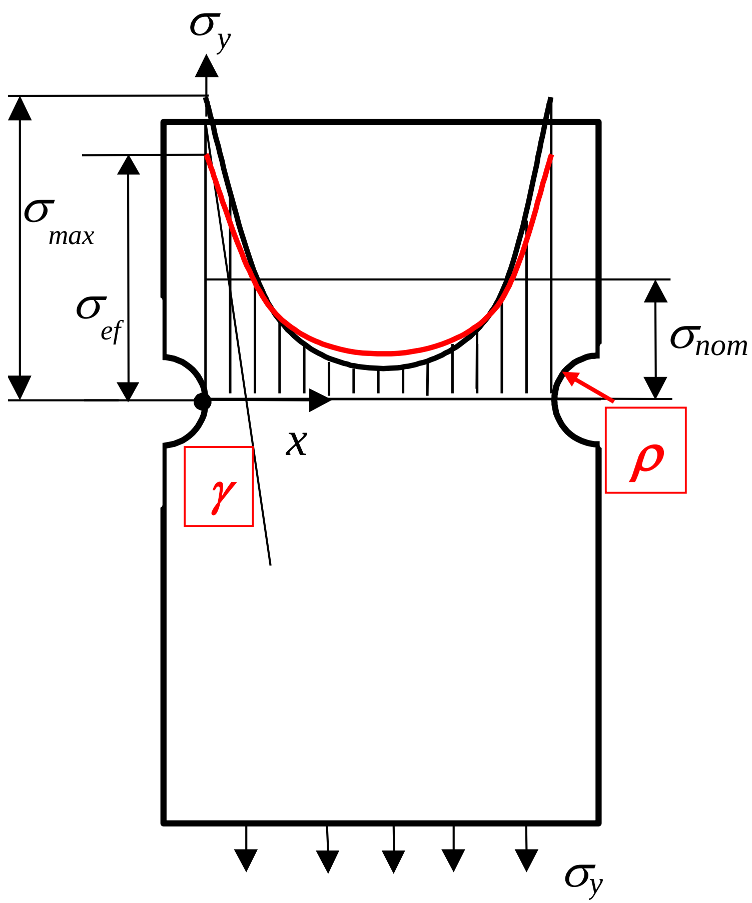
Stress concentration factor

$$K_t \equiv \alpha = \frac{\sigma_{max}}{\sigma_{nom}}$$

Notch factor

$$K_f \equiv \beta = \frac{\sigma_C}{\sigma_C^*}$$

Effect of notches – HCF



Notch sensitivity effect

$$q = \frac{\sigma_{ef} - \sigma_{nom}}{\sigma_{max} - \sigma_{nom}} = \frac{\beta - 1}{\alpha - 1} \equiv \frac{K_f - 1}{K_t - 1}$$

Relation between both coefficients

$$K_f = 1 + (K_t - 1) \cdot q = \frac{K_t}{n}$$

Fatigue factor

$$n = \frac{K_t}{K_f}$$

Two approaches:

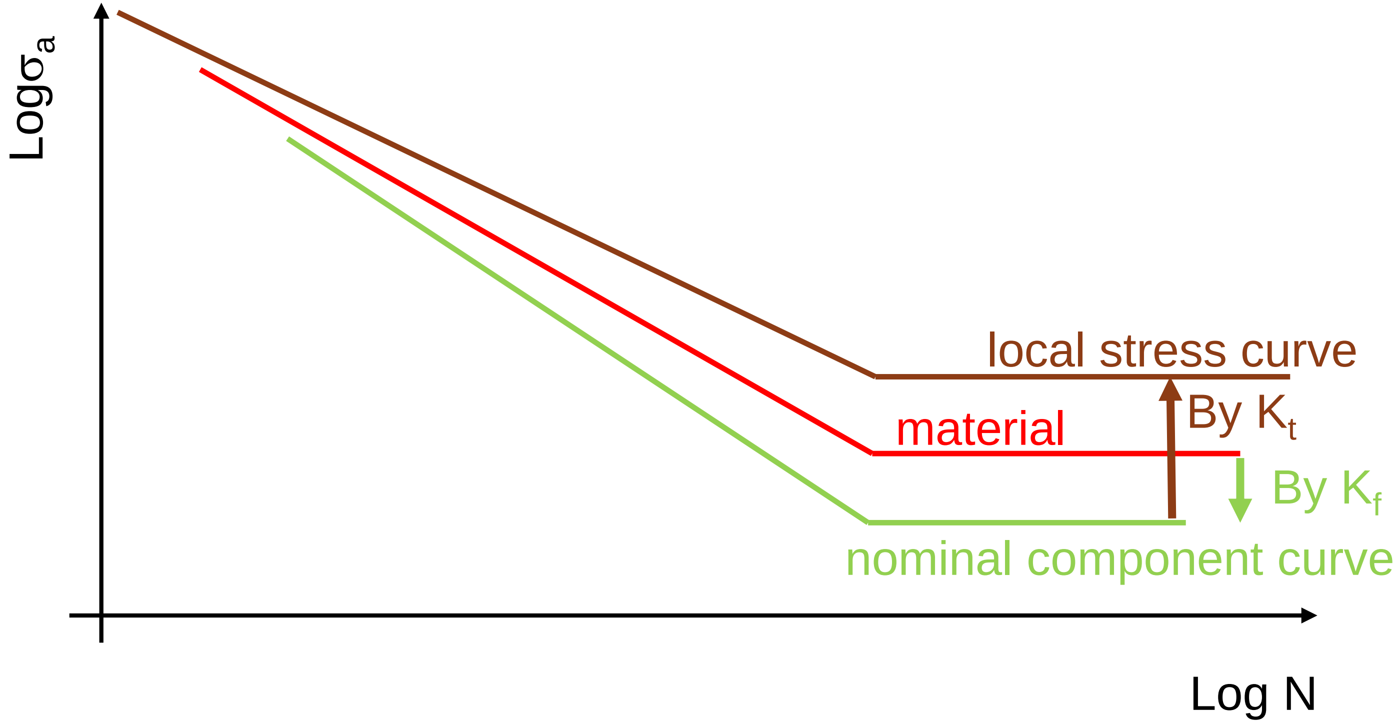
$$n = f(\rho, R_m) \equiv n_\rho$$

$$n = f(\gamma, R_m) \equiv n_\gamma$$

Relative stress gradient $\gamma = \frac{1}{\sigma}\left(\frac{\partial \sigma_y}{\partial x}\right)_{x=0}$

Notch factor modification

The S-N curve has to be modified to cover the transformation: MATERIAL -> COMPONENT



Fatigue limit:

