Lecture 3

Are local approaches easy to apply in engineering applications?

Some case studies

Influencing factors - fatigue of welded connections

Material

Weld geometry:

Overfill height, weld angle, misalignment

Weld defects:

Cracks

Pores

Weld stresses (= residual stresses)

Tensile stresses (bad!)

Compressive stresses (good!)

Primary factors influencing fatigue strength of welded joints

Material: *Small influence – all steel grades have same S-N curves. But different alloys have different S-N curves (Aluminium, titanium)*

Type of loading: *Tension, bending: Small influence*

Special S-N curves for shear stress

Mean stress: *Small or no influence except for stress relieved structures*

Geometry: *Large scale geometry accounted for by SCF (= Stress Concentration Factor)*

Weld geometry, notches, weld defects, surface condition: *Included in S-N curves*

Primary factors influencing fatigue strength of welded joints (cont'd):

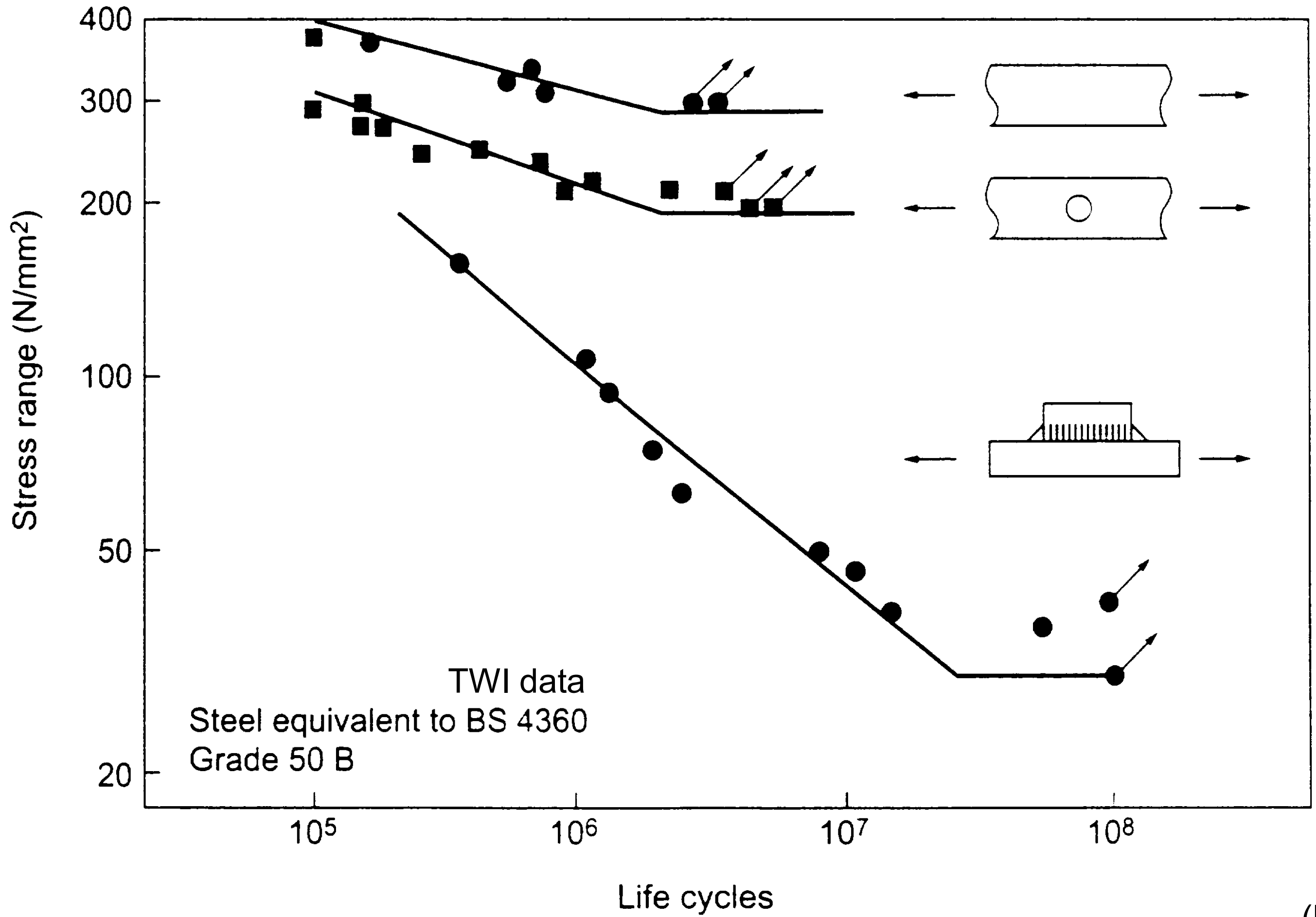
Size: *S-N curves lower when t > 25 mm*

Environment: (corrosion):

- **Temperature:** *No effects below ~200 ºC for steel*
- **Corrosion:** *Strong effects for carbon steels*

Residual stresses: *Included in S-N curves*

The fatigue strength of welded joints is much lower than for non-welded components due to *early crack initiation* and *high tensile welding stresses*

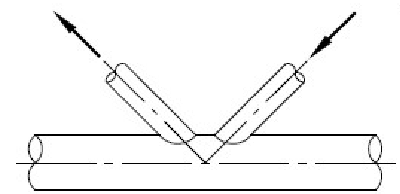


(Maddox, 1991)

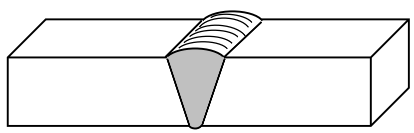
Types of welded connections

Planar connections

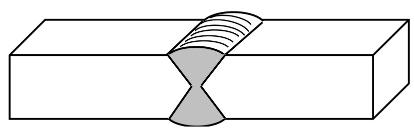
Tubular connections



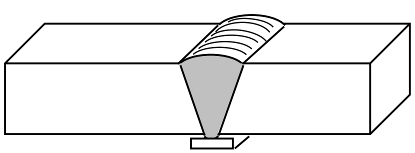
Butt welds



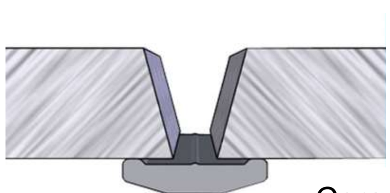
Butt weld made from one side



Butt weld made from two sides

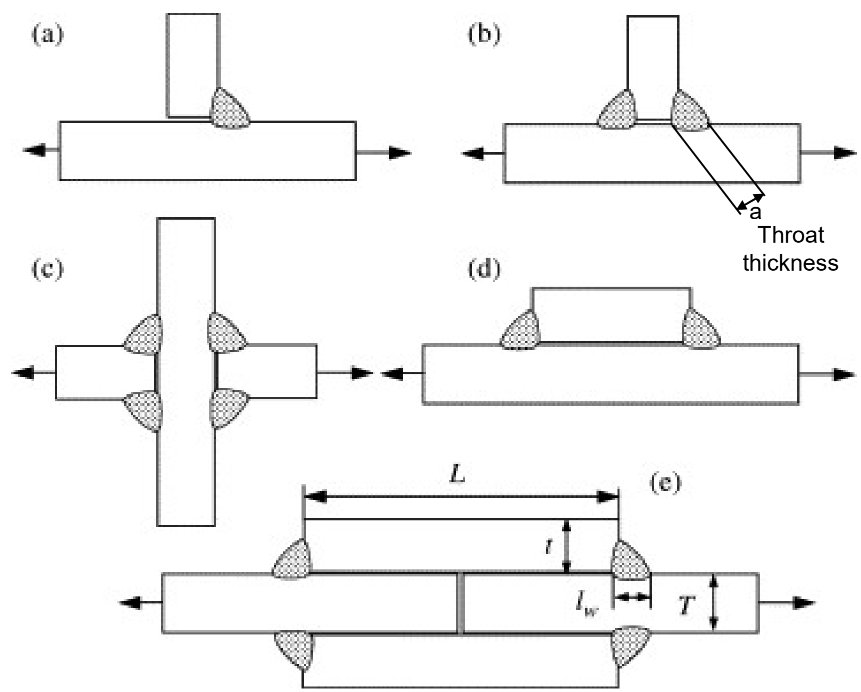


Butt weld made from one side on a backing bar

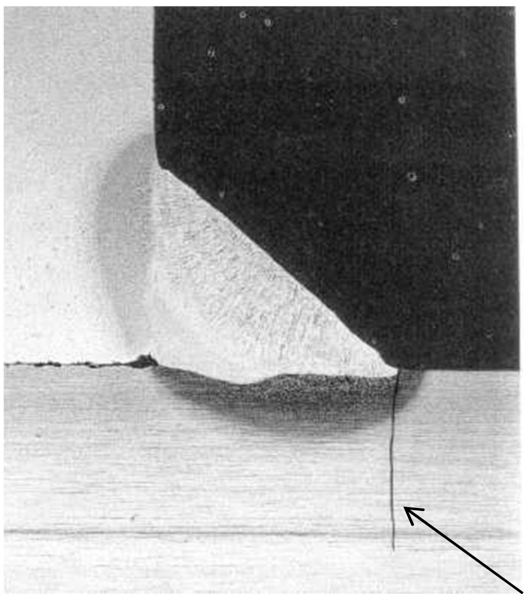


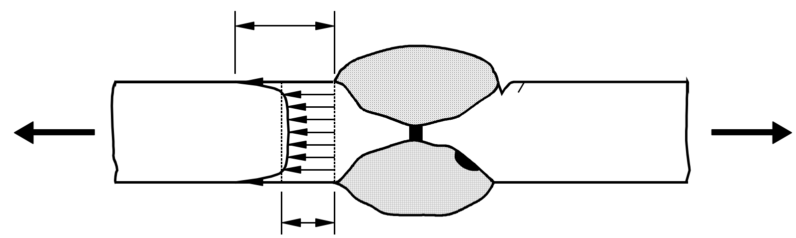
Ceramic backing strip

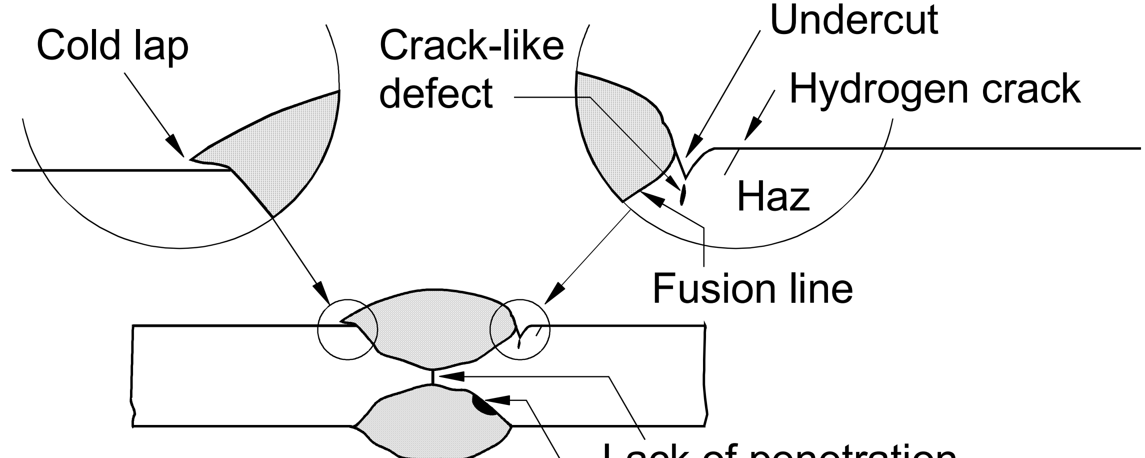
Fillet welds



Welded components have stress raisers and defects

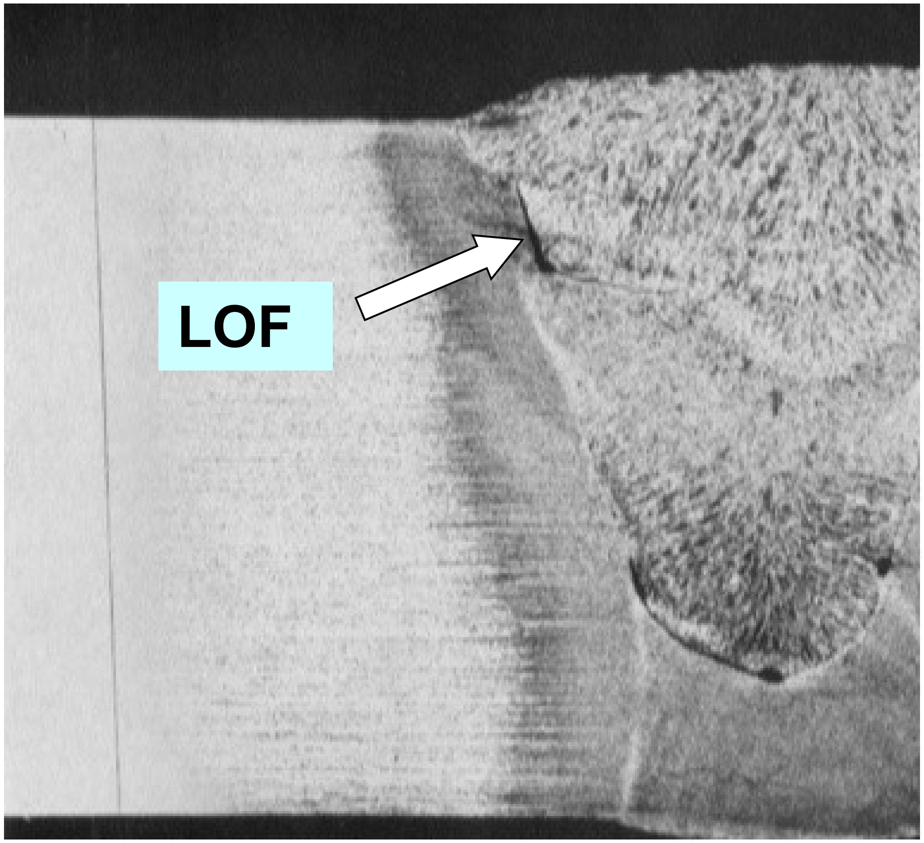






Unacceptable defects

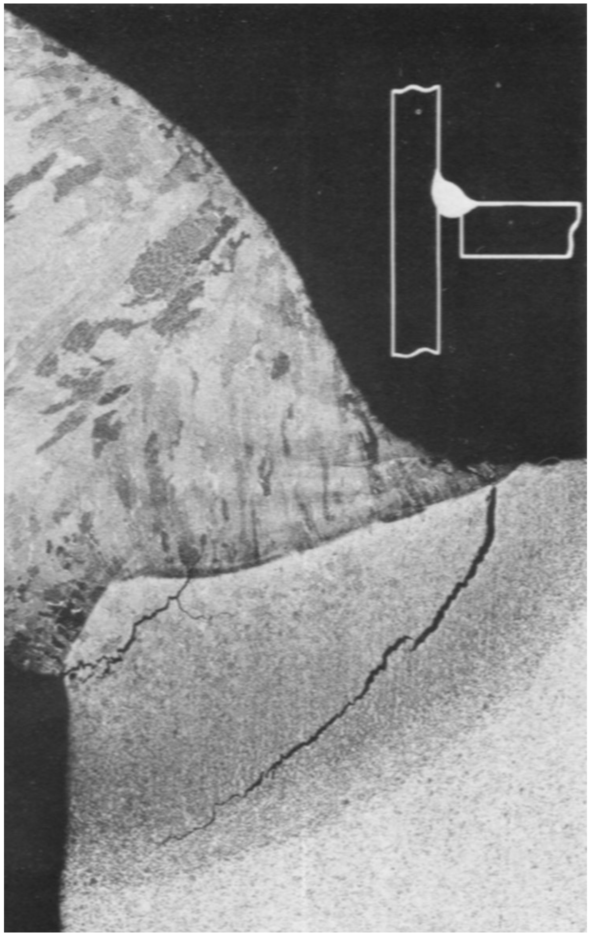
Lack of fusion is an example of defects resulting from incorrect welding conditions



Lack of fusion is caused by e.g. too low current or incorrect torch angle.

Unacceptable defects

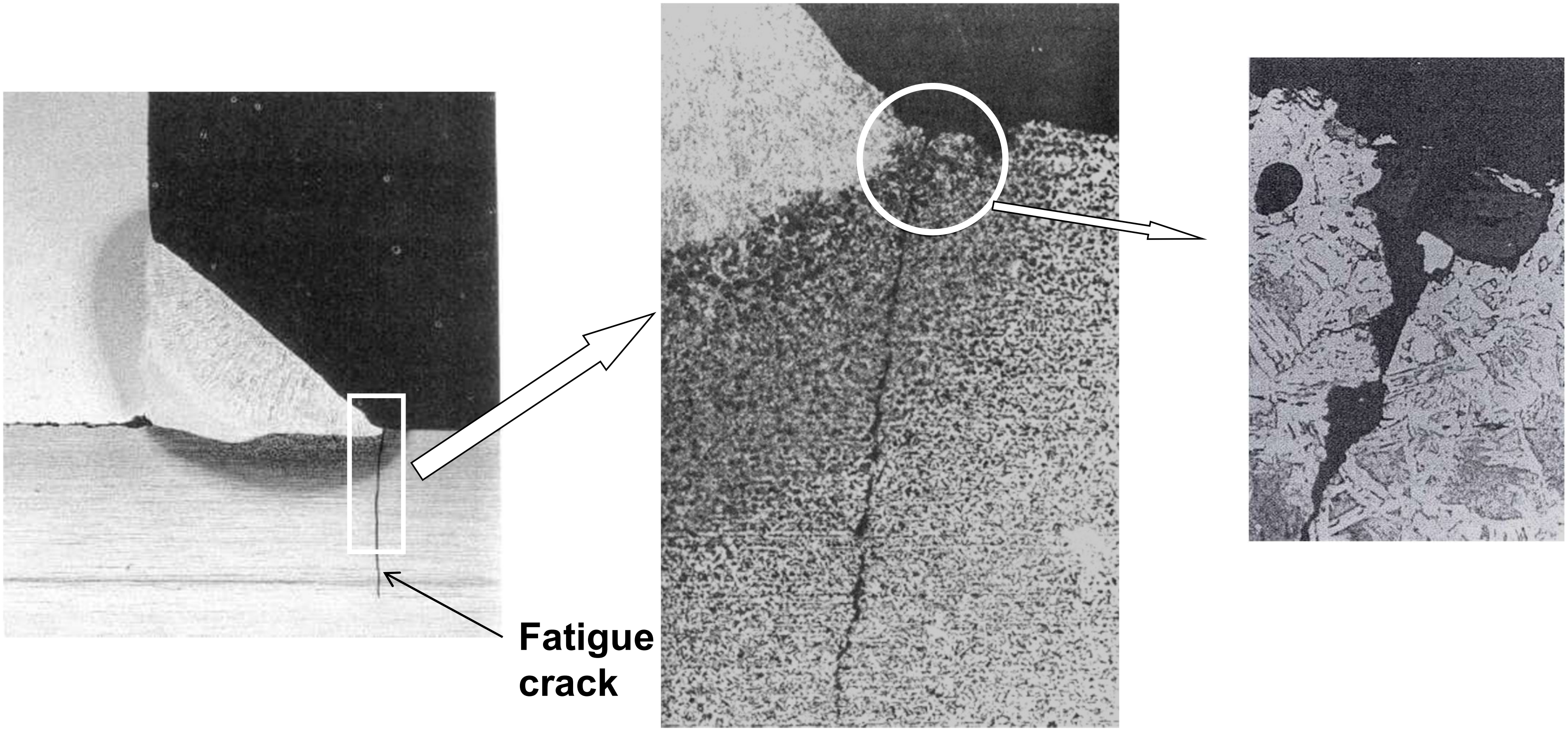
Hydrogen cracks are examples of defects resulting from incorrect welding conditions or bad choice of materials



Hydrogen induced cracking is influenced by factors such as high hardness in heat affected zone (HAZ), high residual stresses, and rapid cooling which does not allow hydrogen to diffuse out

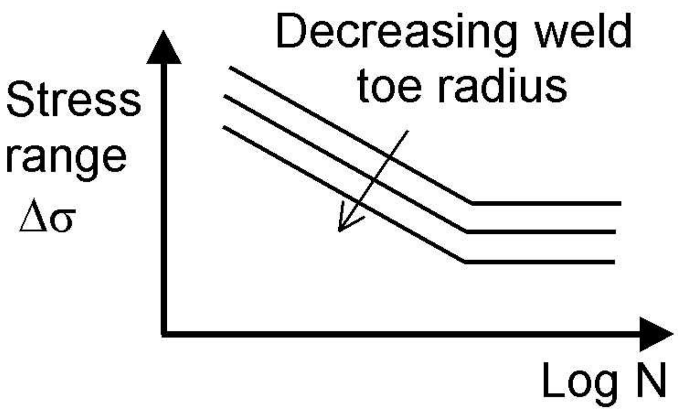
Defects at the weld toe

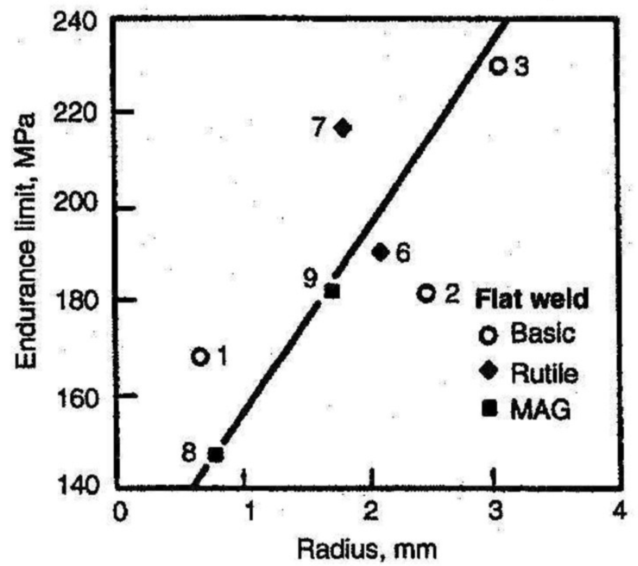
Small defects at the toe are a result of the welding process and cannot be avoided

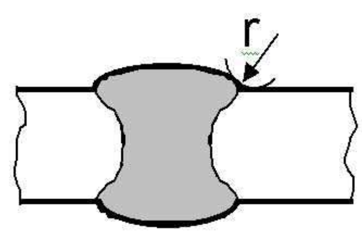


Fatigue strength depends of the local geometry at the weld toe

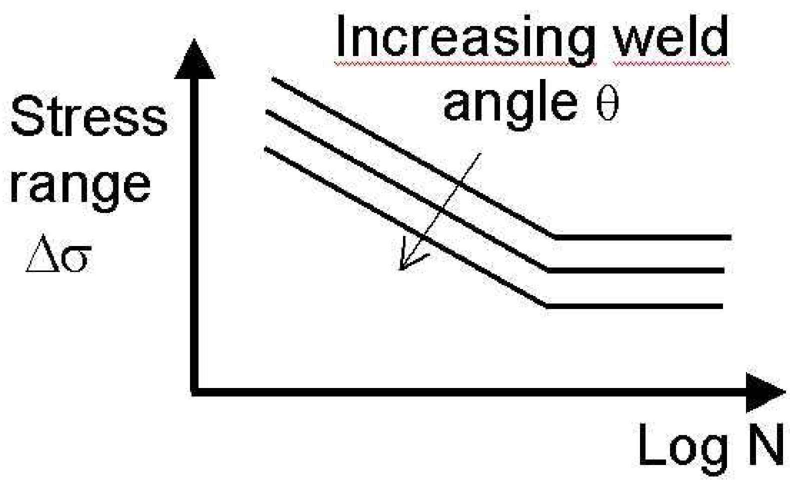
Influence weld toe radius

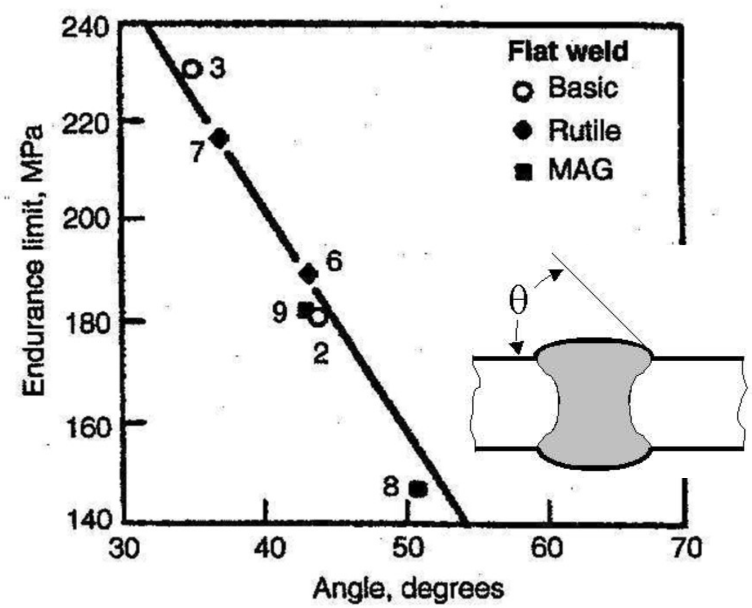




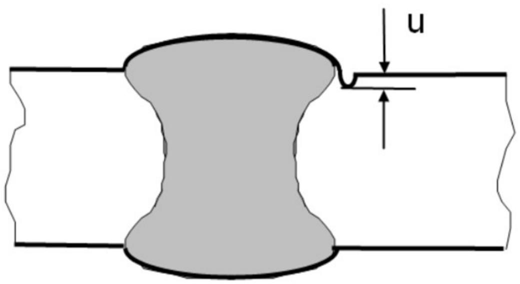


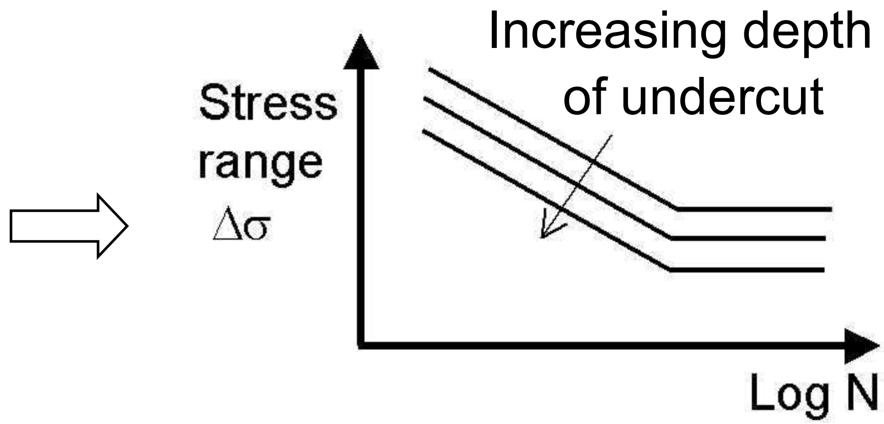
Influence of weld angle θ

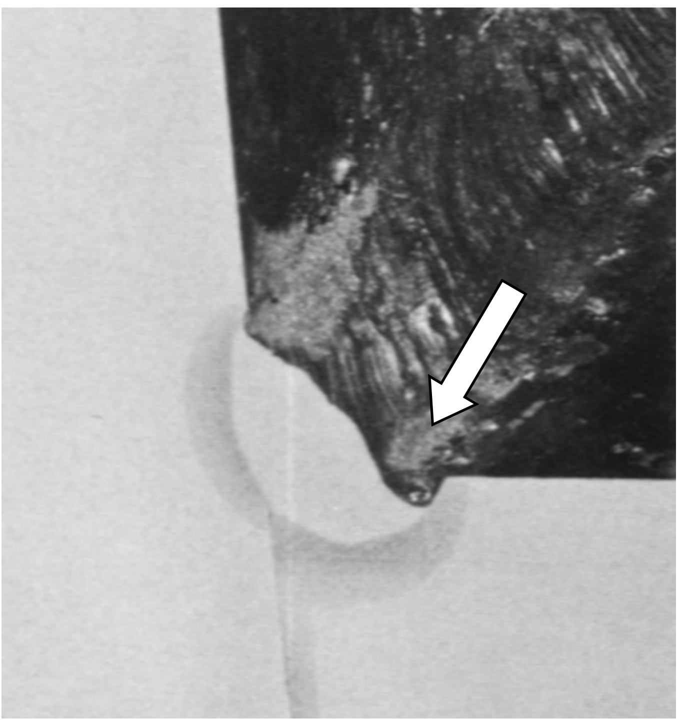




Influence of undercut







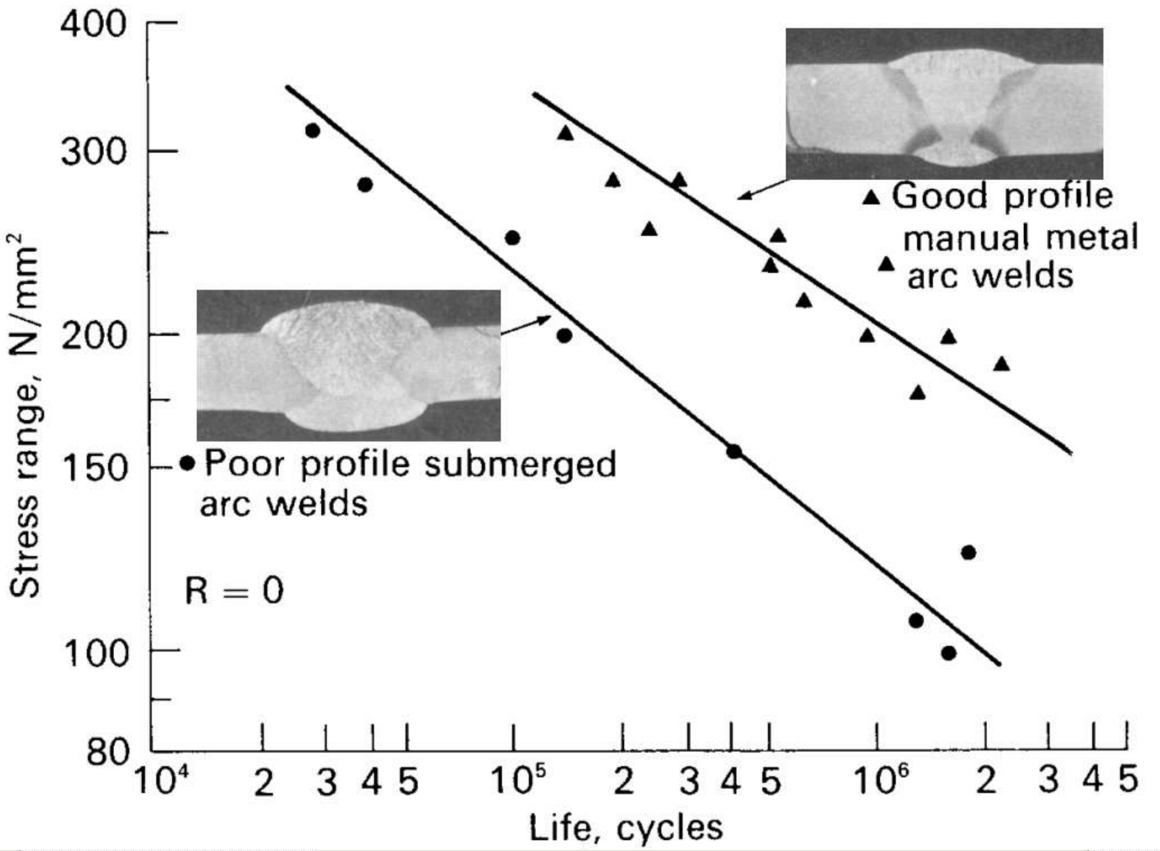
IIW acceptance levels for undercuts

Fatigue class	Allowable undercut u/t	
	butt welds	fillet welds
100	0.025	not applicable
90	0.05	not applicable
80	0.075	0.05
71	0.10	0.075
63	0.10	0.10
56 and lower	0.10	0.10

Notes: a) undercut deeper than 1 mm shall be assessed as a crack-like imperfection.
b) the table is only applicable for plate thicknesses from 10 to 20 mm

Effect of welding process

Different welding processes can give large variations in fatigue strength, depending on the shape on the welds and the defects.

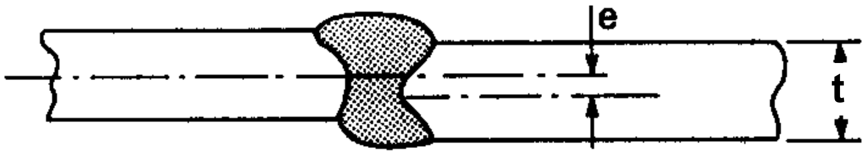


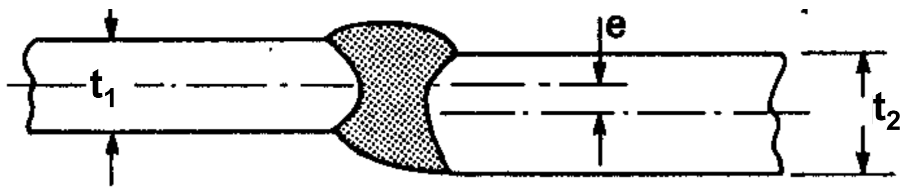
(Maddox, 1991)

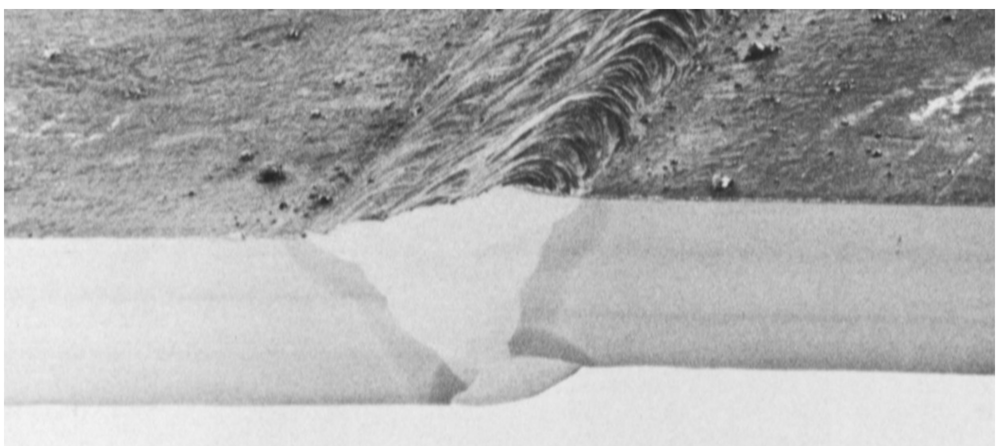
However, if the quality requirements are satisfied, all S-N curves are the same for a given detail, irrespective of welding process

Linear misalignment (eccentricity) or high-low

Misalignment is one of the main causes of the low fatigue strength of welded joints

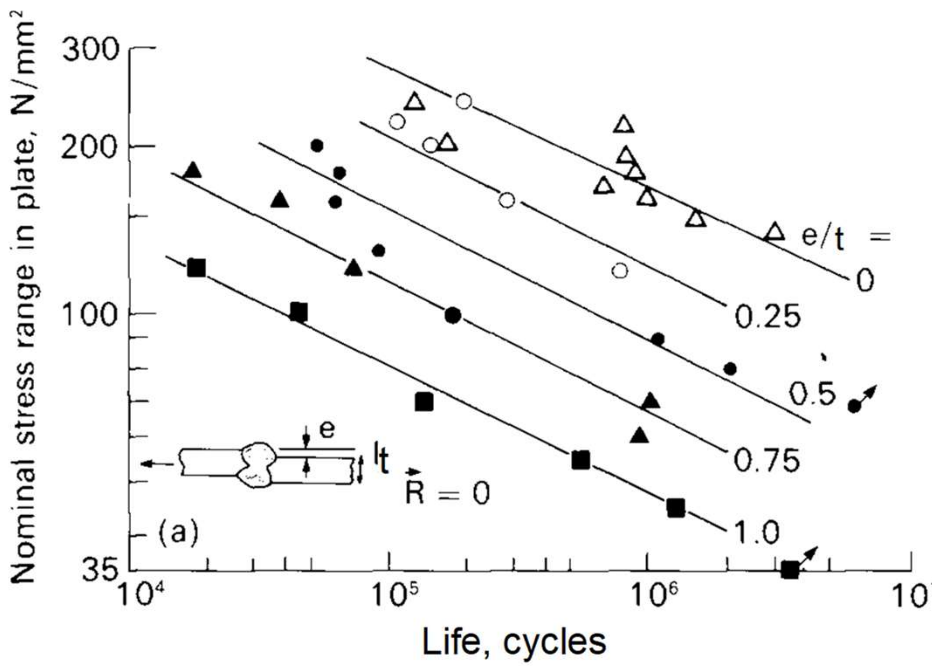


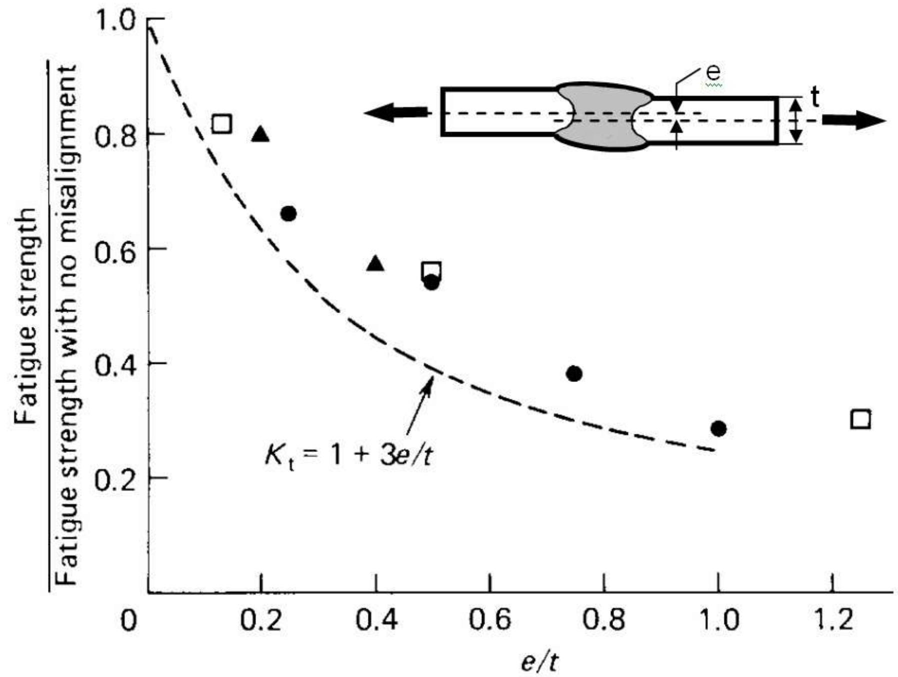




Examples of linear misalignment (eccentricity)

Effect of misalignment on fatigue strength





(Maddox, 1991)

Test data for misaligned joints can be correlated on the basis of the stress concentration factor (SCF) K_t = 1+3e/t caused by linear misalignment.

Compensating for misalignment

Misalignment is unavoidable in normal production welding. It is therefore assumed that the welds on which design S-N curves are based contain some misalignment:

Butt welds: 10% eccentricity (δ_0 /t =0.10)

Fillet welds: 15% eccentricity (δ_0 /t =0.15)

The S-N curves for welds that are inspected should only be downgraded if the eccentricities are higher than these values

Example, plate butt welds:

Increase applied stress by multiplying stress range by SCF:

$$SCF = 1 + \frac{3 + (\delta_m - \delta_0)}{t}$$

where δ_m is misalignment, t is plate thickness and

δ_0 = 0.1 t is misalignment inherent in original test data for butt welds

Equations for other joints are given in DNV RP-C203 and BS 7910

Effect of material on fatigue strength of welded joints

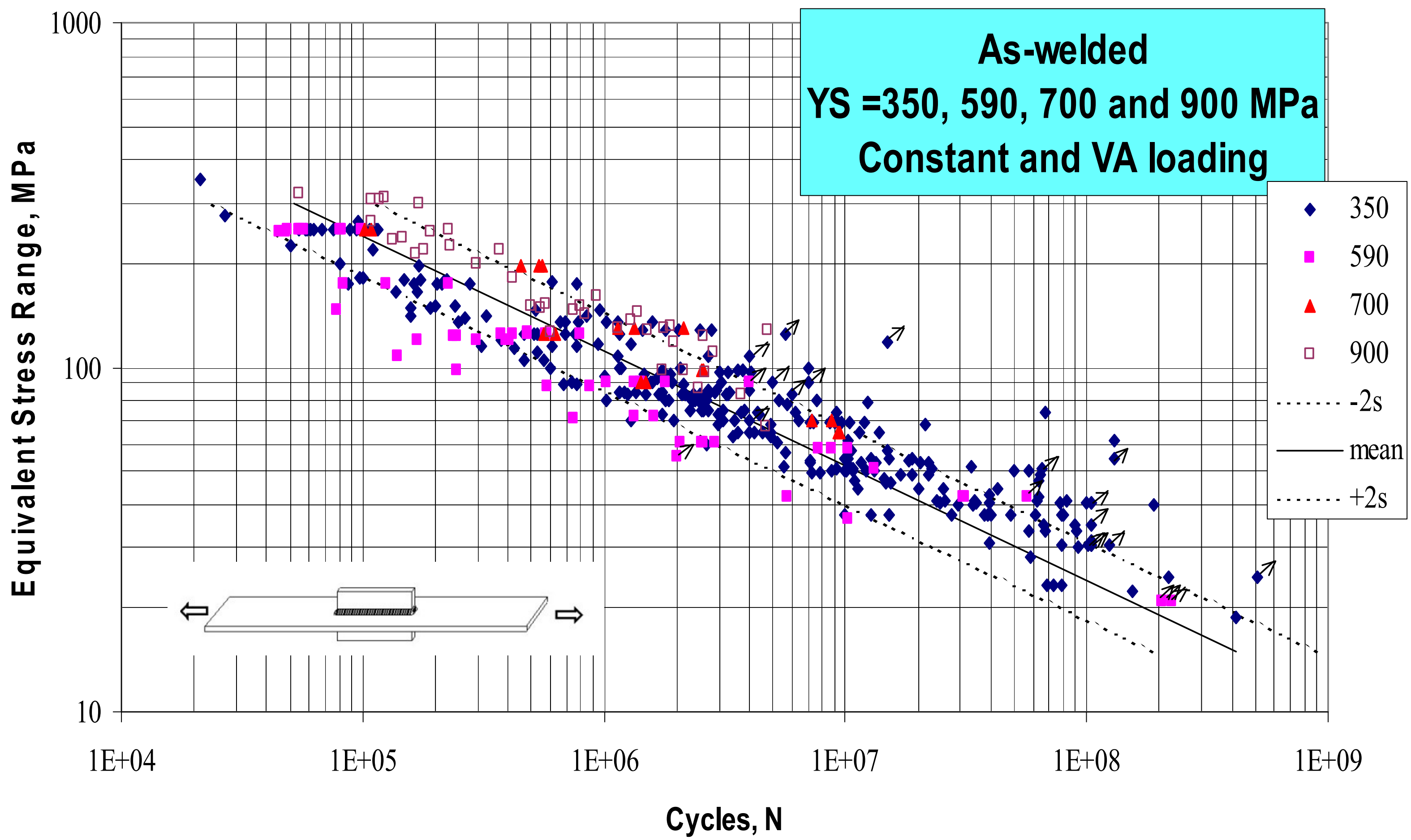
Within each group of alloys, e.g. steels there is almost no effect of base material strength

The reasons:

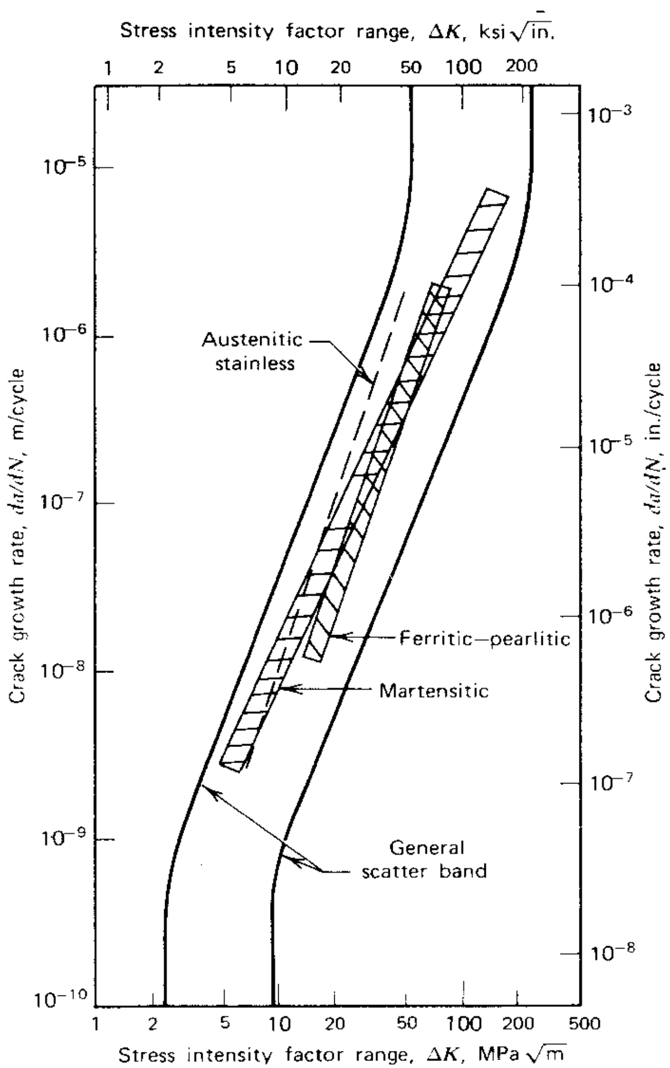
1. Due to high local stresses and defects at the weld toe a crack starts to grow very early
2. A crack grows equally fast in a high strength steel as in lower grades

