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Design of dynamically loaded welded structures

Dynamically loaded welded structures

Prepared for

Universities

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1. INTRODUCTION

This document presents the class notes for the fatigue design of dynamically loaded welded structures.

2. STUDY MATERIAL

The student shall arrange access to the following documents:

- BS EN 1993-1-9:2005. Eurocode 3: Design of steel structures. Part 1-9: Fatigue. *British Standards Institution.*
- IIW Bulletin 520.

Other material is referenced and used in the slides.

3. DESIGN OF DYNAMIC LOADED WELDED STRUCTURES

3.1. Objectives

The objective of this section is to understand in detail the different fatigue design methods in the range of application.

3.2. Scope

The scope of theory covered is:

1. Range of application:
    a. Bridges
    b. Cranes
    c. Machines
    d. Ships and offshore constructions
    e. Chimneys
    f. Towers and masts
    g. Vehicles (cars, trucks, railway vehicles, etc.)
2. Acceptance criteria
3. Dimensioning according to different standards and specifications.
4. Worked examples
5. Calculation methods.

3.3. Outcome

After completion of this section you will be able to:

1. Interpret and apply the principles in design.
2. Design welded joints in accordance with given details.
3. Detail the influence of notch effects on the classification of welded joints.
4. Interpret appropriate standards.
5. Compare details in different standards and classify them.

4. WELD FATIGUE DESIGN ACCORDING TO BS EN 1993-1-9, SANS 10162-1 & IIW BULLETIN 520

The purpose of this section is to introduce the fatigue design of weld detail according to the requirements of BS EN 1993-1-9, SANS 10162-1 & IIW 520.

No additional notes are applicable.

Your copy of the EN 1993-1-9 standard is used as reference.

Presentation used in class:	Investmech - Structural Integrity (Fatigue design according to BS EN 1993-1-9, SANS 10162-1 & IIW Bulletin 520) R0.0

Compulsary study material

1. EN 1993-1-0. CCYY. Eurocode 3: Design of steel structures – Par 1-9: Fatigue. *European Standard.*
2. BS 7608.
3. IIW Bulletin 520
   - Investmech course:
     - Investmech will issue a legitimate copy to every delegate
   - University course:
     - The student shall obtain a legitimate copy for this course
     - No notes or extractions from these documents will be made available to delegates
       - The copy made available to delegates will reference tables and figures

Scope of BS EN 1993-1-9: Eurocode 3

Part 9: Fatigue

- Gives methods for assessment of fatigue resistance of members, connections and joints subject to variable amplitude loading
  - Derived from fatigue tests with large scale specimens
  - Include effects of geometrical & structural imperfections from material production & execution
    - E.g. Tolerances & residual stress from welding
- Execution shall conform with EN 1090
  - BS EN 1090-2:2008+A1:2011. Execution of steel structures and aluminium structures. Technical requirements for steel structures
    - Define products: steels, welding consumables, mechanical fasteners
    - Preparation, welding, testing, erection of structural systems
    - Inspection & correction to ensure maximum levels of quality control
- Applicable to all grades of:
  - Structural steels
  - Stainless steels
  - Unprotected weathering steels

Steel must conform to toughness requirement of BS EN 1993-1-10

- Fatigue assessment **methods not covered:**
  - Fracture mechanics
  - Notch strain

See IIW Bulletin 520 for comprehensive coverage of these methods

- Improvement techniques:

Covered: Stress relieving

Not covered: Toe dressing, burring, peening, etc.

- Environment not covered:
  - Structures operating under normal atmospheric conditions
  - Sufficient corrosion protection
  - Regular maintenance
- Environment not covered:
  - Effect of seawater corrosion
  - Microstructural damage from high temperature >150°C

See IIW Bulletin 520 for coverage of these

- Other Standards referred to

BS EN 1090. Execution of steel structures – Technical requirements.

BS EN 1990. Basis of structural design.

BS EN 1991. Actions on structures.

BS EN 1993. Design of steel structures.