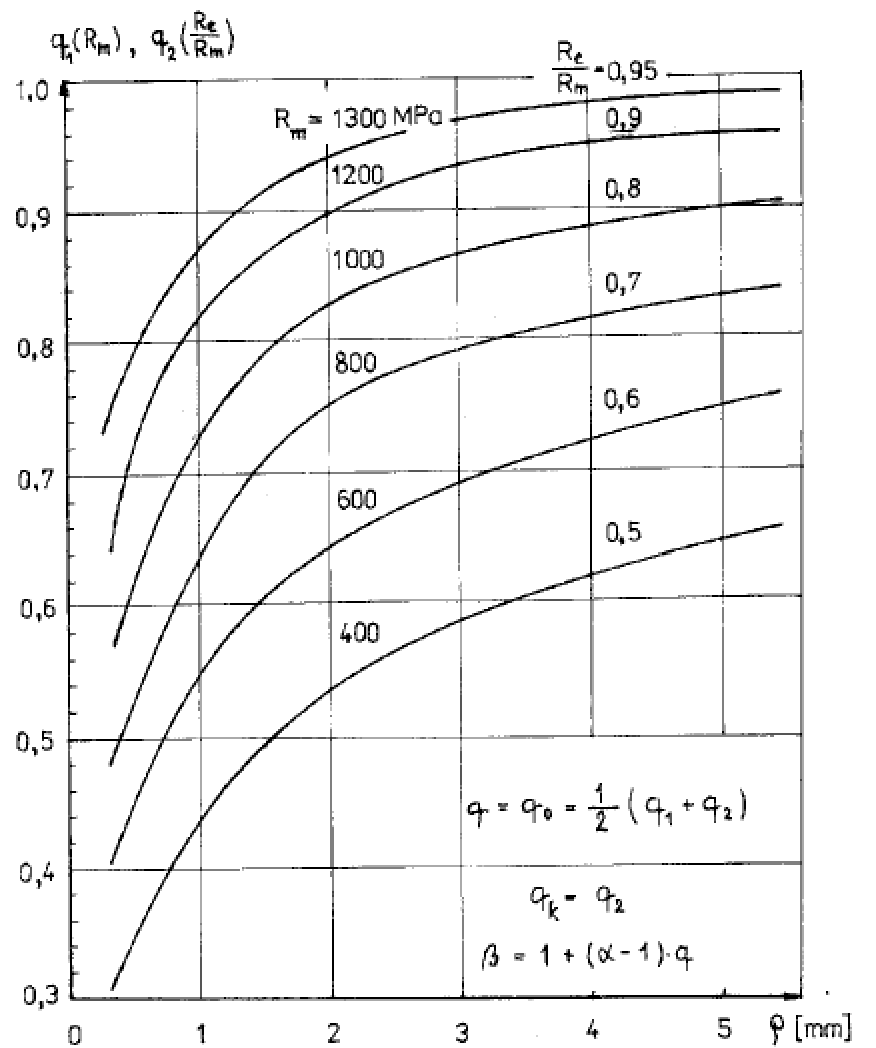
σ_{FL} , σ_C

$$K_f = \frac{\sigma_{FL,mat}}{\sigma_{FL,notched}} < K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

Notch sensitivity *q*

$$K_f = 1 + \left(K_t - 1\right) \cdot q = \frac{K_t}{n}$$

- If q = 1 then $K_t = K_f$
- The more thermo-mechanically treated material, the higher notch sensitivity expectable
- It is not a material parameter



Poloměr vrubu

Various Formulas to Include Notch Effect

$$n = \frac{K_t}{K_f}$$

	Notch factor	Fatigue factor $n = f\left(\rho, R_m\right) \equiv n_\rho$
Thum	$K_f = 1 + \left(K_t - 1\right) \cdot q = \frac{K_t}{n}$	$n_\rho = \frac{K_t}{1 + \left(K_t - 1\right) \cdot q}$
Neuber	$K_f = 1 + \frac{K_t - 1}{1 + \sqrt{\frac{A}{\rho}}}$	$n_\rho = \frac{K_t\left(1 + \sqrt{\frac{A}{\rho}}\right)}{K_t + \sqrt{\frac{A}{\rho}}}$
Peterson	$K_f = 1 + \frac{K_t - 1}{1 + \frac{a}{\rho}}$	$n_\rho = \frac{K_t\left(1 + K_t \frac{a}{\rho}\right)}{K_t + \frac{a}{\rho}}$
Heywood	$K_f = \frac{K_t}{1 + 2\frac{K_t - 1}{K_t}\sqrt{\frac{a'}{\rho}}}$	$n_\rho = 1 + 2\frac{K_t - 1}{K_t}\sqrt{\frac{a'}{\rho}}$

If FEA Results Processed - Fatigue Factor

$$n = f\left(\gamma, R_m\right) \equiv n_\gamma$$

Volejnik, Kogaev, Serensen

$$n_\gamma = 1 + \left(\frac{1}{\nu_\infty} - 1\right) \cdot \left(\frac{L/\gamma}{2\pi d_0 \big/ \frac{2}{d_0}} \right)^{-\mu}$$

Eichlseder (FEMFAT)

$$n_\gamma = 1 + \left(\frac{\sigma_{-1,b}}{\sigma_{-1}} - 1\right) \cdot \left(\frac{\gamma}{2/d_0}\right)^{K_D}$$

Siebel, Stiller

$$n_\gamma = 1 + \sqrt{c \cdot \gamma}$$

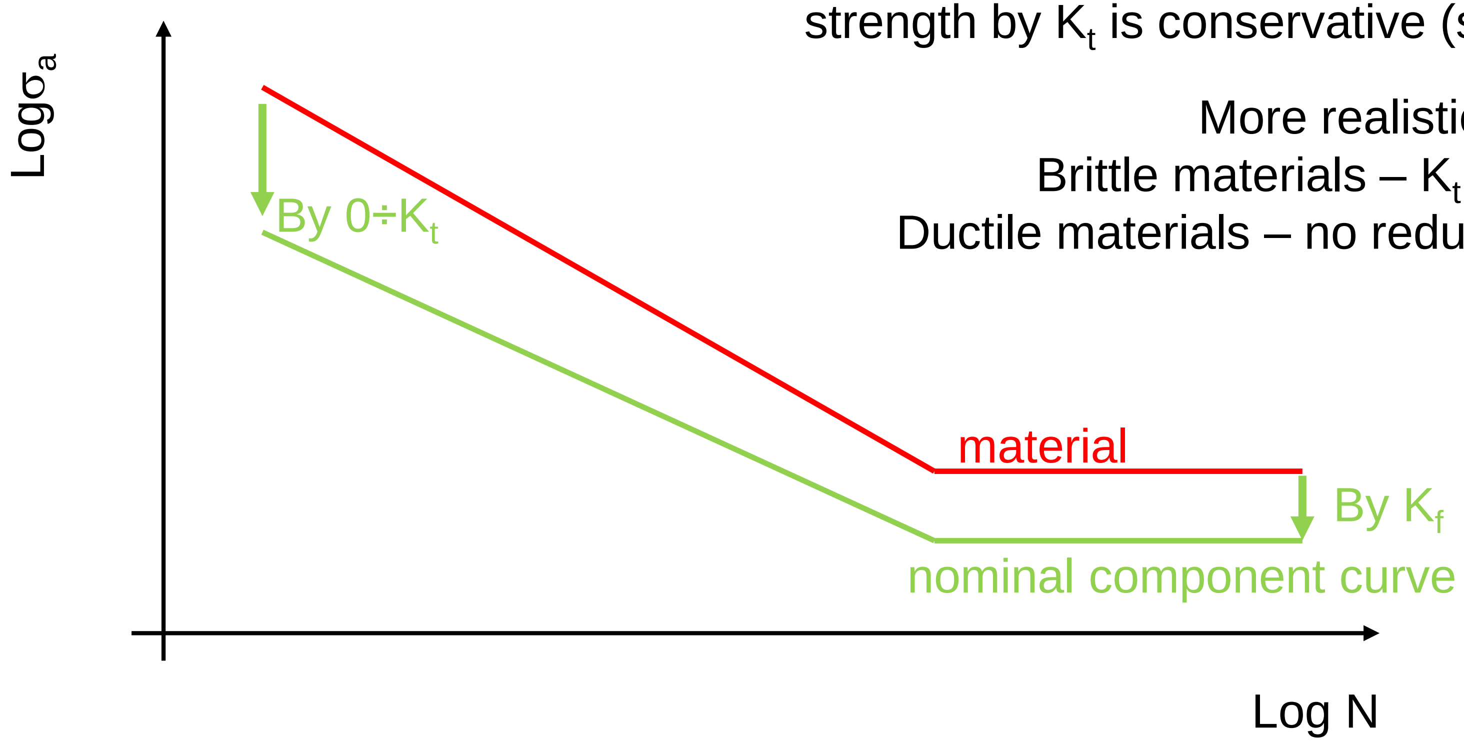
FKM-Richtlinie

$\gamma' \le 0{,}1 mm^{-1}$	$0{,}1 mm^{-1} < \gamma' \le 1 mm^{-1}$	$1 mm^{-1} < \gamma' \le 100 mm^{-1}$
$n_\gamma = 1 + \gamma'.10^{-\left(a_G - 0{,}5 + \frac{R_m}{b_G}\right)}$	$n_\gamma = 1 + \sqrt{\gamma'}.10^{-\left(a_G + \frac{R_m}{b_G}\right)}$	$n_\gamma = 1 + \sqrt[4]{\gamma'}.10^{-\left(a_G + \frac{R_m}{b_G}\right)}$

Lower lifetimes?