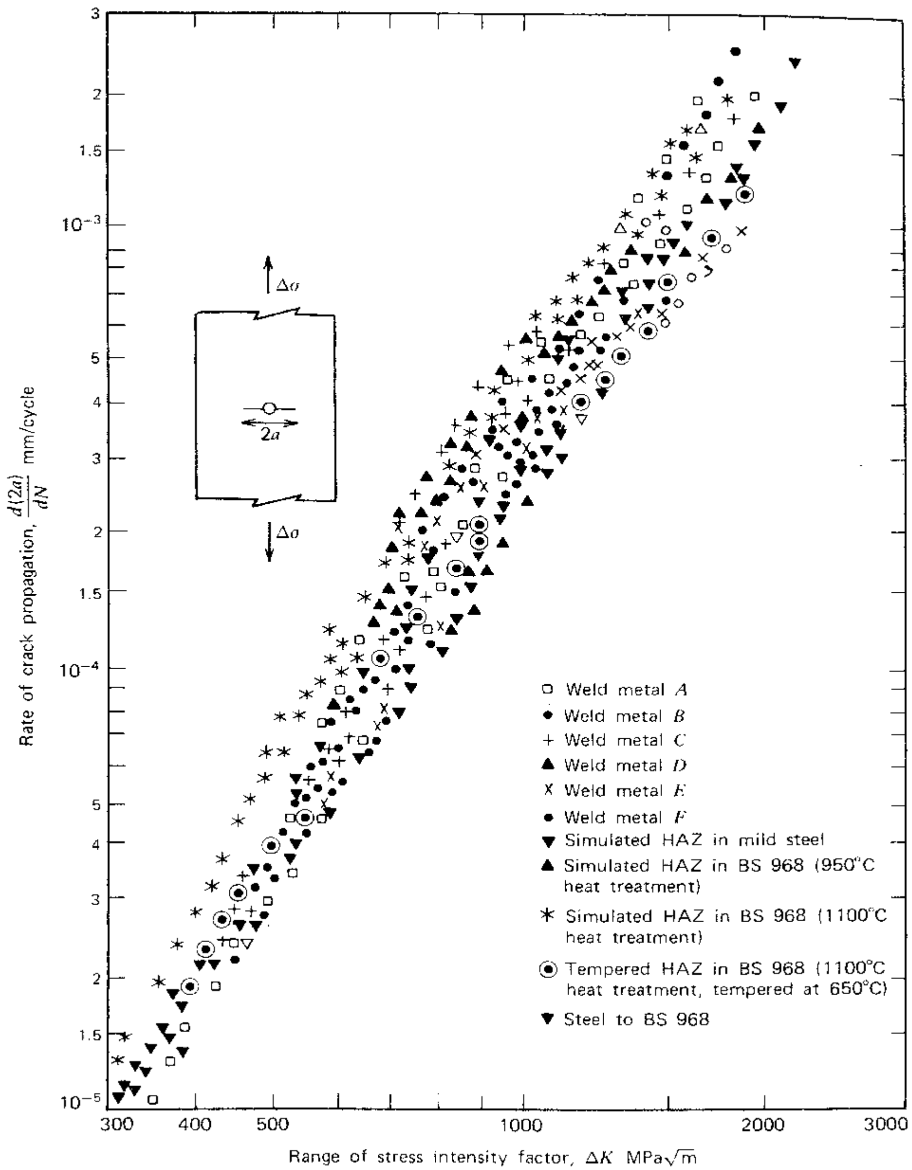
Use of high strength steels – high cycle fatigue:

No advantage – fatigue strength same as for lower strength materials

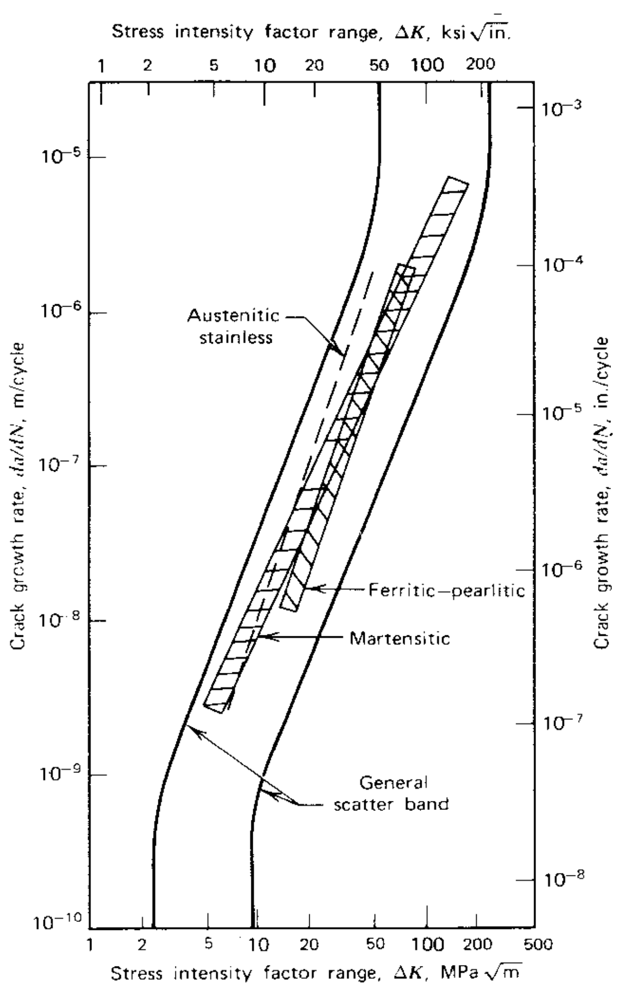


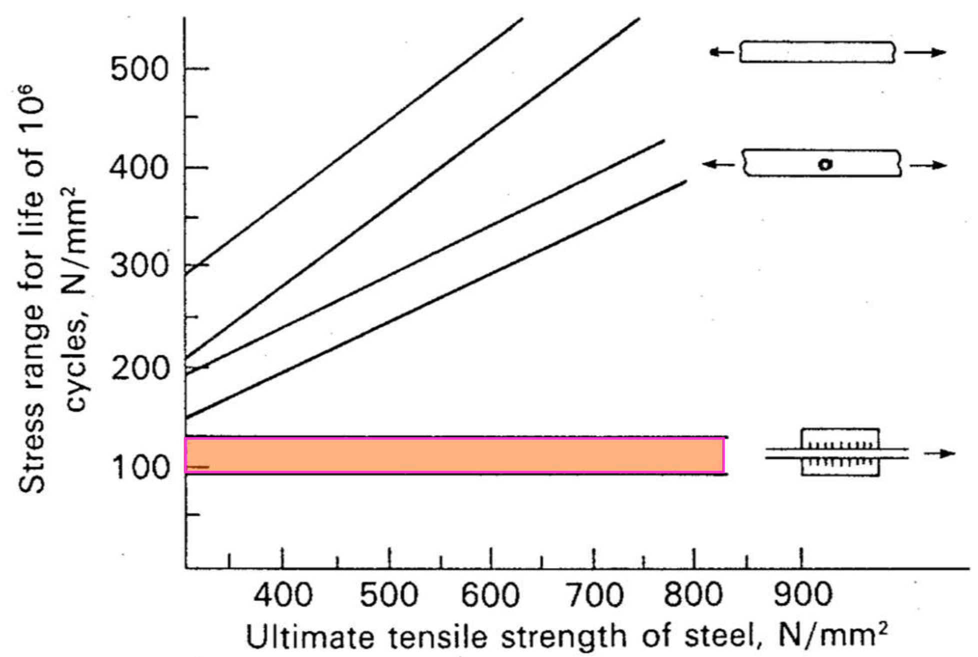
Crack growth speed is almost independent of type of steel



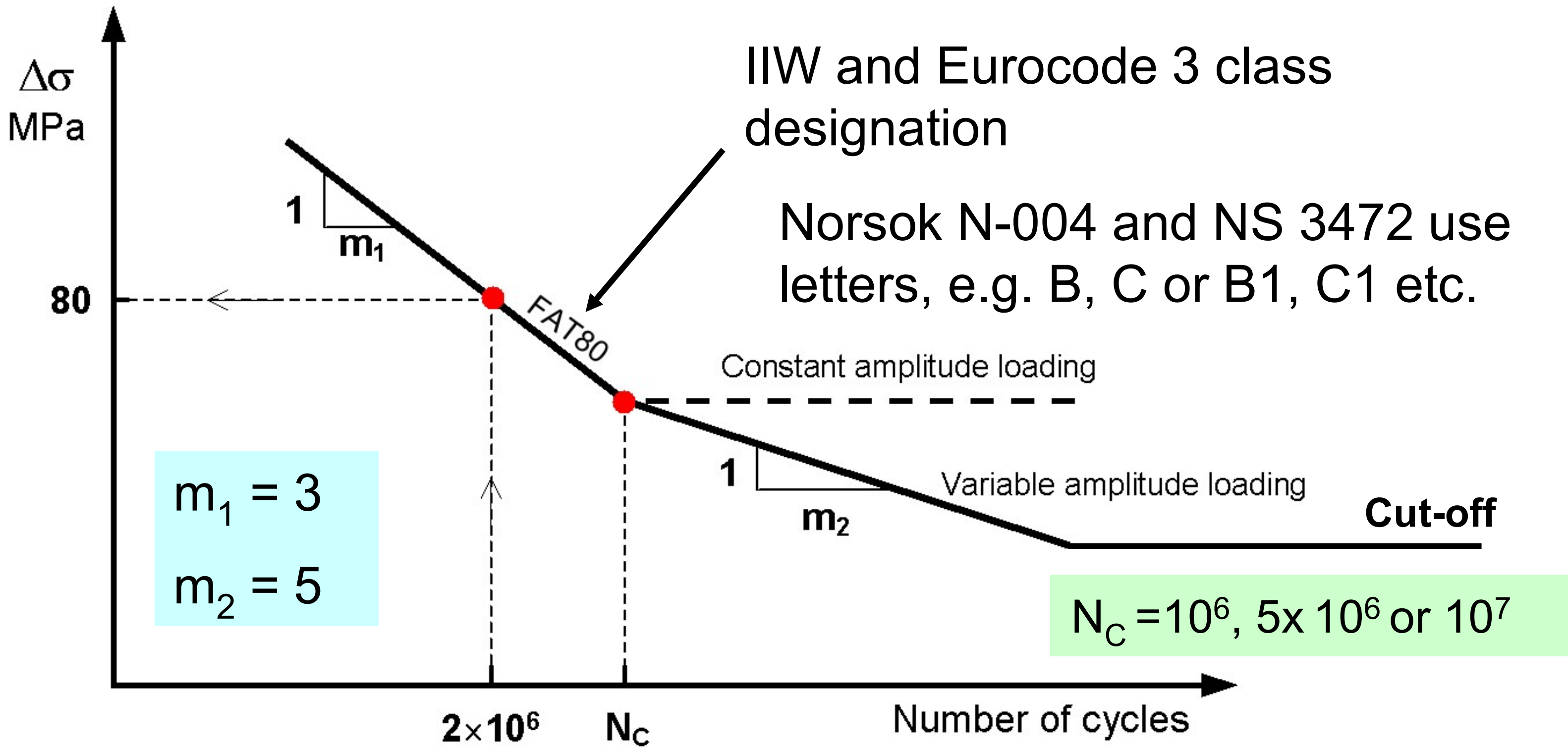


Crack growth speed is almost independent of steel strength therefore the fatigue strength of welded joints is the same for low and high strength steels, in contrast to machined components





Some fatigue terminology,
S-N curves for welded joints in design standards



Basic equations for S-N curves

$$N(\Delta\sigma)^m = C$$

or

$$\log N = \log C - m\log(\Delta\sigma)$$

S-N curves for welded joints

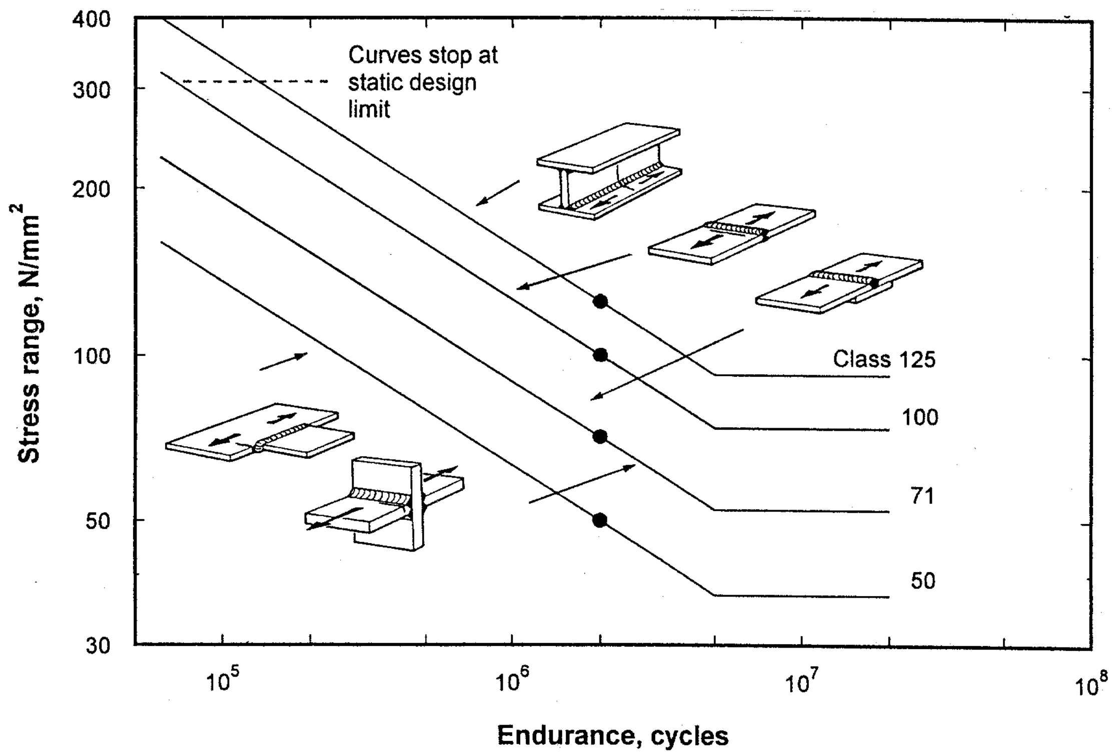
The weld class or category depends on:

Geometry

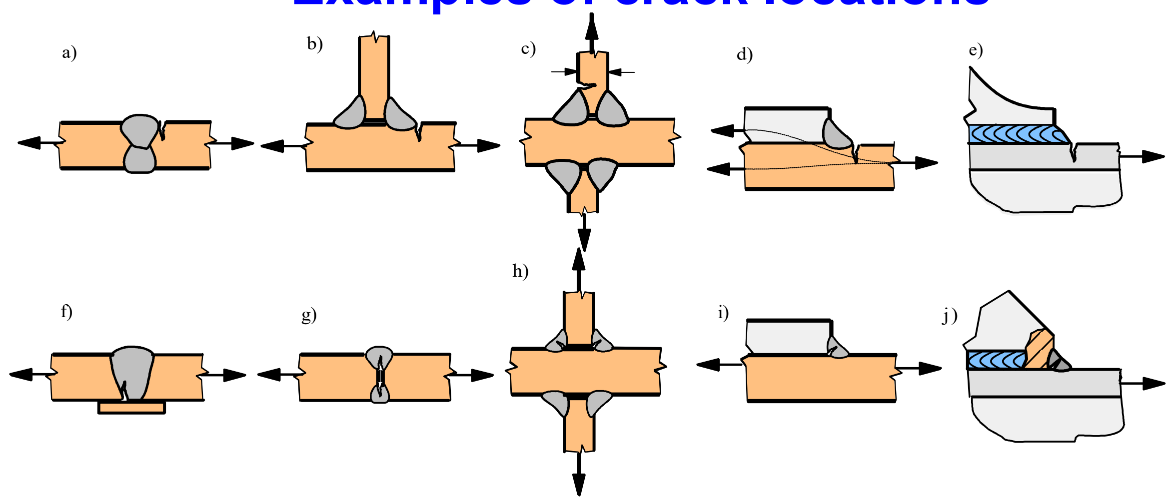
Direction of loading

Crack location

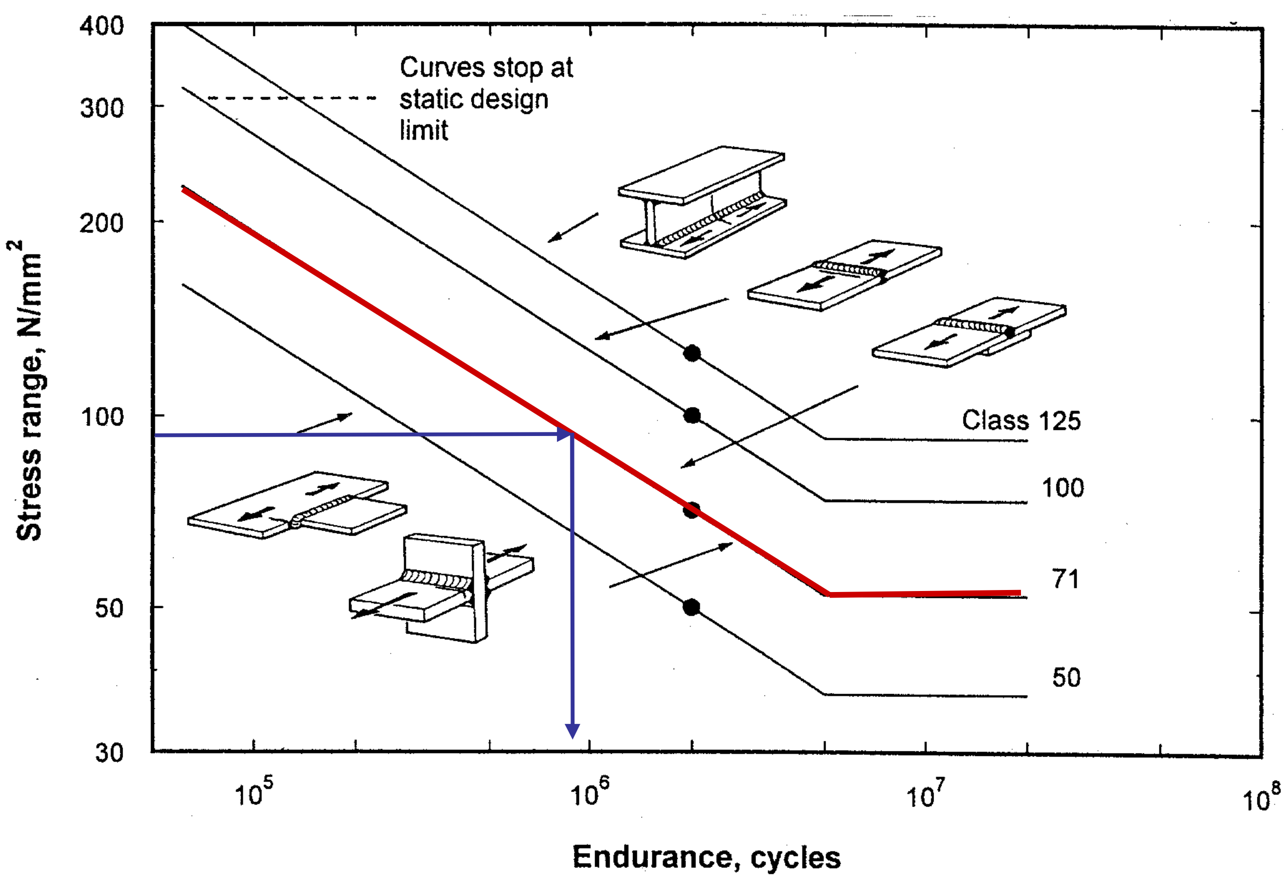
Fabrication & inspection



Examples of crack locations



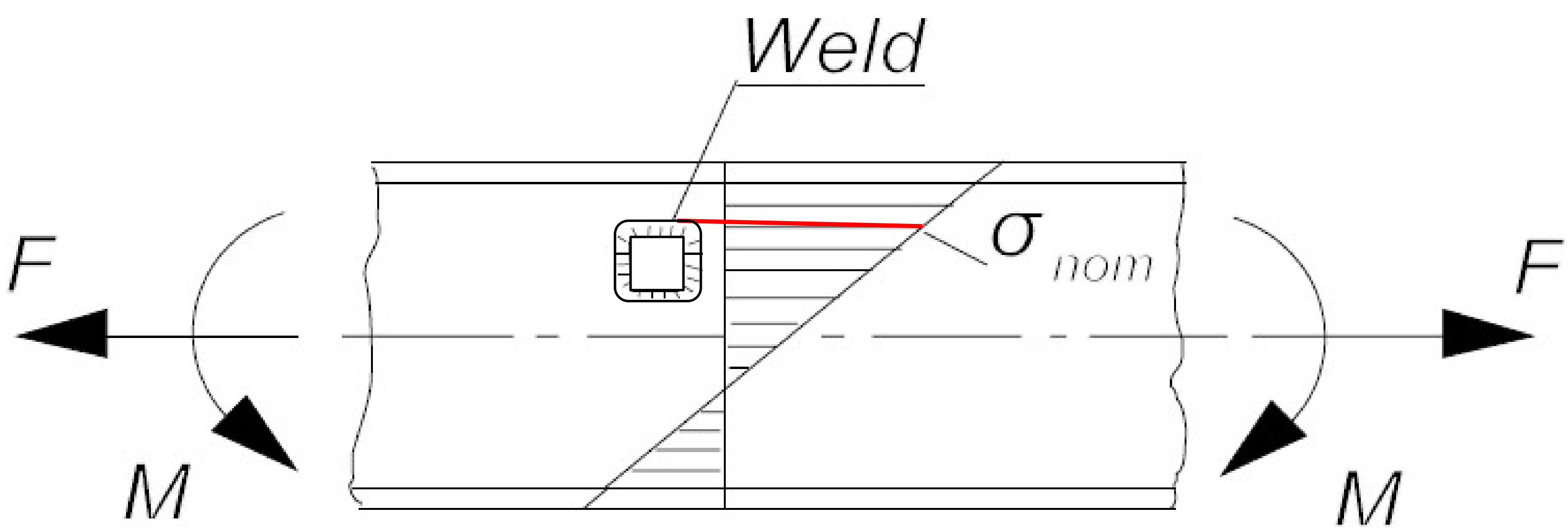
Fatigue life calculation – nominal stress method



1. Choose weld class
2. Calculate nominal stress range
3. Correct stress range for thickness effect and ?
3. Determine cycles to failure from S-N curve
4. Use Miner rule to calculate damage and life

Nominal stress calculations

Nominal stress is the stress calculated in the sectional area under consideration, disregarding the local stress raising effects of the welded joint, but including the stress raising effects of the macro-geometric shape of the component in the vicinity of the joint, such as e.g. large cut-outs. Overall elastic behaviour is assumed.



Calculation of nominal stress

In simple components the nominal stress can be determined using elementary theories of structural mechanics based on linear-elastic behaviour.

In other cases, finite element method (FEM) modelling may be used. This is primarily the case in:

a) complicated statically over-determined (redundant) structures

b) structural components incorporating macro-geometric discontinuities, for which no analytical solutions are available

Using FEM, meshing can be simple and coarse. However, care must be taken to ensure that all stress raising effects of the structural detail of the welded joint are excluded when calculating the modified (local) nominal stress.

Modification of basic S-N curves

The basic S-N curves may need to be modified for the following influencing factors:

Misalignment, axial and angular
Effects of stress relief
Plate thickness, for t > 25 mm
Effect of weld length
Effects of corrosion (special curves)

Temperature
Effects of high and low stresses in the spectrum

Material:* *Different S-N curves for steel, aluminium, titanium

Thickness effects in welded connections:

$$\boxed{S / S_0 = (t / t_0)^k}$$

Exponent k depends on weld class:

0.1< n <0.3 (IIW design guidance)

0 < n <0.25 (0.3 for tubular joints with high SCF's

0.25 for bolts) (DNV-RP-C203)

The hot spot stress method

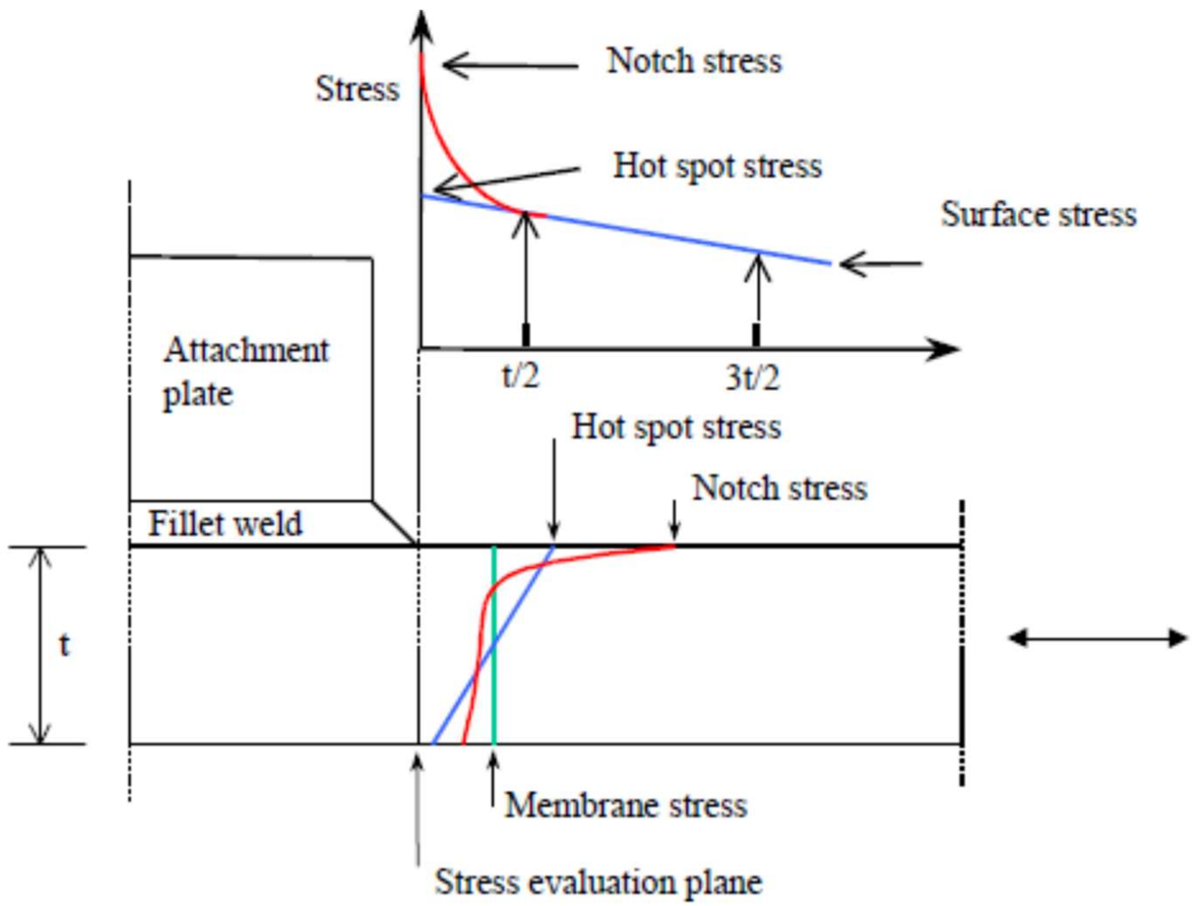
The hot spot stress is a local stress at the weld toe, taking into account the overall geometry of the joint, except the shape of the weld. It is therefore sometimes called the ***structural*** or ***geometrical*** stress.

It is used when it is difficult to define a nominal stress, e.g. in complicated plate structures.

Originally (in the 60's), the stress was measured at a single spot. In the AWS/API at a distance of 1/8" (3.2mm) from the weld toe, while Haibach recommended 2mm.

In recent versions the stress at the weld toe is extrapolated from two or three points near the weld toe. The method is included in DNV's RP-C203, also and IIW (International Institute of Welding)

Definition of the hot spot stress (DNV)



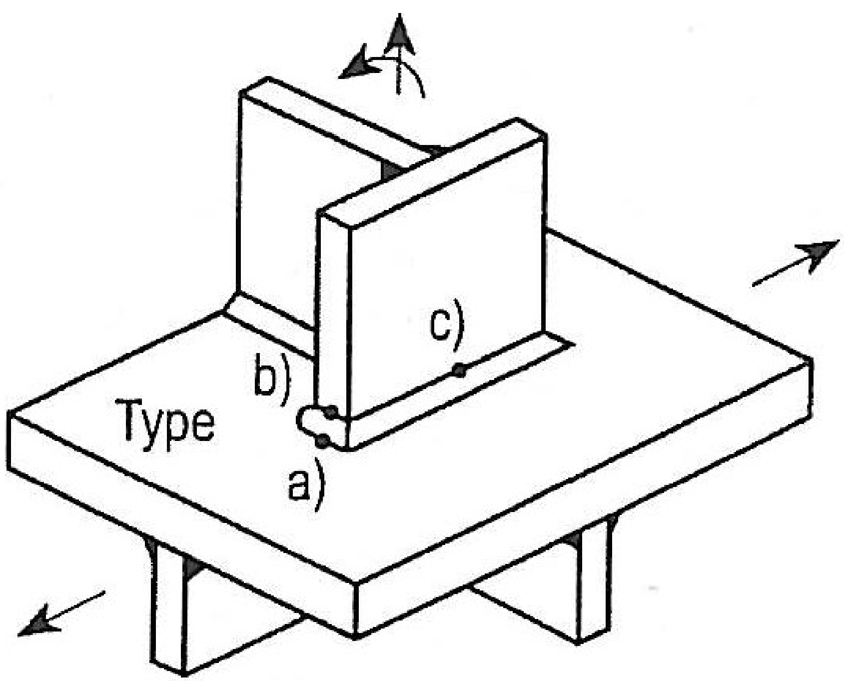
The hot spot stress is a linear extrapolation at distances 0.5t an 1.5t from the weld toe.

In the IIW guidance the to points are at 0.4 and 1.0t. The stress at these two points are obtained from FE analysis or from strain gauge measurements.

Types of hot spot stress

The stresses obtained in FE analyses must include any misalignments or by an appropriate stress concentration factor, SCF.

Two or three types of hot spot stress are usully defined:



a) Weld at end of longitudinal attachment (weld toe or end on loaded plate surface).

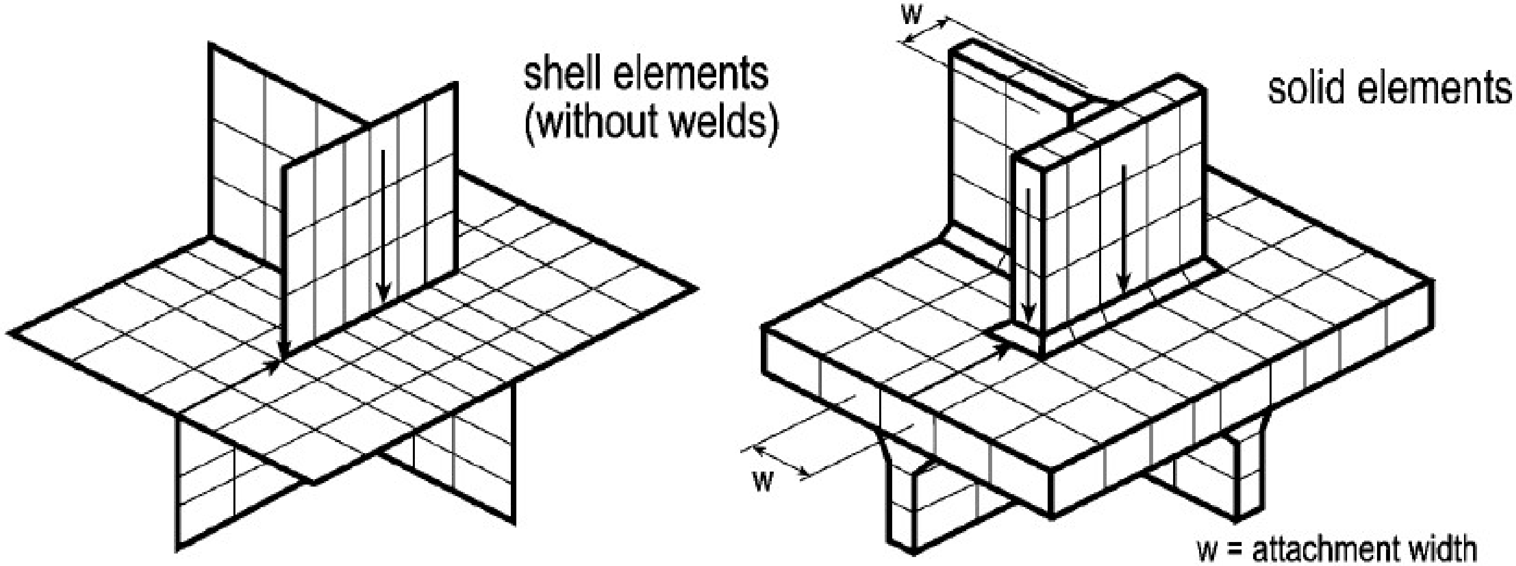
b) Weld on or around a plate edge (weld toe on plate edge).

c) Weld transverse to loading (weld toe on loaded plate surface

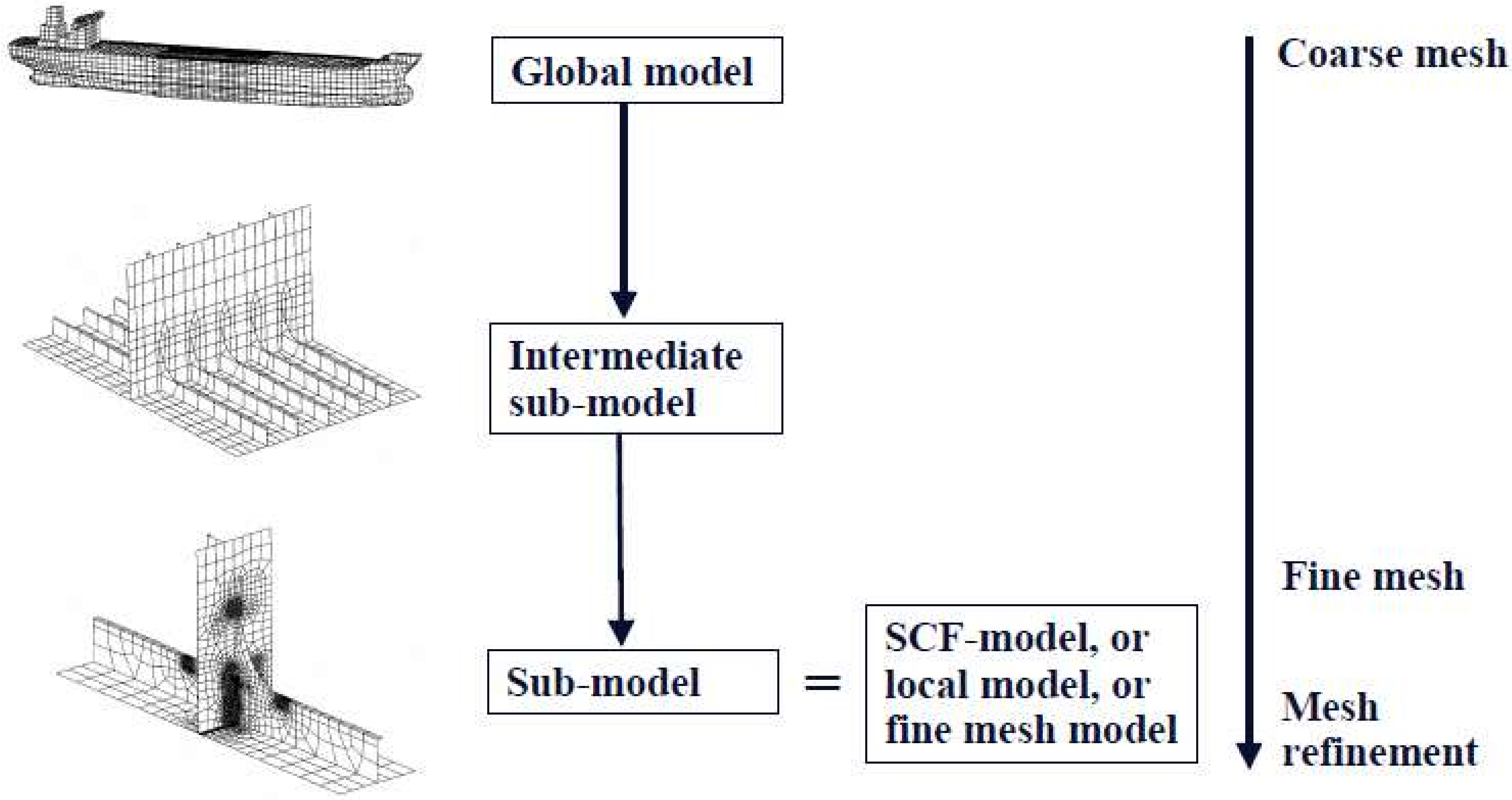
FE modeling - hot spot stress

The stresses obtained in FE analyses must include any misalignments or an appropriate stress concentration factor, SCF.

Shell or solid elements are used in the FE meshing depending on the shape and size of the structure

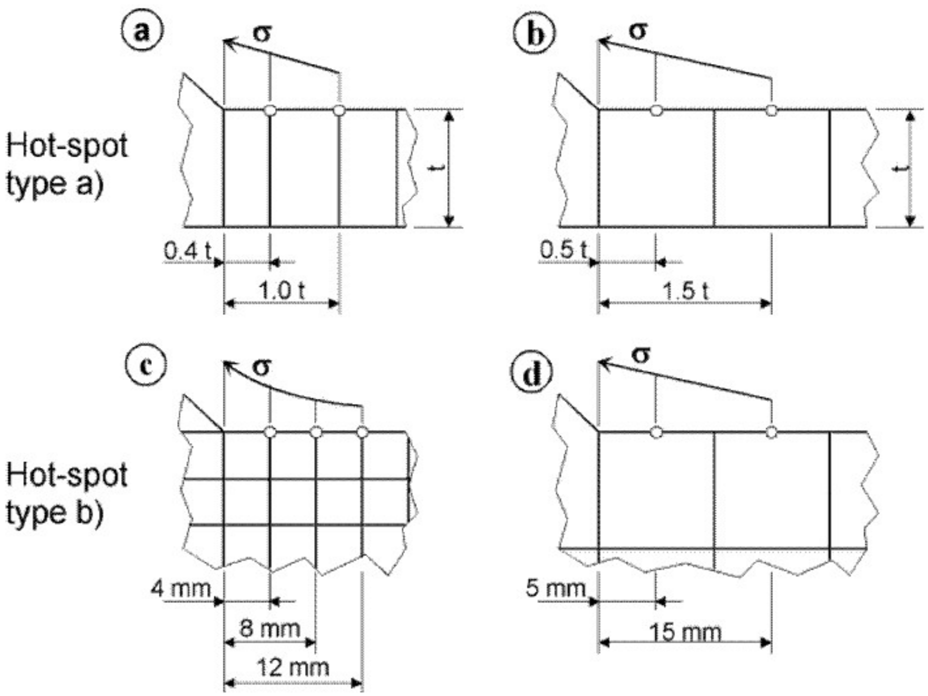


FE stress analysis – ship structure



Meshing rules and determination of hot spot stress

The IIW and DNV fatigue design rules give detailed advice regarding meshing and determination of the hot spot stress



Reference points for different types of meshing

(DNV RP-C-203)

Recommended meshing and extrapolation

Type of model and weld toe		Relatively coase models		Relatively fine models	
		Type a	Type b	Type a	Type b
Element size	Shells	t x t max t x w/2*)	10 x 10 mm	≤0.4 t x t or ≤0.4 t x w/2	≤ 4 x 4 mm
	Solids	t x t max t x w	10 x 10 mm	≤0.4 t x t or ≤0.4 t x w/2	≤ 4 x 4 mm
Extra-polation points	Shells	0.5 t and 1.5 t mid-side points**)	5 and 15 mm mid-side points	0.4 t and 1.0 t nodal points	4, 8 and 12 mm nodal points
	Solids	0.5 and 1.5 t surface center	5 and 15 mm surface center	0.4 t and 1.0 t nodal points	4, 8 and 12 mm nodal points

*) w = longitudinal attachment thickness + 2 weld leg lengths
**) surface center at transverse welds, if the weld below the plate is not modelled (see left part of fig. 2.2-11)

(IIW Recommendations)

At the extrapolation procedures for structural hot spot stress of type **"b", a wall thickness** correction exponent of **n=0.1 shall be applied.**

Calculation of hot spot stress

Since the stresses obtained in FE analyses depend strongly on the type of element and the mesh that are used, detailed guidance is given in the design rules. The degree of bending influences life.

The DNV RP C-203 correction:

$$\Delta\sigma_{\text{e,hot spot}} = \Delta\sigma_{\text{a,hot spot}} + 0.60\Delta\sigma_{\text{b,hot spot}}$$

where

$$\Delta\sigma_{\text{a,hot spot}} = \text{membrane stress}$$
$$\Delta\sigma_{\text{b,hot spot}} = \text{bending stress}$$

A single hot spot S-N curve is used by DNV (in air). This is the T-curve = the D-curve = the FAT 90 curve. This is the S-N curve for a "good" butt weld, welded from both sides.

In IIW the FAT 90 curve is used for *load carrying welds* and FAT 100 for *non-load carrying welds.*

Lecture 3

Local approaches

Some case studies

Fictitious notch rounding approach

Fictitious notch rounding concept for welded joints

Fictitious notch rounding simulating stress averaging over ρ^* in the direction of crack propagation has successfully been applied to the fatigue assessment of welded joints (Radaj 1969, 1975, 1990).

Within a worst case consideration, the parameter values:

- **ρ = 0 (worst case), ρ^* ≈ 0.4 mm (welded steel), s ≈ 2.5**

result in the fictitious notch radius:

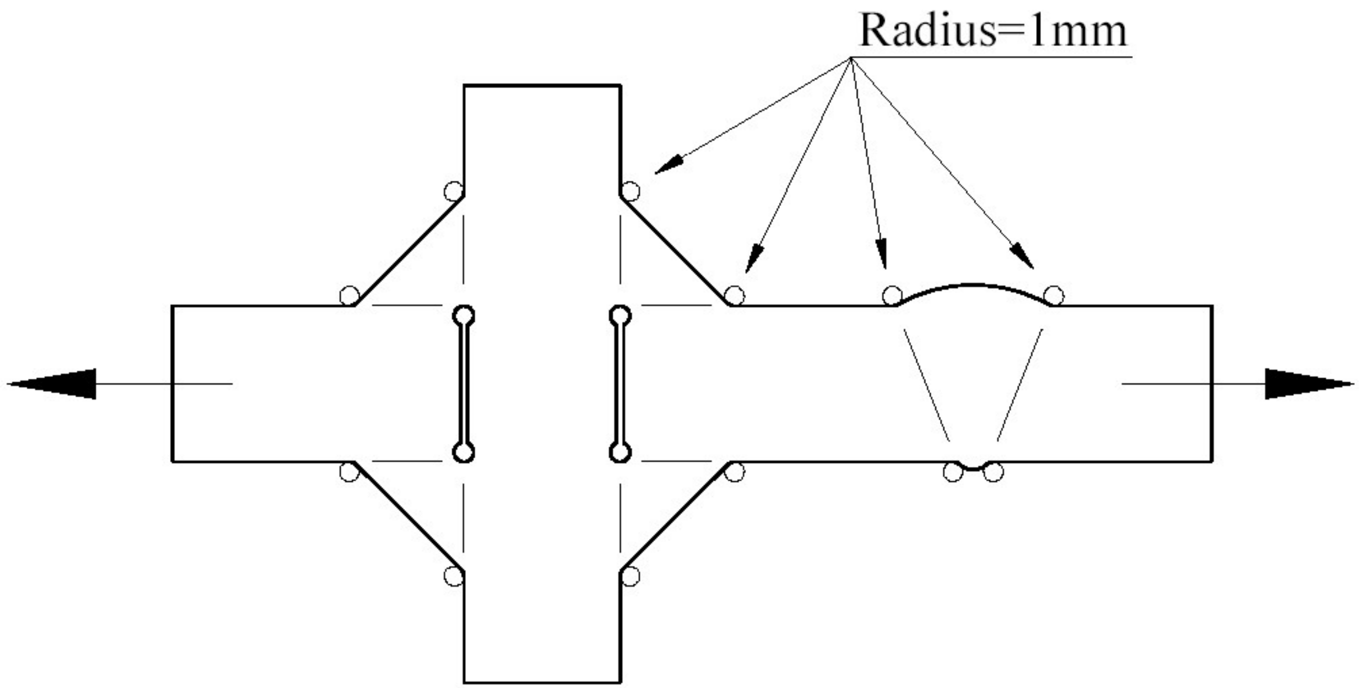
- **$\rho_f = \rho + s\rho^* = 1$ mm**

This very rough estimate is applied to the cross-sectional model of welded joints in the form of a blunt circular notch at the weld toe and a keyhole at the weld root.

The SCFs at these notches are considered as theoretical fatigue notch factors characterising the endurance limit of the joints.