BS EN 1993-2. Design of composite steel and concrete structures: Part2: Bridges.

BS EN 1999. Aluminium.

See the terms and conditions in the notes issued in class

4.1. Basics

- Structural members:
  - Design for fatigue such that there is an acceptable level of probability that their performance will be satisfactorily through their design life
    - Use fatigue actions from BS EN 1991
    - Use fatigue resistance curves from BS EN 1993-1-9
    - Use BS EN 1993-1-9 Annex A for specific loading model if:
      - No fatigue load model is available in BS EN 1991
      - A more realistic fatigue model is required
- Fatigue tests is required to:
  - Determine fatigue strength for details not included in this part of the standard
  - Determine fatigue life of prototypes for:
    - Actual fatigue loads
    - Damage equivalent fatigue loads
  - Take BS EN 1990 into account to determine structural inputs
- Methods for fatigue assessment in BS EN 1993-1-9:
  - Principle of design verification comparing action effects & fatigue strengths
    - Only possible when fatigue actions are determined with parameters of fatigue strengths prescribed
- Fatigue actions:
  - Determined according to requirements of fatigue assessment
  - Are different from actions for ultimate limit state & serviceability limit state verifications
- Crack initiation
  - Do not necessarily mean the end of service life
    - Could be repaired with particular care to **avoid introducing more severe notch** conditions

4.2. Fatigue assessment methods

4.2.1. Damage Tolerant Method

- Provide acceptable reliability that the structure will perform satisfactorily for its design life provided that:
    - Prescribed inspection and maintenance regime for detecting and correcting fatigue damage is implemented throughout the design life
- Apply in the event where fatigue damage occurring, a load distribution between components of structural elements can occur
- Structures assessed to BS EN 1993-1-9 & material according to BS EN 1993-1-10 subjected to regular maintenance = damage tolerant

4.2.2. Safe life method

- Provide acceptable level of reliability that the structure will perform satisfactorily for design life without need for regular in-service inspection for fatigue damage
- Apply in cases where local formation of cracks in one component could rapidly lead to failure of the structural element or structure
- Implies that the structure will resist the ultimate limit state load at the end of its design life

4.2.3. Analysis

- Use partial factor for fatigue strength γ_{Mf} taking into account
    - – Consequences of failure
    - – Design assessment used
    - – This results in a reduced fatigue strength: $\Delta\sigma_{C,red} = \frac{\Delta\sigma_C}{\gamma_{Mf}}$

Table 1: Partial factor for fatigue strength

Assessment method	Consequence of failure	
	Low consequence	High consequence
Damage tolerant	1.00	1.15
Safe life	1.15	1.35
Source: BS EN 1993-1-9, 2005:11		

- Fatigue strengths
    - – Are determined considering:
        - Structural detail
        - Metallurgical and geometric notch effects
        - Probable site of crack initiation
    - – Standard details applicable to nominal stresses
        - Cross-section dimensions that has an effect on the nominal stress
    - – Reference weld configurations applicable to geometric stresses
        - Stress concentrations due to the geometry can result at weld detail that must be included
- **Achieving reliability: Damage Tolerant Method**
    - – Selecting details, materials, stress levels so that in the event of crack initiation
        - low rate of crack propagation result
        - long critical crack length can result
    - – Provision of multiple load path
    - – Provision of crack-arresting detail
    - – Provision of readily inspectable details during regular inspections
- **Achieving reliability: Safe-life method**

– Selecting details and stress levels resulting in a fatigue life sufficient to achieve the β values equal to those for ultimate limit state verifications at the end of the design life

4.3. Stresses from fatigue actions

- Nominal stresses take into account:
  - All actions
  - Distortional effects
  - Linear elastic analysis for members and connections
- Latticed girders made of hollow sections:
  - Model based on <u>simplified truss model with pinned connections</u>
    - Stresses due to external loading applied to members between joints must be taken into account
    - The effects from secondary moments due to the stiffness of the connection can be allowed for by k_1 factors