K_t ~ local stress increase, i.e. decrease of the tensile strength by K_t is conservative (safe)

More realistically:
Brittle materials – K_t ratio
Ductile materials – no reduction



Fatigue limit:

σ_{FL} , σ_C

$$K_f = \frac{\sigma_{FL,mat}}{\sigma_{FL,notched}} < K_t = \frac{\sigma_{max}}{\sigma_{nom}}$$

Stress Gradient Vs Critical Distance

In addition to the nominal approach (using K_f), there are two basic concepts for local stress evalution. **The basic premise:** Stress directly at the notch:

1) Relates to K_t factor

2) Should not be used directly without any modification for fatigue life estimate, which depends on K_f

3) If used, the result is likely to be conservative

Stress gradient:

- Its value helps to compute fatigue factor *n*, which shifts the **effective material curve upwards**

Critical distance:

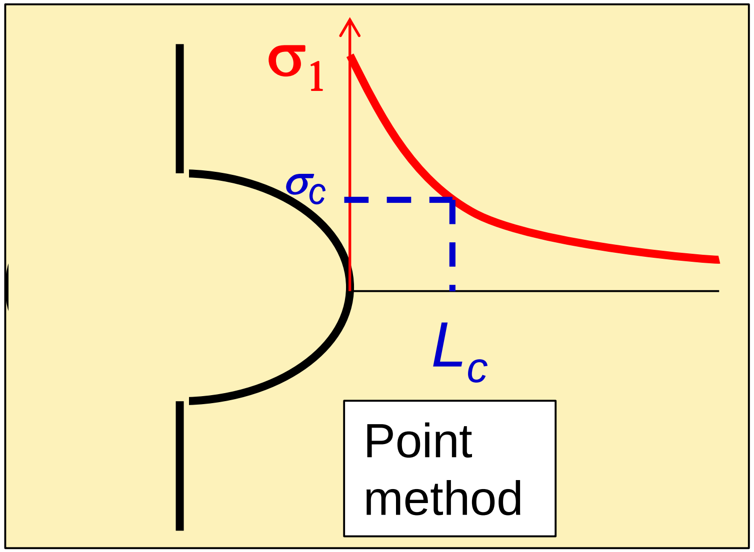
- The stress used for calculating the **fatigue life** is derived **in a** particular **distance** from the notch root
- **the effective stress is** thus **lower,** than it would be at the notch

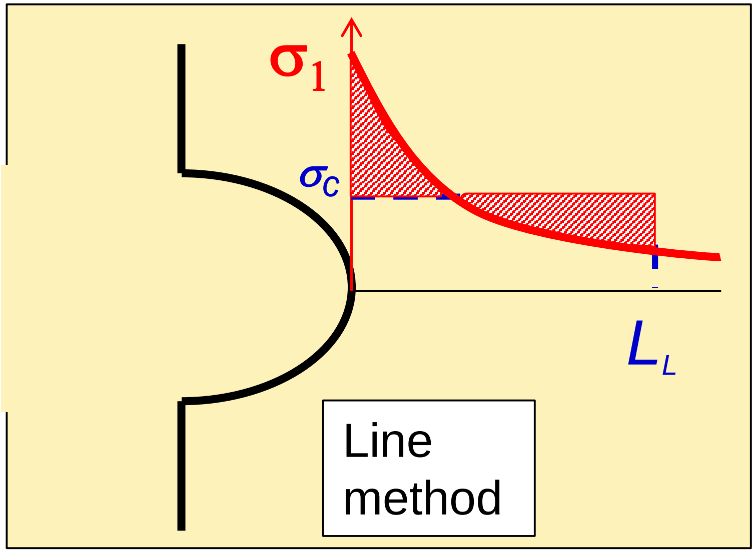
Theory of Critical Distance (TCD)

Simplifies the critical volume method to a 1D problem.

Theorem: The crack initiation at the notch starts in the moment, when some reference value (Sig HMH, Sig1, Damage Parameter P) in a defined depth below the surface reaches critical value:

- Point method (crit. depth L_c)
- Line method (analyses the integral **mean of the stress from the notch root to the distance** L_L)



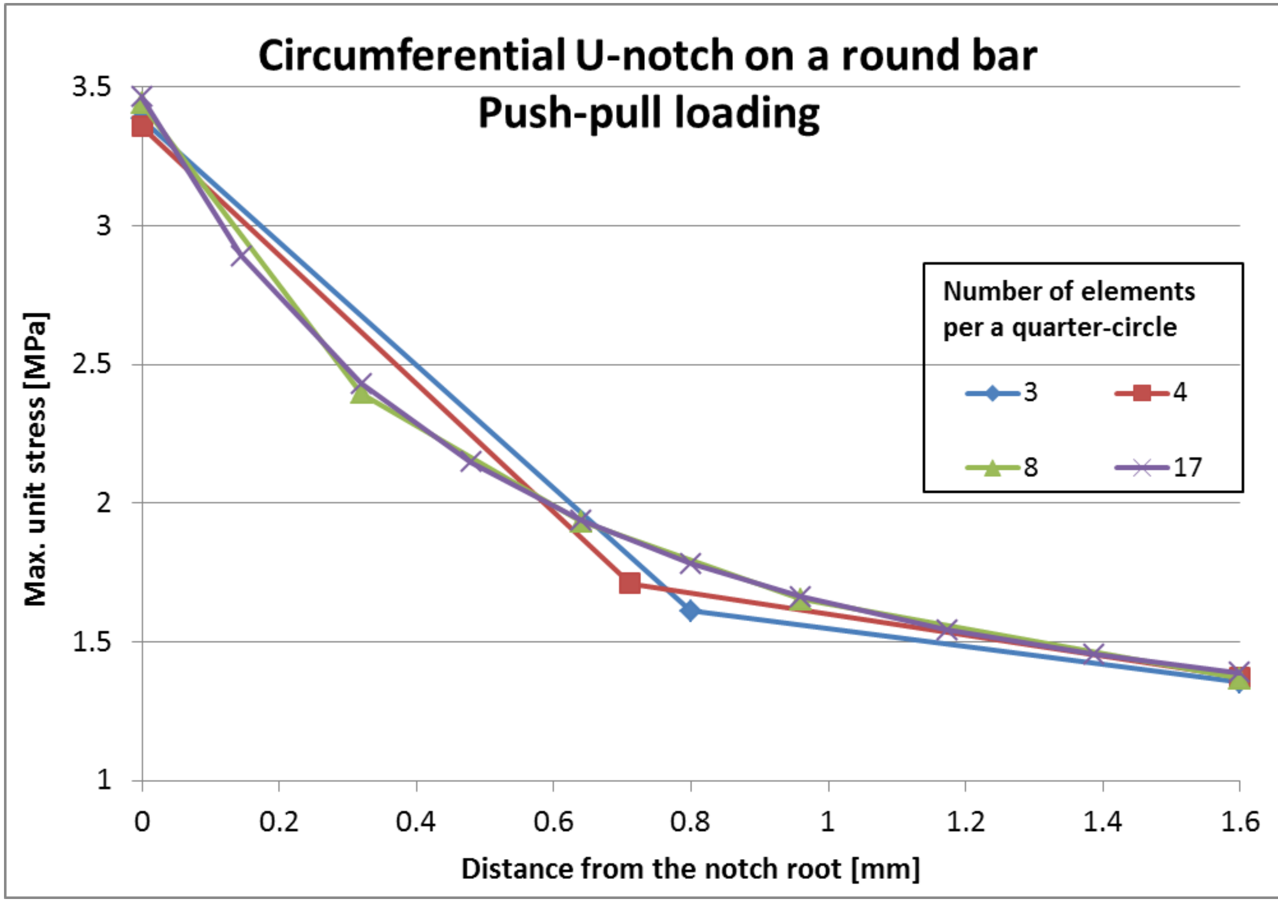


Summary – Notch Effect

- The notches decrease the fatigue life and fatigue strength
- Nominal solution works through notch sensitivity factor q and stress concentration factor K_t to estimate notch factor K_f, and to modify the S-N curve accordingly
- If directly the local stress value read from the FEA-results is used, the resulting life value is likely to be conservatively underestimated.
- This is the reason, why some correction has to happen:
  - TCD: Decrease the processed stress (and use original S-N curve)
  - Stress gradient: Modify the material curve (and use original local stress)
- Though such methods have been already implemented in some commercial fatigue solvers, the extent of validation has to be doubted.