Effective notch stress method

An effective notch radius of 1 mm is assumed in the FE analysis



Main advantages:

- **Only one S-N curve is required, the FAT 225 curve.**
- **Can be used to assess fatigue life for root cracks**

Effective notch stress method

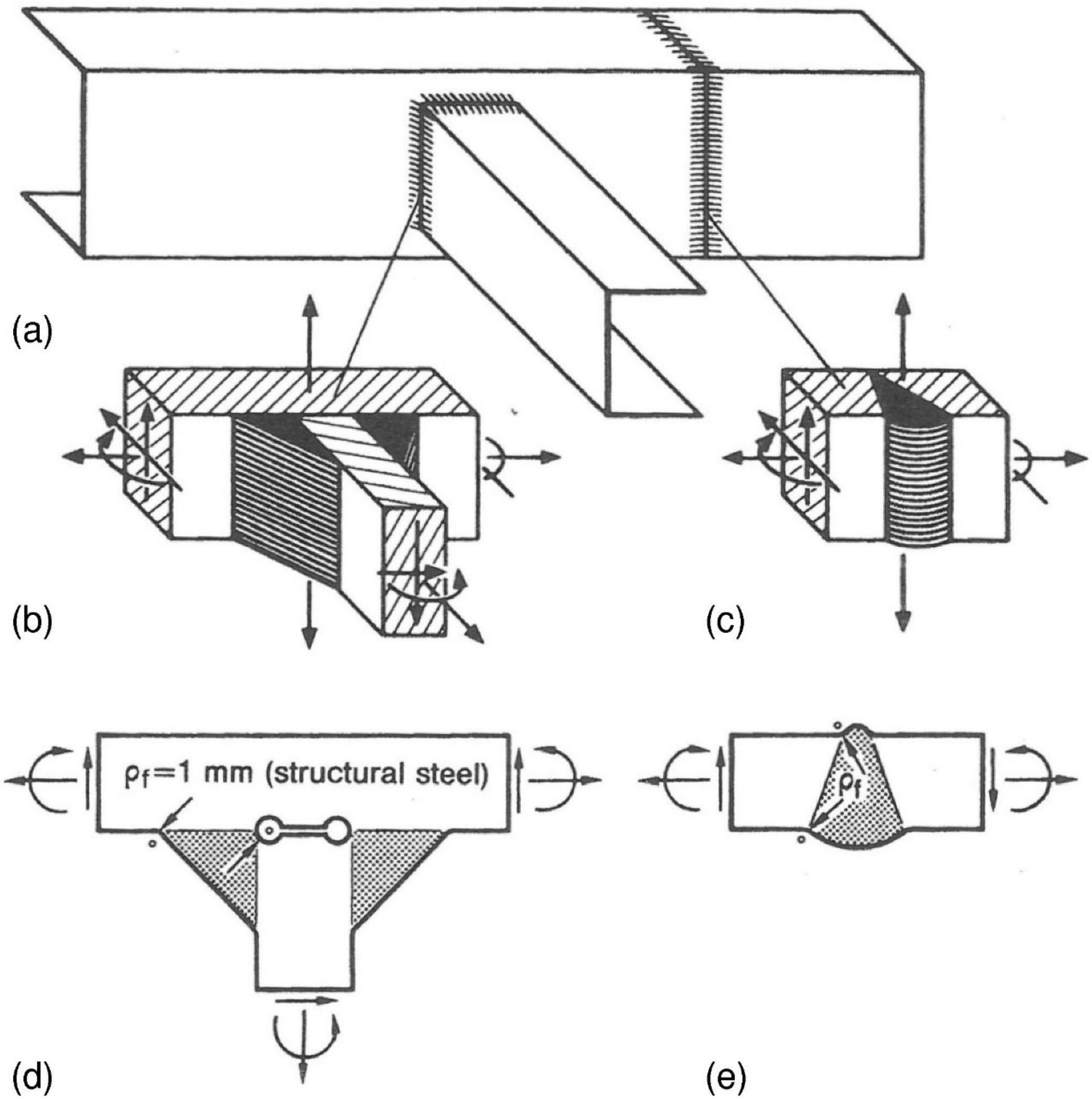
The effective notch stress is the total stress at the root of a notch, obtained assuming linear-elastic material behaviour. For structural steels an effective notch root radius of r = 1 mm in the FE analysis gives consistent results. For fatigue assessment, the effective notch stress is compared with a common fatigue resistance curve.)The method is valid for plate thickness **t> 5 mm**

The FAT 225 (m=3) S-N curve is to be used in this method. For t < 5 mm a radius o

The method is included in DNV's revised RP-C203, April 2010

For t < 5 mm a radius of 0.05 has been proposed (Sonsino 2002) with an S-N curve with FAT 630

Fictitious notch rounding concept for welded joints



Fictitious notch rounding concept for welded joints

It is usual to determine the SCFs of the cross-sectional model by FE or BE analysis with the fictitious notches introduced as real notches within the finite boundaries of the model, thus generating effects which are absent in the infinite-plane notch stress theory.

Most important is the effect of cross-sectional weakening caused by the real fictitious notch. Counter measures are:

- Blunt notches without weakening effect (at weld toe)
- Notch stress reduction guided by structural stress increase
- Micronotches with notch stress reduction according to t/ρ ratio
- Micronotches with notch stress averaging over ρ^*

Another disturbing effect originates from slit-parallel loading:
No stress increase at the ideal slit, but SCF $K_t \approx 3.0$ at the keyhole.

Reference notch concept for welded joints

The deviations from Neuber's concept of fictitious notch rounding, especially real fictitious notches and extension into the medium-cycle fatigue range, suggested a special name:

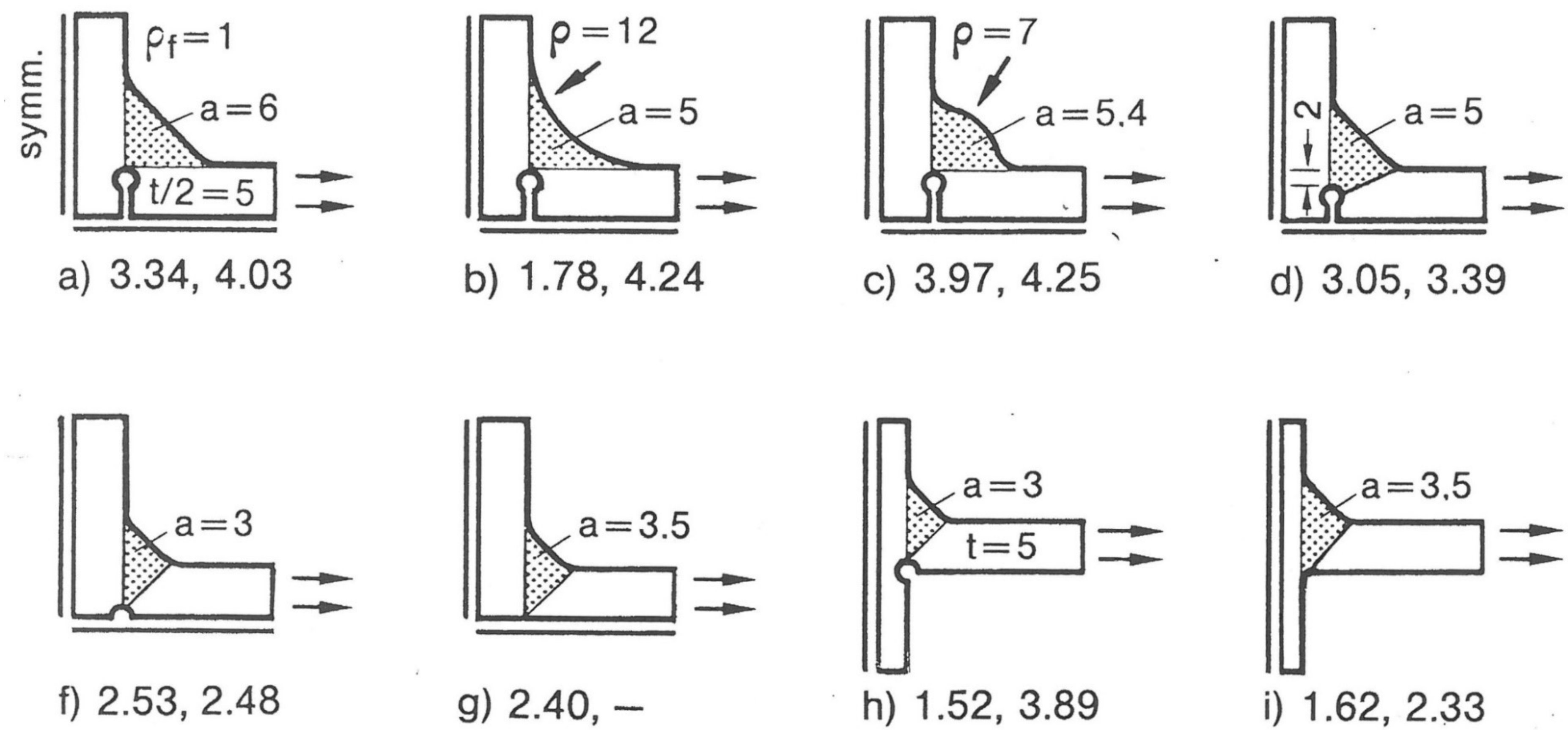
- Reference notch concept

The following concept versions may be distinguished:

- Original version (Radaj 1968) related to high-cycle endurance limit; application to design comparisons
- Modified version (Olivier et al. 1989) with mean values and scatter ranges of high-cycle endurable notch stresses for reference notch radius $\rho_r = 1$ mm
- IIW version (Hobbacher 2009) with extension into the medium-cycle fatigue range
- Pedersen's diagram (Pedersen 2011) with FAT 200 design curve
- Microhole version for thin-sheet lap joints

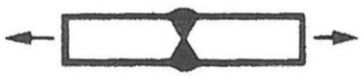
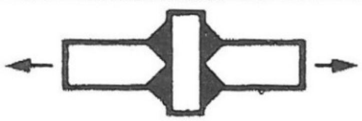
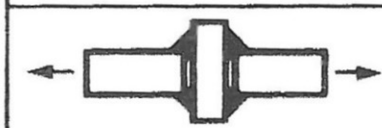
Reference notch concept – design comparisons

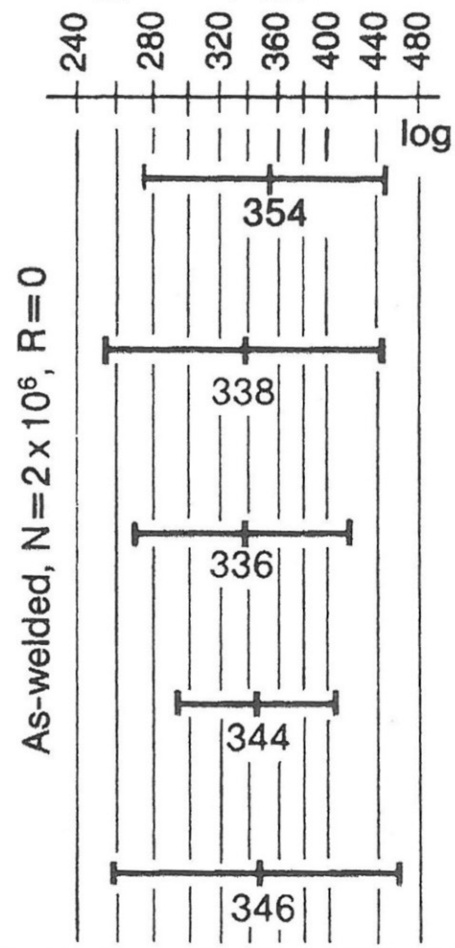
(Radaj 1968, 1975, 1990 etc.)



Reference notch concept – statistical data

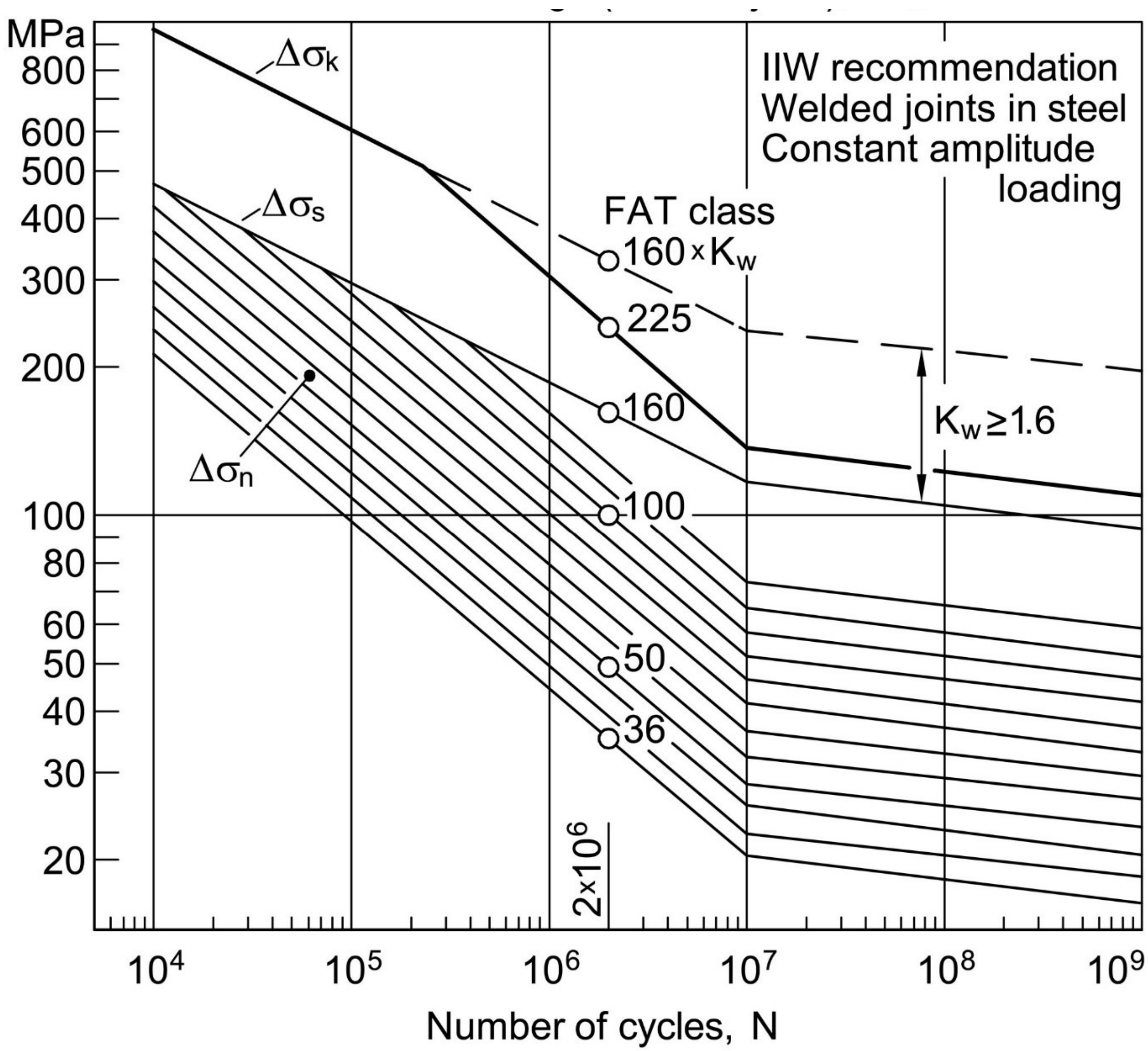
(Olivier et al. 1989, 1994)

Welded joint type (structural steels)	Fatigue notch factor K_f / Fracture initiation	Global fatigue strength (R=0) σ_{nA} [N/mm²], P_f= 10 %	50 %	90 %	Local fatigue strength $\Delta\bar{\sigma}_{kE} = 2K_f\sigma_{nA}$ [N/mm²]
 Butt joint	2.27 Weld toe	61	78	99	354
 Transverse stiffener	2.45 Weld toe	52	69	91	338
 Cruciform joint	2.50 Weld toe	54	67	83	336
 Overlap joint	3.12 Weld toe	47	55	65	344
 Cruciform joint	4.03 Weld root	32	43	57	346



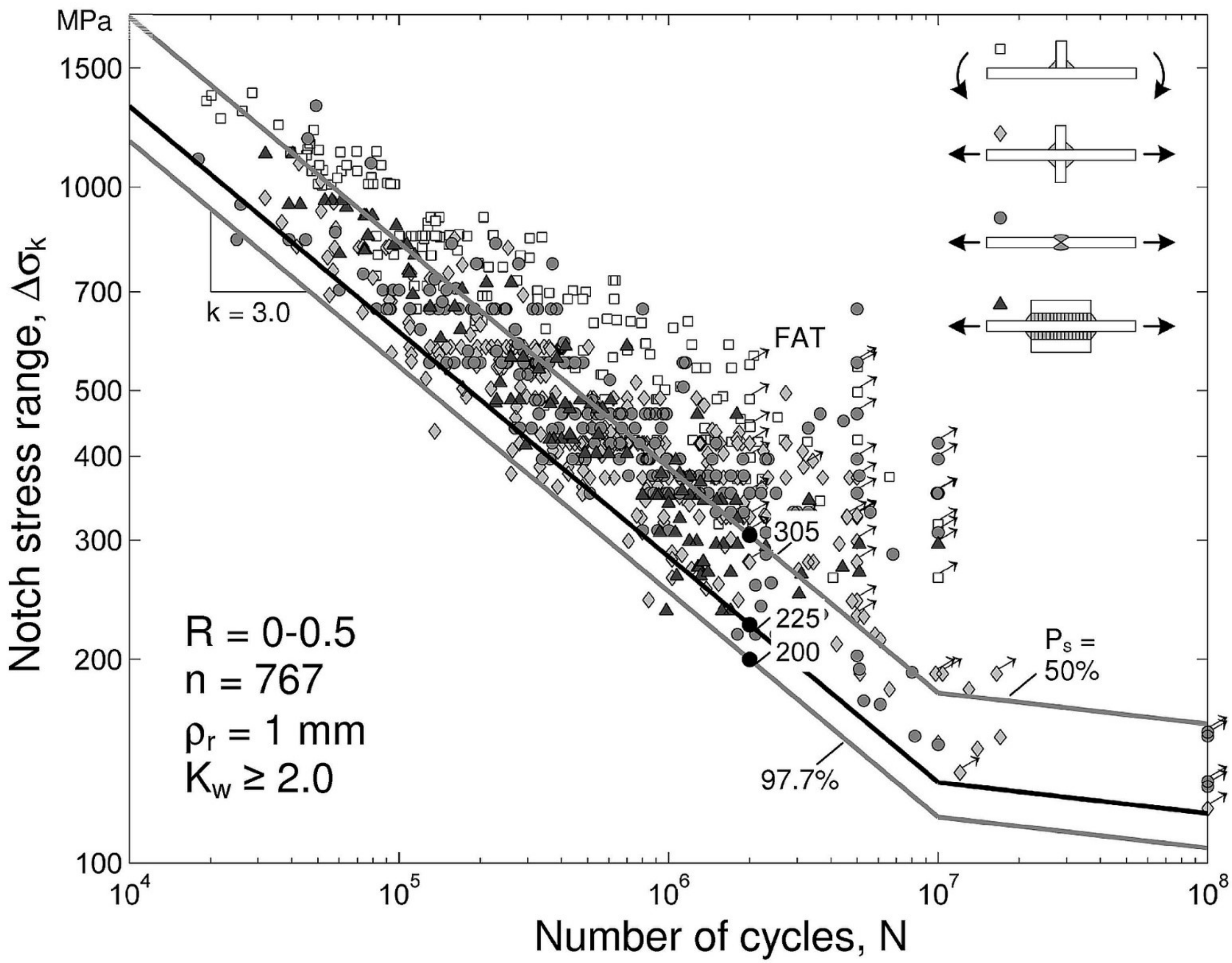
Reference notch concept – IIW version

(Hobbacher 2009)



Reference notch concept – Pedersen's diagram

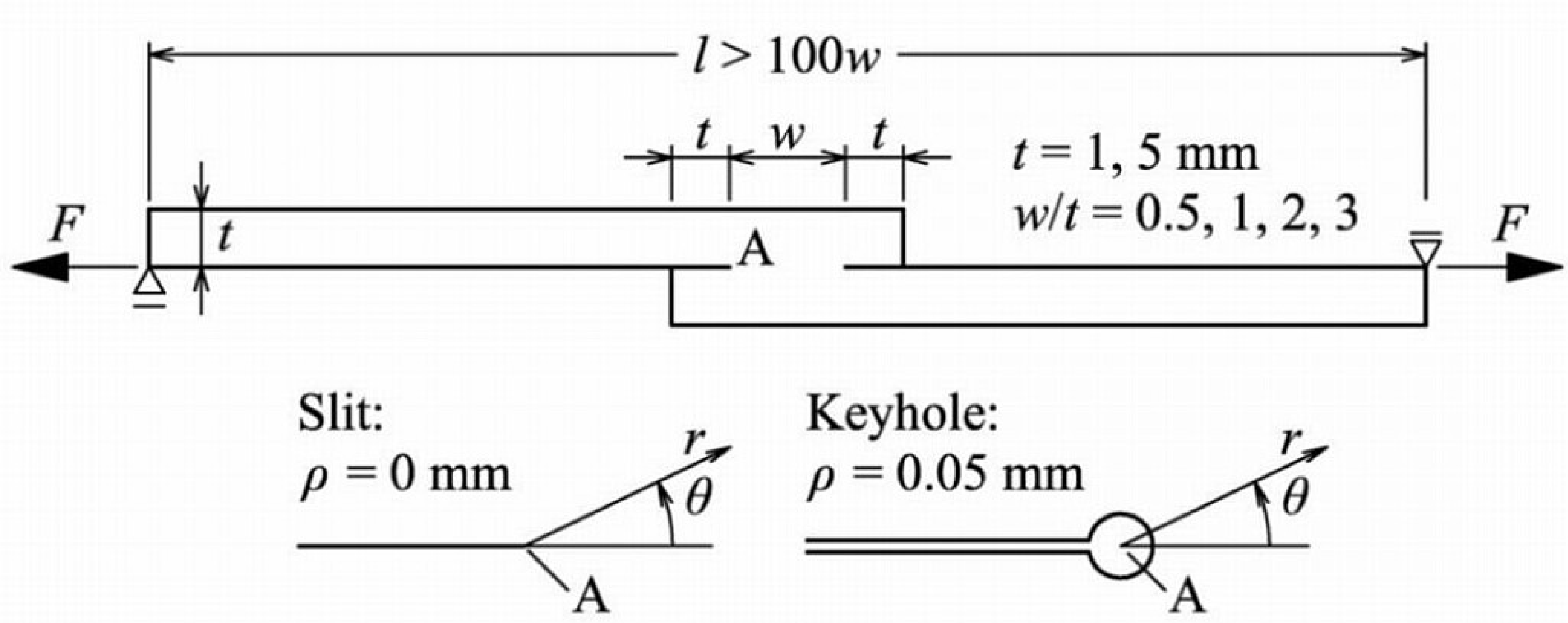
(Pedersen 2011)



Microhole at weld root of thin-sheet lap joint

Thin-sheet lap joints (t = 0.7-5 mm), resistance spot-welded or laser beam seam-welded, require a special procedure because of increasing problems with cross-sectional weakening and slit-parallel loading.

These peculiarities are overcome by application of a microhole at the weld root (ρ = 0.05 mm) followed by notch stress averaging over ρ*.



Microhole at weld root of thin-sheet lap joint

Notch stress formula (Lazzarin and Berto 2009):

$$\sigma_\theta = \frac{K_{1,\rho}}{\sqrt{2\pi\rho}}\left(2\cos\frac{\theta}{2} + \cos\frac{3\theta}{2}\right) - \frac{K_{2,\rho}}{\sqrt{2\pi\rho}}\left(2\sin\frac{\theta}{2} + 3\sin\frac{3\theta}{2}\right) + \frac{3}{2}\bar{T}\left(1 - \cos 2\theta\right)$$

with NSIFs $K_{1,\rho}$, $K_{2,\rho}$ and T-stress $\bar{T}$ gained from FE analysis.

Notch stress formula (Radaj 2010):

$$\sigma_\theta = \frac{K_{\mathrm{I}}}{\sqrt{2\pi\rho}}\left(2\cos\frac{\theta}{2} + \cos\frac{3\theta}{2}\right) - \frac{K_{\mathrm{II}}}{\sqrt{2\pi\rho}}\left(2\sin\frac{\theta}{2} + 3\sin\frac{3\theta}{2}\right) + \bar{T}\left(1 - 2\cos 2\theta\right)$$

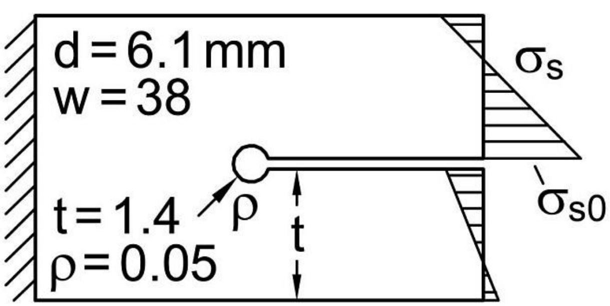
with simple, equilibrium-based formulae for SIFs K_{I}, K_{II} and T-stress $\bar{T}$.

Theoretical *S*-*N* curve of spot-welded lap joint

The *S*-*N* curve of a spot-welded lap joint has been determined theoretically based on the microhole concept in combination with Neuber's microsupport and macrosupport concepts (Seeger et al. 2005).

Procedural steps of the microsupport analysis:

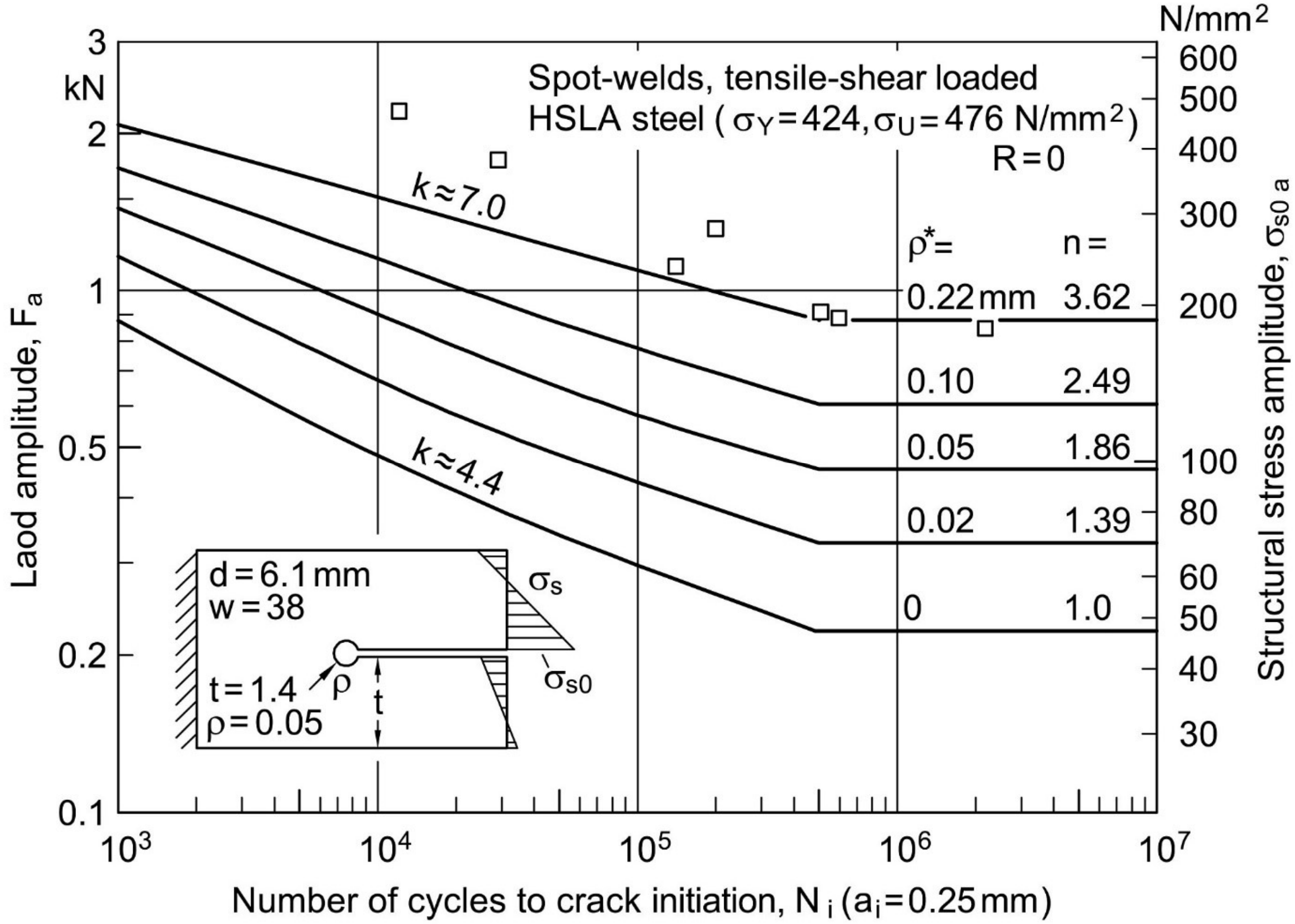
- FE model of lap joint specimen, plane shell elements
- Cross-sectional model with microkeyhole (ρ **= 0.05 mm**) at front side of weld spot subjected to the above membrane and bending stresses
- Notch stress field at microkeyhole according to notch stress formula
- Averaged notch stress σ over $\bar{\rho}^*$ in crack propagation direction
- Notch support index $n = \sigma_{max} / \sigma_{max} = K_t / K_f$ dependent on ρ^*
- Endurance limit $\sigma_{max} = \sigma_A$ (material) results in F_A and $\sigma_{s0,A}$ (lap joint)



Theoretical *S-N* curve of spot-welded lap joint

The theoretical *S-N* curves are compared with experimental results from the literature (MacMahon et al. 1990).

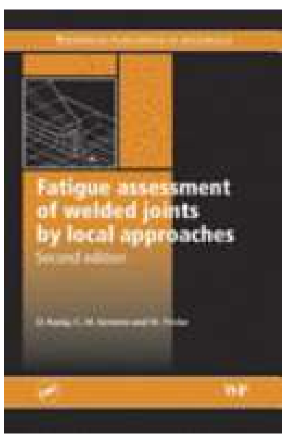
The comparison indicates that $\rho^* = 0.22$ mm might be a reasonable value, but only at the expense of a too large slope exponent, k ≈ 7.0.



Theoretical *S*-*N* curve of spot-welded lap joint

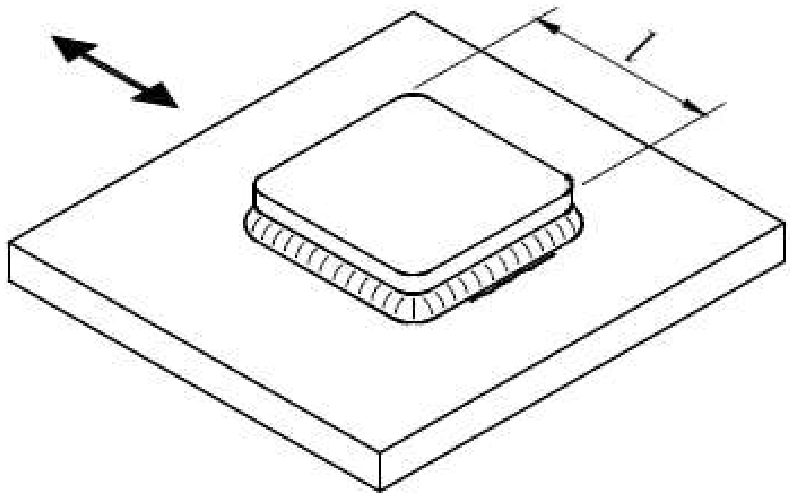
Procedural steps of the macrosupport analysis:

- Notch stresses and strains according to Neuber rule applied to Ramberg-Osgood cyclic stress-strain relationship (ρ = 0.05 mm, $K_t = \sigma_{max} / \sigma_{s0} \approx 4.0$, $K_f = f(K_t, \rho^*) < K_t$)
- Endurable strain amplitude for crack initiation (a_i = 0.25 mm), Manson-Coffin-Morrow strain *S*-*N* curve
- Smith-Watson-Topper damage parameter expressing the mean stress influence
- Cyclic material parameters according to Seeger's uniform material law



Fatigue assessment of welded joints by local approaches

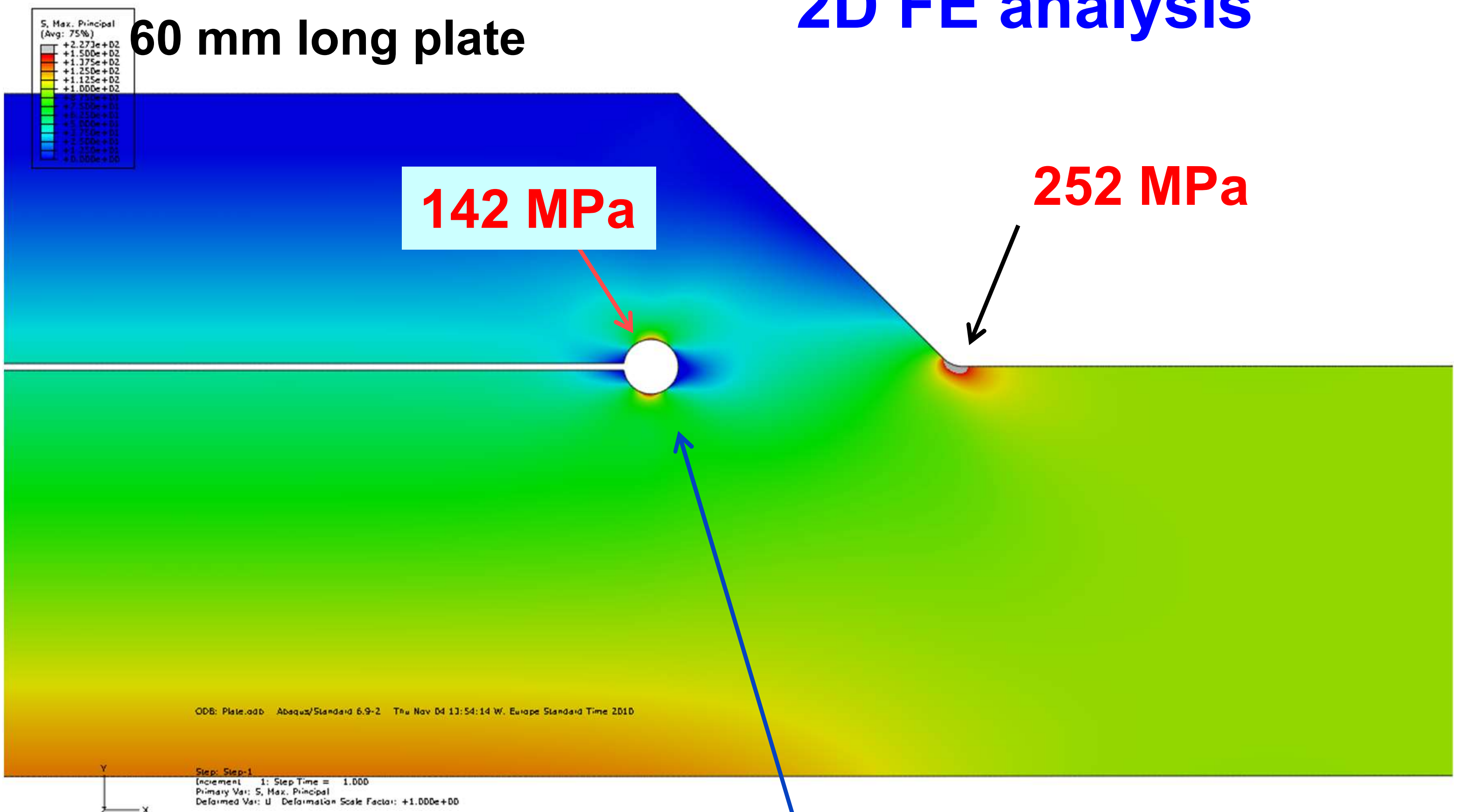
Example of stress analysis of cover plate which can fail from the weld toe or the root

<table>
<tr><td></td><td>2.
</td></tr>
<tr><td>E</td><td>$l \leq 50$ mm</td></tr>
<tr><td>F</td><td>$50 < l \leq 120$ mm</td></tr>
<tr><td>F1</td><td>$120 < l \leq 300$ mm</td></tr>
<tr><td>F3</td><td>$l > 300$ mm</td></tr>
</table>

Example of effective notch stress analysis

2D FE analysis

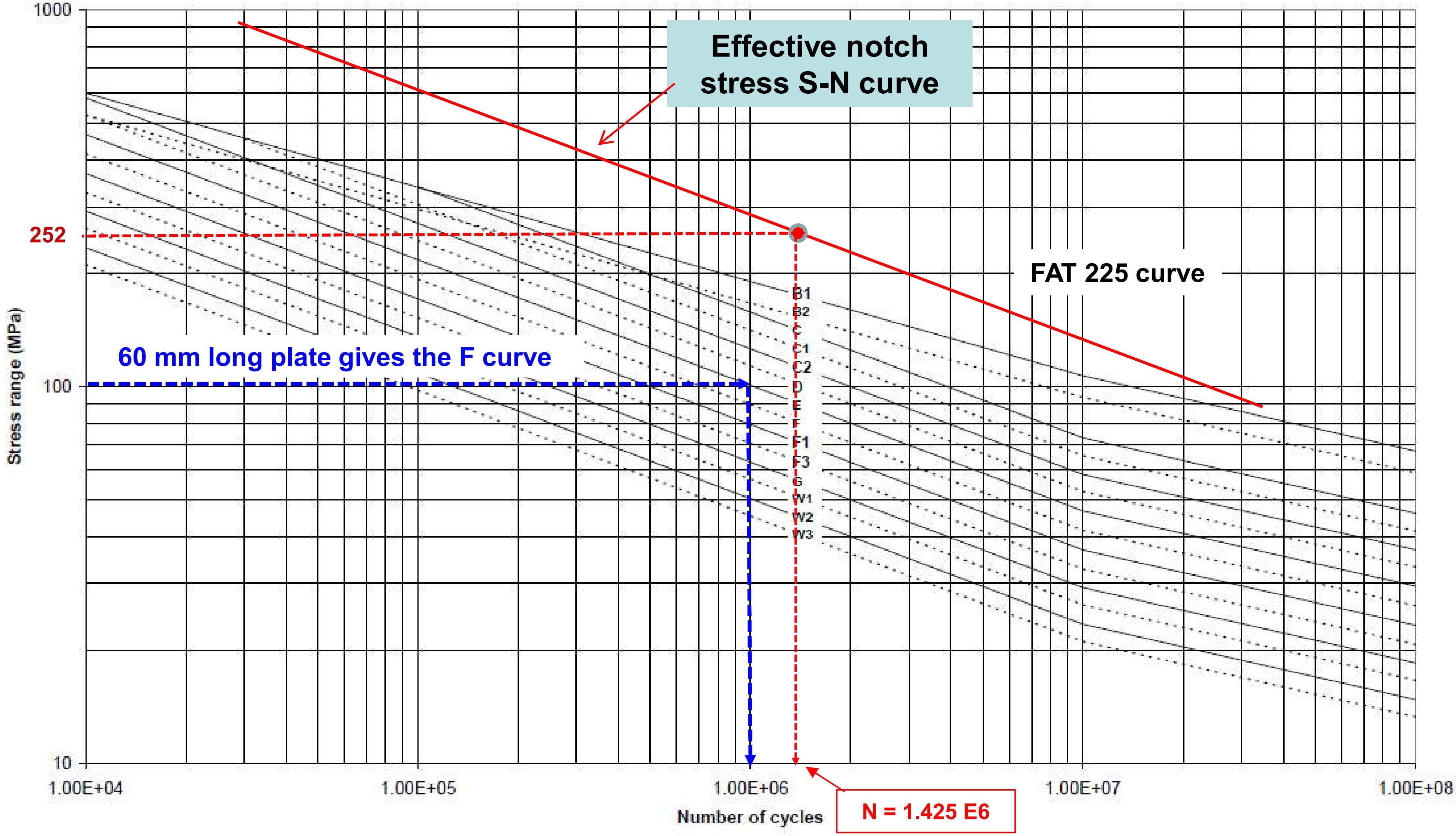
60 mm long plate



Ref. Stress = 100 MPa

Small risk of root cracking

Comparison with nominal stress method

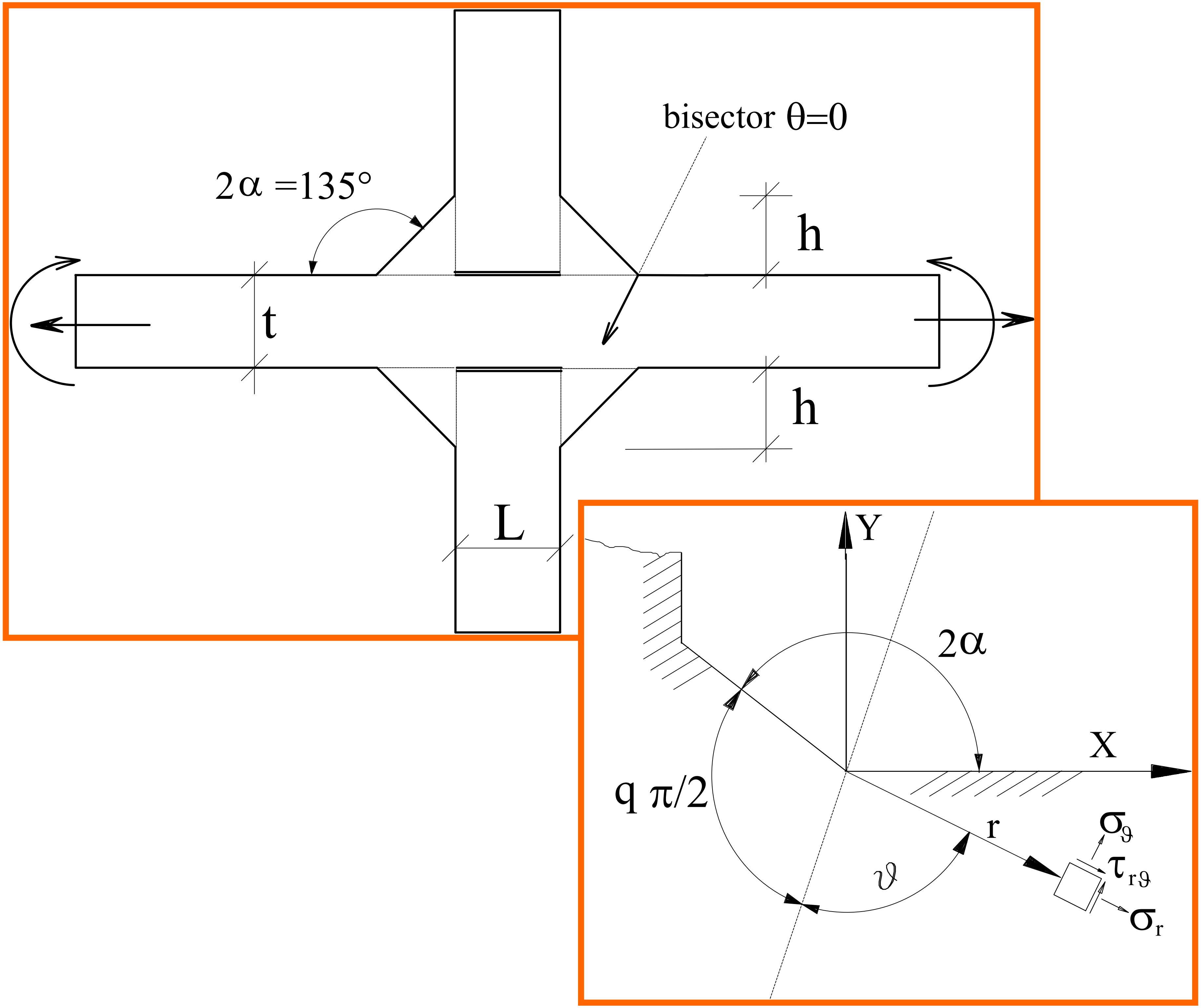


Lecture 3

Local approaches

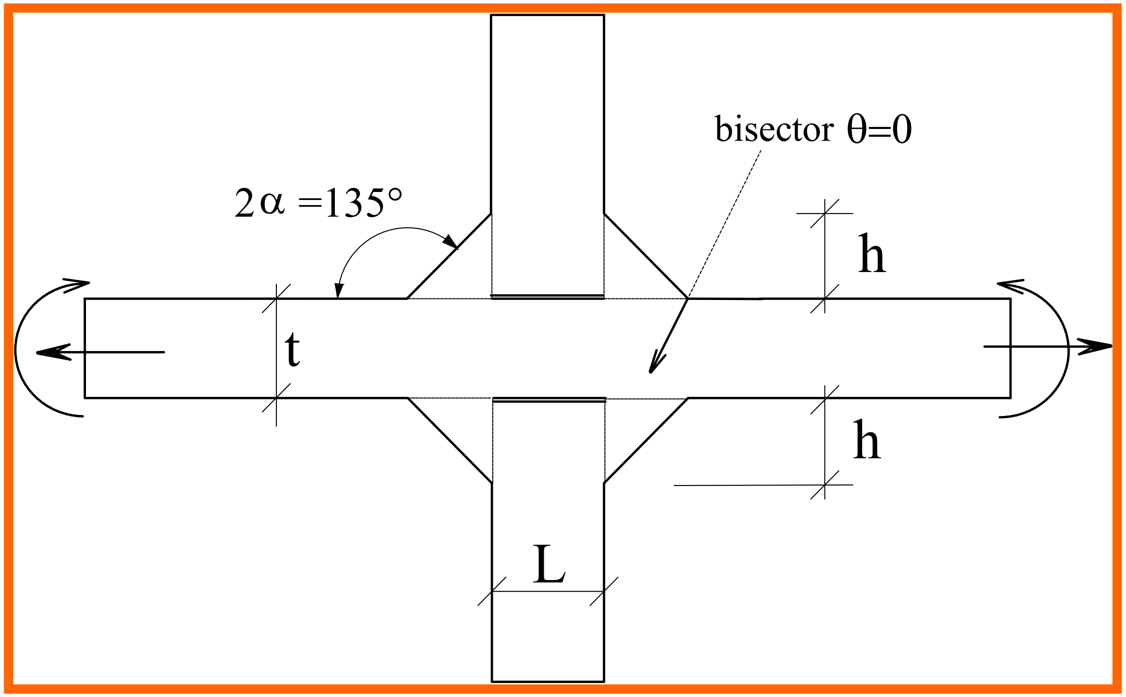
Some case studies

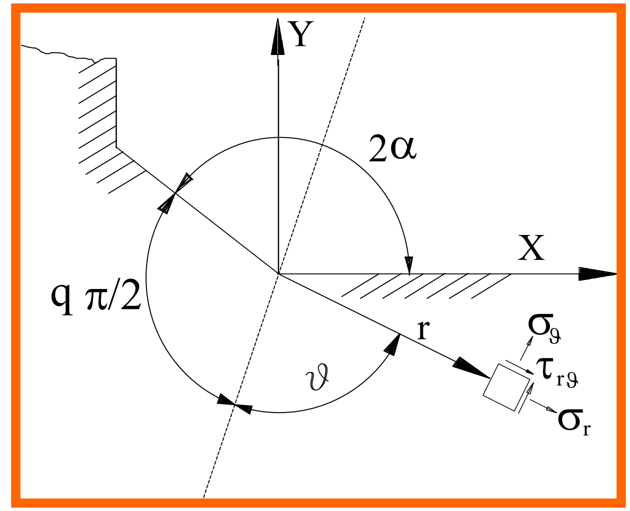
Notch stress intensity factor



bisector θ=0
2α =135°
h
t
h
L
Y
2α
X
q π/2
r
ϑ
σ_ϑ
$\tau_{r\vartheta}$
σ_r

WELDED JOINTS



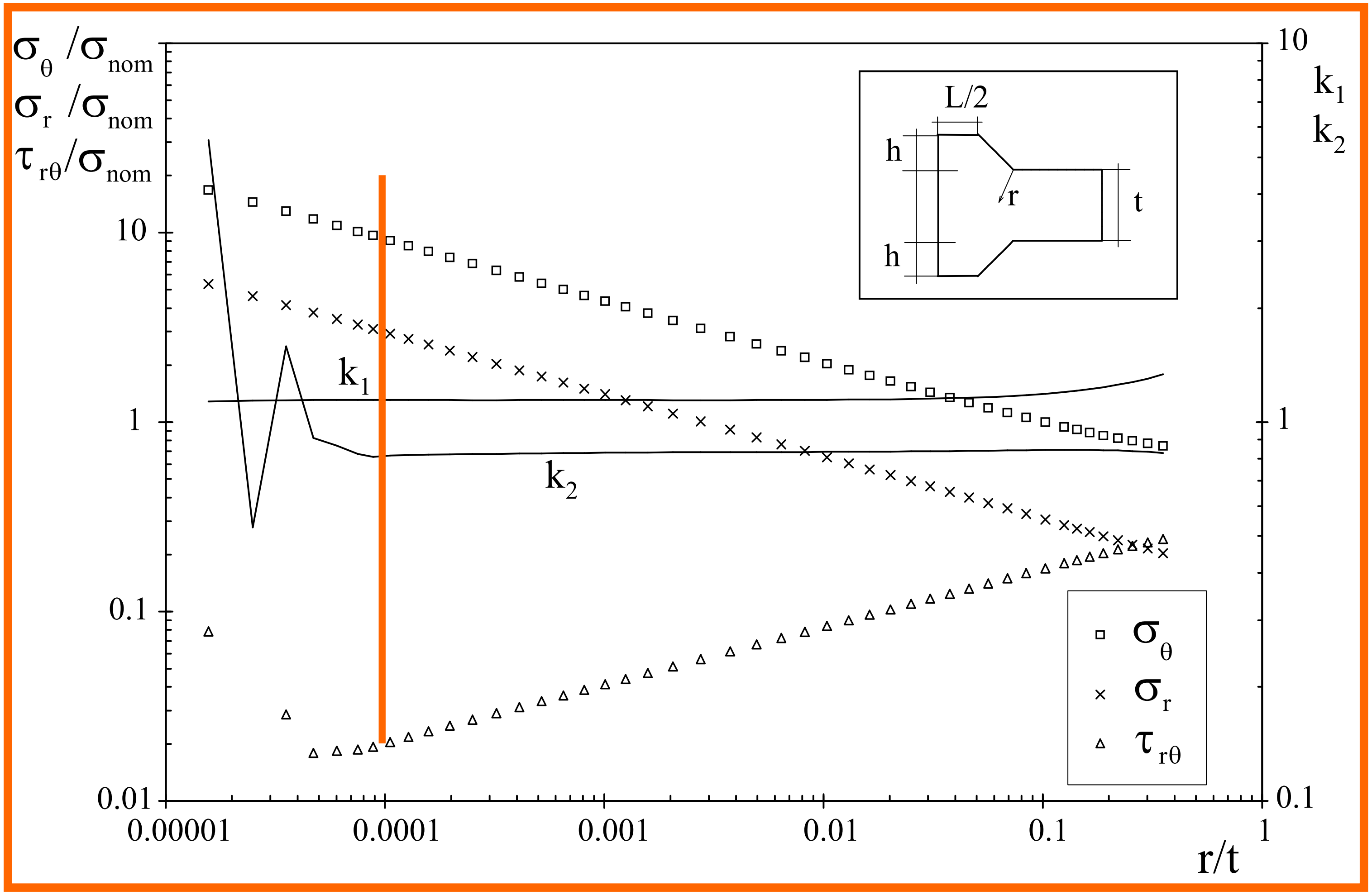


Coordinate system and geometrical parameters for the analyses of the welded joints