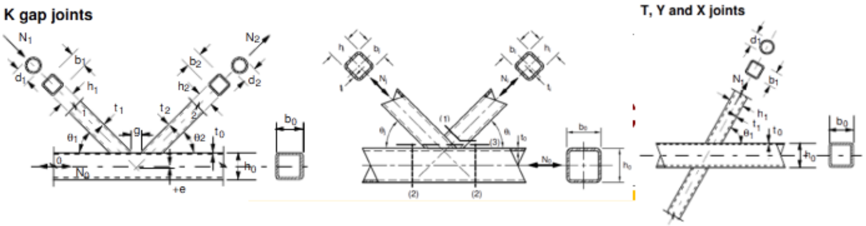


Table 4.1: k_1-factors for circular hollow sections under in-plane loading

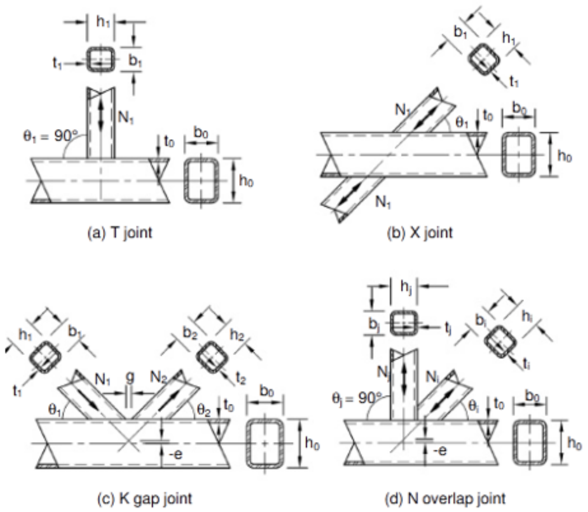
Type of joint		Chords	Verticals	Diagonals
Gap joints	K type	1,5	1,0	1,3
	N type / KT type	1,5	1,8	1,4
Overlap joints	K type	1,5	1,0	1,2
	N type / KT type	1,5	1,65	1,25

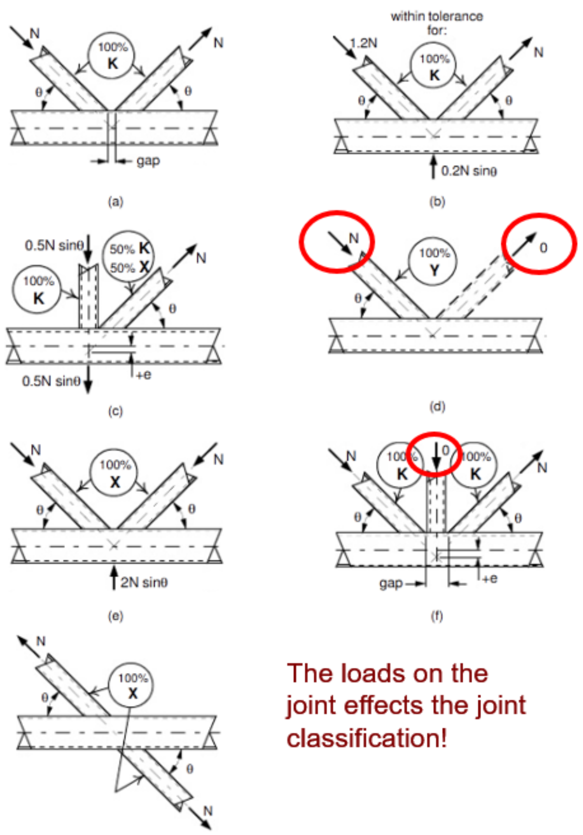
Reference: BS EN 1993-1-9, 2005:11

Table 4.2: k_1-factors for rectangular hollow sections under in-plane loading

Type of joint		Chords	Verticals	Diagonals
Gap joints	K type	1,5	1,0	1,5
	N type / KT type	1,5	2,2	1,6
Overlap joints	K type	1,5	1,0	1,3
	N type / KT type	1,5	2,0	1,4