Effect of the FE-model quality

- Best results for equi-sided elements - bricks
- The sizes of element sides can differ for some variants here – this is the reason for only slight difference between variants with 3 and 4 elements

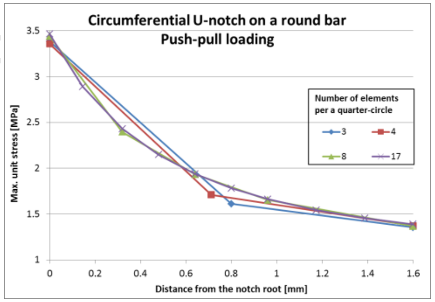


- Common mesh quality in static analyses of airplanes in Evektor – 2 elements per a quarter-circle

Push-pull load case

- Relative stress gradient (RSG):

$$\gamma' = \frac{\gamma}{\sigma_{max}} = \frac{1}{\sigma_{max}} \frac{d\sigma}{dx}$$



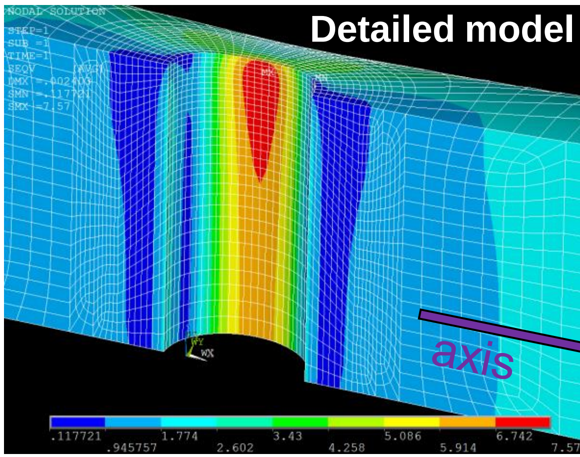
Note:
Only minor differences in results of maximum stress are likely to be caused by a simple load mode with the obvious hot-spot location, in which the node is positioned.

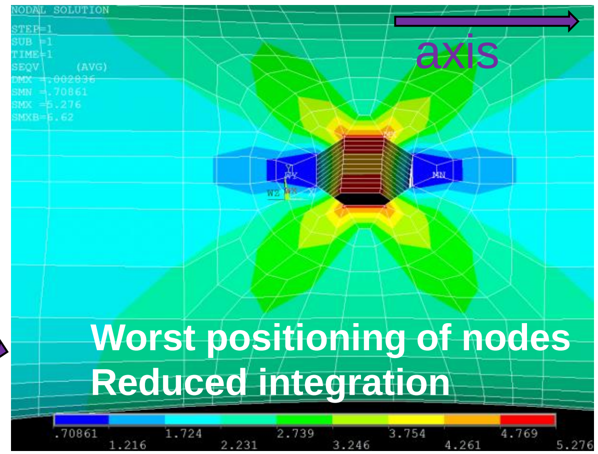
- If we compare various notches here:
  - RSG by common models are close to one half of the right value
  - Minimum effect of the mesh quality on the SCF (see $K_{t,net}$)
  - First principal stress results in less scattered analysis error compared with von Mises stress

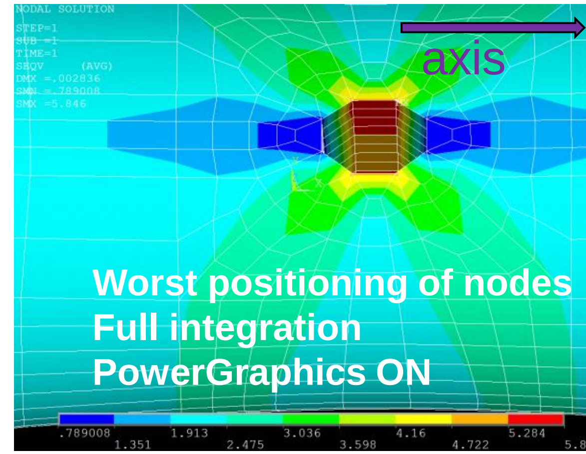
Specimen type (radius of the notch root)	equivalent stress	parameter	ideal FE-model	Evektor FE-model	relative deviation from the ideal value
U-notch (R=1.6 mm), real model uses **3** elements per a quarter-circle	von Mises stress	r.s.g. [1/mm]	1.610	0.836	-48%
		Kt, net [-]	2.133	2.076	-3%
	1st principal stress	r.s.g. [1/mm]	1.203	0.655	-46%
		Kt, net [-]	2.360	2.306	-2%
Fillet (R=0.4 mm), real model uses **3** elements per a quarter-circle	von Mises stress	r.s.g. [1/mm]	5.953	2.821	-53%
		Kt, net [-]	2.469	2.229	-10%
	1st principal stress	r.s.g. [1/mm]	4.618	2.452	-47%
		Kt, net [-]	2.717	2.543	-6%
V-notch (R=1.6 mm), real model uses **3** elements per a quarter-circle	von Mises stress	r.s.g. [1/mm]	1.582	0.778	-51%
		Kt, net [-]	2.242	2.177	-3%
	1st principal stress	r.s.g. [1/mm]	1.141	0.603	-47%
		Kt, net [-]	2.506	2.453	-2%
Hole (R=2.0 mm), real model uses **2** elements per a quarter-circle	von Mises stress	r.s.g. [1/mm]	1.056	0.486	-54%
		Kt, net [-]	2.513	2.476	-1%
	1st principal stress	r.s.g. [1/mm]	0.814	0.420	-48%
		Kt, net [-]	2.628	2.609	-1%

Push-pull load case

Beware: Small maximum stress errors are likely to be caused by simple load modes and a priori known critical location. If a node is not placed to the maximum stress location, the output can be much worse.



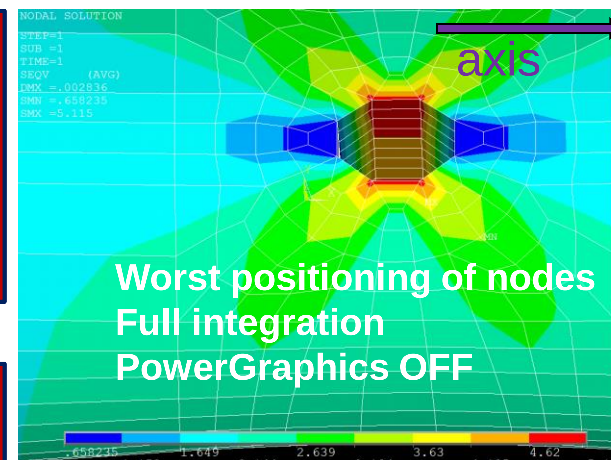




Output: At unknown critical place, stress can be underestimated by

$$\mathit{Relative\ error} = \frac{7.57 - 5.115}{5.115} = 48.0\%,$$

while neglecting other errors (wrong critical place location, much smaller stress gradient).