According to Gross and Mendelson's definition (1972), the N-SIFs related to the mode I stress distribution are:

$$\boxed{K_1^N = \sqrt{2\pi} \lim_{r \to 0^+} r^{1-\lambda_1} \sigma_{\theta\theta}(r)}$$

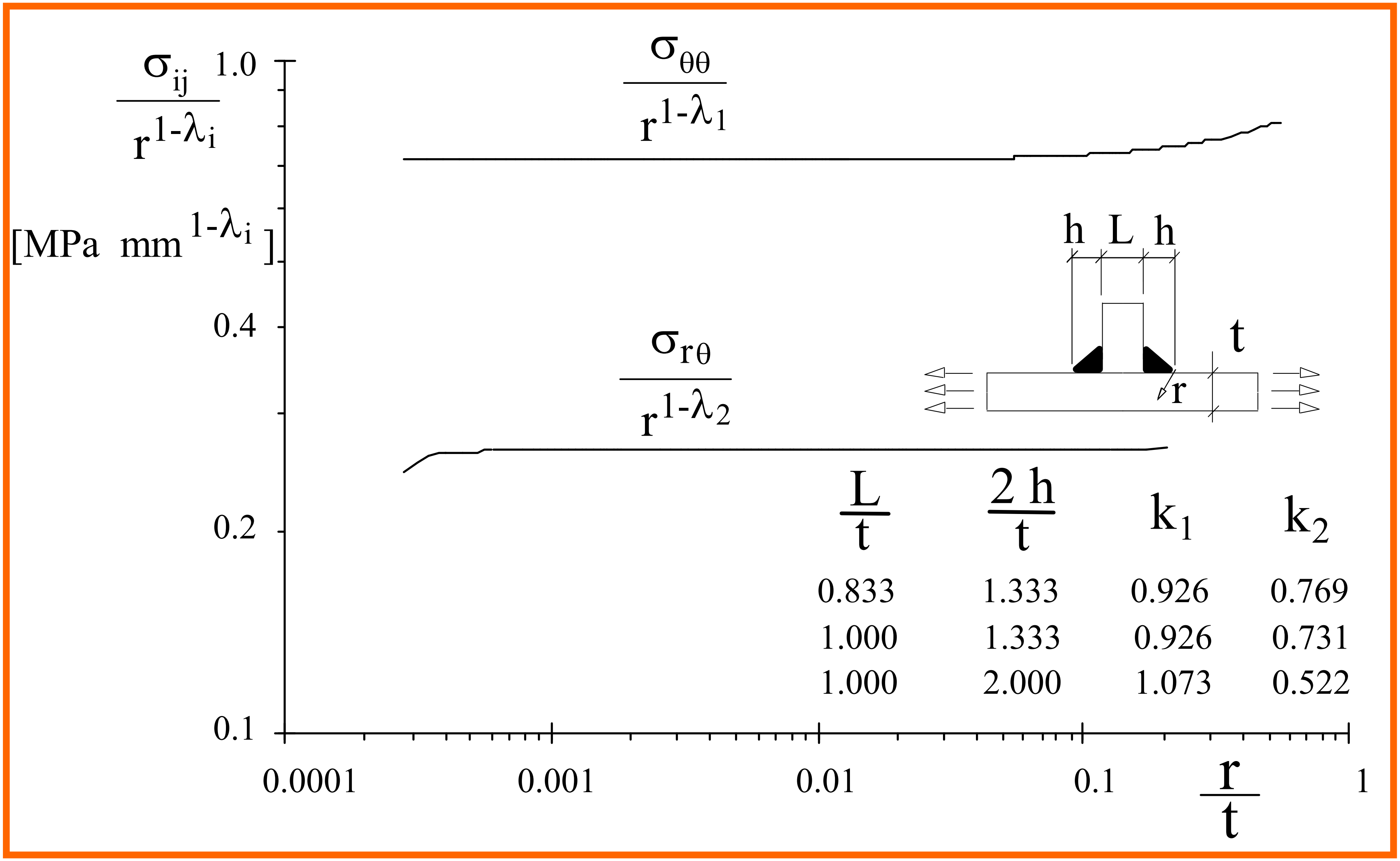
$$\boxed{K_2^N = \sqrt{2\pi} \lim_{r \to 0^+} r^{1-\lambda_2} \sigma_{r\theta}(r)}$$



Stress components along the bisector and k1 and k2 evaluation.

Local NSIFs can be linked to nominal stress according to the expression

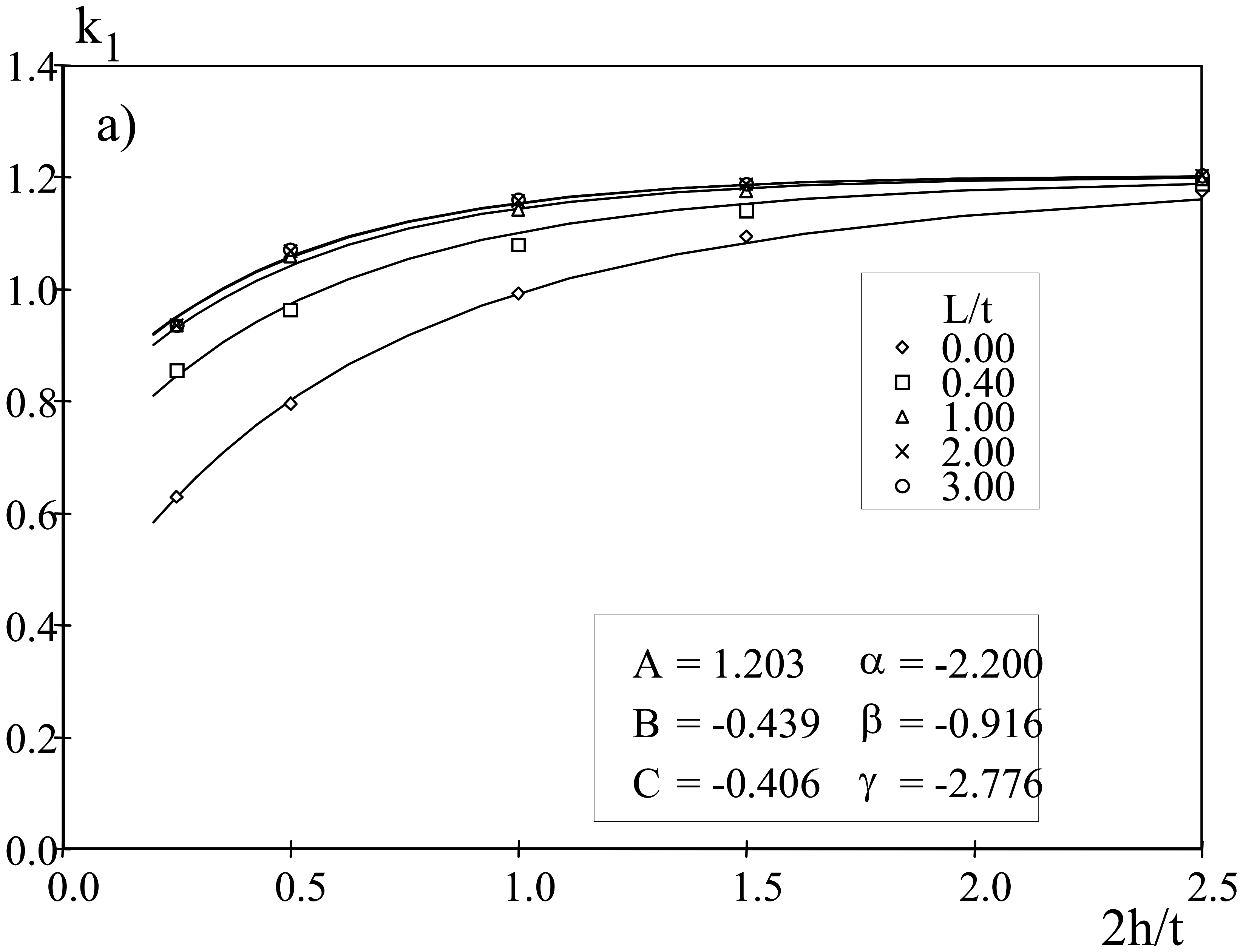
$$\boxed{\Delta K_1^N = k_1 \cdot t^{1-\lambda_1} \cdot \Delta\sigma_{nom}}$$



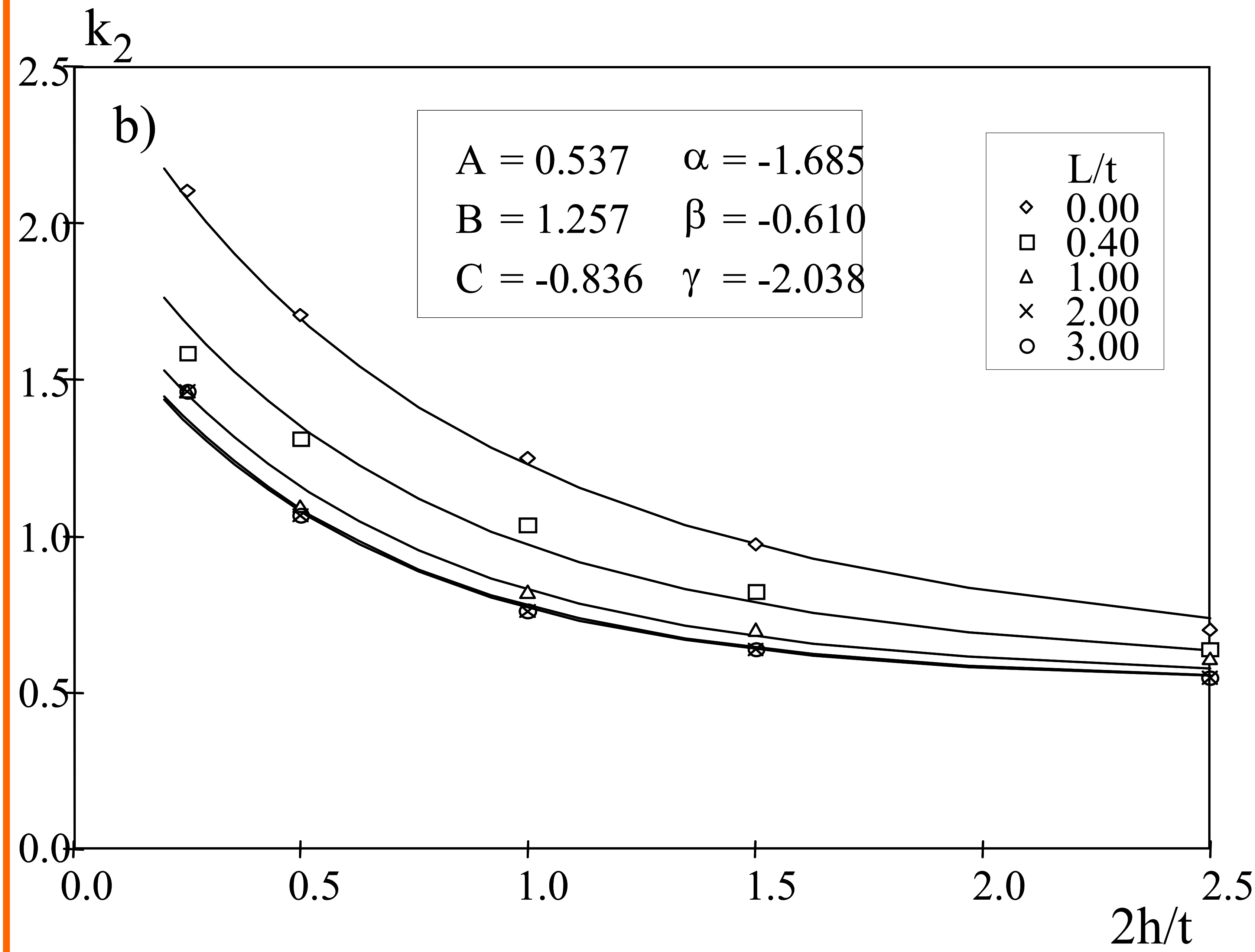
$\frac{L}{t}$	$\frac{2h}{t}$	k_1	k_2
0.833	1.333	0.926	0.769
1.000	1.333	0.926	0.731
1.000	2.000	1.073	0.522

$$k_{1,j} = \sqrt{2\pi}\,\frac{\sigma_{\theta,j}}{\sigma_0}\,(r_j/t)^{1-\lambda_1}$$

$$k_{2,j} = \sqrt{2\pi}\,\frac{\tau_{r\theta,j}}{\sigma_0}\,(r_j/t)^{1-\lambda_2}$$



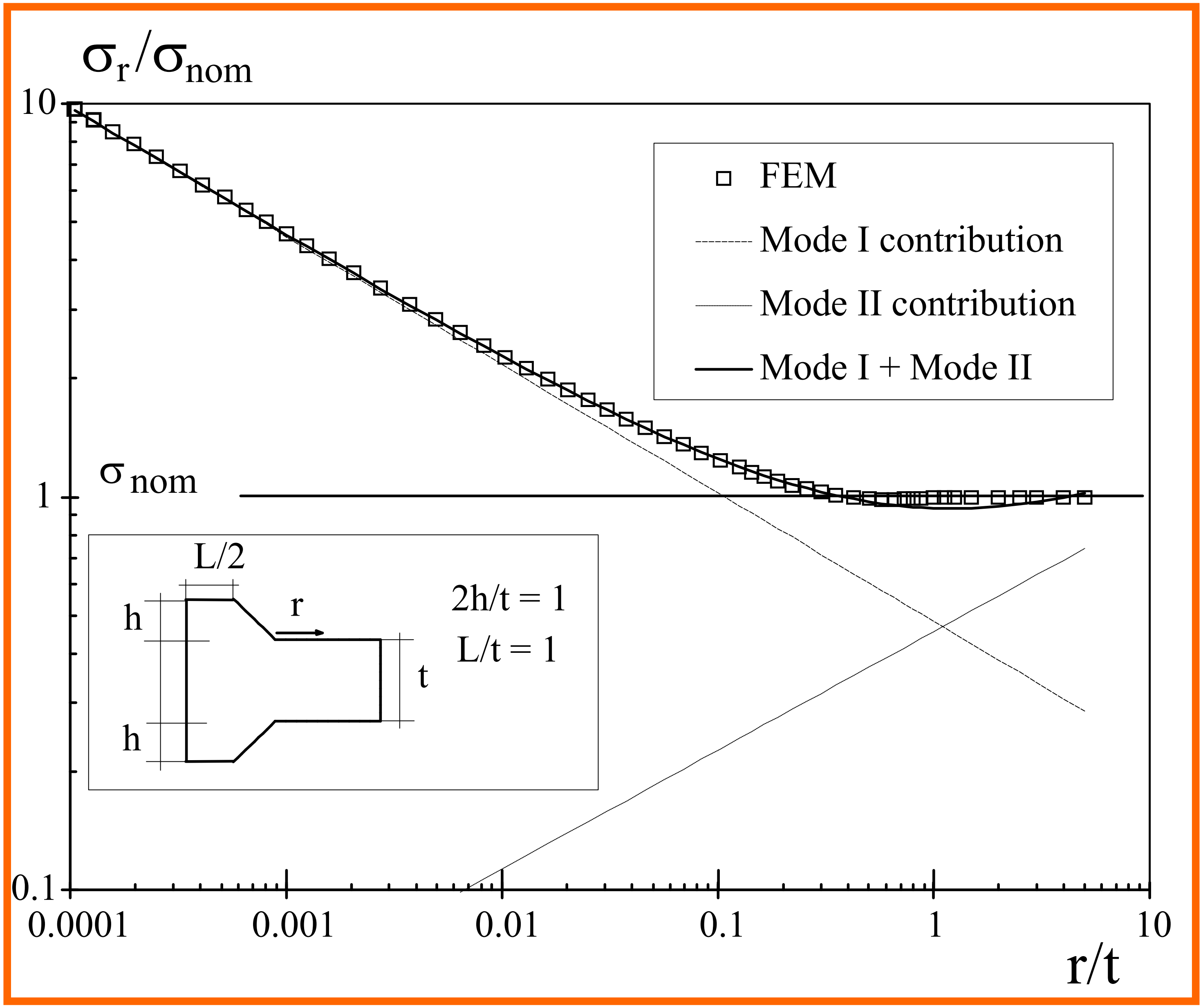
$$k_i = A_i + B_i \cdot e^{\alpha_i(2h/t)} + C_i \cdot e^{\beta_i(2h/t) + \gamma_i(L/t)}$$



$$k_i = A_i + B_i \cdot \mathrm{e}^{\alpha_i (2h/t)} + C_i \cdot \mathrm{e}^{\beta_i (2h/t) + \gamma_i (L/t)}$$

Table 2. Coefficients k_1 and k_2 for 'weld-like' geometries

	k_1				
	L/t				
$2h/t$	0.00	0.40	1.00	2.00	3.00
0.25	0.630	0.885	0.936	0.936	0.936
0.50	0.795	0.963	1.059	1.068	1.071
1.00	0.993	1.080	1.142	1.157	1.160
1.50	1.095	1.139	1.175	1.187	1.187
2.50	1.175	1.187	1.196	1.202	1.202
	k_2				
	L/t				
$2h/t$	0.00	0.40	1.00	2.00	3.00
0.25	2.102	1.584	1.462	1.462	1.462
0.50	1.706	1.310	1.097	1.066	1.066
1.00	1.249	1.036	0.823	0.762	0.762
1.50	0.975	0.823	0.701	0.640	0.640
2.50	0.701	0.640	0.609	0.548	0.548



Contribution of mode I and mode II to the radial stress along the free edge

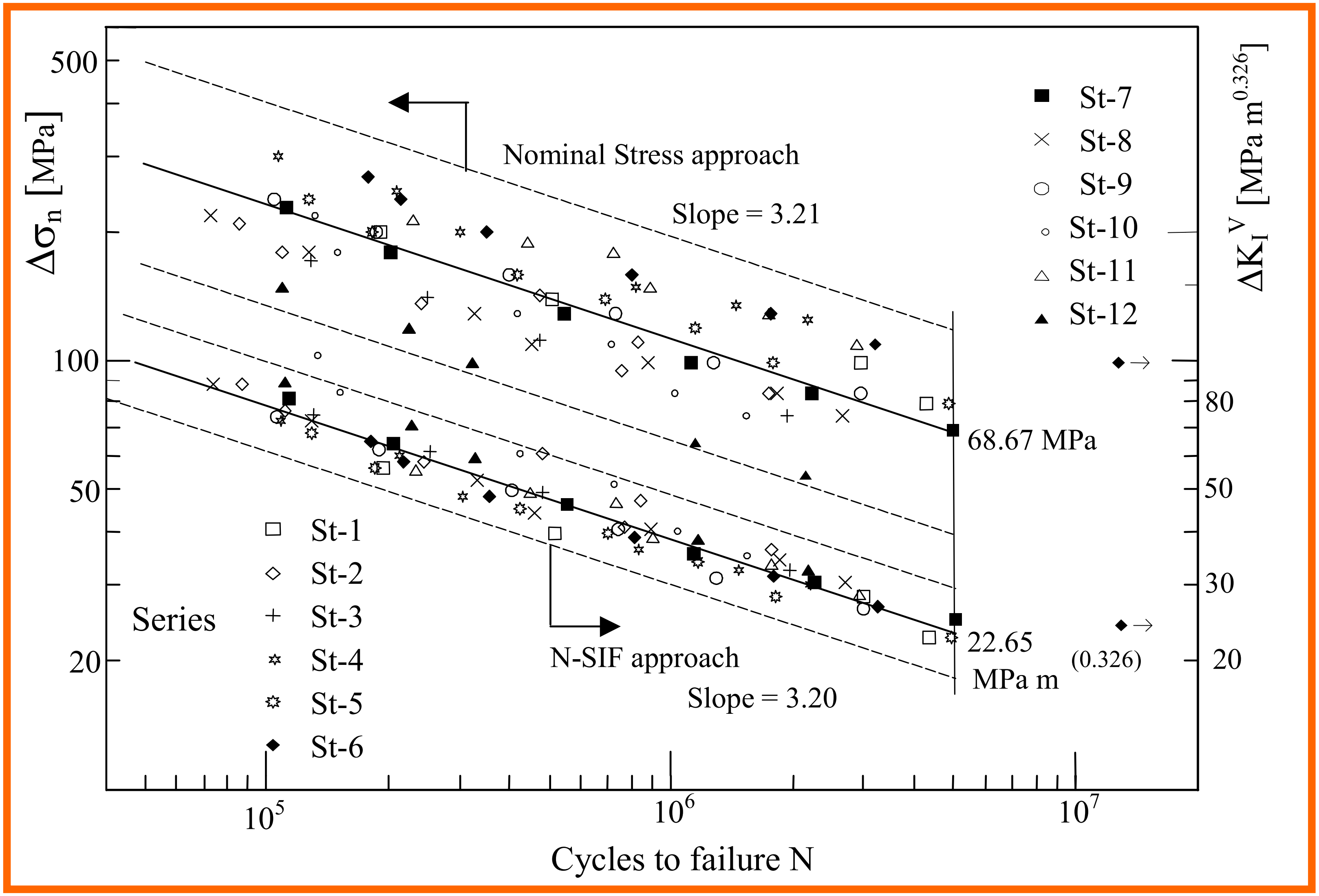
Original data from Gurney (1991) and Maddox (1987)

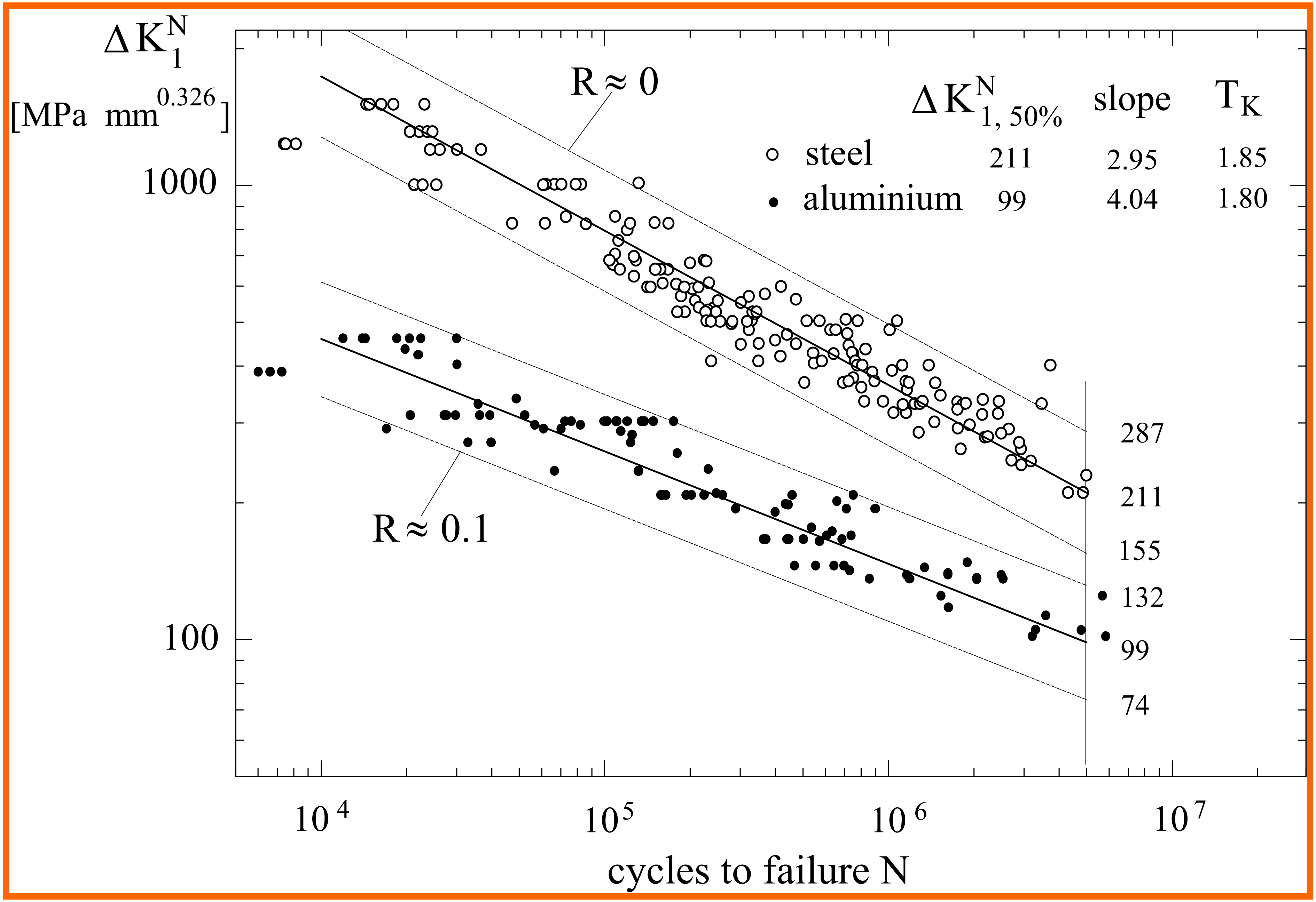
Table 3. Transverse non-load-carrying welded joints (Refs [10,11]) giving geometrical parameters, fatigue strength ranges at 5×10^6 cycles and differences in percent with respect to the mean value of the overall distribution

Series	t (mm)	h (mm)	L (mm)	$\Delta\sigma$ (MPa)	$\Delta\%$	ΔK_1 (MPa m$^{0.326}$)	$\Delta\%$
1	13	8	10	79.52	16	22.04	−3
2	50	16	50	59.64	−13	25.02	10
3	100	16	50	55.47	−19	23.56	4
4	13	5	3	91.70	34	21.62	−5
5	13	10	8	76.68	12	21.24	−6
6	25	5	3	93.92	37	22.19	−2
7	25	9	32	66.02	−4	23.04	2
8	25	15	220	59.72	−13	23.59	4
9	38	8	13	68.89	0	20.84	−8
10	38	15	220	45.46	−34	20.86	−8
11	100	5	3	95.70	39	24.34	7
12	100	15	220	40.09	−42	23.22	2

Original data from Gurney (1991) and Maddox (1987)

- Main Plate thickness ranging from 6 to 100 mm
- Transverse plate thickness ranging from 3.0 to 200 mm



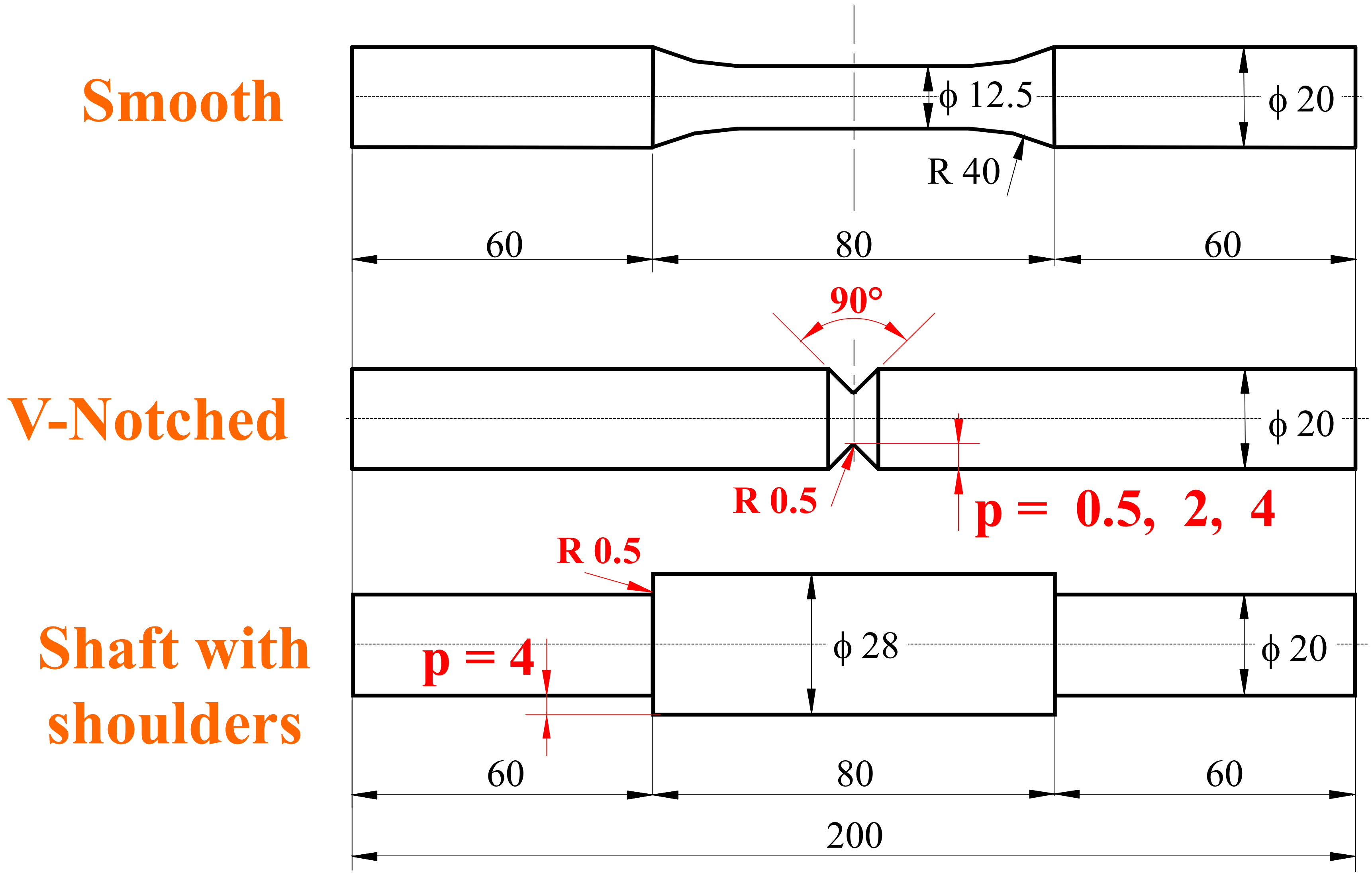


Fatigue strength of aluminium and steel welded joints as a function of Mode I Notch Stress Intensity Factor. Scatter band related to mean values plus/minus 2 standard deviations

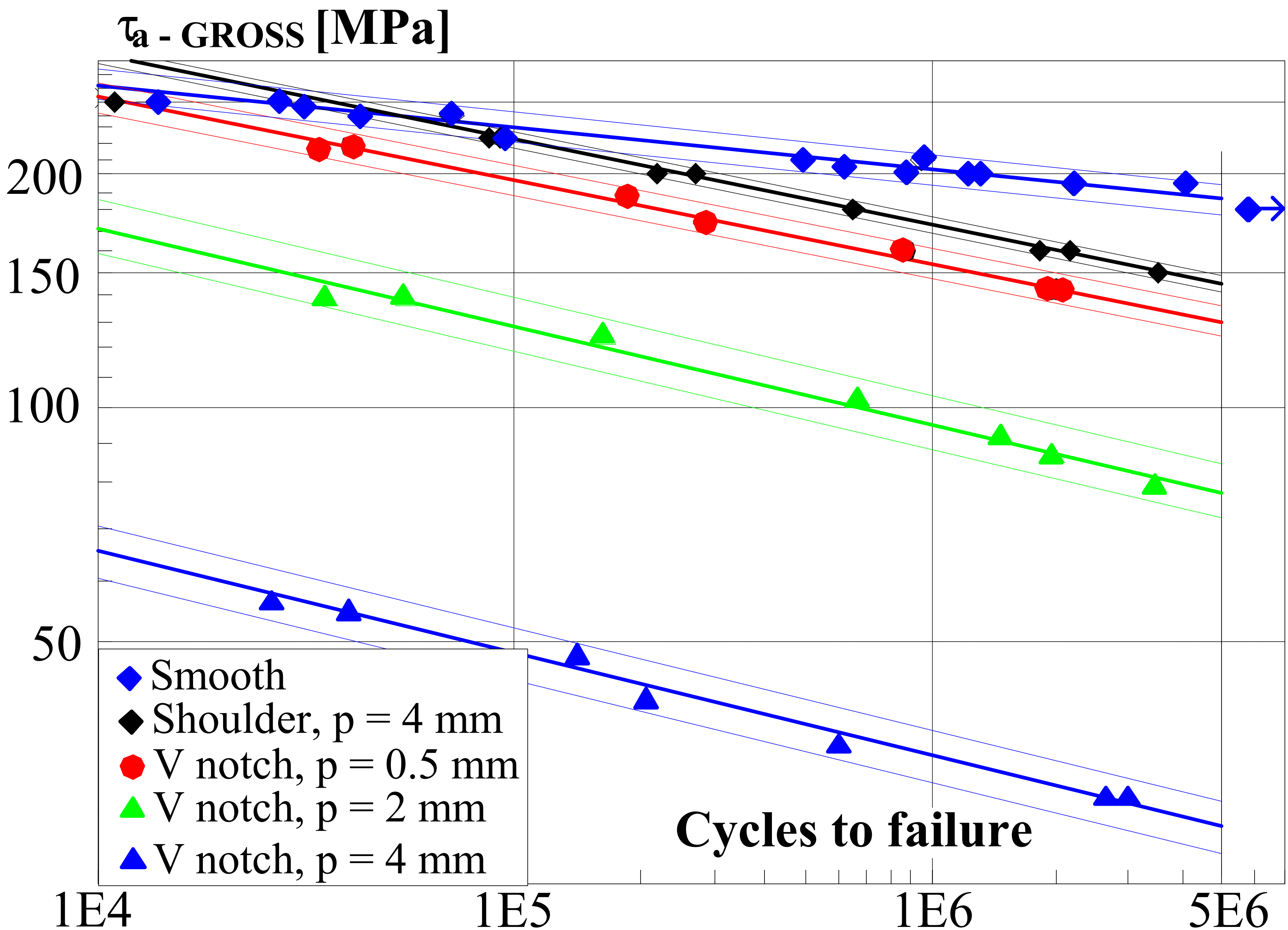
MATERIAL AND SPECIMEN GEOMETRY

MATERIAL: normalised C40 structural steel

E= 206 GPa, $\sigma_{p0,2}$=537 MPa, σ_{uts} = 715 MPa



FATIGUE TEST RESULTS (stress on gross section)



SEM FAILURE SURFACE - HIGH STRESS LEVEL

V-notch p = 2 mm, $\tau_{a,nom}$ = 240 MPa, N = 35274

FAILURE SURFACES - p=0.5 mm

V-notch p = 0.5 mm,
$\tau_{a,nom}$ = 245 MPa, N = 40743

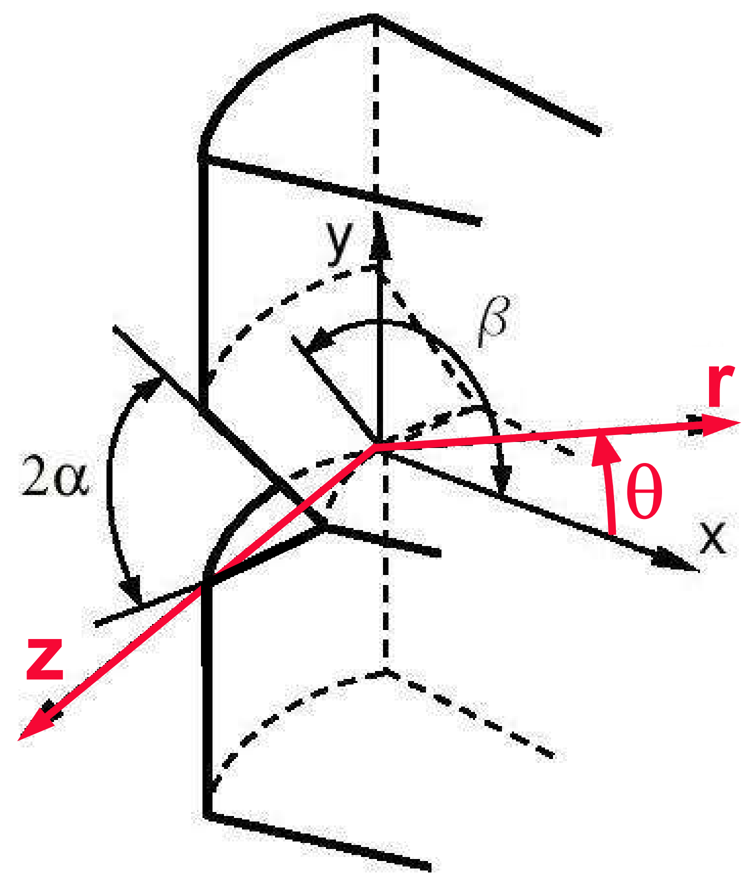
V-notch p = 0.5 mm,
$\tau_{a,nom}$ = 165 MPa, N = 1979392

DEFINITION OF THE MODE III N-SIF

The mode III N-SIF was evaluated on the uncracked geometries, modelling all the notches like re-entrant corners (i.e. sharp V-notches).

$$K_3^N = \sqrt{2\pi} \lim_{r\to 0^+} r^{1-\lambda_3} \cdot \sigma_{\theta z}(r,0)$$

for $2\alpha=\pi/2$, the eigenvalue λ_3 is 2/3



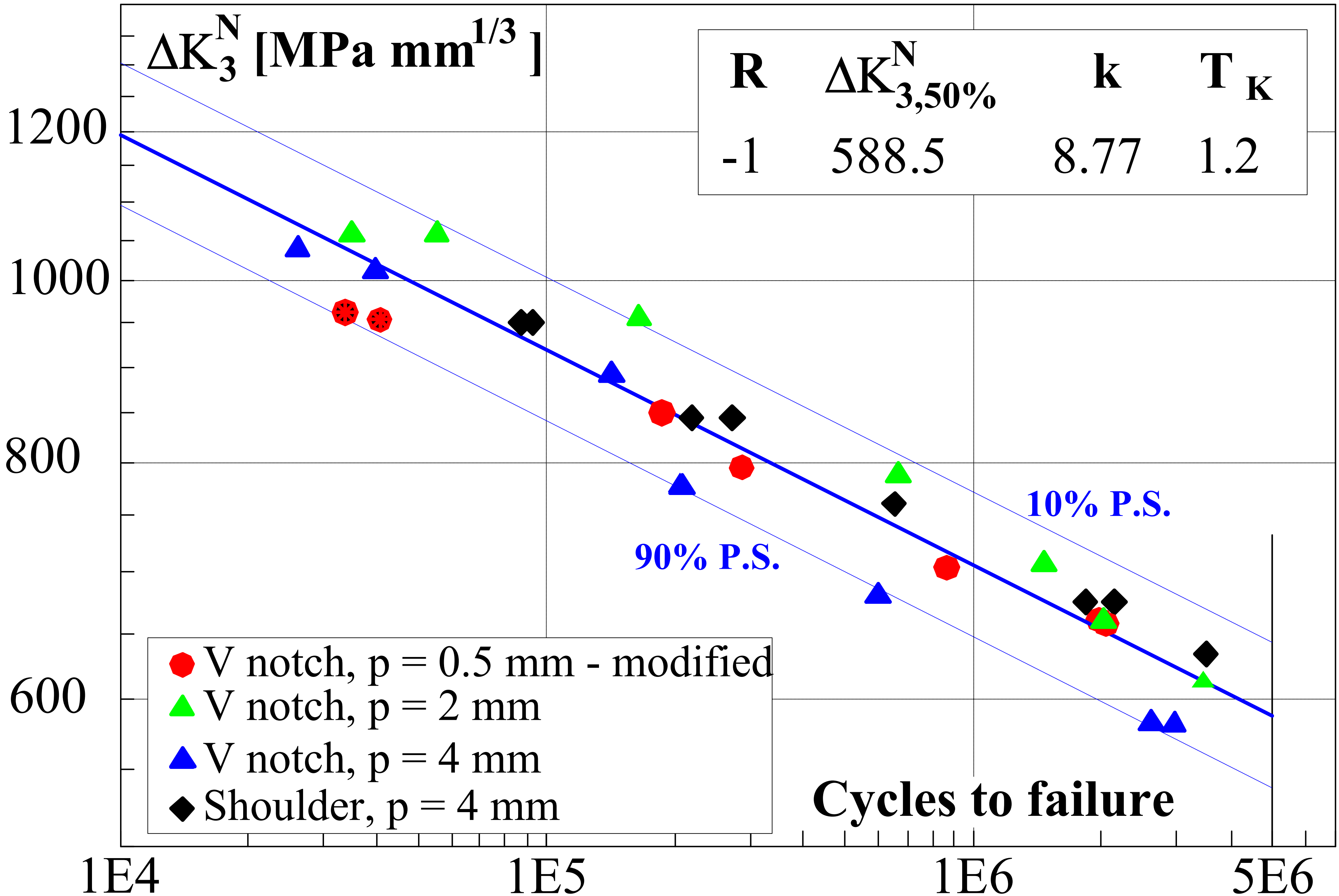
The N-SIF can be expressed as a function of nominal shear stress and notch depth p:

$$\Delta K_3^N = k_3 \, \Delta\tau_{gross} \, p^{1/3}$$

For small-depth notches, the intrinsic defect length p_o can also be introduced:

$$\Delta K_3^N = k_3 \, \Delta\tau_{gross} \, (p + p_o)^{1/3}$$

FATIGUE SCATTER BAND IN TERMS OF K_3 N-SIF



R	$\Delta K^N_{3,50\%}$	k	T_K
-1	588.5	8.77	1.2

Lecture 3

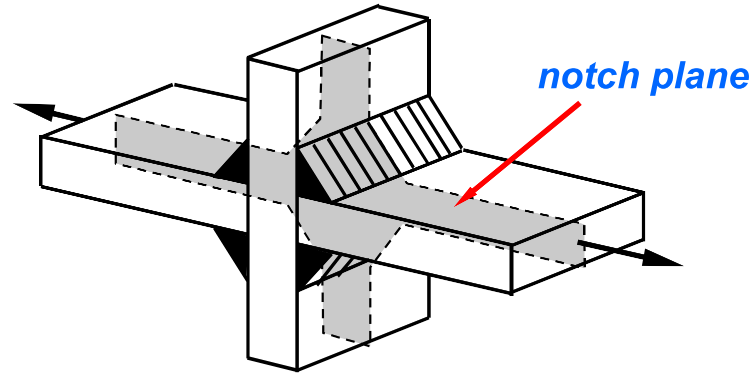
Local approaches

Some case studies: Part 1

STRAIN ENERGY DENSITY

AVERAGED SED AS A FATIGUE PARAMETER

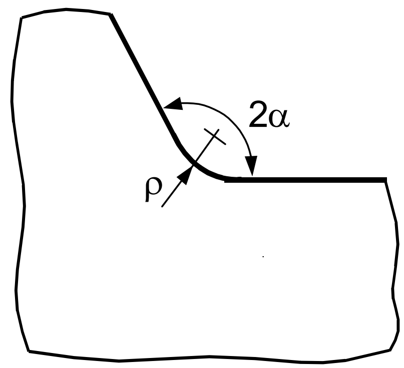
V-shaped notched component: load carrying cruciform steel welded joints



"Real" geometry at the weld toe

typical opening angle $2\alpha \approx 135°$

typical toe radius ρ = 0.2 ÷ 0.8 mm

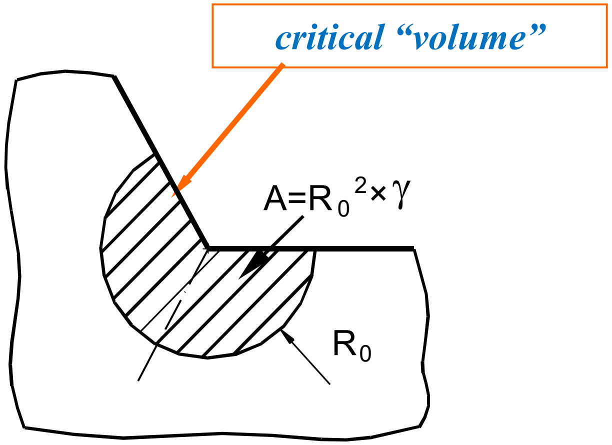


Theoretical model and critical "volume"

opening angle $2\alpha = 135°$

toe radius $\rho = 0$ mm

R_0 characteristic dimension of the critical volume



Livieri P., Lazzarin P. (2005). International Journal of Fracture
Berto F, Lazzarin P. (2009). Theoretical and Applied Fracture Mechanics

SOME CONSIDERATIONS ABOUT NSIF APPROACH APPLIED TO WELDED JOINTS

- **Weld bead geometry cannot be precisely defined. Parameters such as bead shape and toe radius vary.**

- **Conventional welding techniques result in very small values for the toe radius.**

- **Currently the weld toe region is modeled as a sharp, zero radius, V-shaped notch.**

- **The intensity of asymptotic stress distributions (Williams' solution) are quantified by means of the Notch Stress Intensity Factors.**

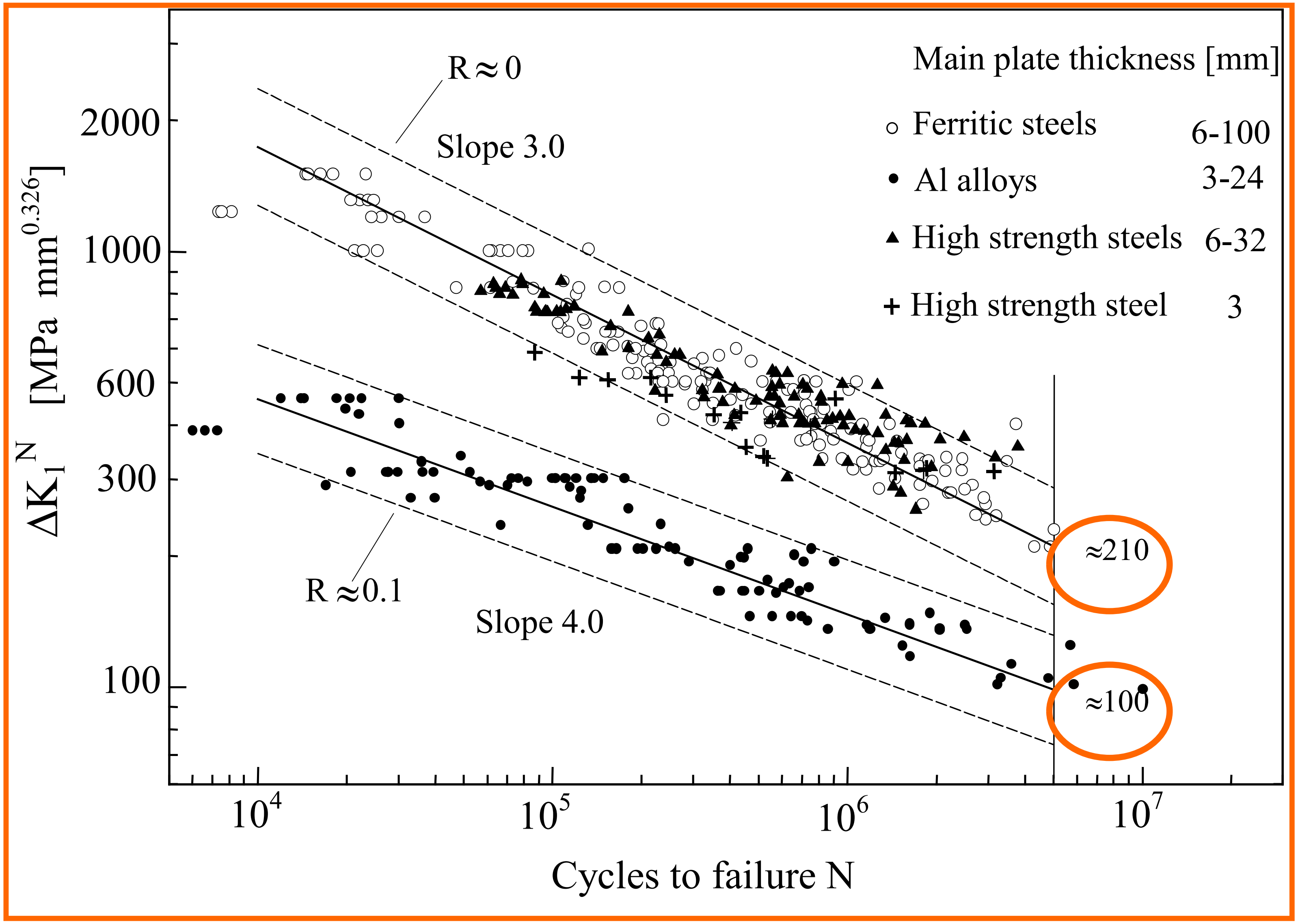
SOME CONSIDERATIONS ABOUT NSIF APPROACH APPLIED TO WELDED JOINTS

- **Originally NSIFs were considered suitable for predicting only the fatigue crack initiation phase. (Pluvinage, 1995.Verreman and Nie, 1996).**

- **Afterwards, NSIFs were applied to fatigue total life assessments. This happens when a large proportion of the component's life span is spent at short crack depth (Lazzarin and Tovo, 1998, Lazzarin and Livieri, 2001).**

- **No demarcation line is drawn between initiation and early propagation. Both phases are thought of as dependent on the stress distribution present in the un-cracked component.**

TOE FAILURES: NSIF APPROACH



Fatigue strength of steel and aluminum fillet welded joints in terms of the Mode I NSIF (Lazzarin and Tovo 1998, Lazzarin and Livieri 2001). Scatter bands defined by mean values of ± 2 standard deviations.