Reference: BS EN 1993-1-9, 2005:12



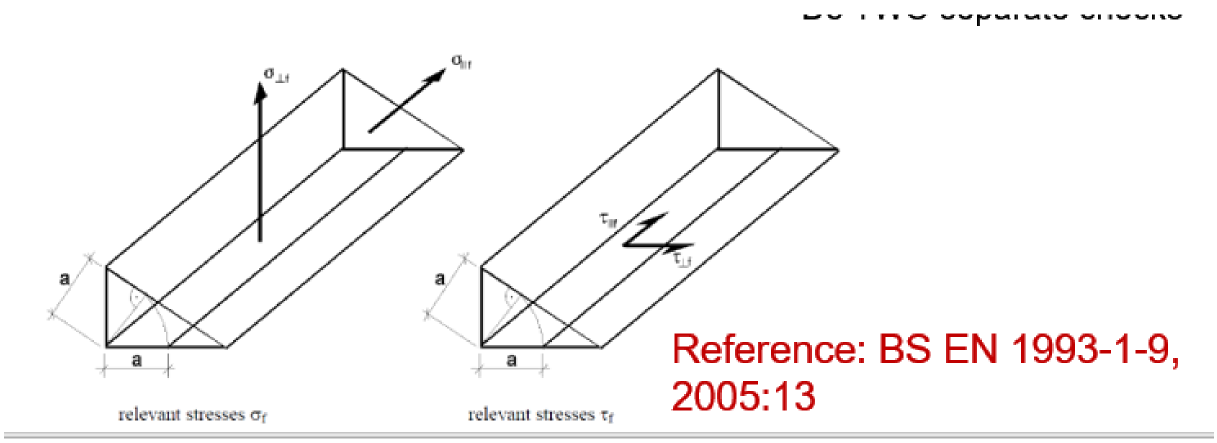


http://www.scribd.com/doc/37761724/25/Joint-resistance-equations-for-T-Y-X-and-K-gap-joints Page 29 and 30
PACKER, J.A., WARDENIER, J., ZHAO, X.L., VAN DER VEGTE, A. & KUROBANE, Y. 2009. Design guide for rectangular hollow section (RHS) under predominantly static loading. Cidect.

The loads on the joint effects the joint classification!

4.4. Calculation of stresses

- Calculate at the serviceability limit state
- Class 4 cross-sections
  - According to BS EN 1993-1-5
    - See BS EN 1993-2 to BS EN 1993-6
- Calculate nominal stress at site of potential fatigue initiation
  - Account for stress concentrations at detail other than those in Table 8.1 to Table 8.10 by using stress concentration factor according to 6.3 to give *modified nominal stress*
- For geometrical (hot spot) stress approach as per Table B.1 calculate stress as per Section 6.5
- Relevant stresses:
  - Nominal direct stress: σ
  - Nominal shear stress: τ
  - Use combined effect where applicable
- Relevant stresses - equations:
  - Normal stresses transverse to the axis of the weld:
    - $\sigma_{wf} = \sqrt{\sigma_{\perp f}^2 + \tau_{\perp f}^2}$
  - Shear stresses longitudinal to the axis of the weld:
    - $\tau_{wf} = \tau_{\parallel f}$
  - Do TWO separate checks