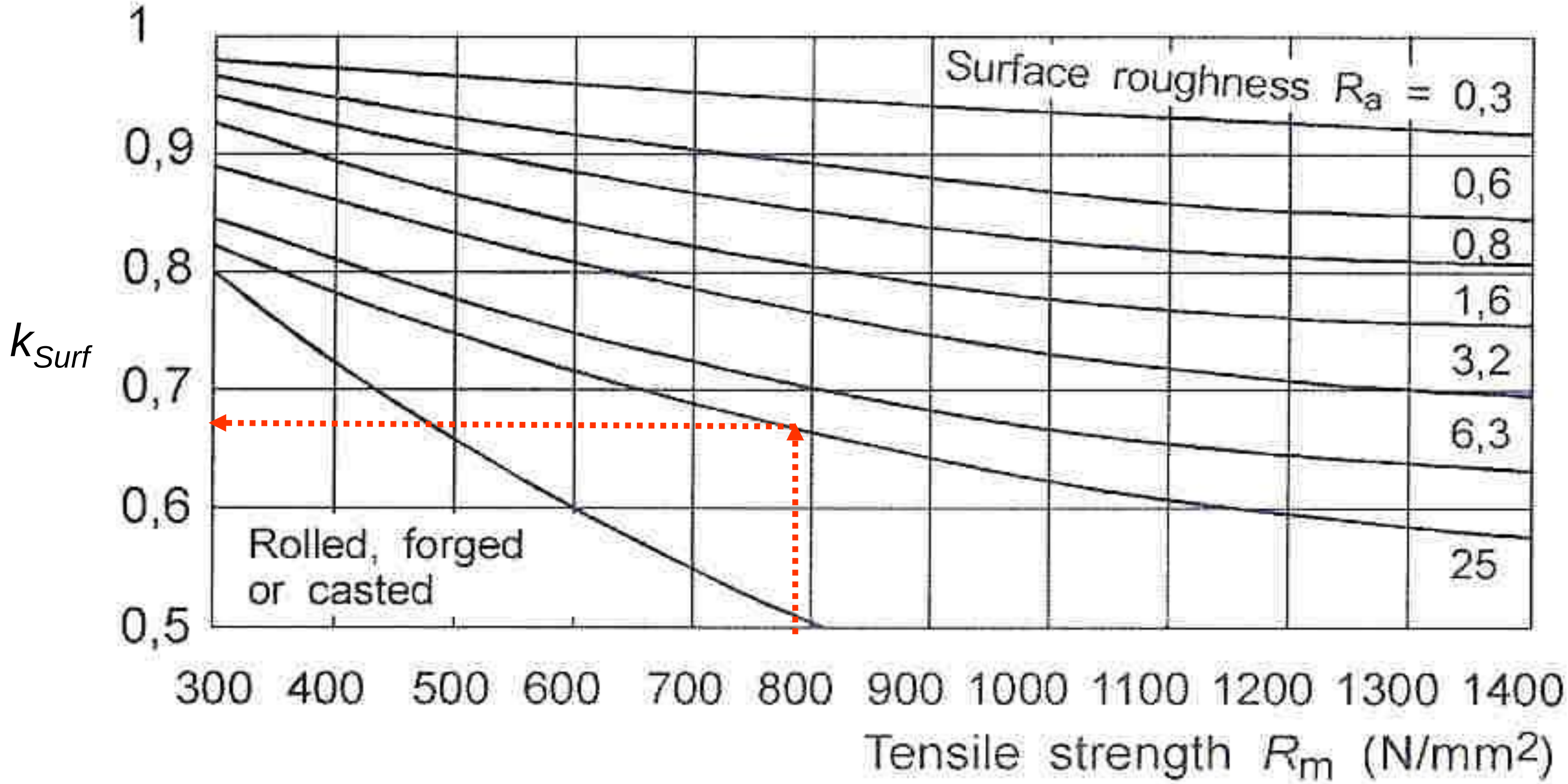


All problems with the setup of postprocessing are **caused by rough mesh**, and should not occur for better meshes**.**

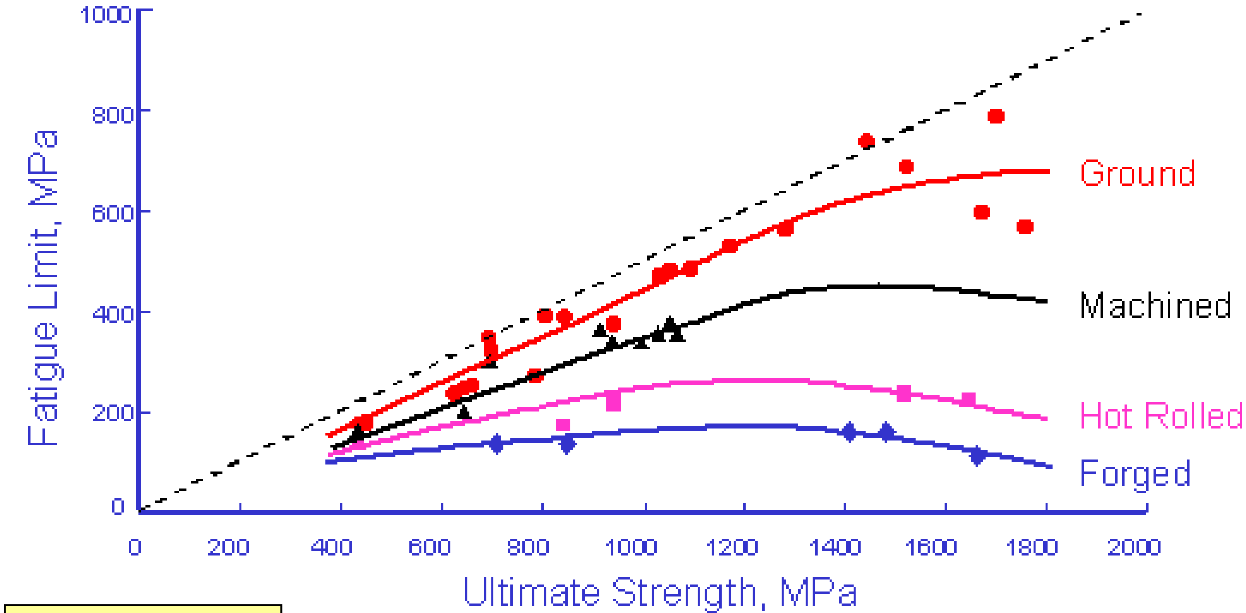
Other fatigue limit modifying factors

Fatigue surface quality

$$k_{Surf} = \frac{\sigma_c^{real}}{\sigma_c^{specimen}}$$



Surface Roughness Effect



Noll and Lipson: Allowable Working Stresses. Society for Experimental Stress Analysis, Vol. III, no. 2, 1949

$$k_{Surf} = a \cdot R_m^{\beta}$$

	a	β
Ground	1.58	-0.085
Machined	4.51	-0.265
Hot rolled	57.7	-0.718
Forged	272	-0.995

These are mean regression curves, not the safe ones!

Source: www.efatigue.com

Surface Roughness Effect – Part II

FKM-Richtlinie

$$k_{Surf} = 1 - a \cdot \log R_z \cdot \log\left(\frac{2 \cdot R_m}{R_{m,N,\min}}\right)$$

Prepared for analyzing normal stresses, if shear stresses concerned, the factor has to be multiplied by a parameter $f_{W,t}$

R_z Mean roughness in microns

R_m Tensile strength in MPa

	Steel	Cast steel	Cast iron with spheroidal graphite	Tempered cast iron	Grey cast iron	Wrought aluminum alloys	Cast aluminum alloys
	Ocel	Litá ocel	Litina s kuličkovým grafitem	Temperovaná litina	Šedá litina	Hliníkové slitiny tvářené	Hliníkové slitiny lité
a	0.22	0.2	0.16	0.12	0.06	0.22	0.2
$R_{m,N,min}$	400	400	400	350	100	133	133
$f_{W,t}$	0.577	0.577	0.65	0.75	1	0.577	0.75

Effect of Thermo-Mechanical Treatment - k_T

$$k_{Tec} = \frac{\sigma_c^{real}}{\sigma_c^{specimen}}$$

FKM-Guideline: Analytical Strength Assessment of Components in Mechanical Engineering. 5th revised edition. Frankfurt/Main, Forschungskuratorium Maschinenbau (FKM) 2003.

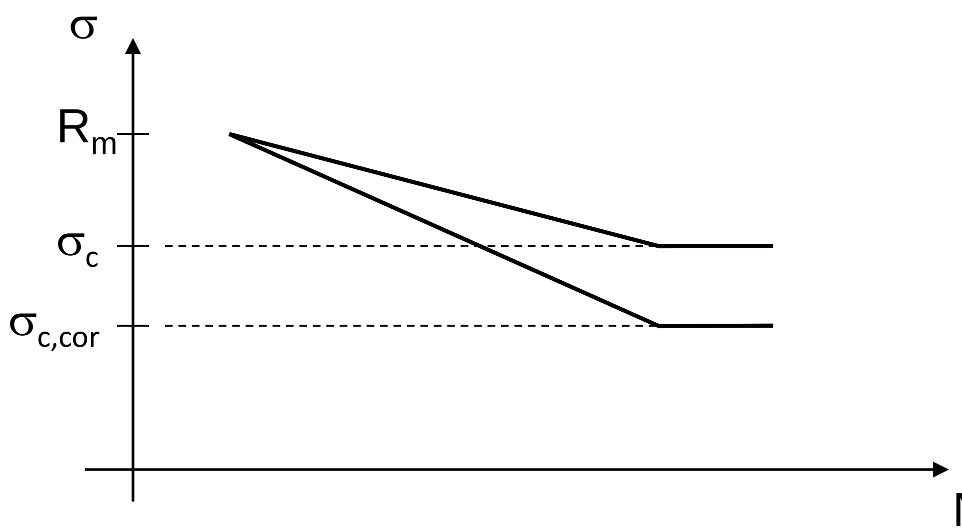
	unnotched	notched
Steel		
Chemo-thermal treatments		
Nitriding Depth of case 0,1...0,4 mm Surface hardness 700 to 1000 HV 10	1,10 - 1,15 (1,15 - 1,25)	1,30 - 2,00 (1,90 - 3,00)
Case hardening Depth of case 0,2 ... 0,8 mm Surface hardness 670 to 750 HV 10	1,10 - 1,50 (1,20 - 2,00)	1,20 - 2,00 (1,50 - 2,50)
Carbo-nitriding Depth of case 0,2 ... 0,8 mm Surface hardness 670 to 750 HV 10	(1,80)	
Mechanical treatment		
Cold rolling	1,10 - 1,25 (1,20 - 1,40)	1,30 - 1,80 (1,50 - 2,20)
Shot peening	1,10 - 1,20 (1,10 - 1,30)	1,10 - 1,50 (1,40 - 2,50)
Thermal treatment		
Inductive hardening Flame-hardening Depth of case 0,9 ... 1,5 mm Surface hardness 51 to 64 HRC	1,20 - 1,50 (1,30 - 1,60)	1,50 - 2,50 (1,60 - 2,80)