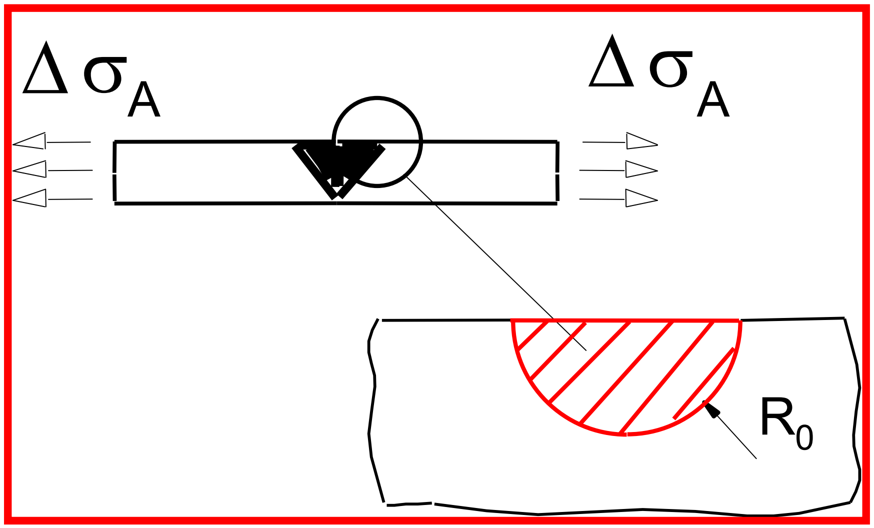
ADVANTAGES OF A LOCAL-ENERGY APPROACH BASED ON NSIFs

- Permits consideration of the scale effect.
- Permits consideration of the contribution of different Modes.
- Permits consideration of the cycle nominal load ratio.
- Overcomes the complex problem tied to the different NSIF units of measure in the case of crack initiation at the toe ($2\alpha=135°$) or root ($2\alpha=0°$).
- Overcomes the problem of multiple crack initiation and their interaction.
- SED can be evaluated with coarse meshes
- It directly takes into account the T-stress
- It directly includes three-dimensional effects

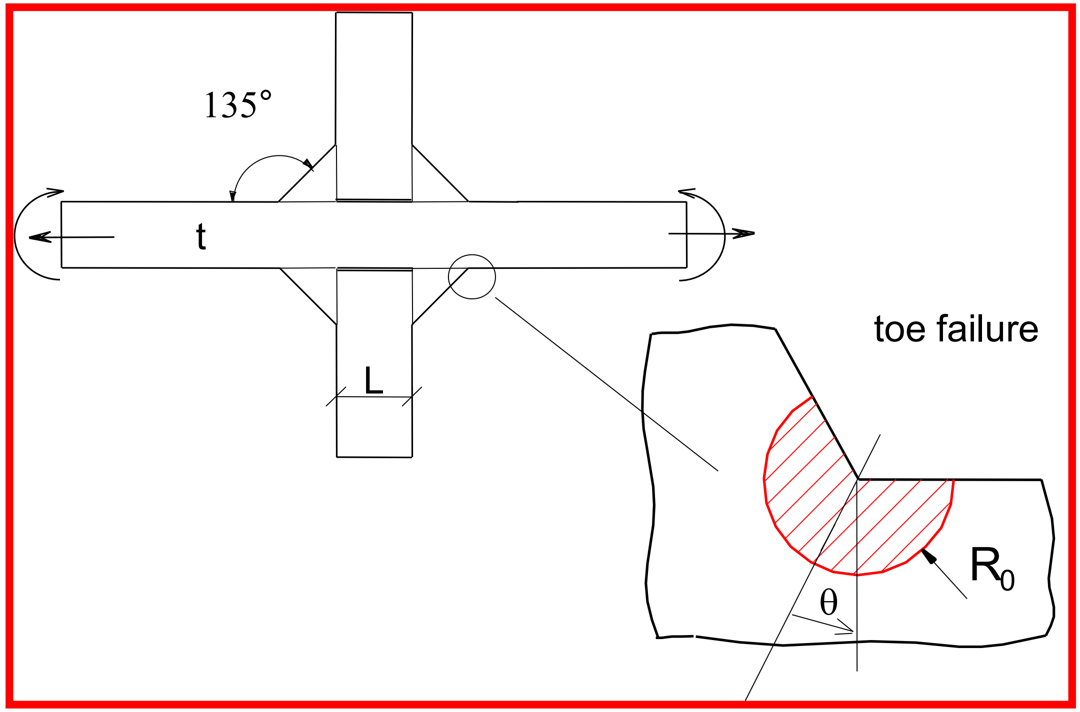
CRITICAL RADIUS EVALUATION

$2\alpha=180°$

ground butt welded joints



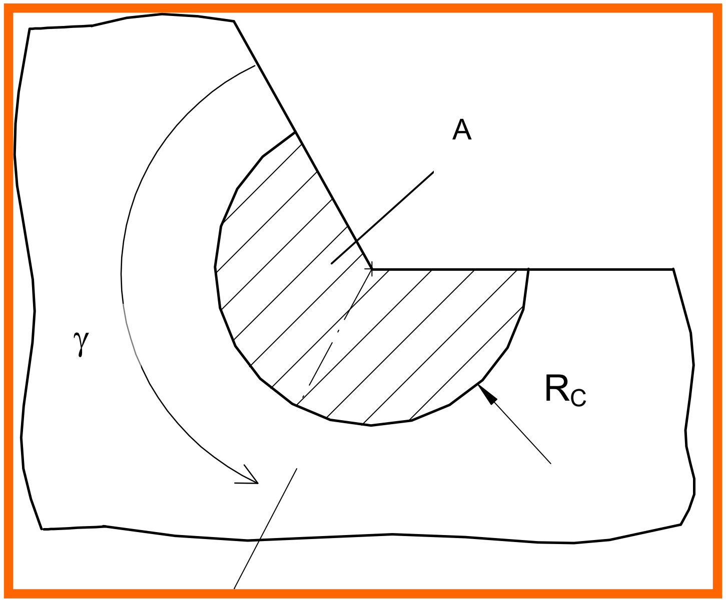
$2\alpha=135°$



$$R_0 = \left(\frac{\sqrt{2e_1}\,\Delta K_{1A}^{N}}{\Delta\sigma_A} \right)^{\frac{1}{1-\lambda_1}}$$

211 MPa mm$^{0.326}$

155 MPa



The critical value of the radius R_C mainly depends on the material. The more brittle is the material, the smaller R_C is

R_C depends also on the failure hypothesis

By using the Beltrami hypothesis, a convenient expression is

$$\boxed{R_C = \left(\frac{\Delta K_{1A}^{N}}{f_1 \ \Delta\sigma_A} \right)^{\frac{1}{1-\lambda_1}}}$$

where λ_1 and f_1 depend on the V-notch opening angle 2α ($f_1 = 2.065$ and $\lambda_1 = 0.674$ when $2\alpha = 135$ degrees)

At $N_A = 5 \cdot 10^6$ cycles, under $R = 0$

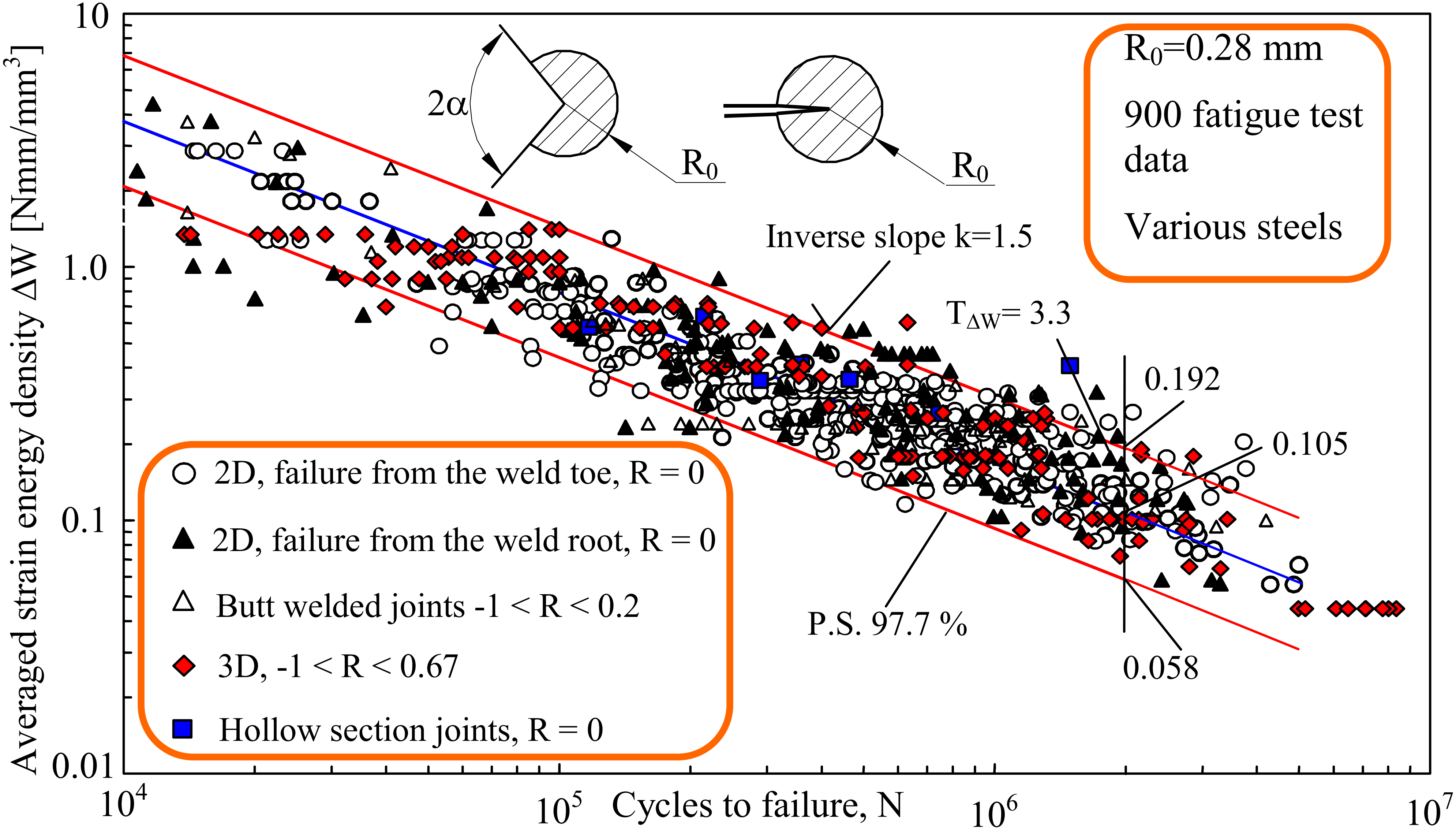
$$\Delta K_{1A}^{N} = 211 \text{ MPa mm}^{0.326}$$

$\Delta\sigma_A = 155$ MPa for butt ground welds made in structural steels (Atzori and Dattoma, 1982, Taylor, 2002).

Then for steel welded joints **$R_C = 0.28$ mm**

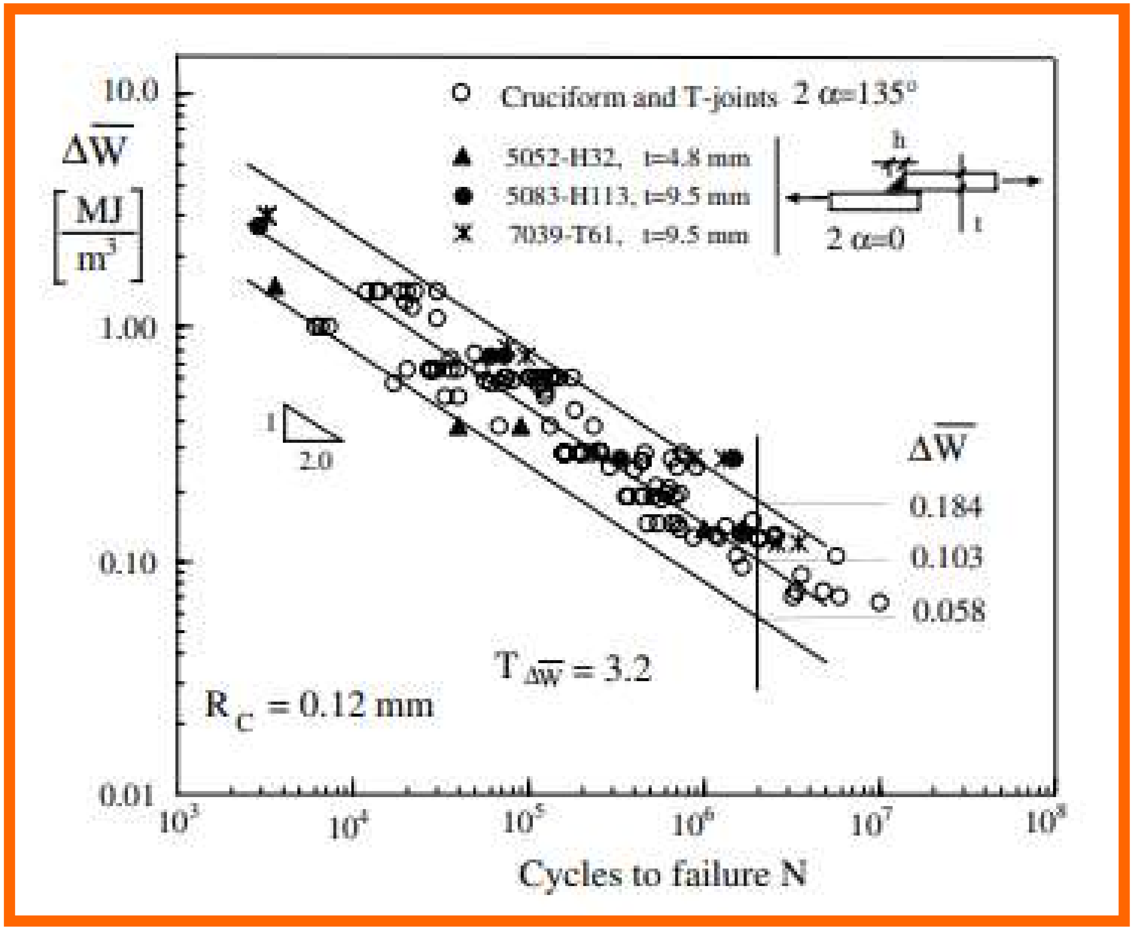
For aluminium welded joints **$R_C = 0.12$ mm**

AVERAGED SED AS A FATIGUE PARAMETER

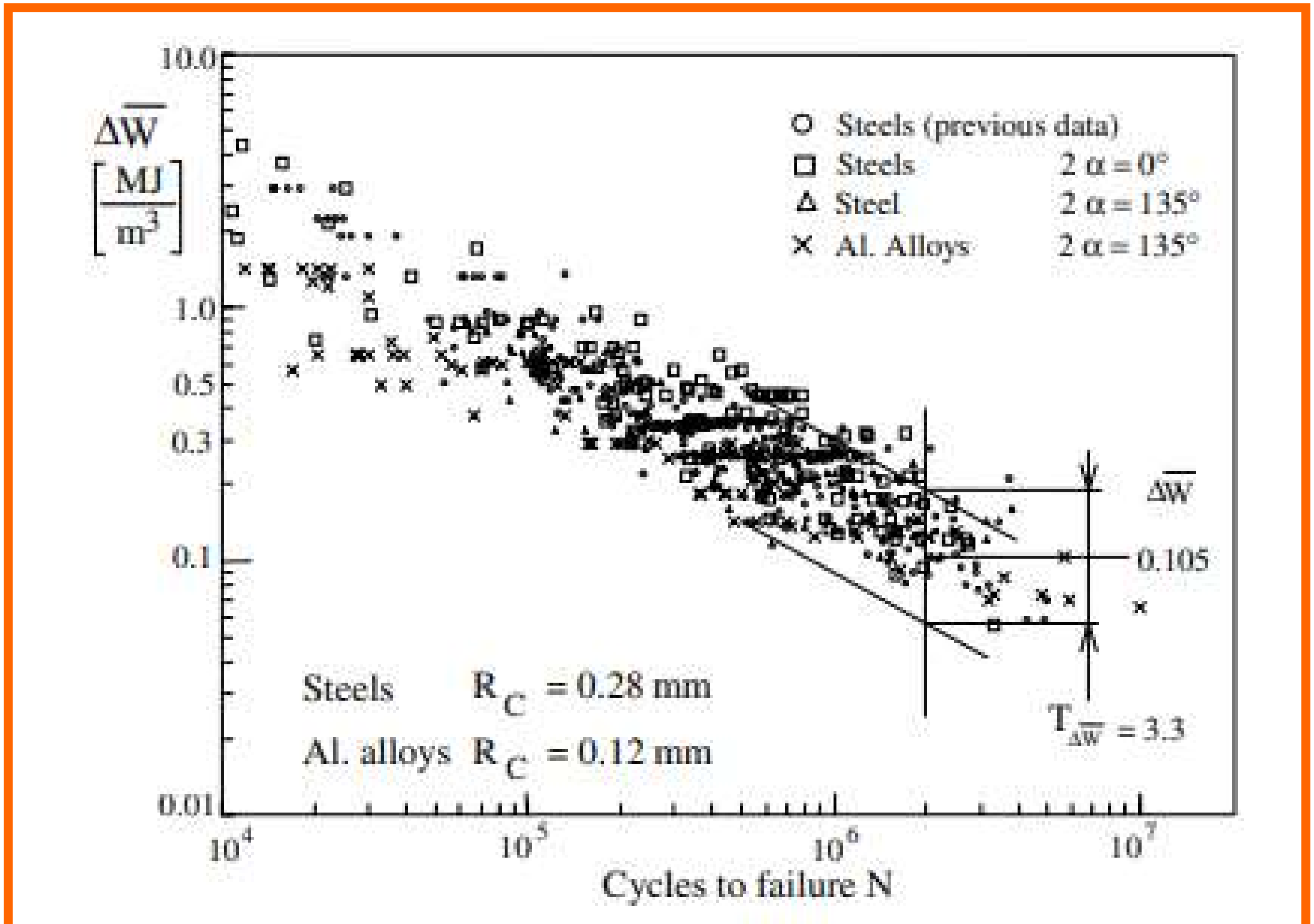


Fatigue strength of welded joints as a function of the averaged local strain energy density; *R* is the nominal load ratio (Berto and Lazzarin 2014)

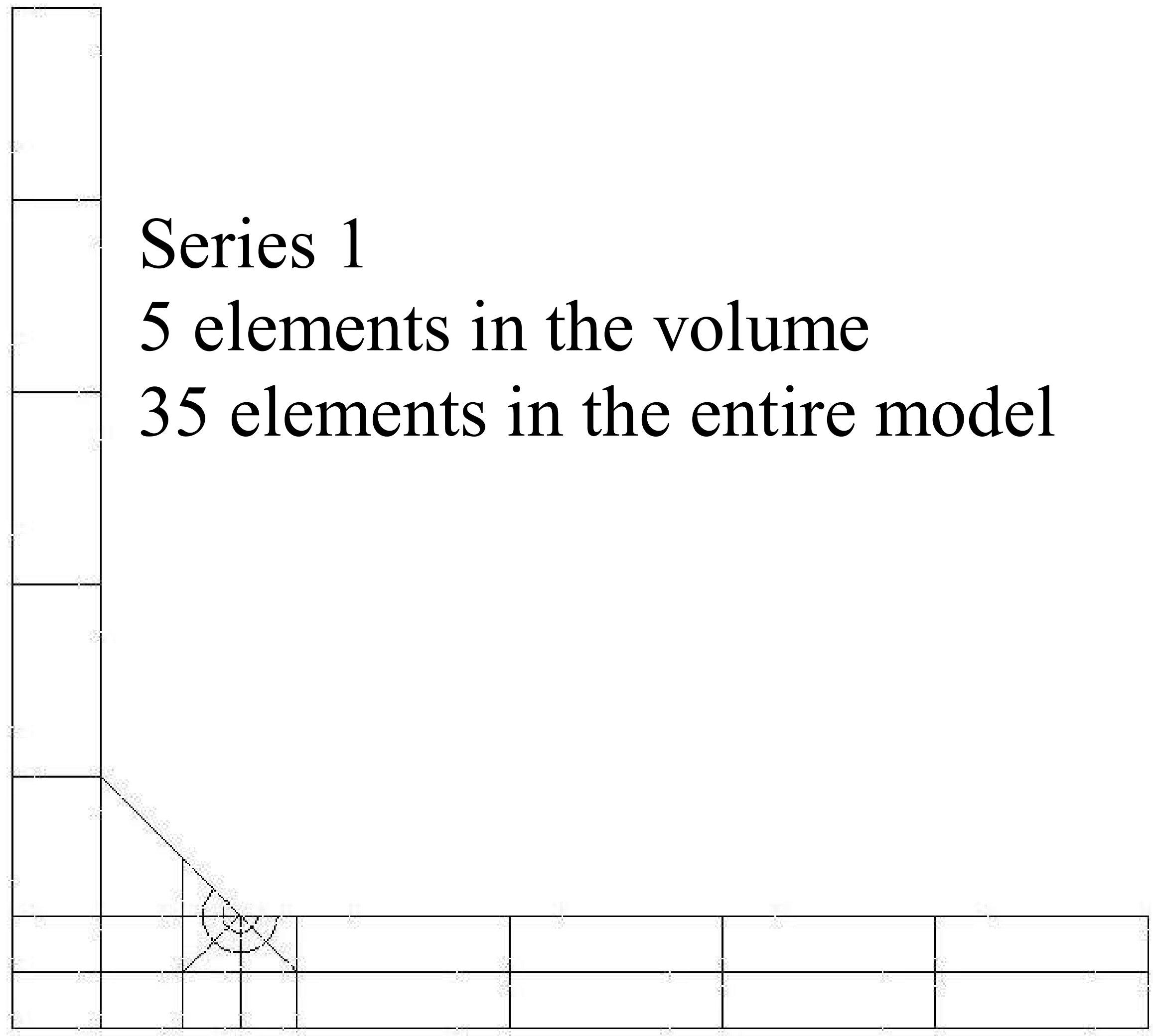
AVERAGED SED AS A FATIGUE PARAMETER



AVERAGED SED AS A FATIGUE PARAMETER



Coarse mesh: example

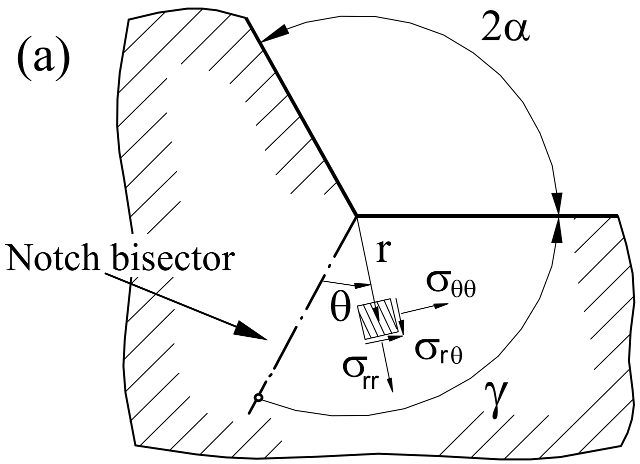


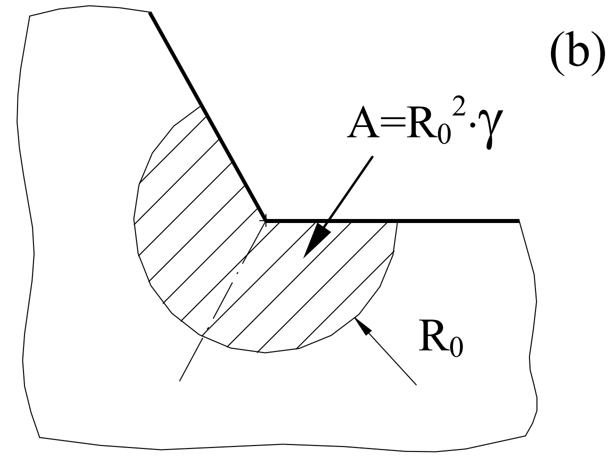
Series 1
5 elements in the volume
35 elements in the entire model

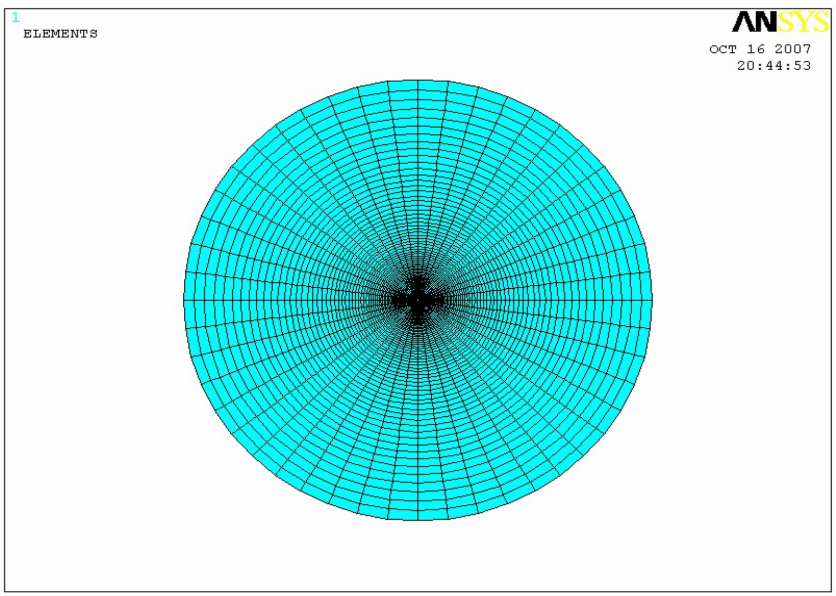
The SED can be accurately evaluated by using coarse meshes.
The NSIFs evaluation requires fine mesh with concentration keypoint.

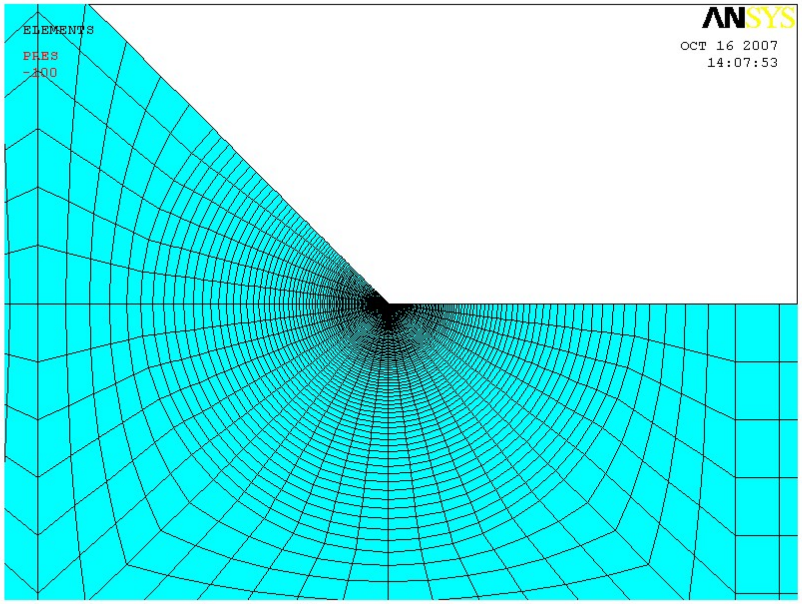
Lazzarin P., Berto et al. Int J Fatigue, 2008

FINE MESHES USUALLY USED FOR NSIFs EVALUATION

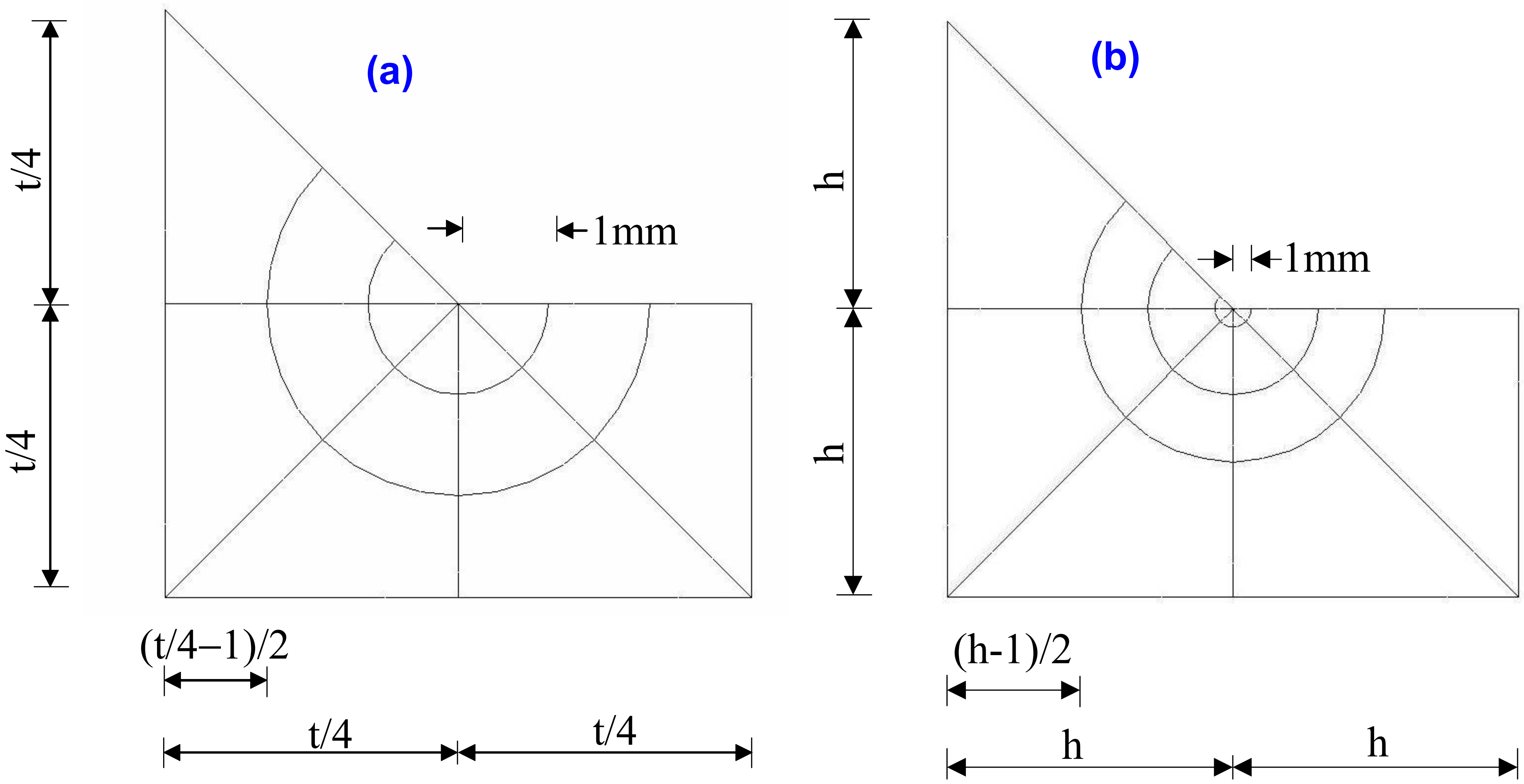






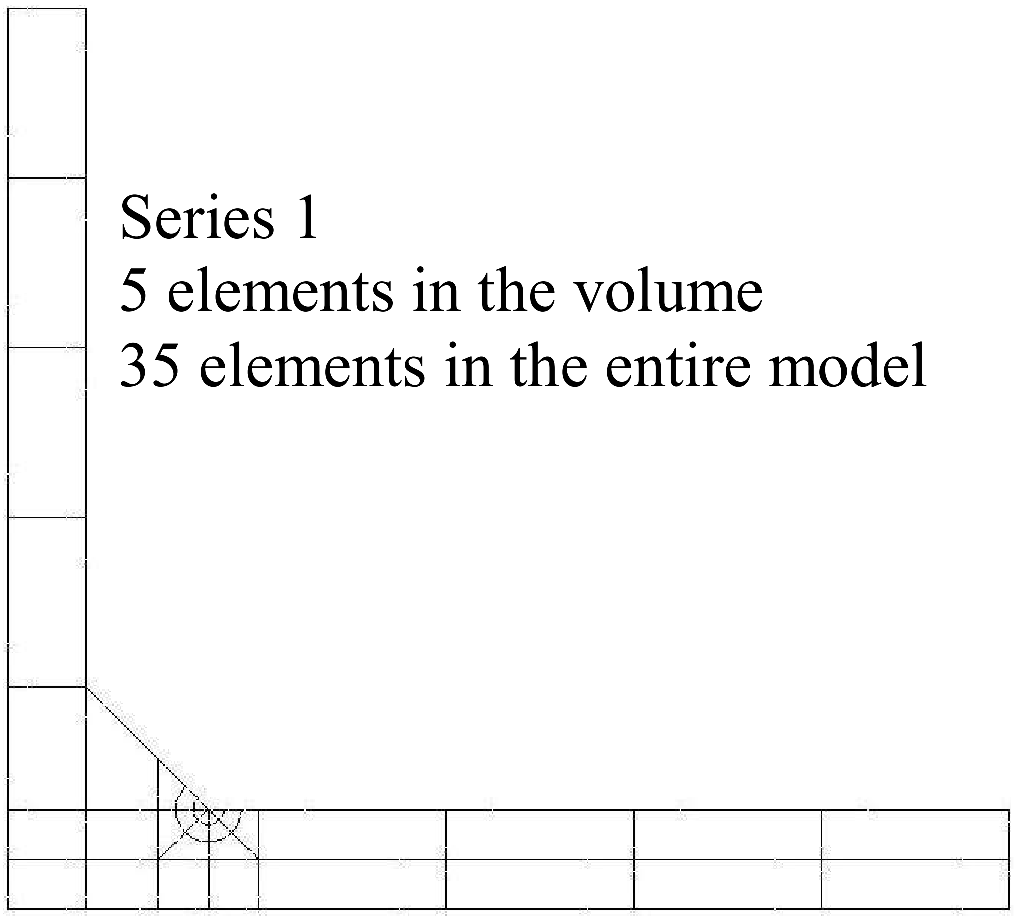


COARSE MESHES

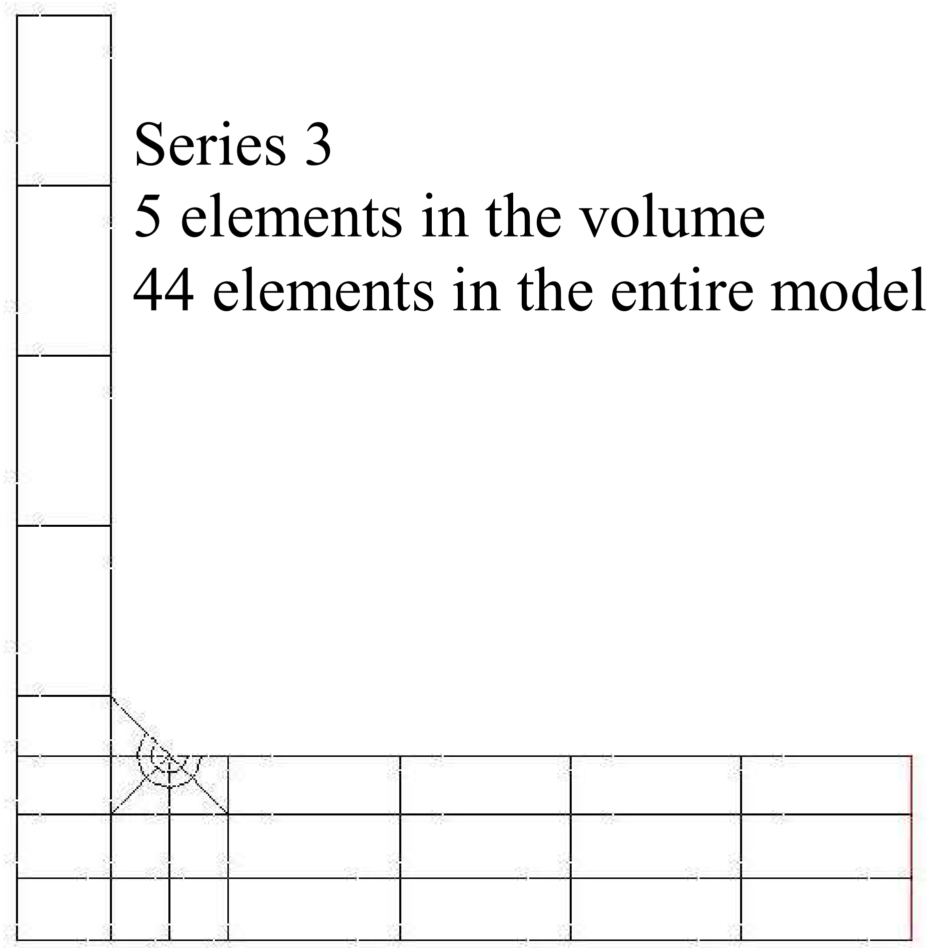


Modulus used for the geometries with *h*>*t*/2 (a); modulus used when *h*<*t*/2 (b)

EXAMPLE OF MODELS WITH COARSE MESHES



Series 1
5 elements in the volume
35 elements in the entire model



Series 3
5 elements in the volume
44 elements in the entire model

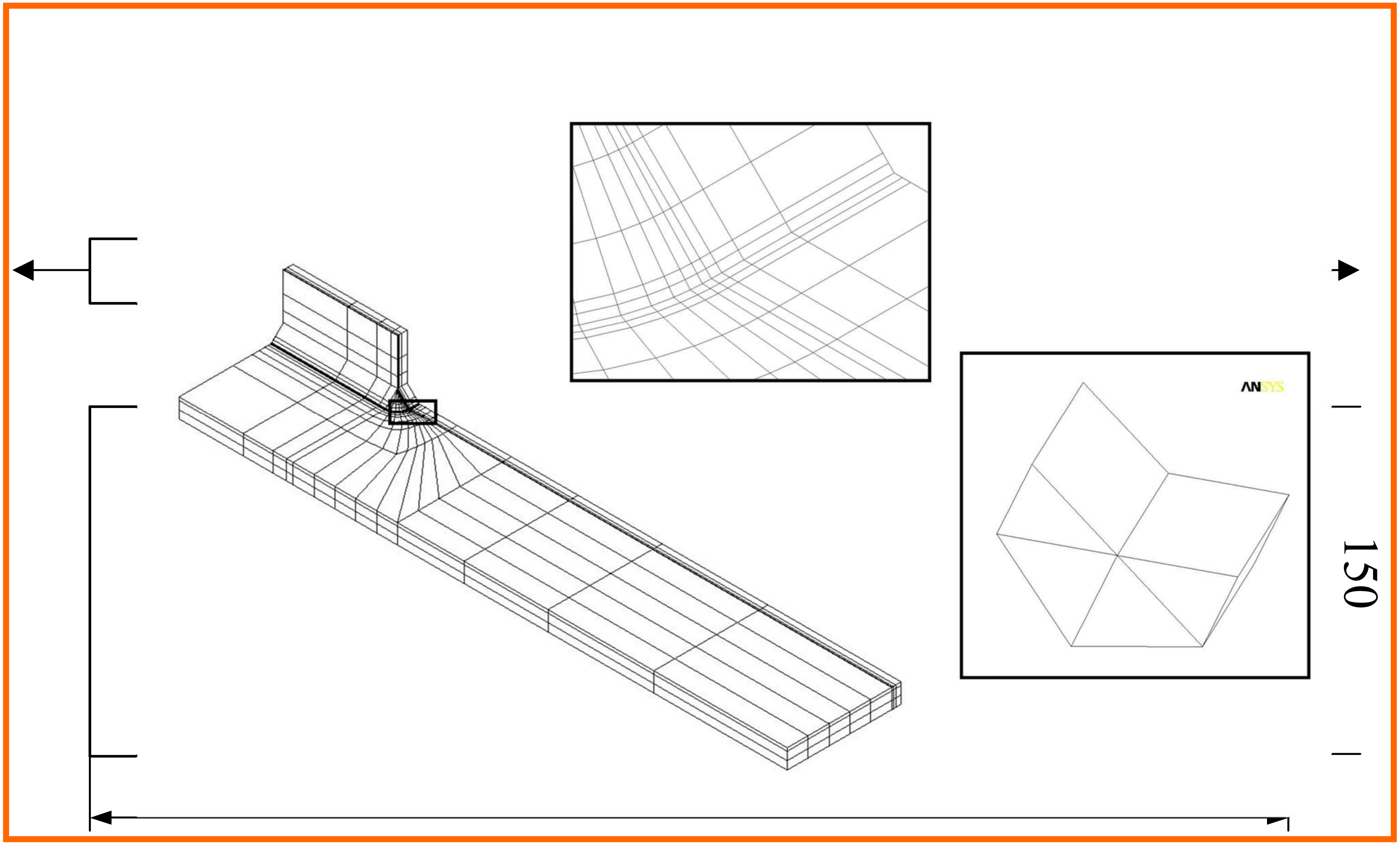
$$K_1 = (R^*)^{1-\lambda_1}\sqrt{\frac{E\,\overline{\overline{W}}}{e_1}} = 2.921\sqrt{E\,\overline{\overline{W}}}$$