Summary: Effects of Surface State

The most influencing for high number of cycles

Thermo-mechanical processing of the surface layer affects its properties

- **Mild changes in static properties**
- **Pronounced effect around fatigue limit**
- **i.e. the effect increases with increasing the desired fatigue life**



Fatigue Limit of a Notched Part

$$\sigma_C^{\text{real}} = \frac{\sigma_C^{\text{smooth specimen in push-pull}}}{K_{f,tot}}$$

Němec, Puchner, Linhart

$$K_{f,tot} = \frac{K_f}{k_S \cdot k_{Surf} \cdot k_{Tec}}$$

Volejnik, Kogaev, Serensen

$$K_{f,tot} = \left(\frac{K_f}{k_S} + \frac{1}{k_{Surf}} - 1 \right) \cdot \frac{1}{k_{Tec}}$$

FKM Guideline

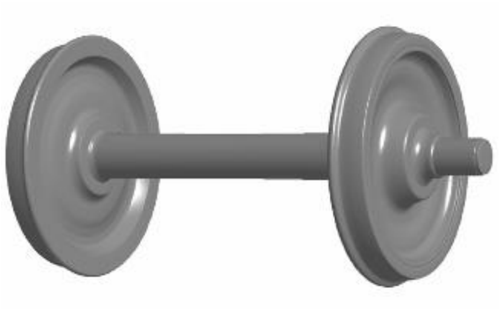
$$K_{f,tot} = \left(K_f + \frac{1}{k_{Surf}} - 1 \right) \cdot \frac{1}{k_S \cdot k_{Tec}}$$

Eichlseder

$$K_{f,tot} = \frac{k_{statistic}}{k_S \cdot k_{Tec} \cdot k_{temp} \sqrt{k_{grad} - 1 + k_{Surf}^2}}$$

Fatigue limit of a real part

Train wheel set:



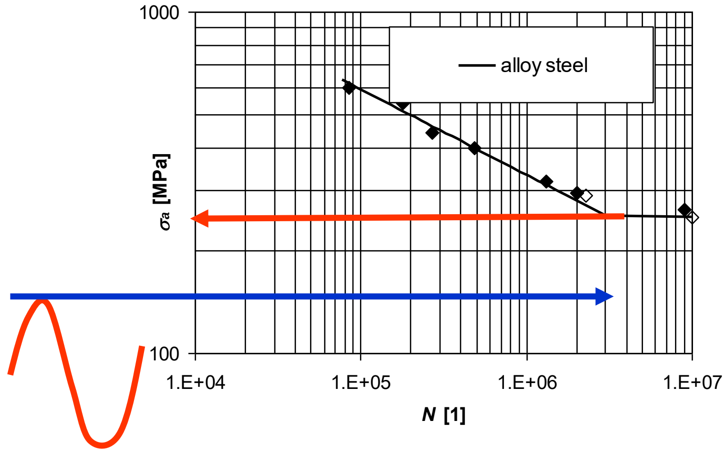
Target life?

$$\sigma_{FL,N} = \frac{\sigma_{FL} \cdot k_L \cdot k_{SF} \cdot k_S \cdot k_T}{K_f}$$

Safety factor for unlimited fatigue life

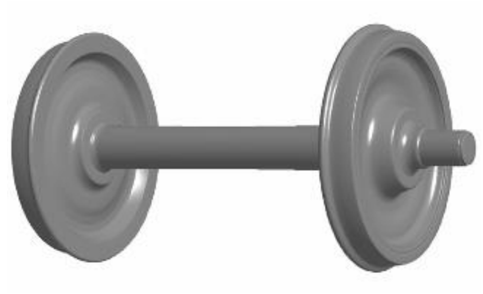
1. Alternating stress (R=-1)

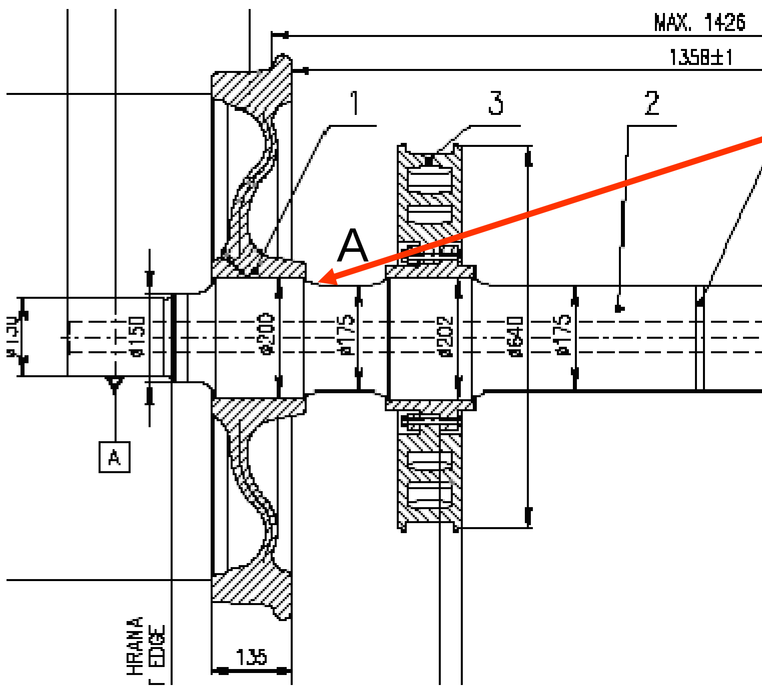
- In-service loading stress amplitude σ_a
- fatigue limit of the real part in the critical cross section area $\sigma_{FL,N}$



$$n_\sigma = \frac{\sigma_{FL,N}}{\sigma_a}$$

Example – Fatigue safe factor calculation





Problem description:

Railway axle

Material: alloy steel 25CrMo4, ASTM 4130

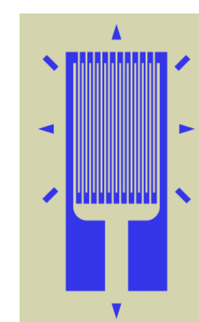
Point A of the potential crack initiation

ε

Experimentally measured strain amplitude (in the point A):

$$\varepsilon_{a,\max} = 312\left[\text{microstrain}\right]$$

Strain gauge (measures resistance changes):



Example – continuation

http://www.efatigue.com

Material Property Finder

☐ Crack Growth
☐ Strain-Life
☒ **Stress-Life**

Specification: Show All

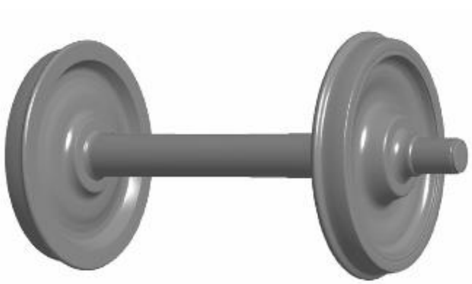
Update Material List Filters

Steel 1045, Q&T, BHN=336
Steel 1045, Q&T, BHN=390
Steel 1045, Q&T, BHN=410
Steel 1045, Q&T, BHN=500
Steel 1045, Q&T, BHN=563
Steel 1045, Q&T, BHN=595
Steel 300M, Su=1958.2
Steel 4130 sheet, Su=1241.1
Steel 4130 sheet, Su=806.7
Steel 4130, BHN=259

Steel 4130, BHN=259

Technology		Constant Amplitude Stress-Life
Owner		public
Material Type		steel
Material Specification		AISI 4130
Material Alloy		4130
Brinell Hardness Number		259
Elastic Modulus	E=	200000 MPa
Ultimate Strength	S_u=	778 MPa
Curve Intercept	S_f'=	1195 MPa
Curve Slope	b=	-0.077
Material Reference		SAE J1099 - June 1998 (from eN data)

Material Property Estimator

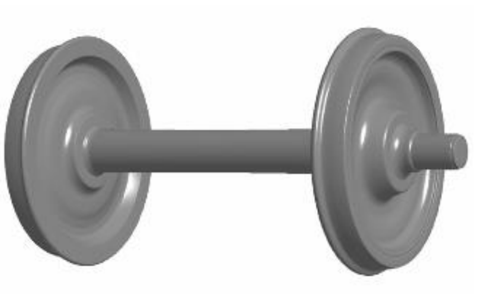


$$\sigma_a = \sigma'_f \cdot (2N)^b$$

$$\sigma_{FL} = 1195 \cdot (2 \cdot 10^7)^{-0.077} = 327.5\ \text{MPa}$$

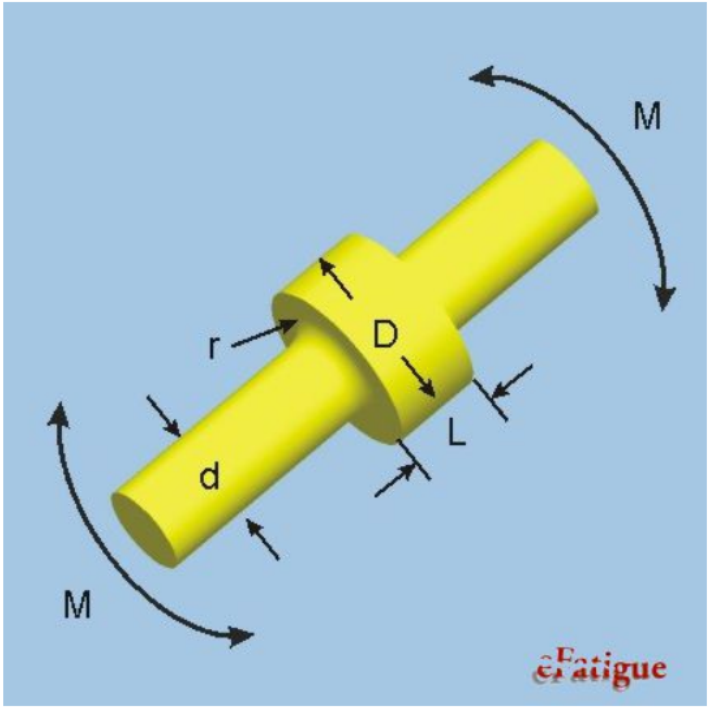
Example – continuation

http://www.efatigue.com



Round Shaft with Double Fillets

Tension | Bending | Torsion



Variables

Net Section Stress

D 250 mm

d 175 mm

r 15 mm

L 75 mm

where

0.01 < d/D

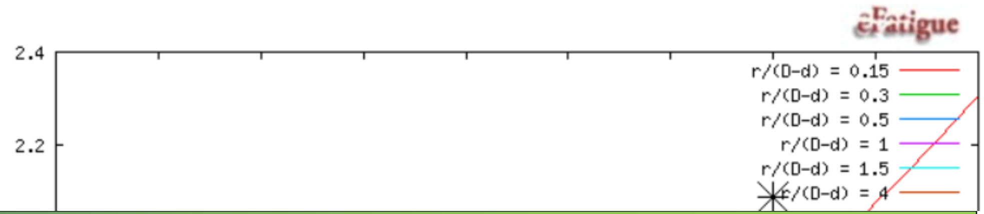
0.1 < r/(D-d) < 5

Calculate Kt | Select Different Geometry

Results

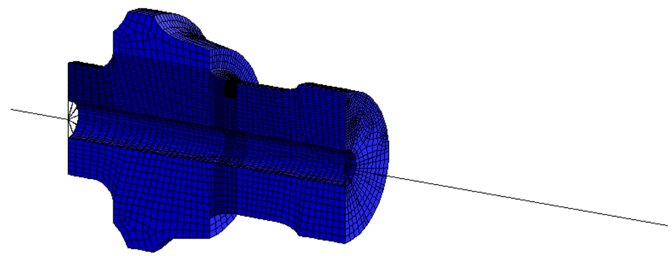
K_t = 2.09

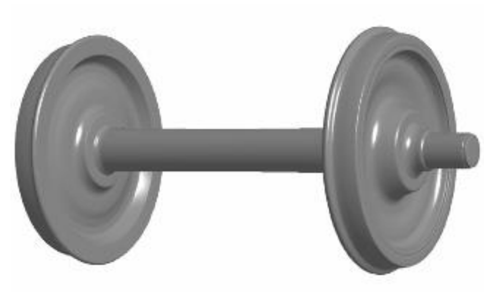
Peterson Plot

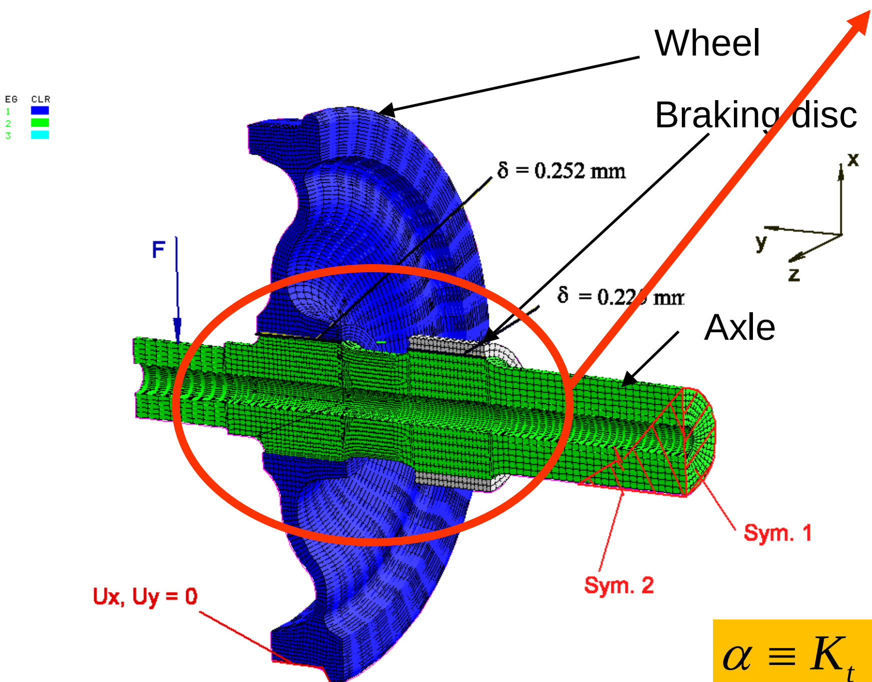


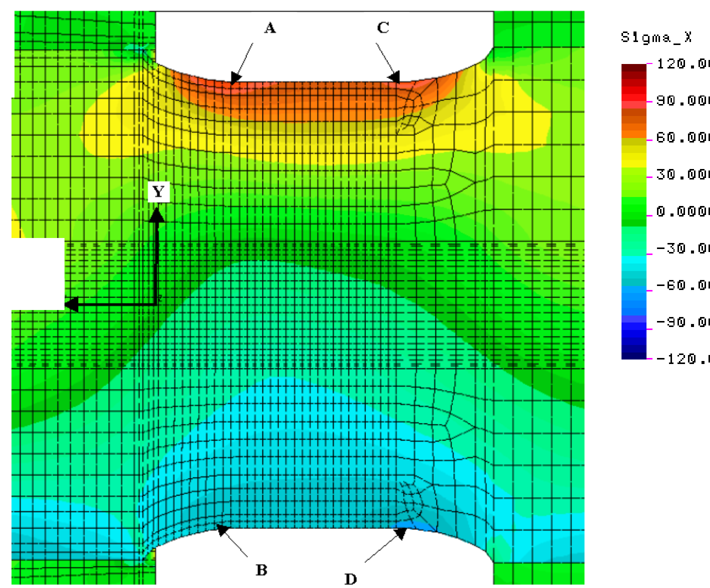
Example – continuation

FEA calculation @ CTU in Prague



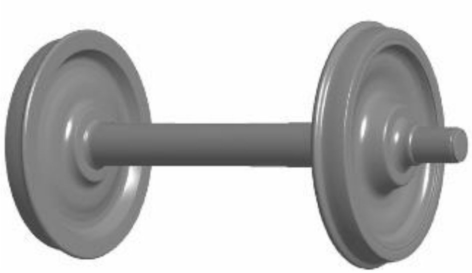


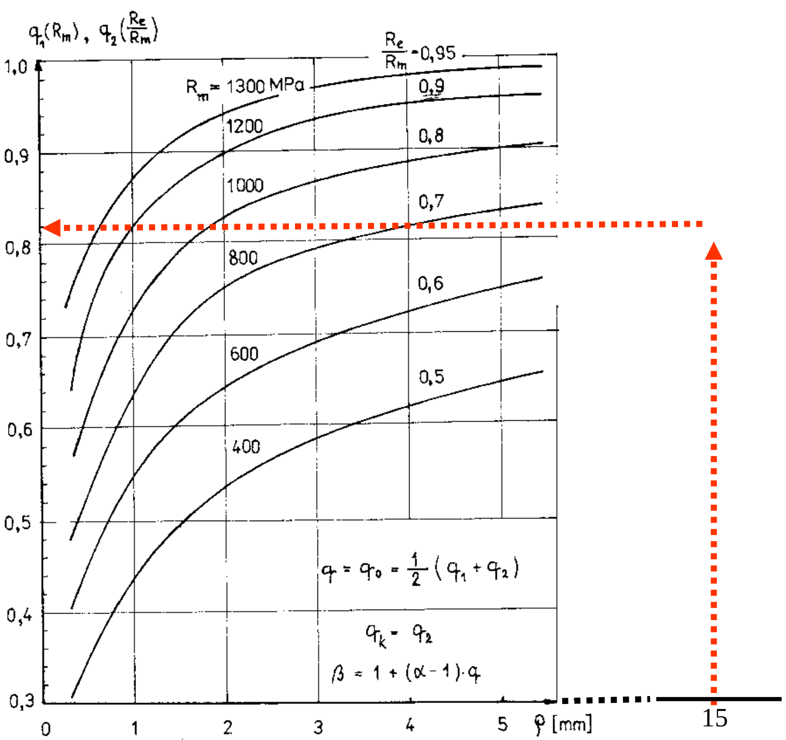




$$\alpha \equiv K_t = 1.95$$

Example – continuation





www.fadoff.cz

Material - 25CrMo4

Basis | Chemical composition | Treatments | **Material parameters**

Mark	Value	Unit	Title	Group
A	12	%	Elongation at fracture	Static data
E	220000	MPa	Tensile elasticity modulus	Static data
f0	600	MPa	Fatigue limit	Repeated tension
f1	361	MPa	Fatigue limit	Fully reversed push-pull
f_d	34	mm	Diameter of specimen at active cross-section	Fully reversed push-pull
HV	332	-	Hardness (acc. to Vickers)	Static data
Ncf1	2000000	-	Number of cycles at fatigue limit	Fully reversed push-pull
RA	73	%	Reduction of area at fracture	Static data
sig_u	780	MPa	Ultimate tensile strength	Static data
sig_y	660	MPa	Tensile yield stress	Static data
T	20	degC	Ambient temperature	Static data
t1	228	MPa	Fatigue limit	Fully reversed torsion

$q_1 \sim 0.83$

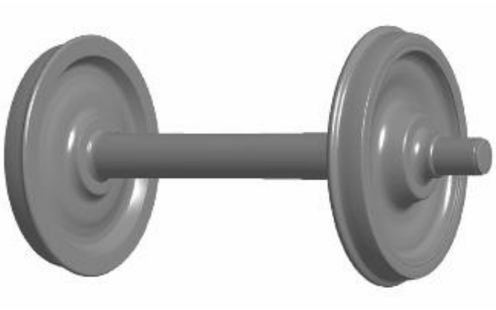
$\frac{R_e}{R_m} = 0.85 \qquad i.e.: q_2 \sim 0.93$

$\alpha \equiv K_t = 1.95$

$\beta = K_f = 1 + (K_t - 1) \cdot q = 1 + (1.95 - 1) \cdot (0.82+0.93)/2 = 1.83$

Example – continuation

Estimation of the fatigue limit of a real part