$2\alpha=135°$, $1-\lambda_1=0.326$, $e_1= 0.1172$, $R^*=1.0$ mm

COMPARISON OF K_1 OBTAINED WITH FINE AND COARSE MESHES

Series	t [mm]	h [mm]	L [mm]	Fine mesh K_1 [MPa mm$^{0.326}$]	Parabolic FE (Coarse mesh) $\overline{W}$ [N mm/mm^3]	K_1 [MPa mm$^{0.326}$]	Δ %
1	13	8	10	265.0	4.28×10^{-2}	274.3	3.5
2	50	16	50	396.	9.07×10^{-2}	399.3	0.7
3	100	16	50	413.0	9.94×10^{-2}	417.9	1.2
4	13	5	3	228.8	3.25×10^{-2}	238.9	4.4
5	13	10	8	267.5	4.23×10^{-2}	272.8	2.0
6	25	5	3	231.0	3.32×10^{-2}	241.6	4.6
7	25	9	32	329.5	6.11×10^{-2}	327.7	-0.5
8	25	15	220	405.0	9.08×10^{-2}	399.4	-1.4
9	38	8	13	296.7	5.21×10^{-2}	302.5	2.0
10	38	15	220	476.0	1.25×10^{-1}	469.0	-1.5
11	100	5	3	228.1	3.28×10^{-2}	240.2	5.3
12	100	15	220	589.5	1.87×10^{-1}	573.0	-2.8

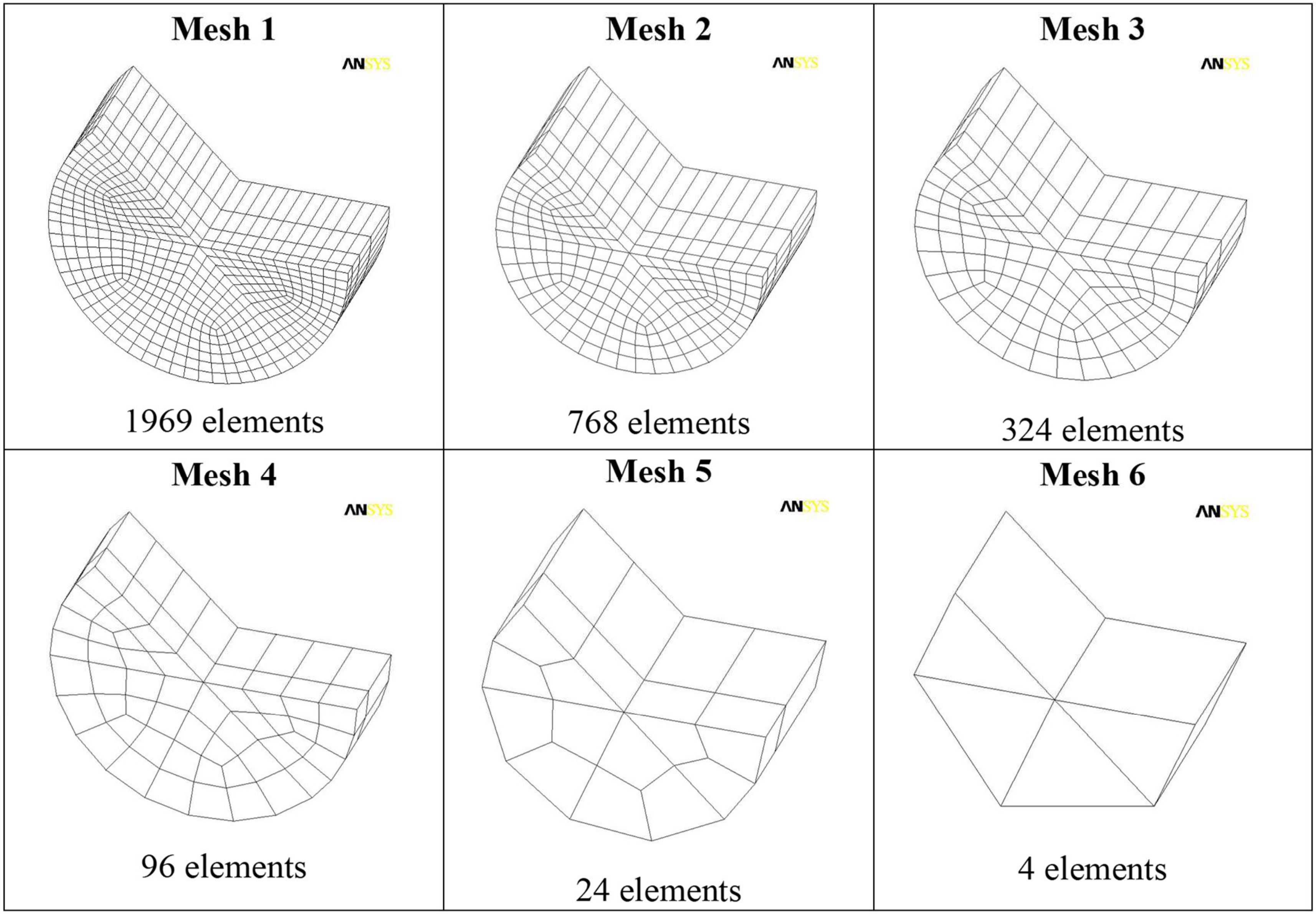
THREE DIMENSIONAL MODELS



Geometry of the welded joints with a longitudinal stiffener tested by Maddox

Maddox SJ. Influence of tensile residual stresses on the fatigue behavior of welded joints in steel. ASTM STP. 1982; 776: 63-96.

DIFFERENT MESHES FOR THREE DIMENSIONAL MODELS



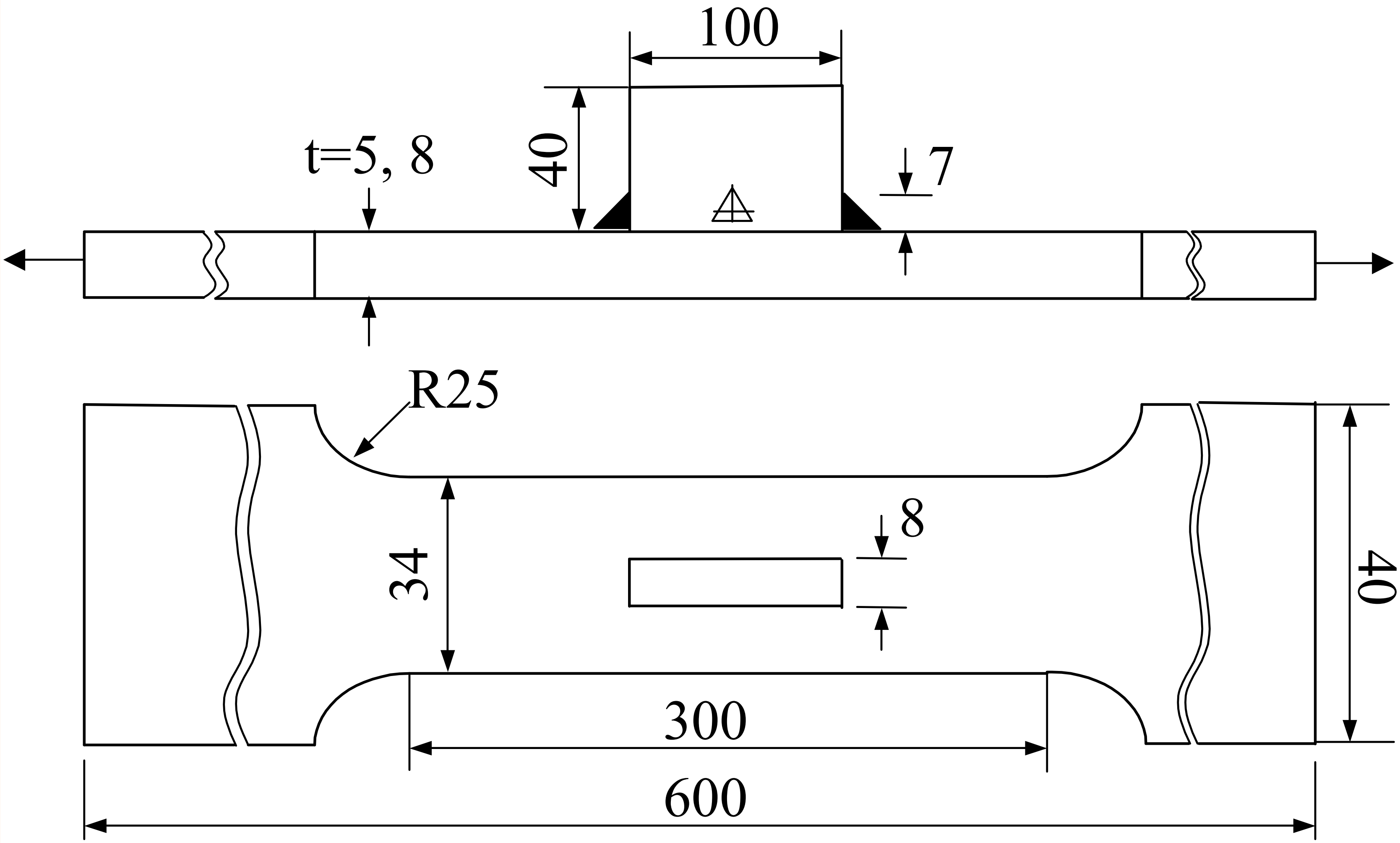
DIFFERENT MESHES FOR THREE DIMENSIONAL MODELS

3D models	Number of FE in the volume	Degrees of freedom (complete model)	$\overline{W}$ Nmm/mm^3	K_1 [MPa mm$^{0.326}$]	Δ%
1	1696	$8.6\cdot10^5$	0.07937	373.5	0
2	768	$4.6\cdot10^5$	0.07903	372.7	0.21
3	324	$2.5\cdot10^5$	0.07896	372.5	0.26
4	96	$1.7\cdot10^5$	0.07895	372.5	0.26
5	24	$4.5\cdot10^4$	0.07790	370.0	0.93
6	4	$1.1\cdot10^4$	0.07594	365.3	2.18

THREE DIMENSIONAL MODELS

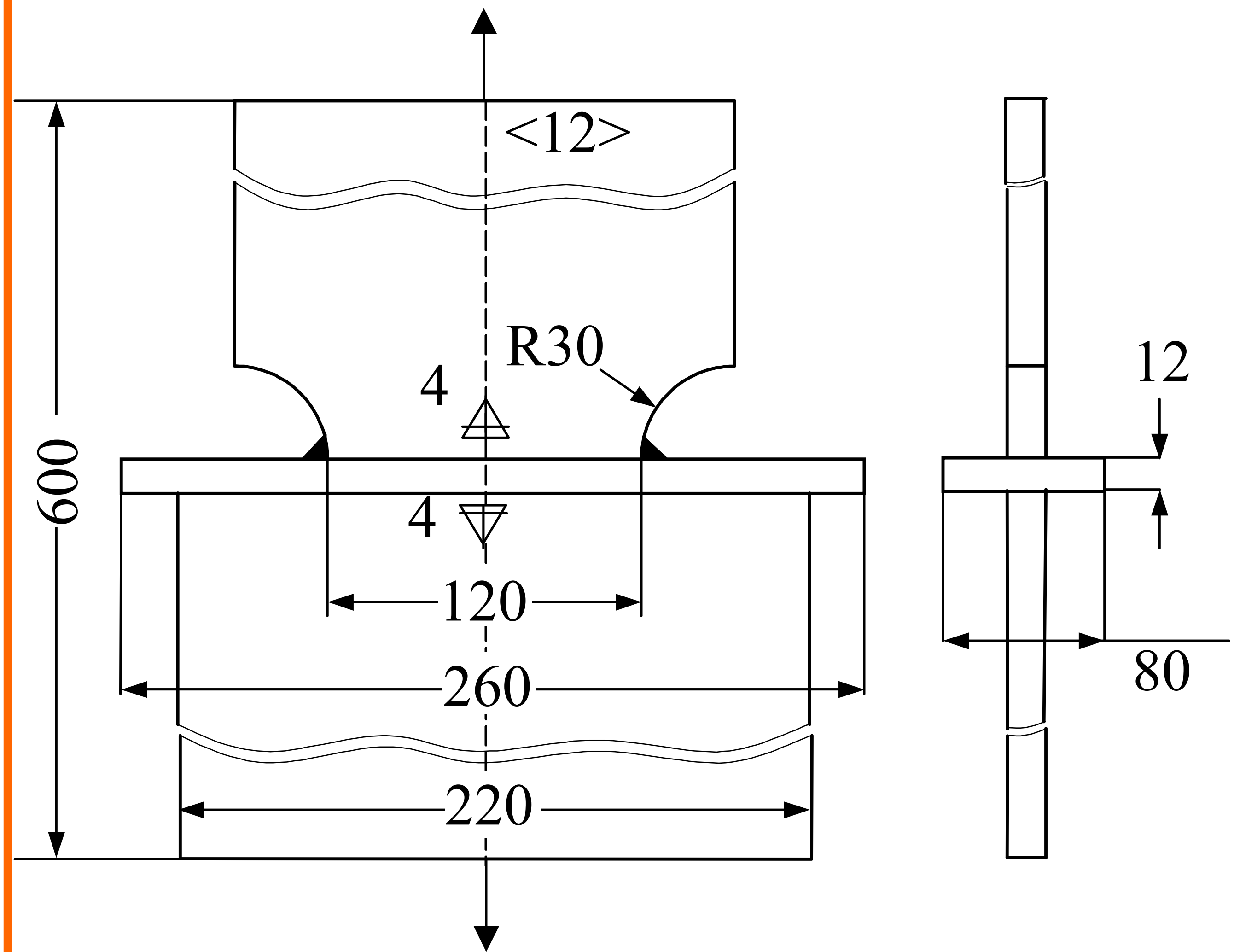
Lihavainen and Marquis, 2003

S355 steel



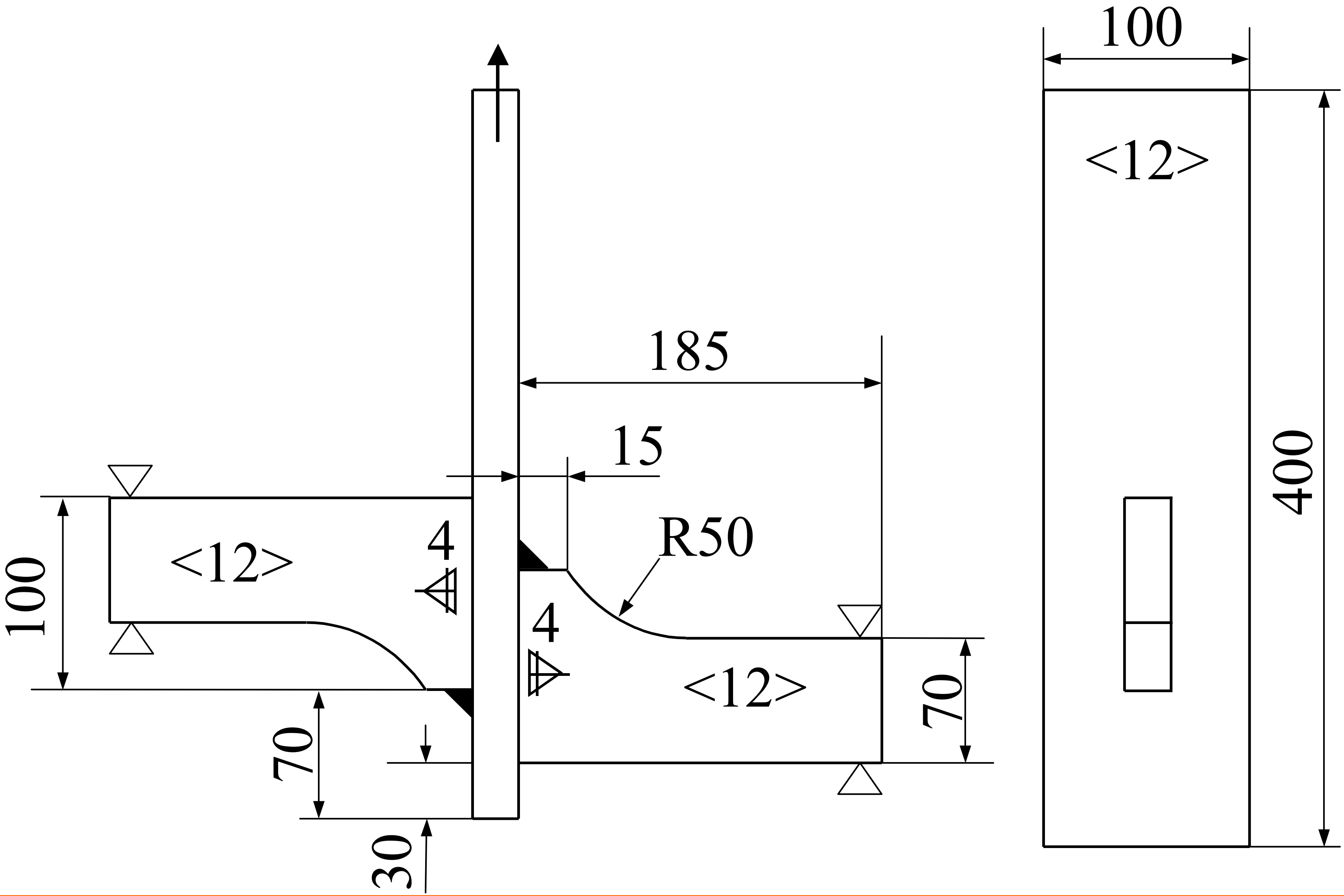
THREE DIMENSIONAL MODELS

Fricke and Doerk, 2006

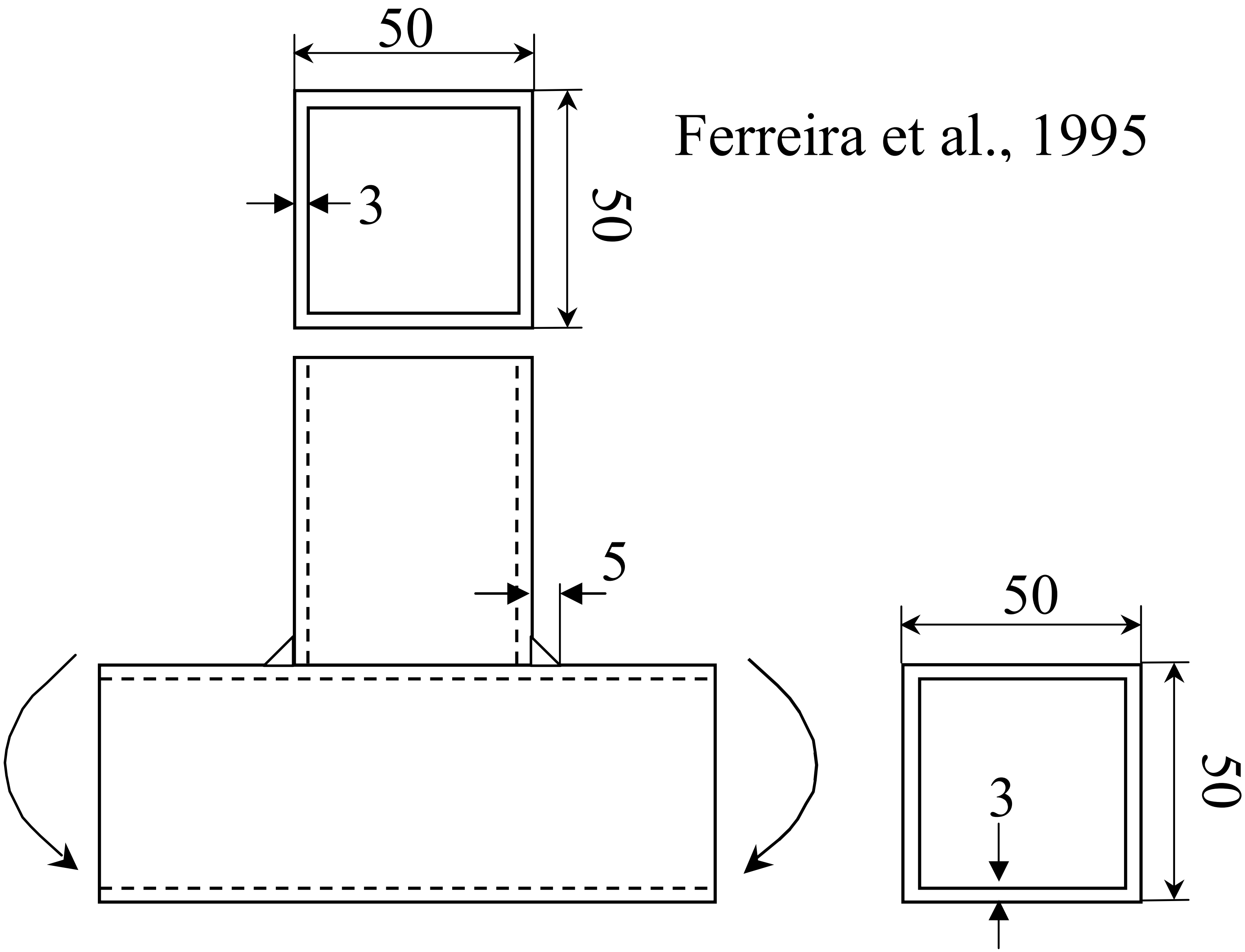


THREE DIMENSIONAL MODELS

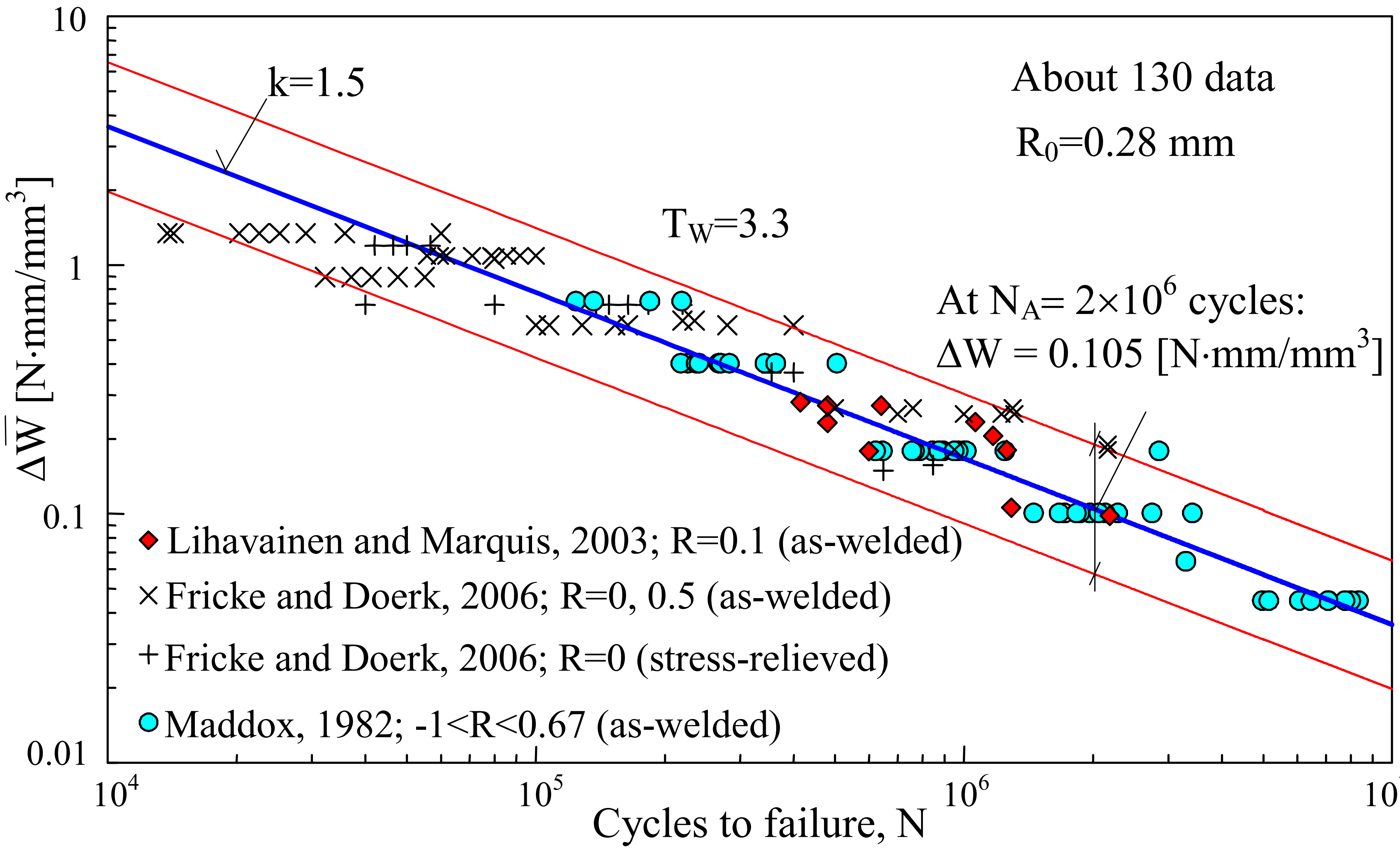
Fricke and Doerk, 2006



THREE DIMENSIONAL MODELS



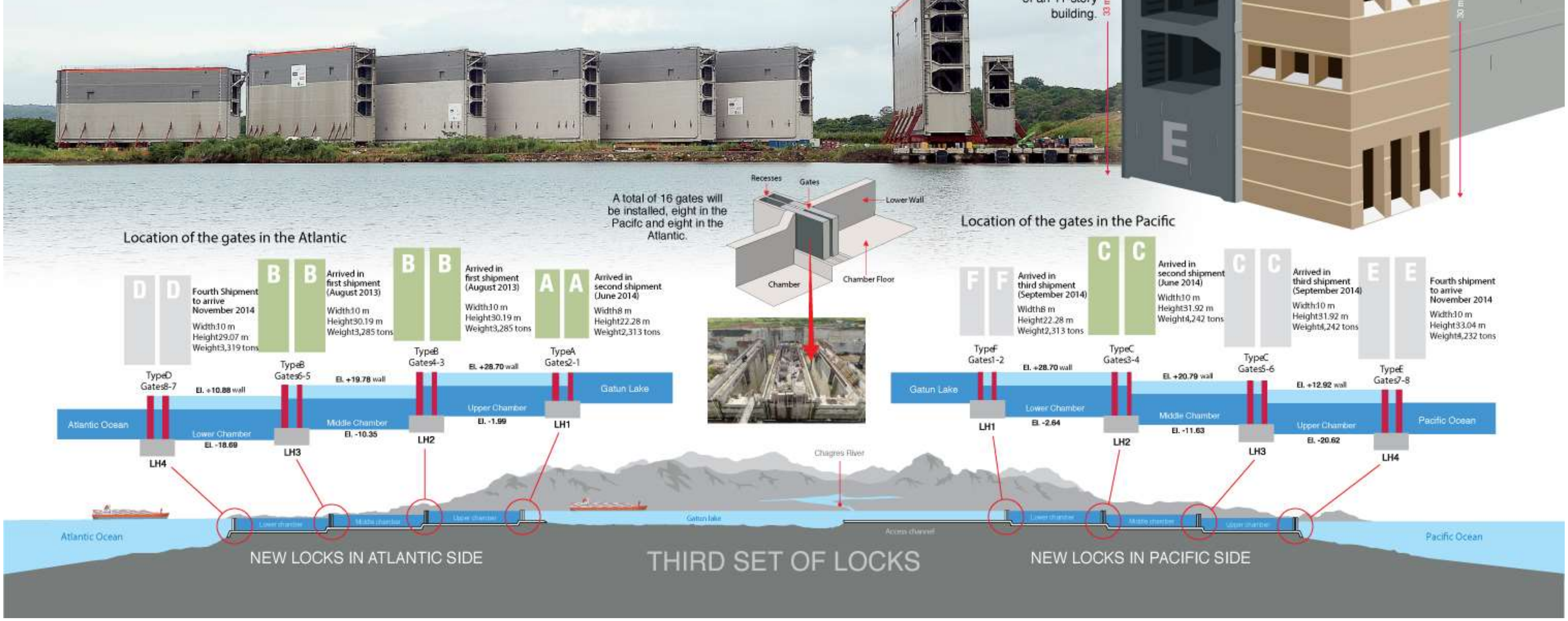
SYNTHESIS BASED ON SED



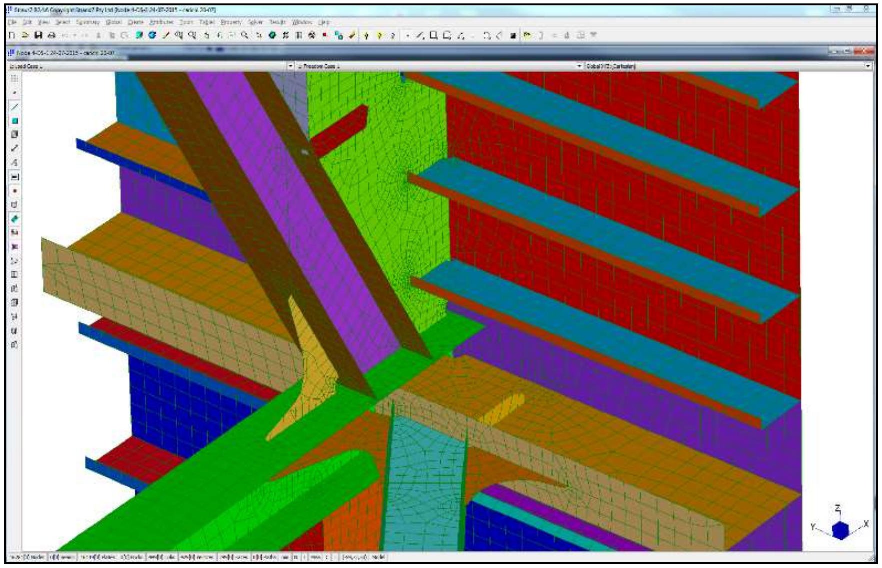
Fatigue strength of the welded joints made of structural steel in terms of local strain energy density; comparison with the scatter band previously proposed

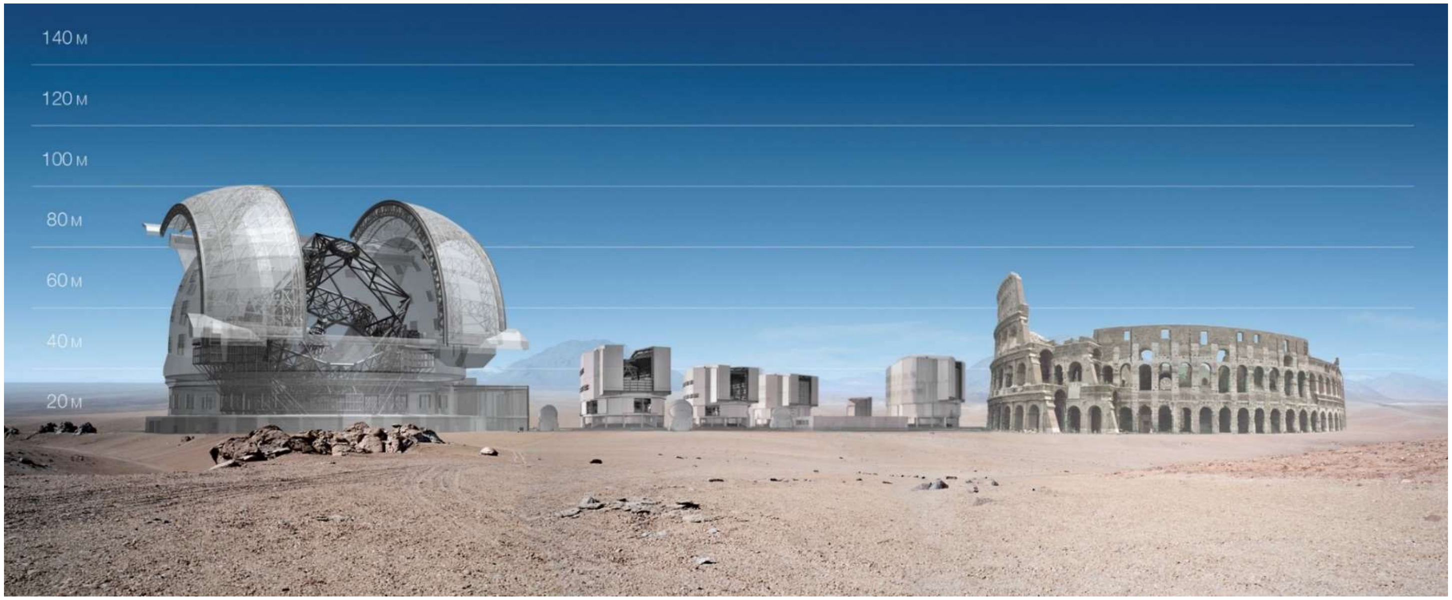
STEELGIANTS

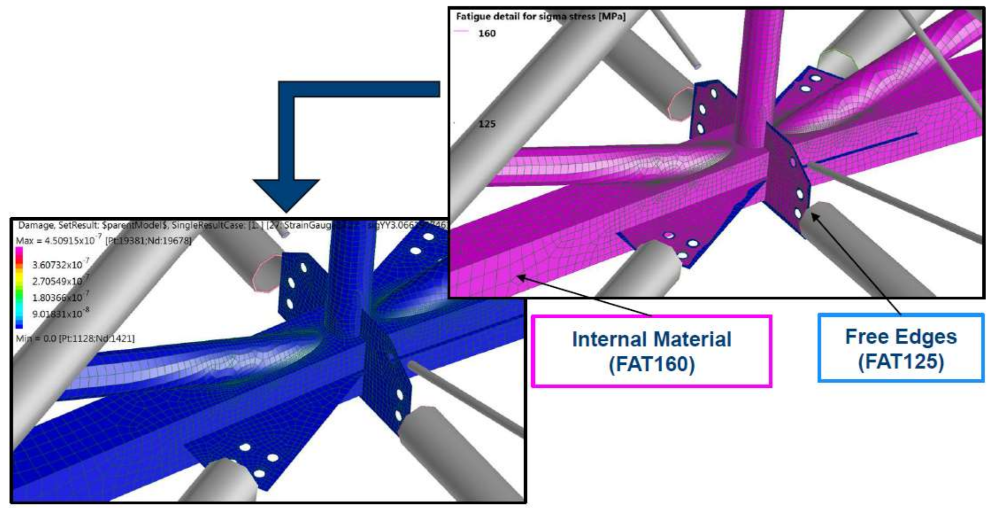
The new locks of the Panama Canal will have 16 rolling gates, fabricated in Italy by Cimolai SpA. The gates have been arriving to Panama since August 2013 in staggered shipments of four at a time. The rolling system facilitates gate maintenance.

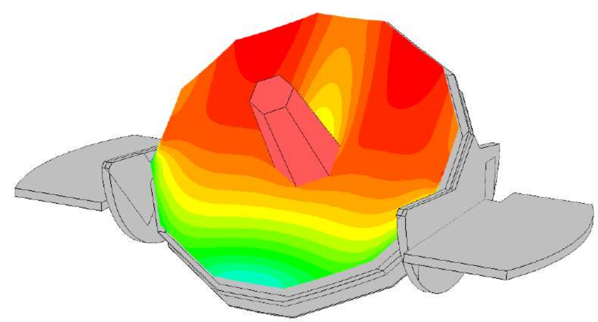


About 250.000 welds x 16 different gates + 1.300.000 notches

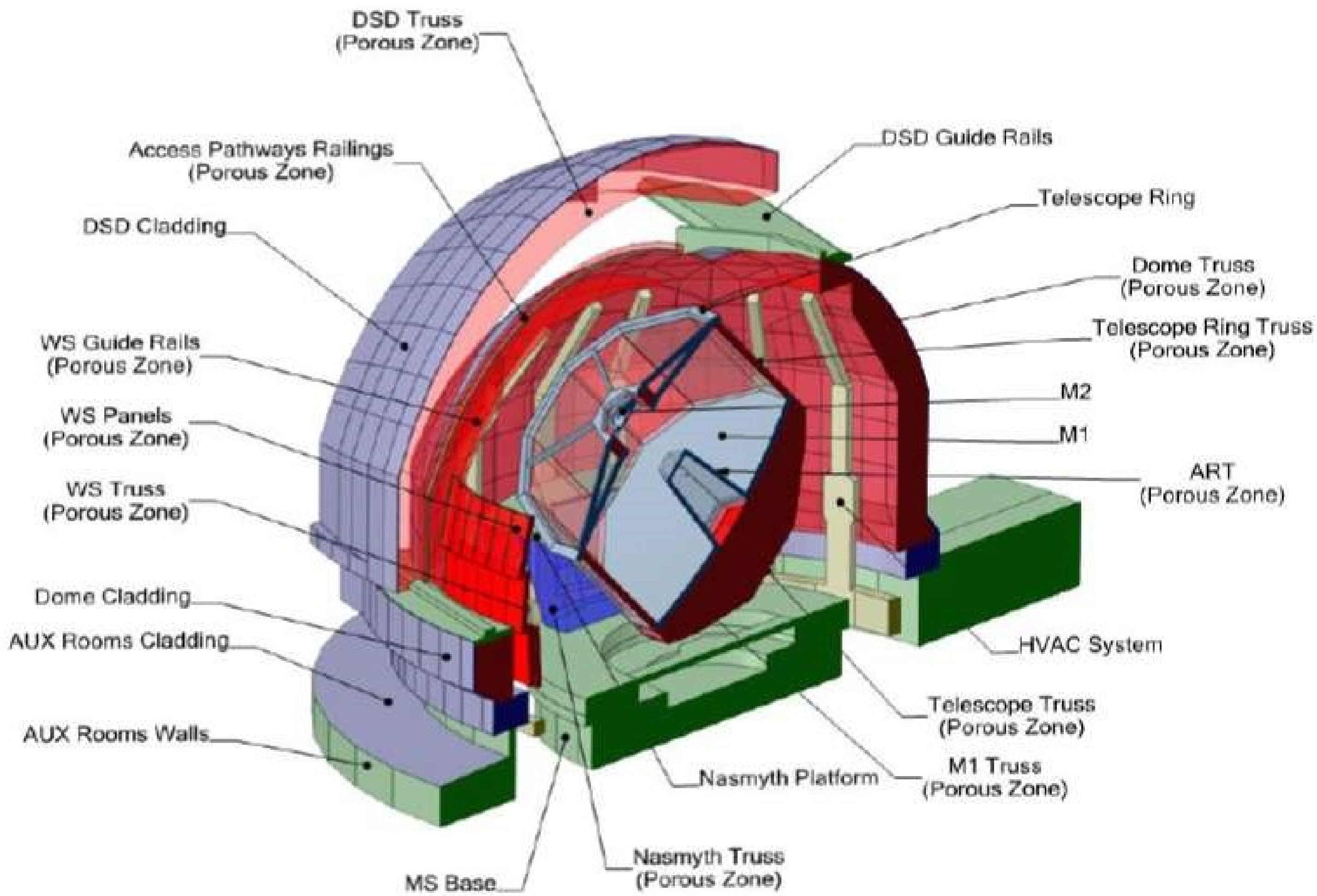



140 M
120 M
100 M
80 M
60 M
40 M
20 M


Fatigue detail for sigma stress [MPa]
160
125
Max = 4.50915x10-7 [Pt:19381;Nd:19678]
3.60732x10-7
2.70549x10-7
1.80366x10-7
9.01831x10-8
Min = 0.0 [Pt:1128;Nd:1421]
Internal Material (FAT160)
Free Edges (FAT125)



Fluent Geometric Model

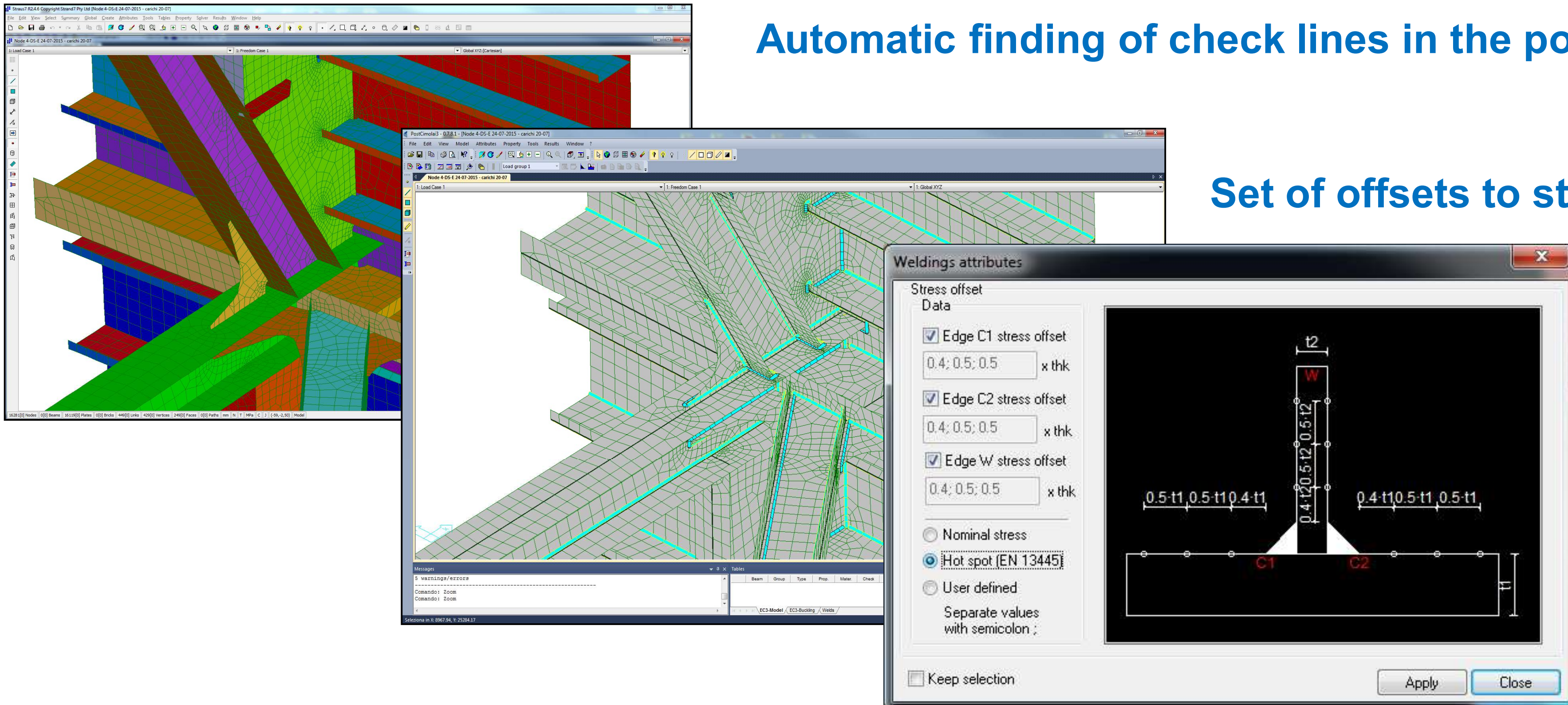


Automatic calculation of the fatigue strength

Definition of a shell finite element model

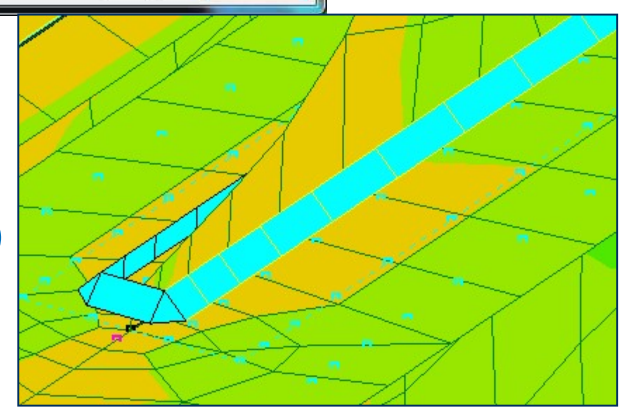
Automatic finding of check lines in the postprocessor