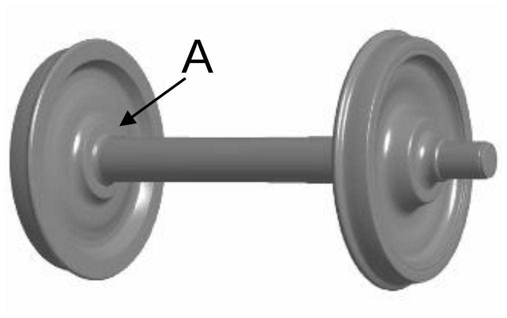
factor	*k*	value
loading	k_L	1.00
surface finish	k_{SF}	0.67
size factor	k_S	0.70
size factor	kT	1.00

$$\sigma_{FL,N} = \frac{\sigma_{FL} \cdot k_L \cdot k_{SF} \cdot k_S \cdot k_T}{K_f}$$

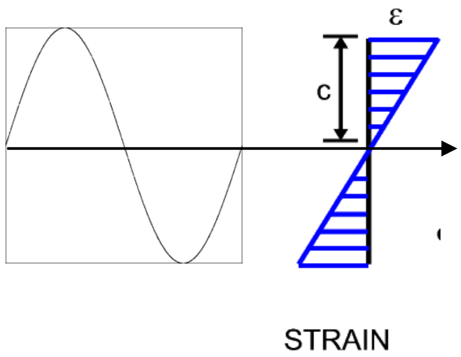
$$\sigma_{FL,N} = \frac{327.5 \cdot 1.00 \cdot 0.67 \cdot 0.70 \cdot 1.00}{1.83} = 83.9\ \mathrm{MPa}$$

Example – continuation

Estimation of the nominal stress amplitude



Experimental strain amplitude measurement (in the point A):

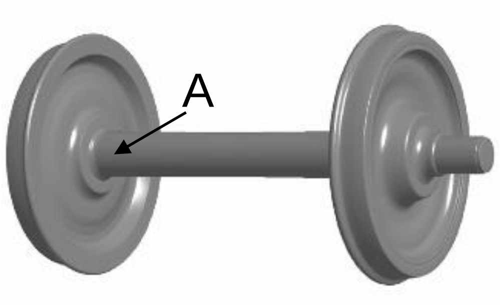


$$\varepsilon_{a,\max} = 312\left[\text{microstrain}\right]$$

$$\sigma_a = E \cdot \varepsilon = 220000 \cdot 0.000312 = 68.6\ \text{MPa}$$

Example – continuation

Estimating the safety factor n_{FL}



Loading: $\sigma_a = 68.6\ \mathrm{MPa}$

Structure properties: $\sigma_{FL,N} = 83.9\ \mathrm{MPa}$

$$n_{FL} = \frac{\sigma_{FL,N}}{\sigma_a} = \frac{83.9}{68.6} = 1.22$$

Table 4.5.1 Safety factors for steel *[3] (not for GS) and for ductile wrought aluminum alloys (A≥ 12,5 %).

j_D		Consequences of failure	
		severe	moderate ⟡1
regular inspections	no	**1,5**	1,3
	yes⟡2	1,35	1,2

⟡1 Moderate consequences of failure of a less important component in the sense of "non catastrophic" effects of a failure; for example because of a load redistribution towards other members of a statical indeterminate system. Reduction by about 15 %.

⟡2 Regular inspection in the sense of damage monitoring. Reduction by about 10 %.

FKM-Guideline: Analytical Strength Assessment of Components in Mechanical Engineering. 5th revised edition. Frankfurt/Main, Forschungskuratorium Maschinenbau (FKM) 2003.

Questions and problems II

1. What is the difference between the stress concentration factor and the notch factor? Write their relevant formulas.
2. Define the notch sensitivity factor of material and write its formula (as a function of a stress concentration factor and of a notch factor).
3. Is the stress concentration factor of metals a material parameter? And what about the notch factor?
4. Is the fatigue limit of a real part the same as the fatigue limit of a basic material? What other factors could be taken in the account by an expression of such fatigue limit?
5. Which shaft size results in a higher k_s size factor? Shaft with higher or smaller diameter?