Set of offsets to stress reading



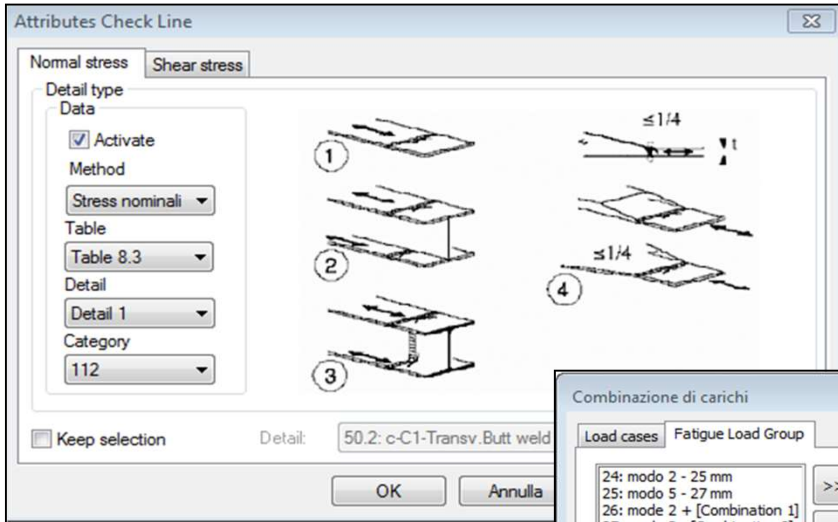
<u>OFFSET STRESS READING:</u>

✓Nominal stress: default 1.5t
(DVS 1612 and Hobbacher)
✓Hot-Spot stress: default 0.4t;0.5t;0.5t (EN 13445)
✓User defined (SED and other local approaches)

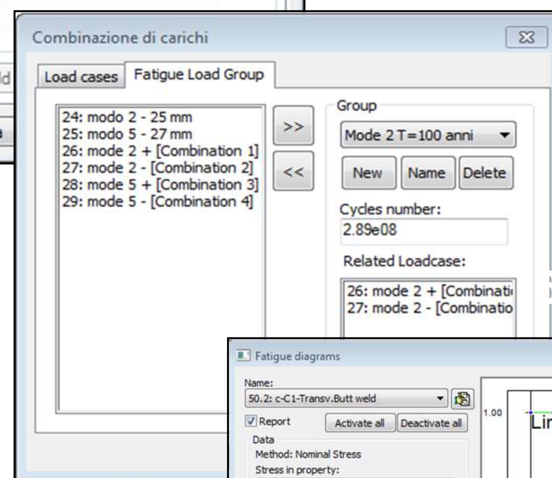


Automatic calculation of the fatigue strength

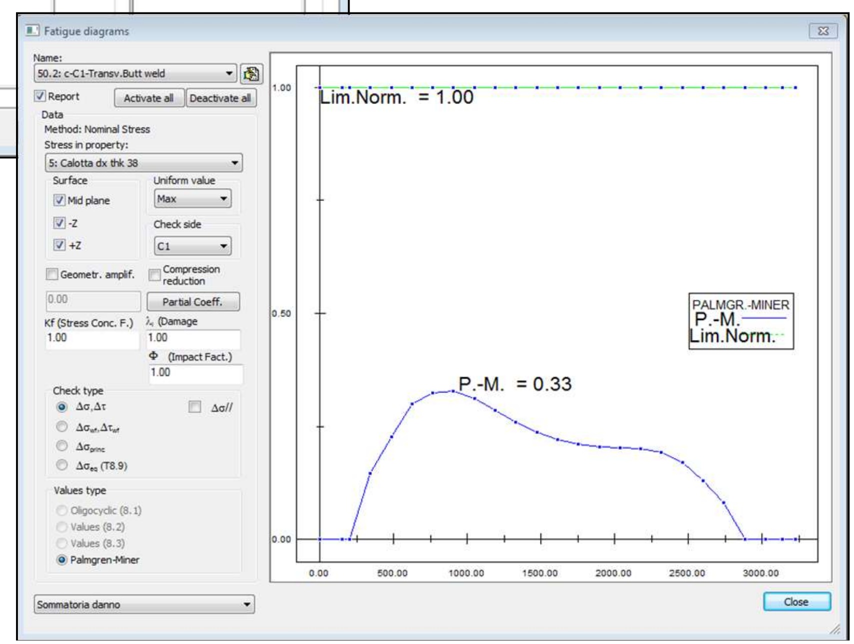
Set the fatigue class details



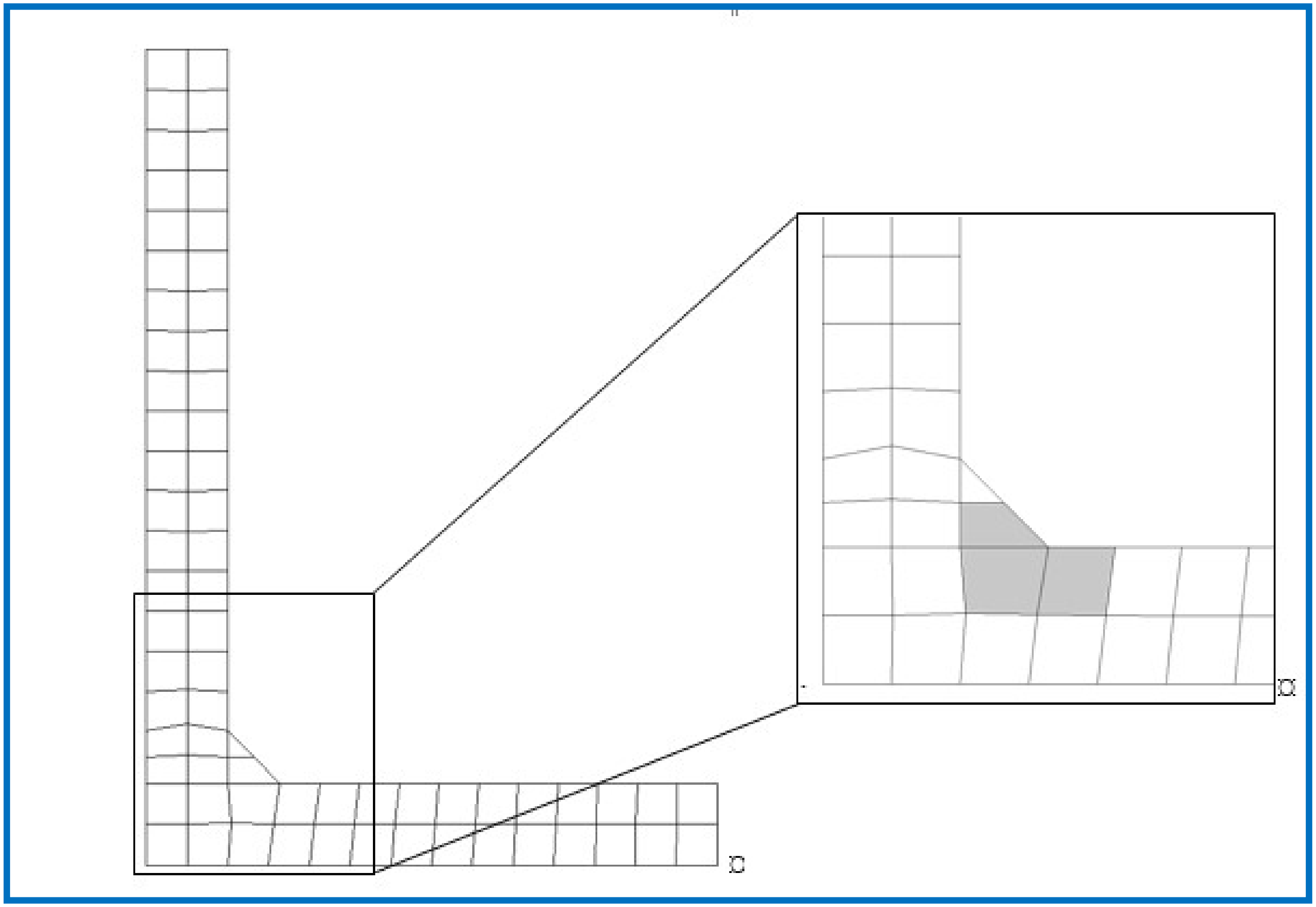
Load spectrum



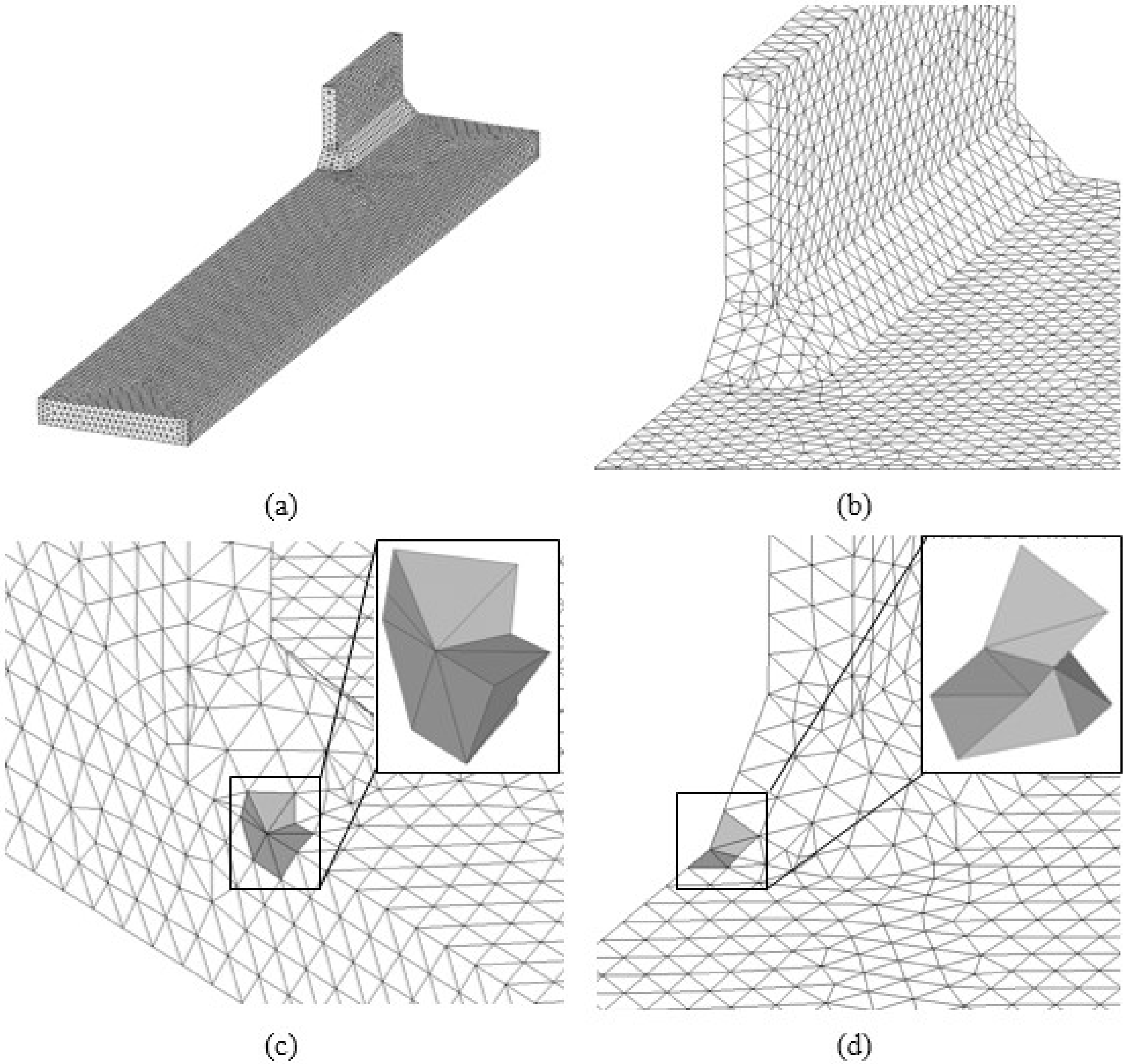
Fatigue check



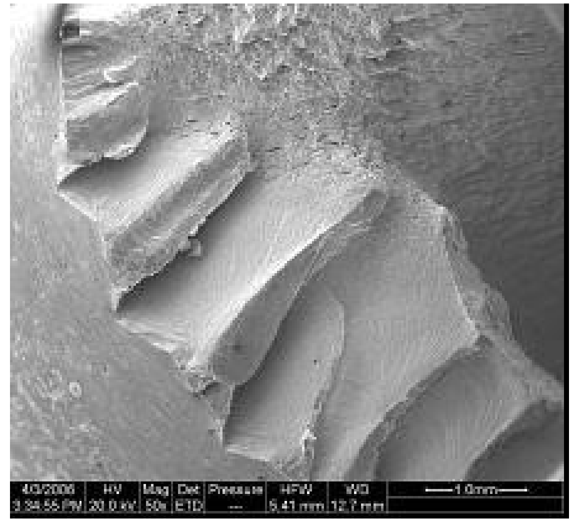
SED Volume Free

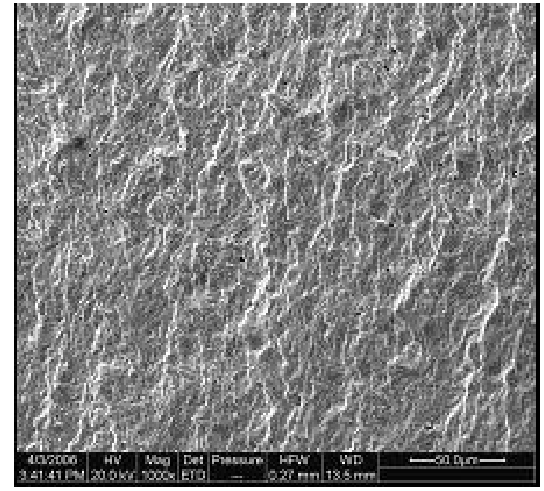


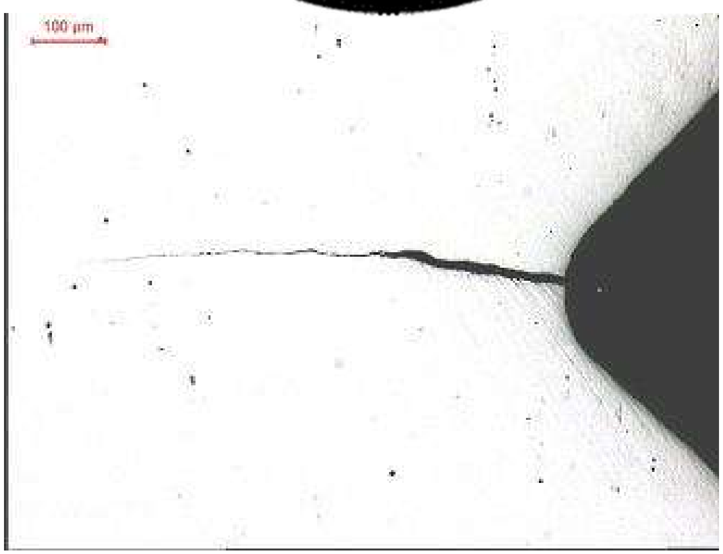
SED Volume Free

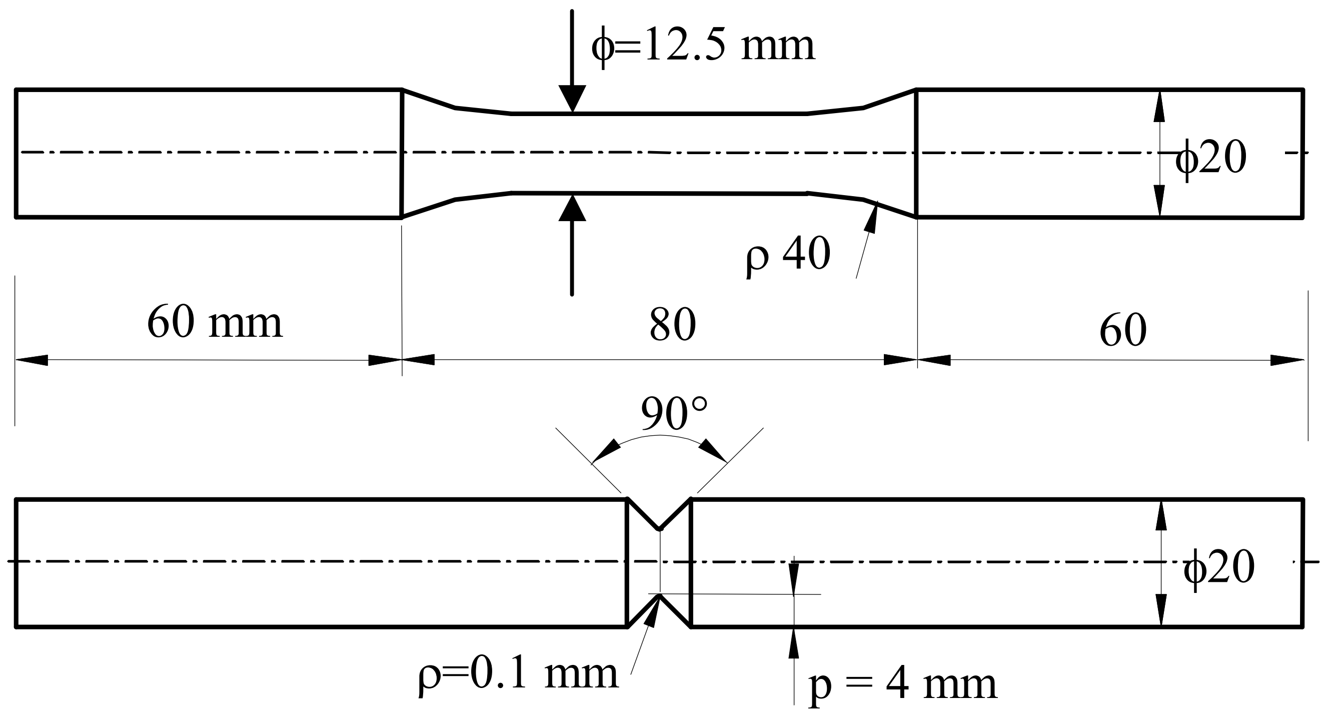


Multiaxial Fatigue









SOME RECENT TESTS (Berto, Lazzarin, Yates, 2010)

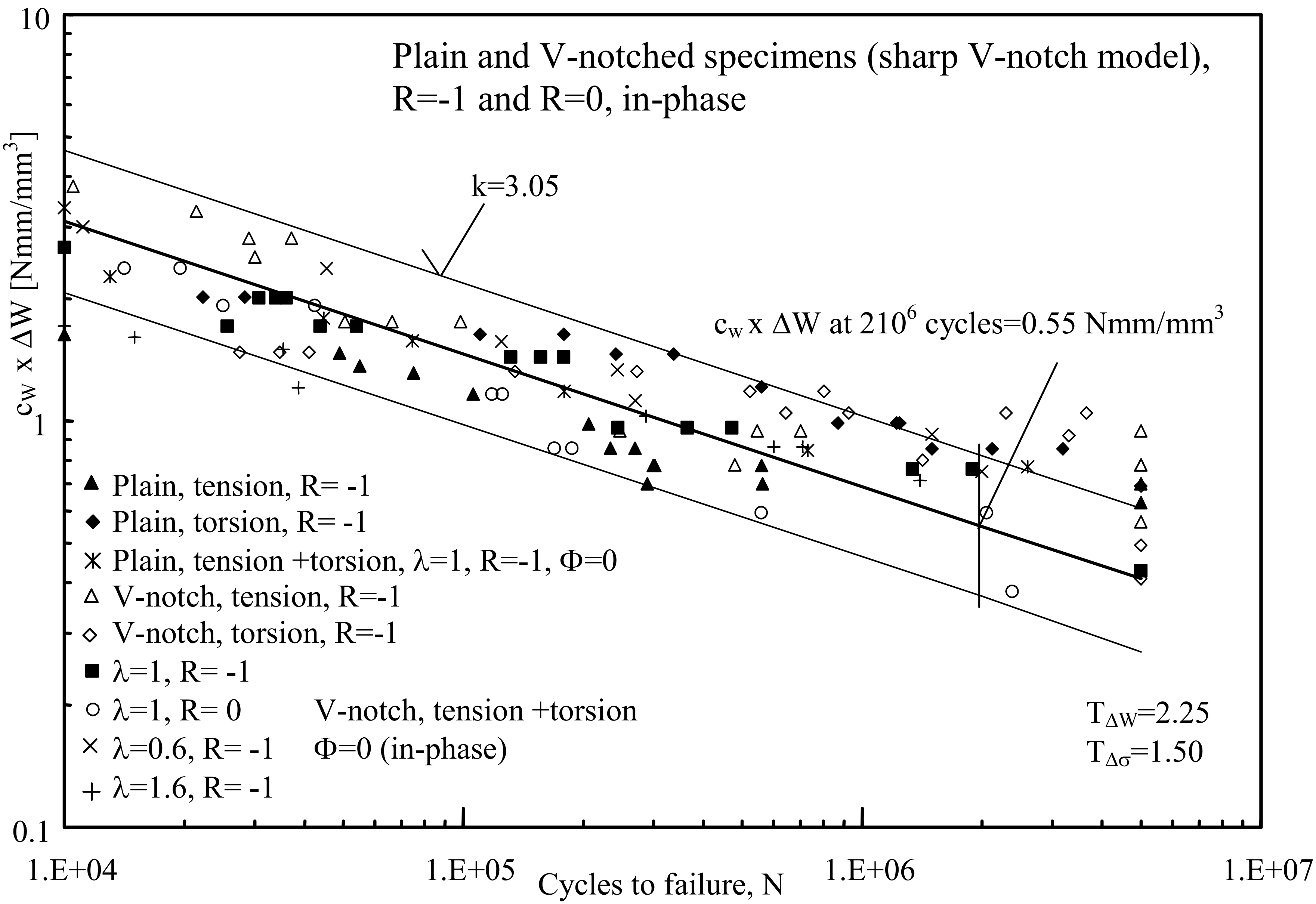
MATERIAL 39NiCrMo3 hardened and tempered state

Multiaxial

	Series	No Specim.	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=10^6$	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=2\times10^6$	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=5\times10^6$	k	T_τ
Tension	A	15	346.90	(315.10)	–	7.21	1.26
Torsion	B	13	285.34	265.30	240.96	9.52	1.18
Combined tension and torsion ($\lambda=1$)	C	6	221.76	205.15	185.08	8.90	1.23

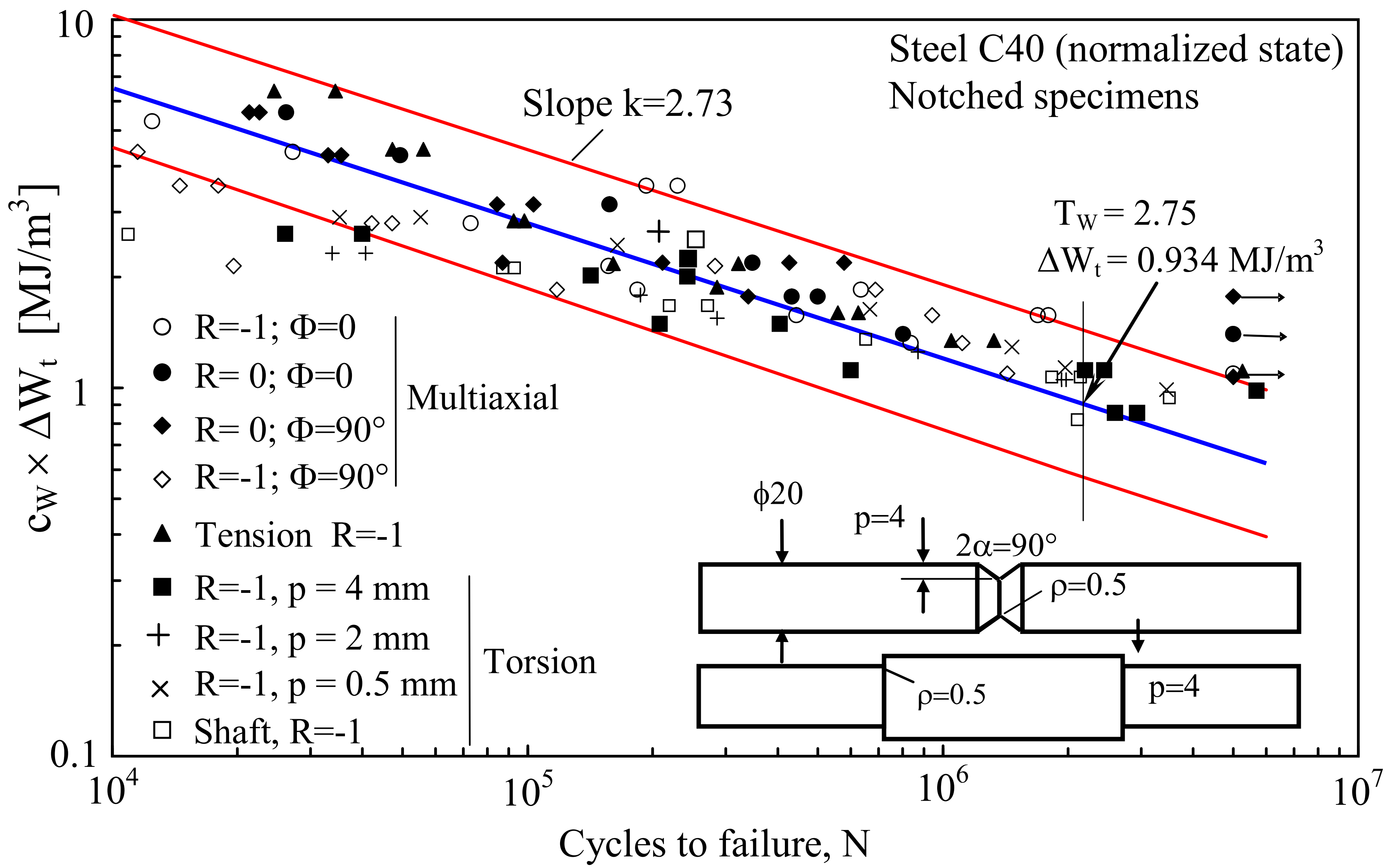
Loading	Series	No Spec.	R	λ	Φ	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=10^6$	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=2\cdot10^6$	$\sigma_{A50\%}$ [MPa] ($\tau_{A50\%}$ [MPa]) at $N=5\cdot10^6$	k	T_σ
Tension	D	16	-1			180.97	(157.14)	–	4.91	1.36
Torsion	E	16	-1			309.17	293.55	274.10	13.37	1.28
Multi-axial	F	16	-1	1	0	163.87	149.63	132.67	7.62	1.22
	G	11	-1	1	90°	128.27	117.07	103.76	7.59	1.22
	H	11	0	1	0	95.02	83.79	70.95	5.51	1.41
	I	16	0	1	90°	95.89	88.98	80.61	9.27	1.32
	L	8	-1	0.6	0	197.97	179.73	158.17	7.17	1.26
	M	8	-1	1.6	0	130.79	123.17	113.76	11.54	1.18

Multiaxial



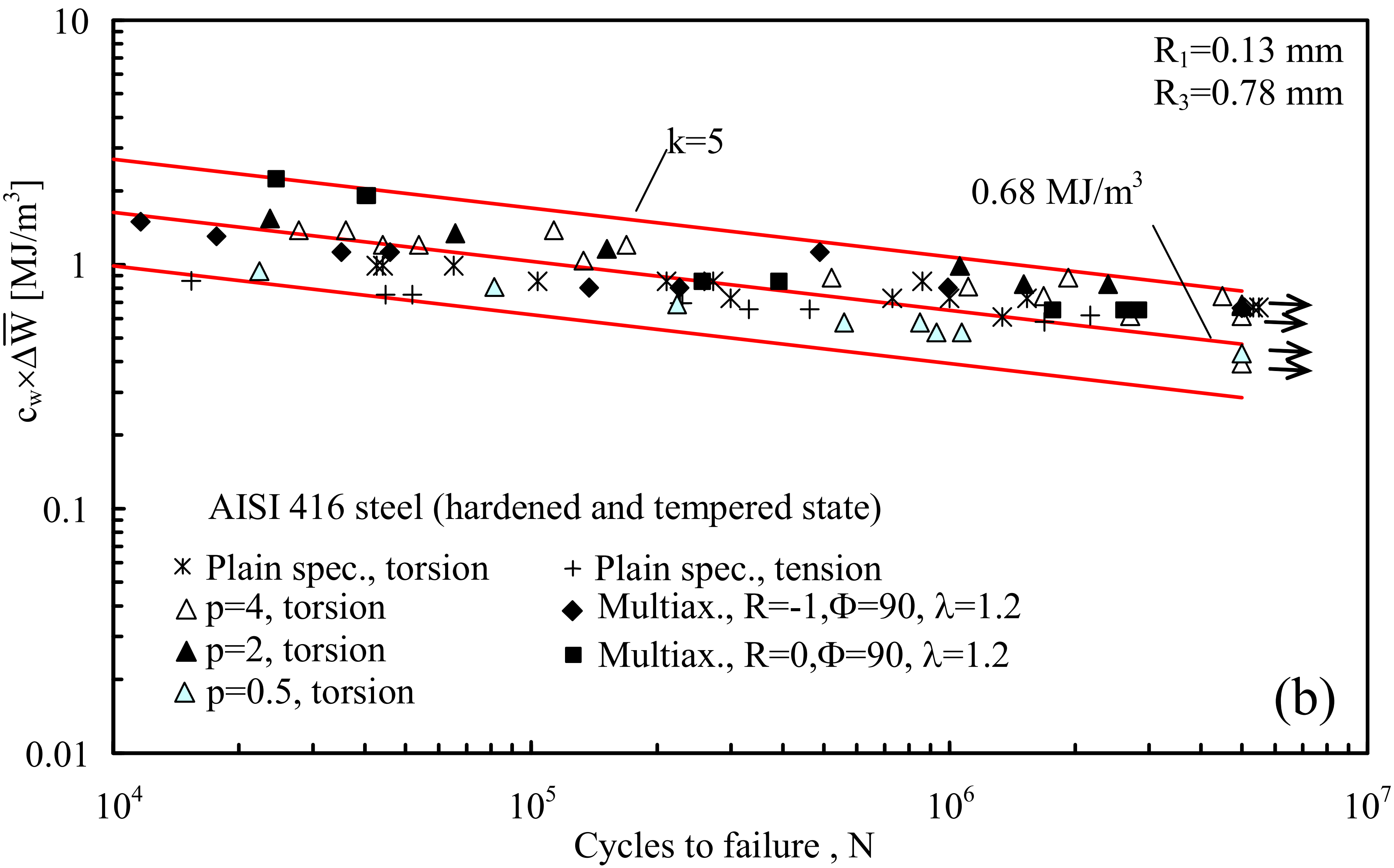
Plain and V-notched specimens (sharp V-notch model),
R=-1 and R=0, in-phase
k=3.05
c_W x ΔW at 210[6] cycles=0.55 Nmm/mm[3]
c_W x ΔW [Nmm/mm[3]]
10
1
0.1
1.E+04
1.E+05
1.E+06
1.E+07
Cycles to failure, N
Plain, tension, R= -1
Plain, torsion, R= -1
Plain, tension +torsion, λ=1, R=-1, Φ=0
V-notch, tension, R=-1
V-notch, torsion, R=-1
λ=1, R= -1
λ=1, R= 0
λ=0.6, R= -1
λ=1.6, R= -1
V-notch, tension +torsion
Φ=0 (in-phase)
$T_{\Delta W}$=2.25
$T_{\Delta\sigma}$=1.50

Multiaxial



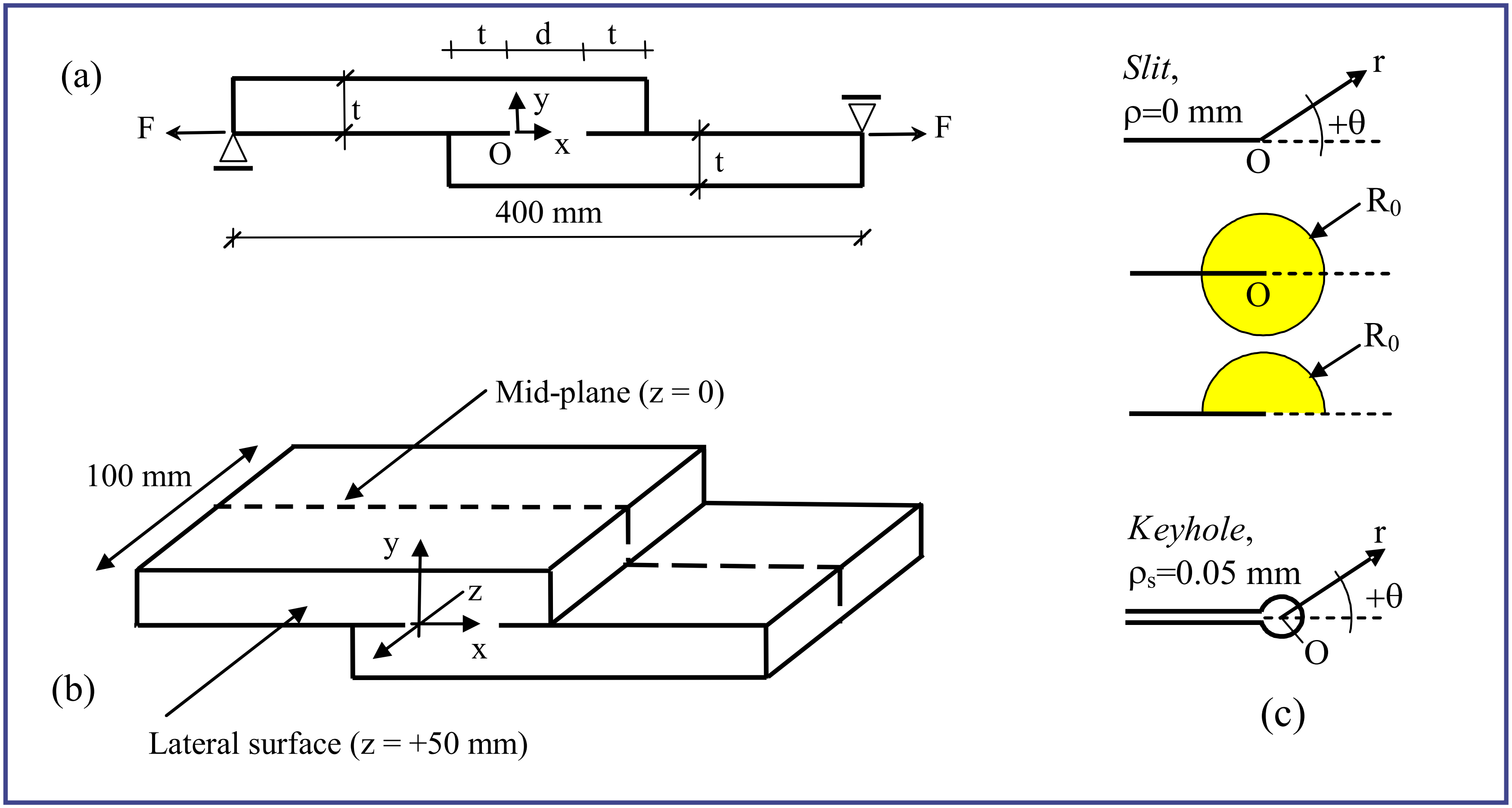
Steel C40 (normalized state)
Notched specimens
Slope k=2.73
$T_W = 2.75$
$\Delta W_t = 0.934\ MJ/m^3$
R=-1; Φ=0
R= 0; Φ=0
R= 0; Φ=90°
R=-1; Φ=90°
Multiaxial
Tension R=-1
R=-1, p = 4 mm
R=-1, p = 2 mm
R=-1, p = 0.5 mm
Shaft, R=-1
Torsion
ϕ20
p=4
2α=90°
ρ=0.5
ρ=0.5
p=4
$c_W \times \Delta W_t\ [MJ/m^3]$
10
1
0.1
10^4
10^5
10^6
10^7
Cycles to failure, N

Multiaxial



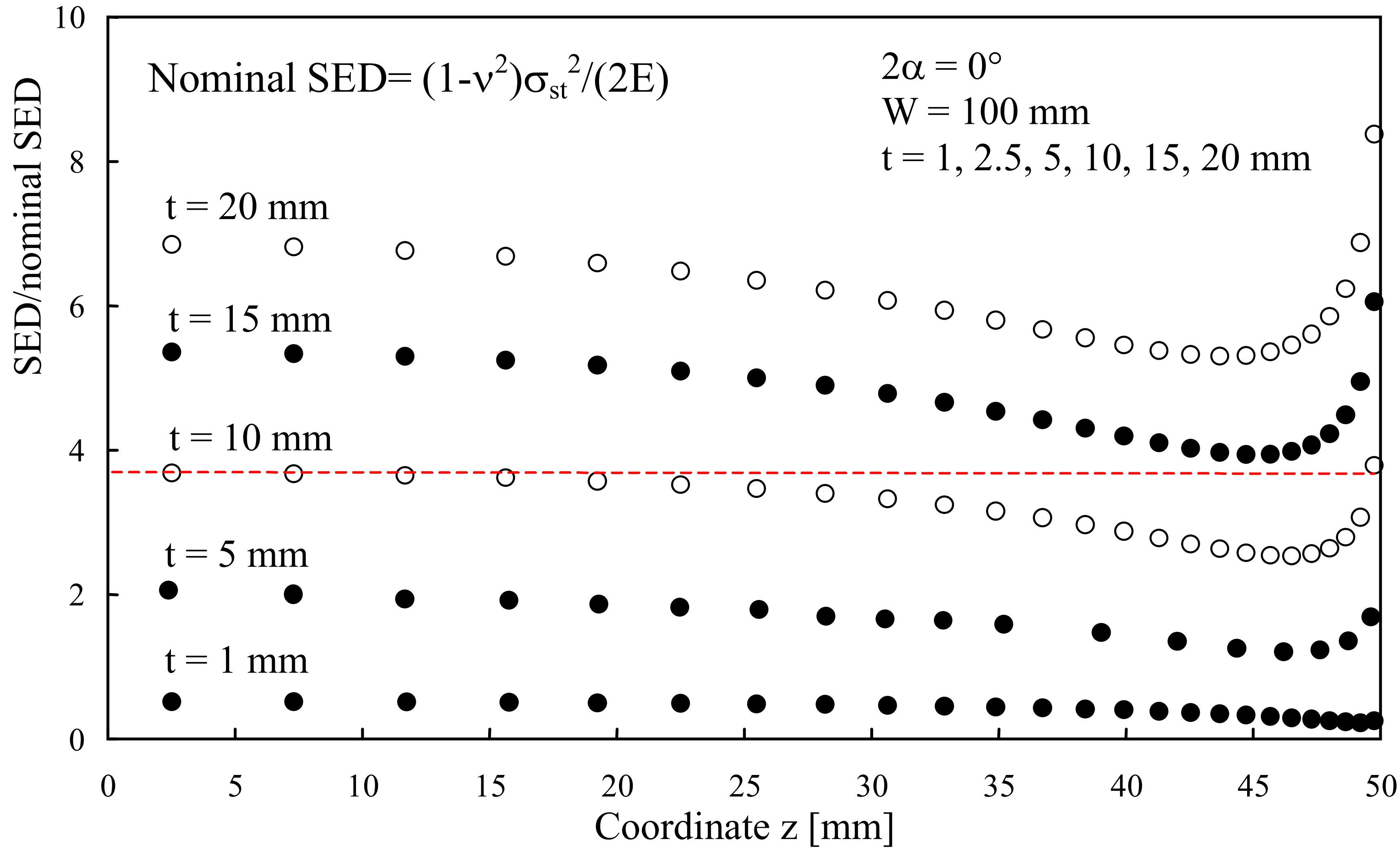
10
1
0.1
0.01
$c_w \times \Delta\overline{W}$ [MJ/m^3]
10^4
10^5
10^6
10^7
Cycles to failure , N
R_1=0.13 mm
R_3=0.78 mm
k=5
0.68 MJ/m^3
AISI 416 steel (hardened and tempered state)
Plain spec., torsion
p=4, torsion
p=2, torsion
p=0.5, torsion
Plain spec., tension
Multiax., R=-1,Φ=90, λ=1.2
Multiax., R=0,Φ=90, λ=1.2
(b)

'OUT-OF-PLANE' SINGULARITY

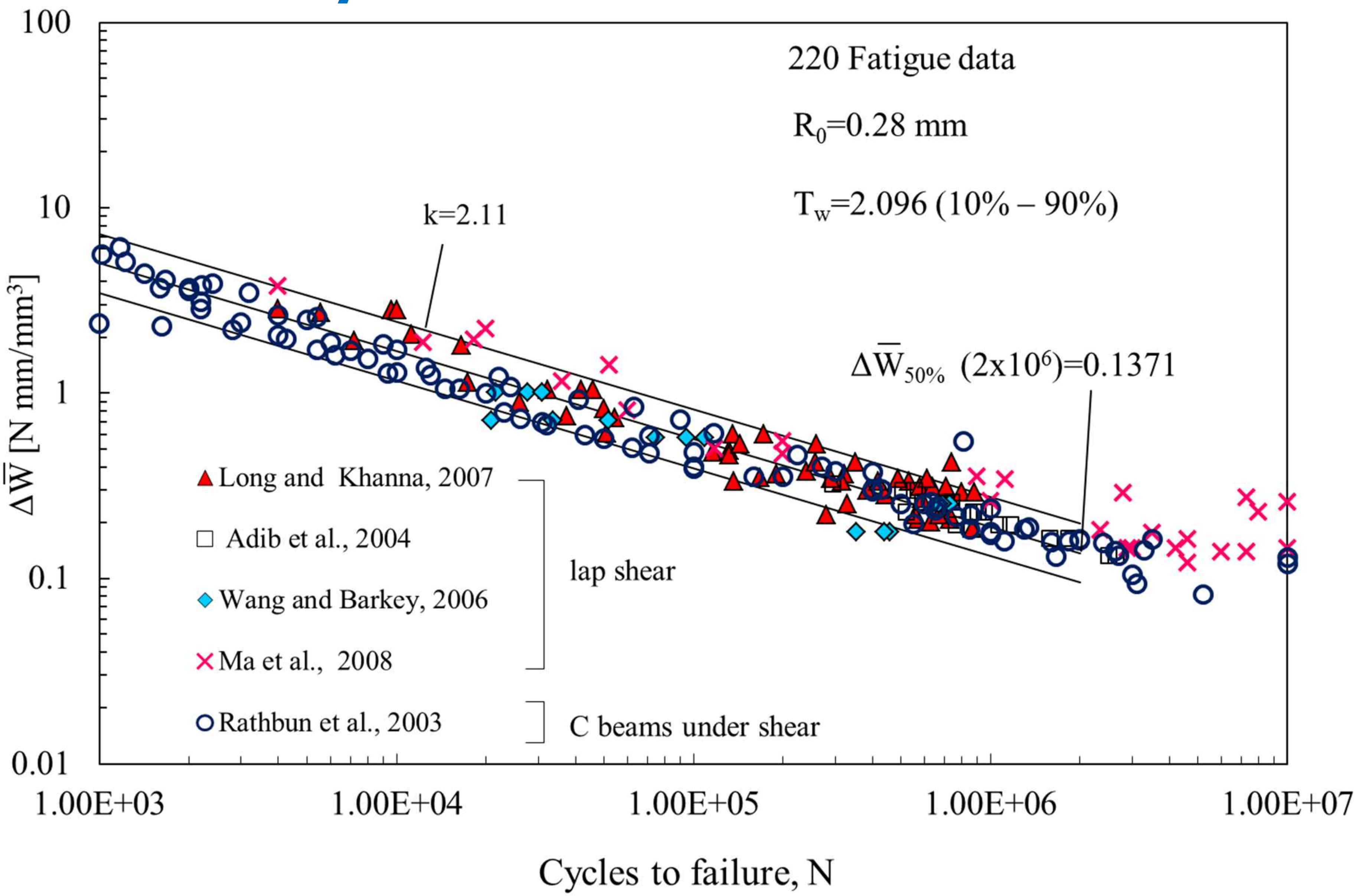


Harding S., Kotousov A., Lazzarin P., Berto F. (2009)

THREE-DIMENSIONAL EFFECTS

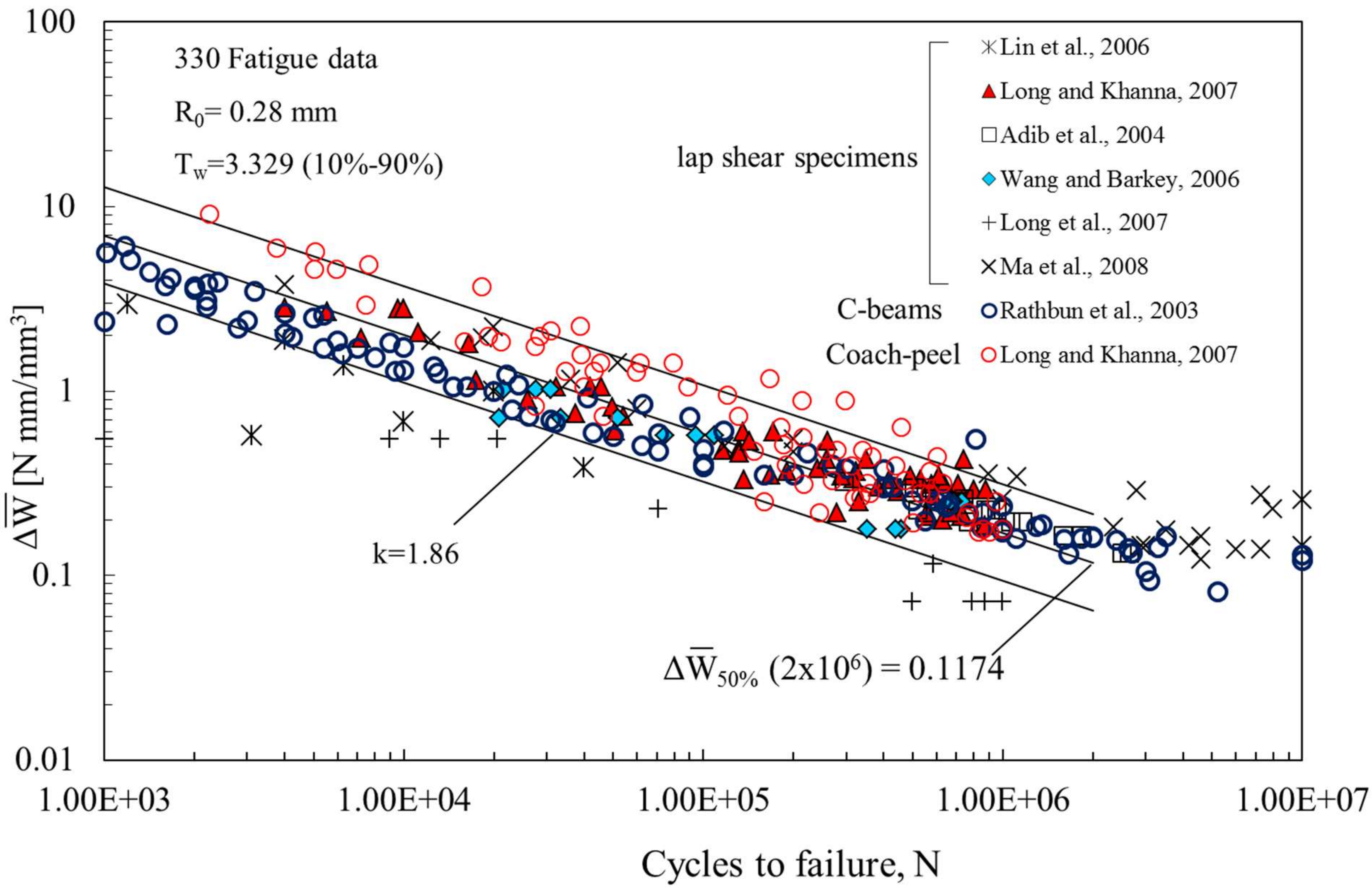


Synthesis based on SED



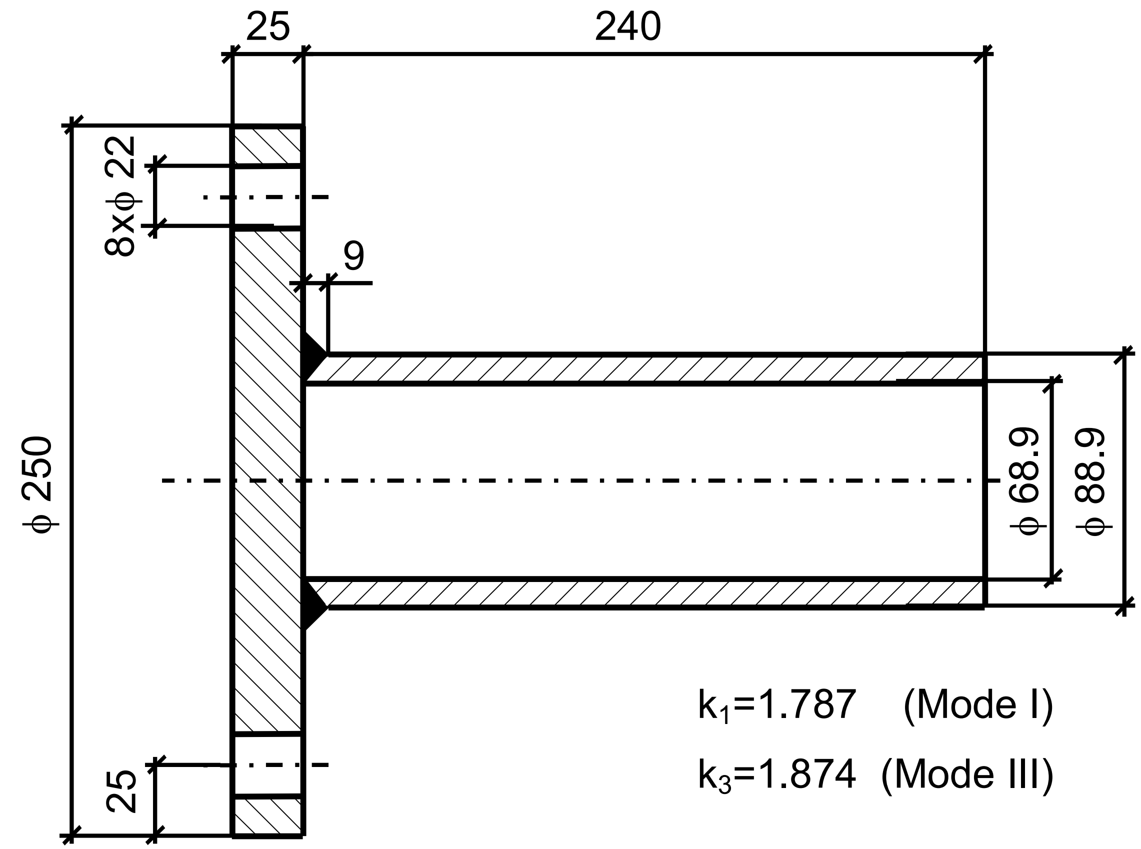
Synthesis of data from spot-welded joints under tension and shear loading. The thickness t ranges from 0.65 to 1.75 mm. SED values have been determined by means of three-dimensional models. The control radius of the toroidal volume is equal to 0.28 mm

Synthesis based on SED



Synthesis of data from lap shear specimens, C-shaped specimens and coach-peel tension specimens (330 data, TW=3.32)

MULTIAXIAL FATIGUE



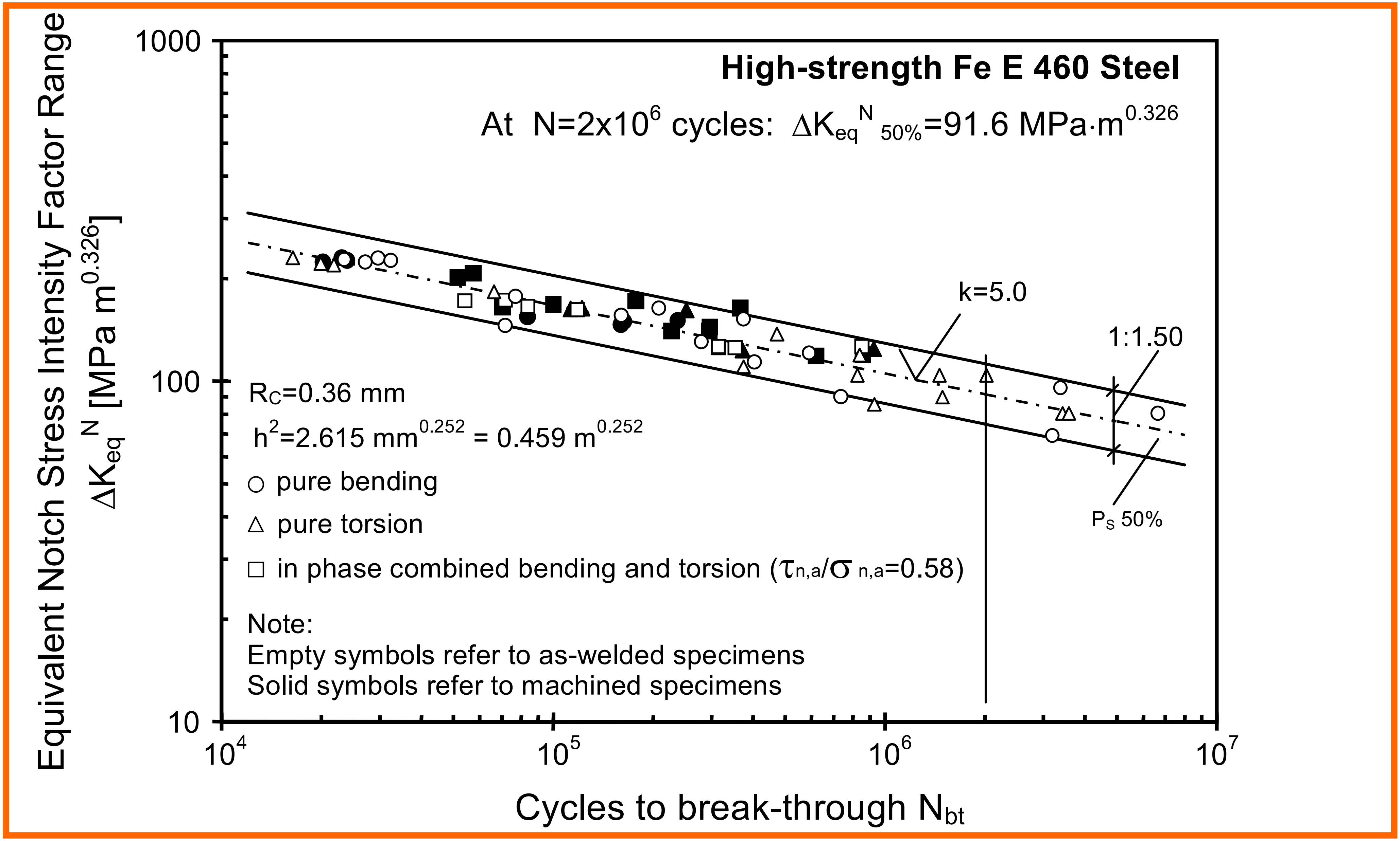
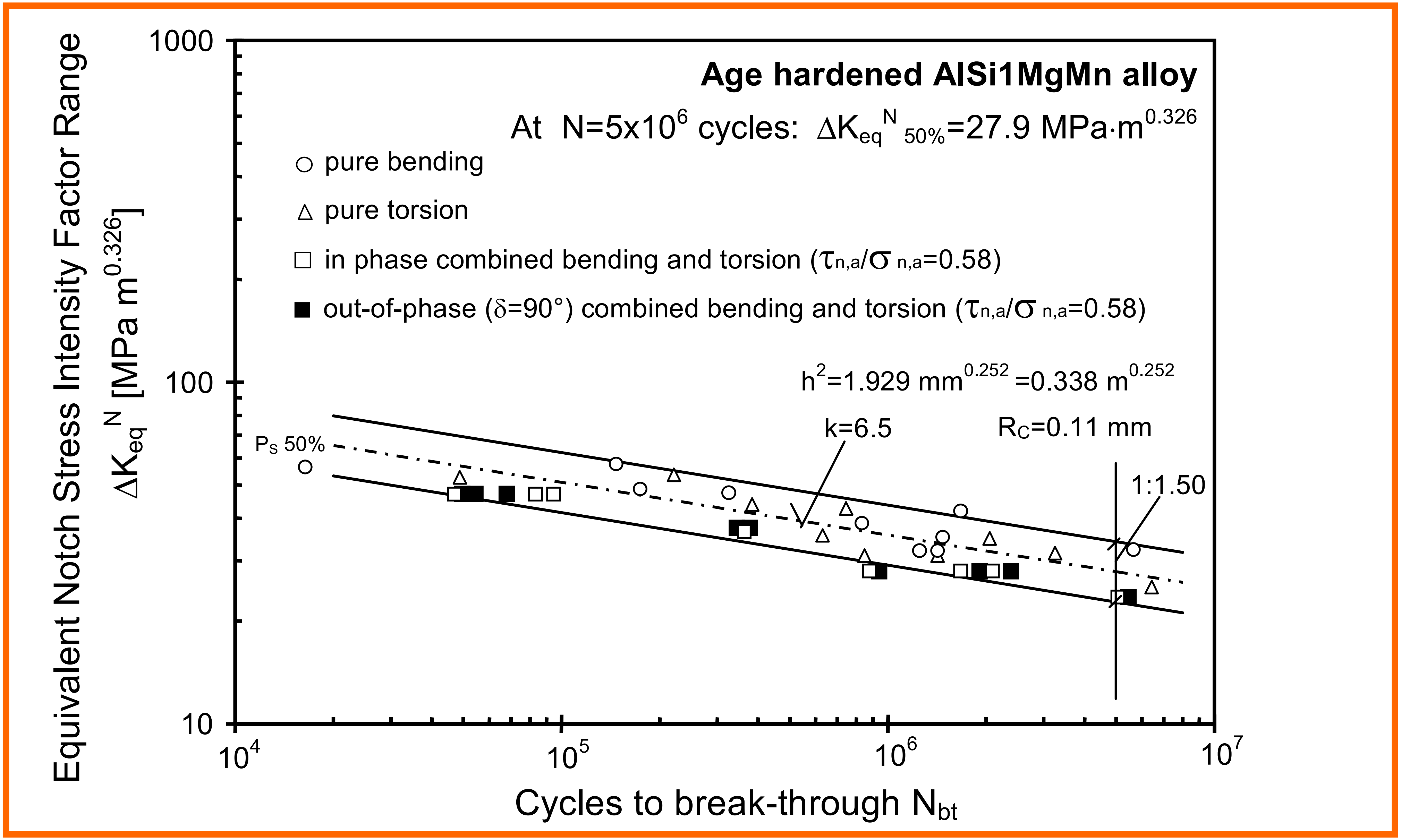
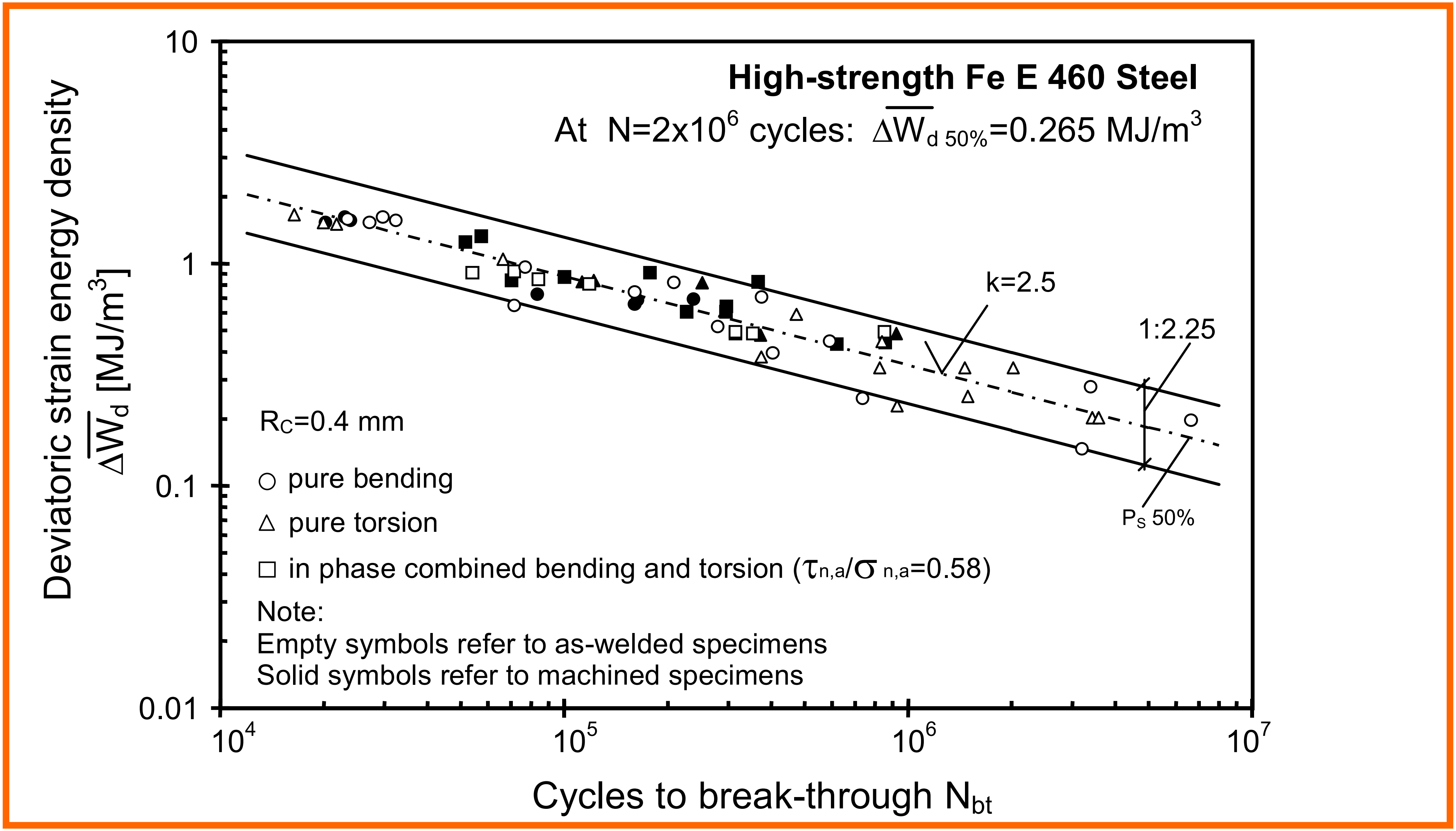


Figure 40: Fatigue test results related to as-welded and machined specimens [33-35].

Nominal load ratio R= -1



Age hardened AlSi1MgMn alloy
At N=5x10^6 cycles: ΔKeq^N 50%=27.9 MPa·m^0.326
pure bending
pure torsion
in phase combined bending and torsion (τn,a/σ n,a=0.58)
out-of-phase (δ=90°) combined bending and torsion (τn,a/σ n,a=0.58)
h^2=1.929 mm^0.252 =0.338 m^0.252
k=6.5
RC=0.11 mm
PS 50%
1:1.50
Equivalent Notch Stress Intensity Factor Range
ΔKeq^N [MPa m^0.326]
Cycles to break-through Nbt
1000
100
10
10^4
10^5
10^6
10^7



Fatigue test results related to as-welded and machined specimens in terms of the mean value of the strain energy density range. Nominal load ratio R= -1

Lecture 3

Local approaches

Some case studies: Part 2

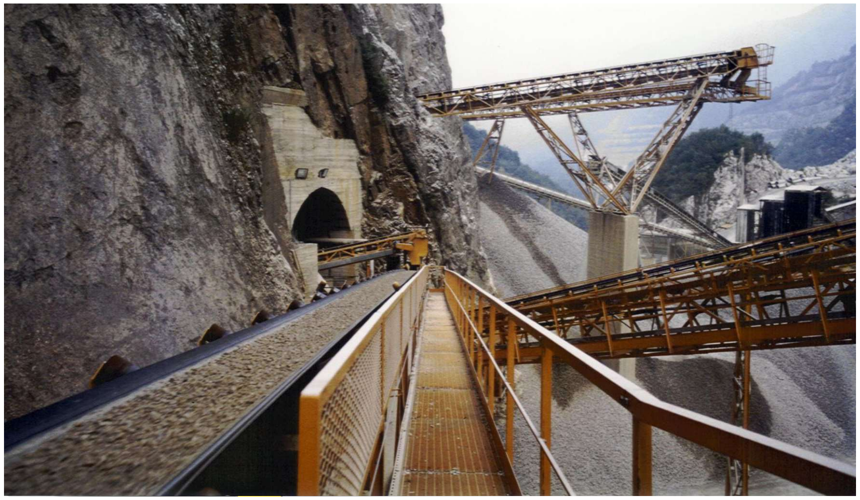
STRAIN ENERGY DENSITY

Rulmeca Group

The world's largest supplier of components for bulk material handling.

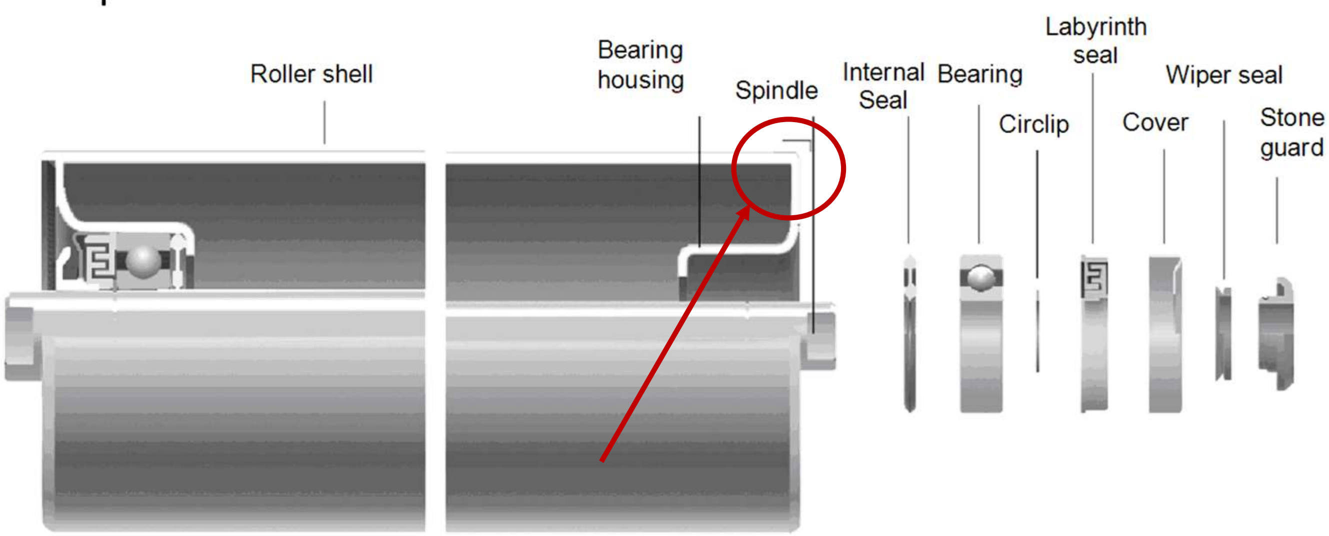
- Based in Bergamo, Italy
- 9 production companies worldwide
- Sales companies in 10 countries
- 1300 employees
- Customers in 85 countries
- Group turnover 2015: € 142 M.
- Family owned with a long term perspective

Output 2015	
Rollers:	*3.600.000*
Transoms/Frames:	*237.000*
Motorized Pulleys:	*36.000*
Belt Pulleys:	*3.500*

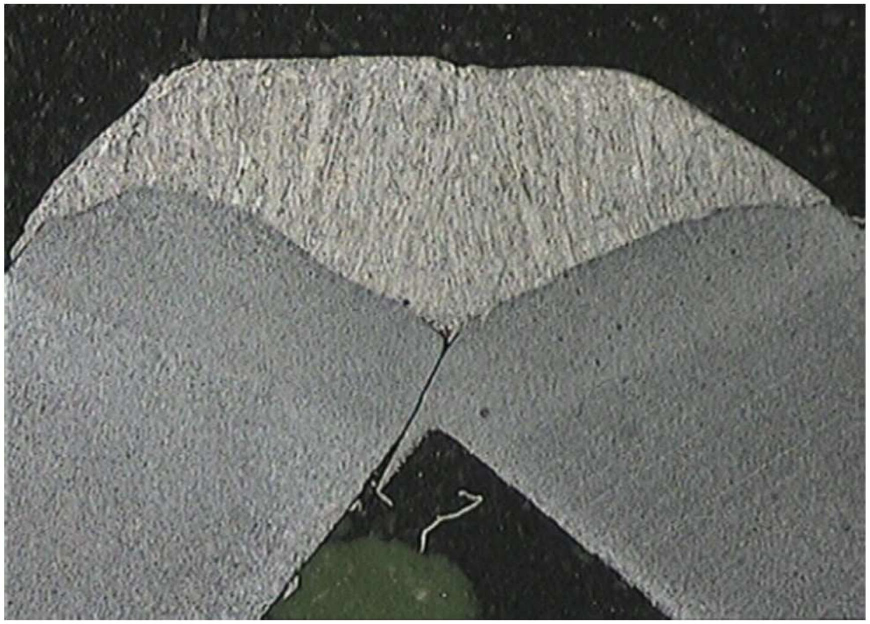


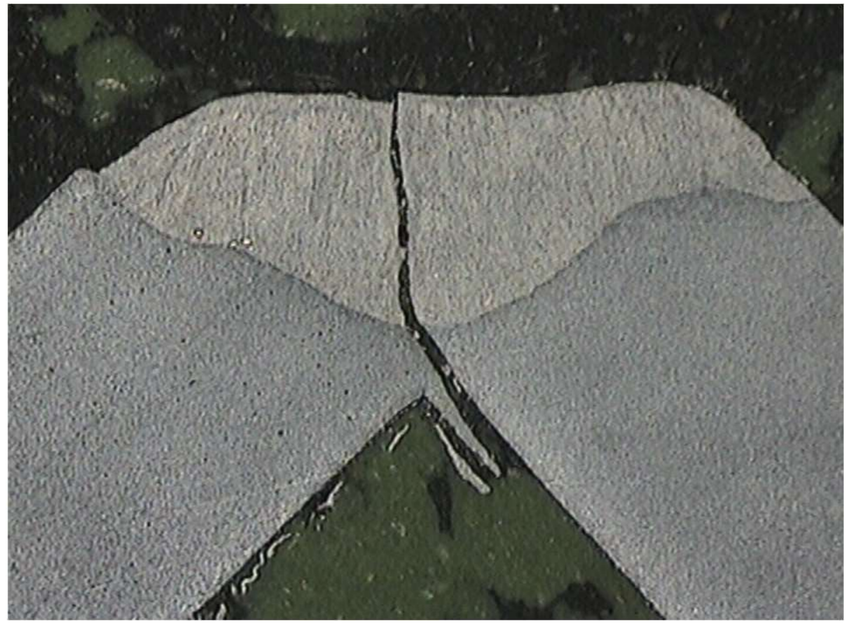
The roller architecture

- Roller material: **steel**
- Typical application: **belt conveyors, mines…**
- The BH and the tube are welded with an auto-centering automatic process (MIG).
- From the point of view of the fatigue behavior, the weakest point of the entire structure is the lack of penetration of the weld root.



Lack of penetration





Project target

The goal for Rulmeca is:

to **reduce the material cost by reducing bearing housing thickness**
guaranteeing **equal performance**

Strain Energy Density (SED)

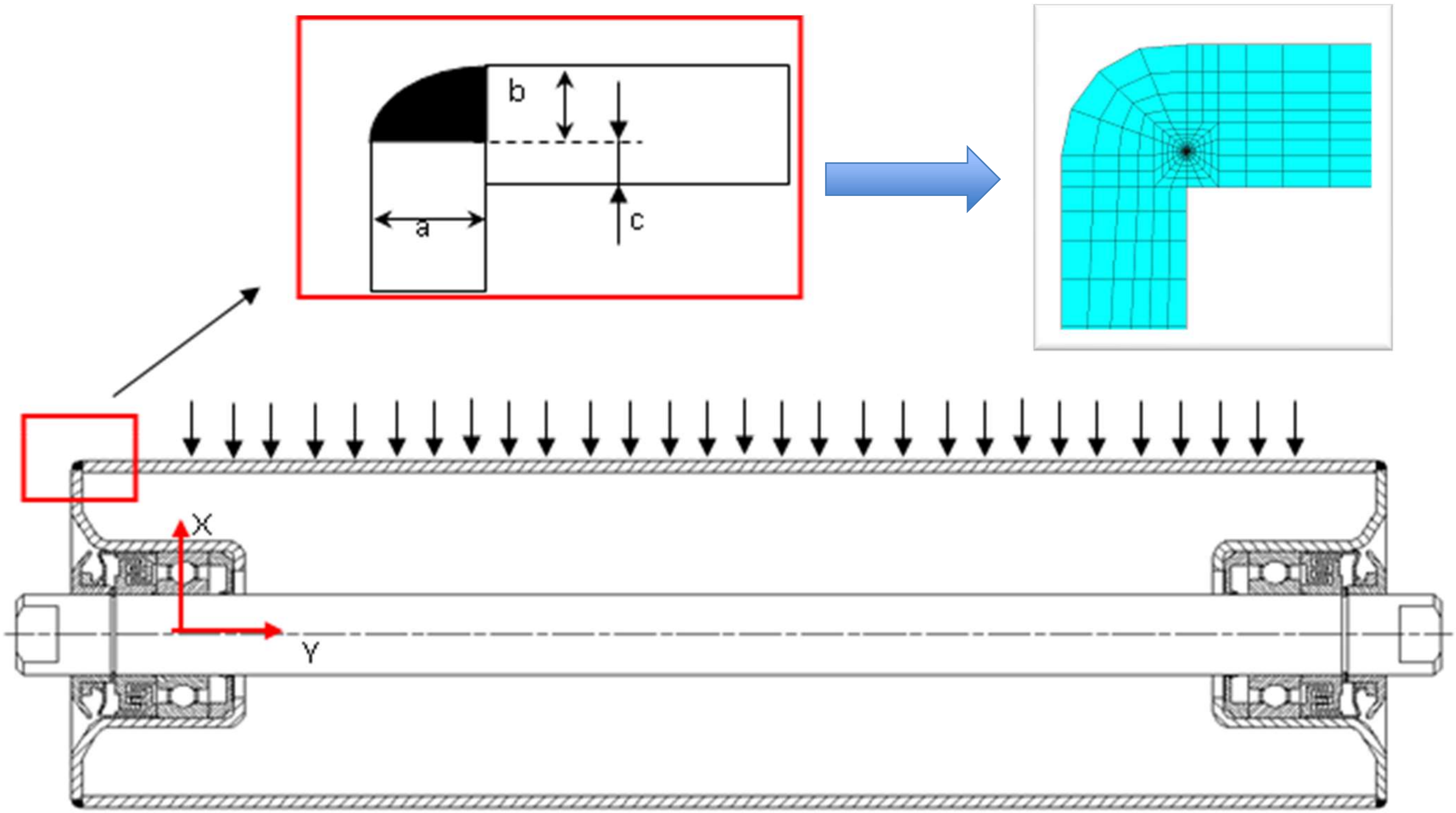
The Strain Energy Density (SED) approach was selected for this case.

- The main novelty of this work is related to the **application of the SED approach to welded joints of SMALL thickness (range 2.5mm-5mm).** Previous literature covers greater welded joints, over **6 mm** thickness.

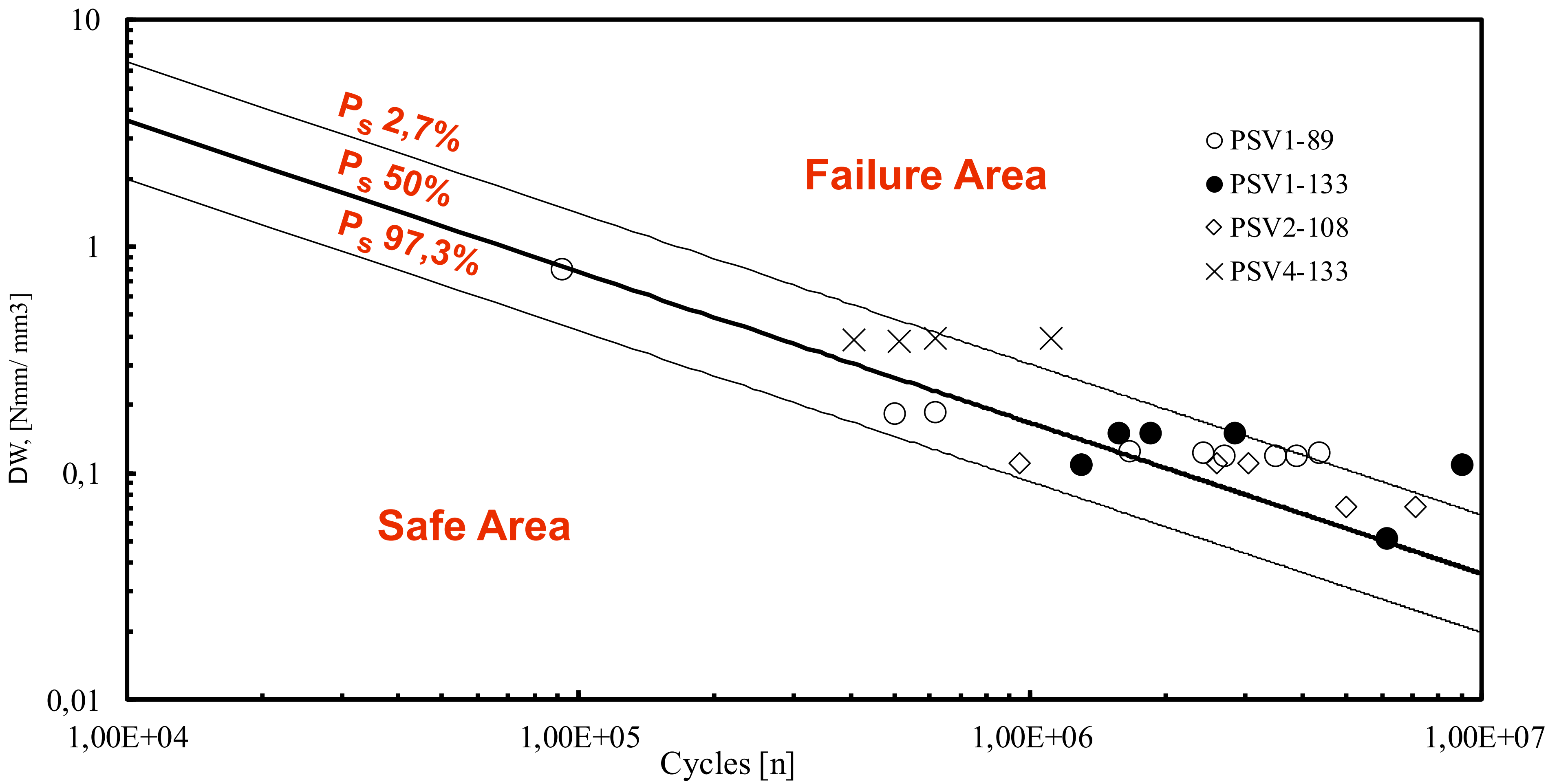
- The approach considers **three dimensional effects** averaged over a control volume (Rc=0,28mm) surrounding the welding critical point.

- The approach requires a **low computation complexity** compared to other methods, so it does need a relatively coarse mesh.

Model definition: load and constraints



SED approach validation (small thickness)

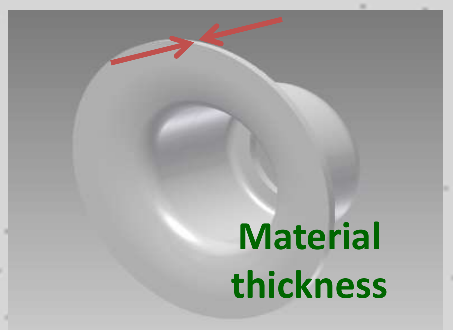


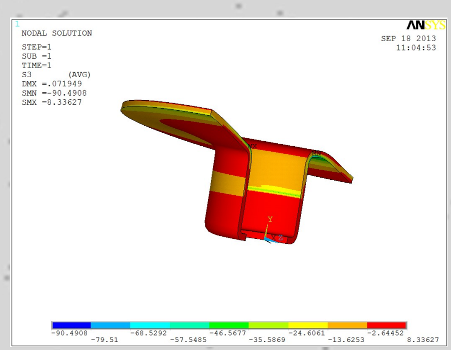
SED application for roller design-review

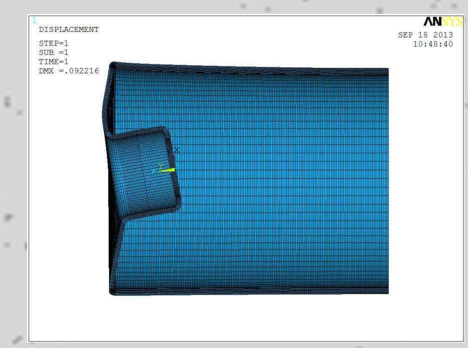
The SED computations have been carried out on roller <u>with reduced thickness (0.5 mm)</u> of bearing housing to confirm roller performance (load bearing capacity) with the new design. Following geometries were analyzed:

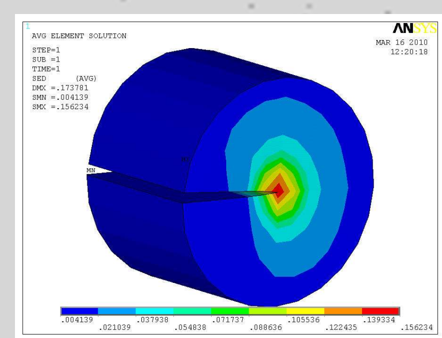
- 5 roller types
- 4 different tube diameters (89 mm, 108 mm, 133 mm, 159 mm)
- 2 different lengths per each combination ("long" and short")

24 computations in total





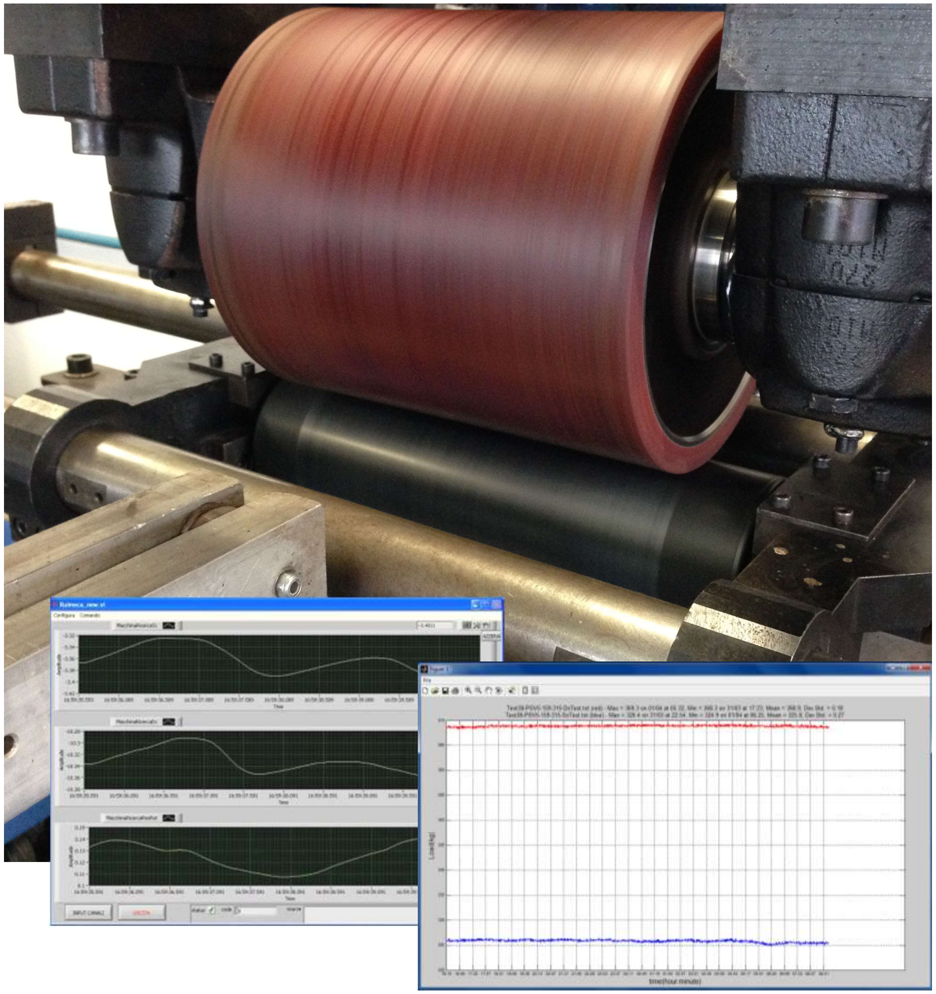




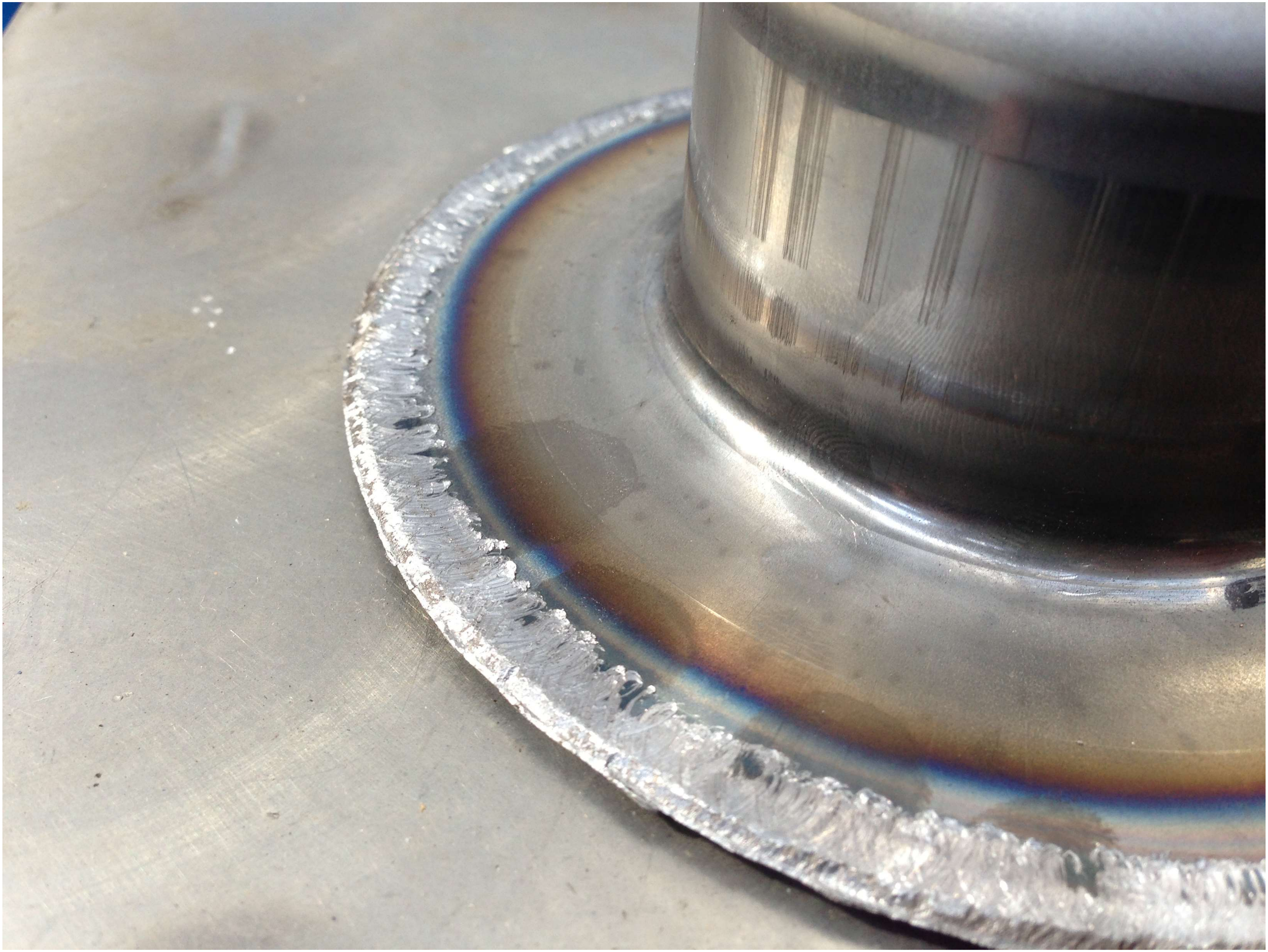
Test campaign on new rollers

In order to confirm the analytic results, in July 2013 we started a test campaign on the reduced thickness.

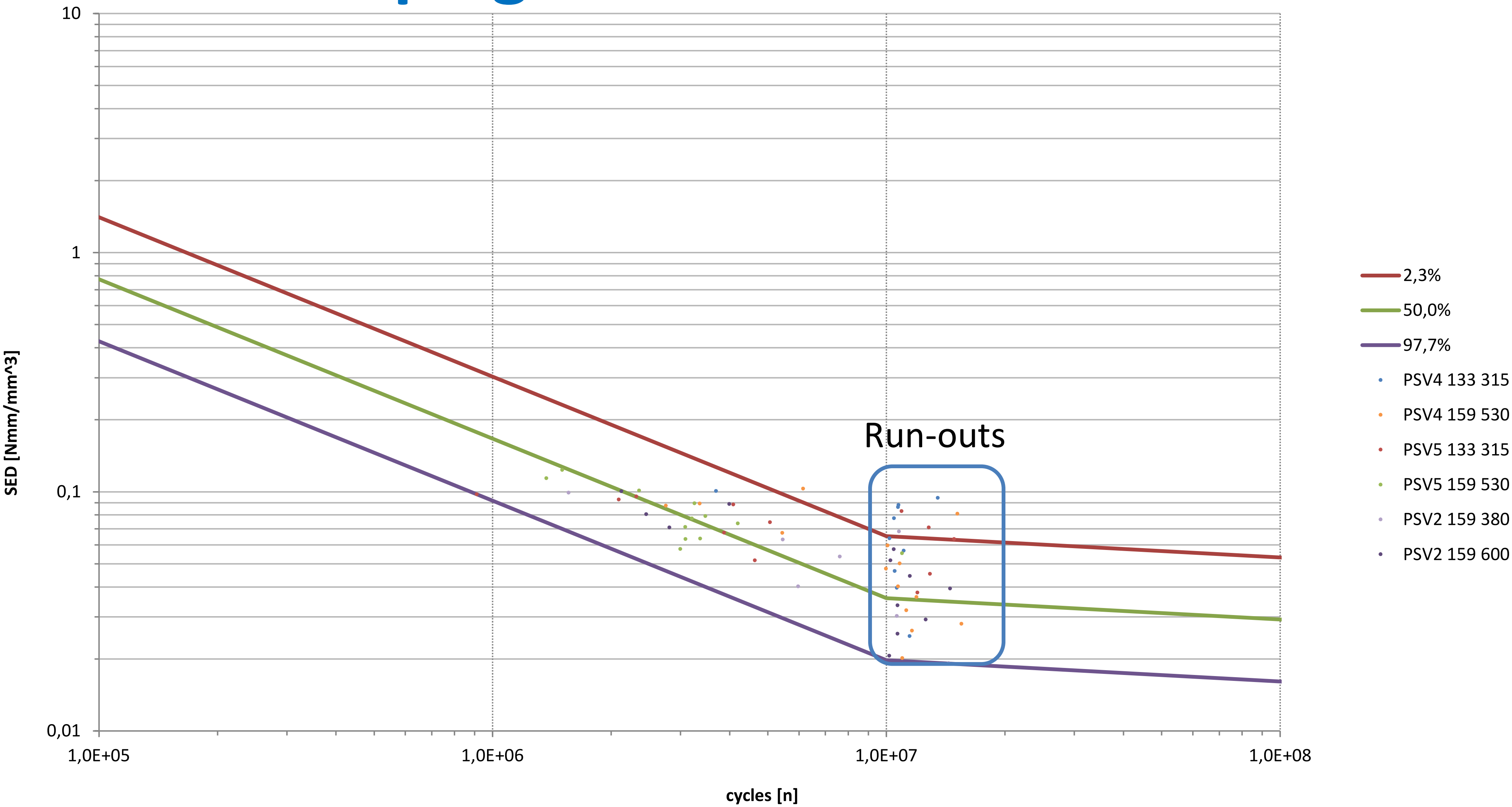
- One machine running 24/7
- 12 samples per each test
- **97 rollers tested so far**
  Still in progress…




RULMECA



Test campaign: results



Test campaign validates the analytic results!

Conclusions

- Test results support the SED approach for Rulmeca case:
  SED can be used to predict the fatigue life of a roller
  with a very good accuracy.

- Rulmeca can successfully take advantage of these results:
  - Increased **knowledge** about rollers.
  - Cost **saving** in the order of 3000 k€ / year.