4.5. Stress ranges

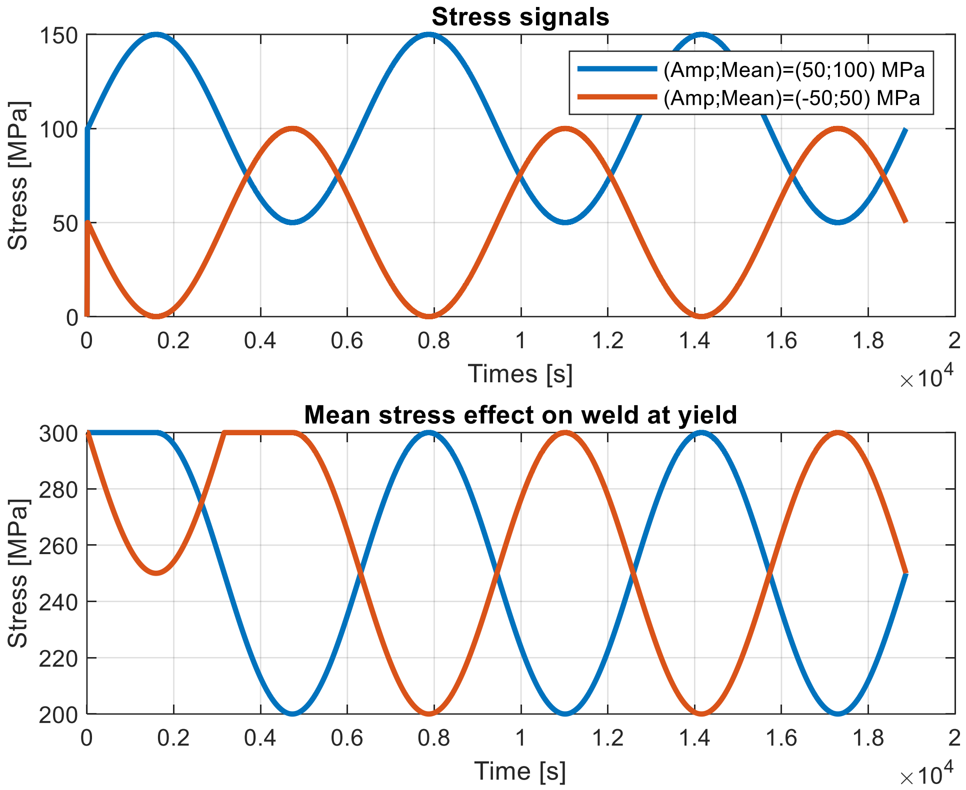
4.5.1. Effect of mean nominal stress on weld stress response

Why is stress range used on the fatigue curves for welded structures?

That is because:

1. The fatigue curves are given for 2 x amplitude (range = 2 x amplitude). So, you can, if you so wish, construct the curves for amplitude. Just be careful in how residual stress, etc. are treated based on empirical data.
    a. Non-heat treated welds have a residual stress in the order of 0.7 x yield strength after welding and cooling.
    b. If the residual stress is taken as yield strength, applying a stress signal with different tensile mean stress will have the same response in the weld (because of yielding under tensile loading). See Figure 1.
2. Then, test results, which forms the basis for the empirically determined fatigue curves, showed that large scale manufactured components can be modelled for fatigue using the stress ranges – as is clear from the fatigue curves that we discussed in the lecture. After stress relieving, a reduced stress range (range = tensile - 60% x compressive) can be used in SOME cases to model fatigue.
3. Machined parts are modelled where Goodman mean stress correction is done and the completely reversed stress amplitude used as ordinate in the fatigue curve (we did this in the first few lectures).
4. The important thing is:
    a. Fatigue curves can be presented as stress range or stress amplitude.
    b. Stress range is normally used where the effect of mean stress is not modelled to the extent that it is done on analysis based on fatigue curves constructed with completely reversed stress amplitude as ordinate, and in which case mean stress effects need to be modelled.

So, as you can see, for construction of the fatigue curve it does not matter whether the ordinate is in range or amplitude. Just apply the detail associated with each fatigue curve.



Source: (Investmech weldmeaneffect.m, 2018)

Figure 1: Stress response in a weld at residual stress equal to yield strength of 300 MPa for two tensile mean nominal stresses

For a mean stress positive and negative, the stress response at the weld initially at yield is shown in Figure 2. This demonstrate the reason why the standards suggest that after stress relieving only 60% of the compressive part of nominal stress can be used in SOME instances to model the beneficial effect of compressive mean stress on SOME weld detail.

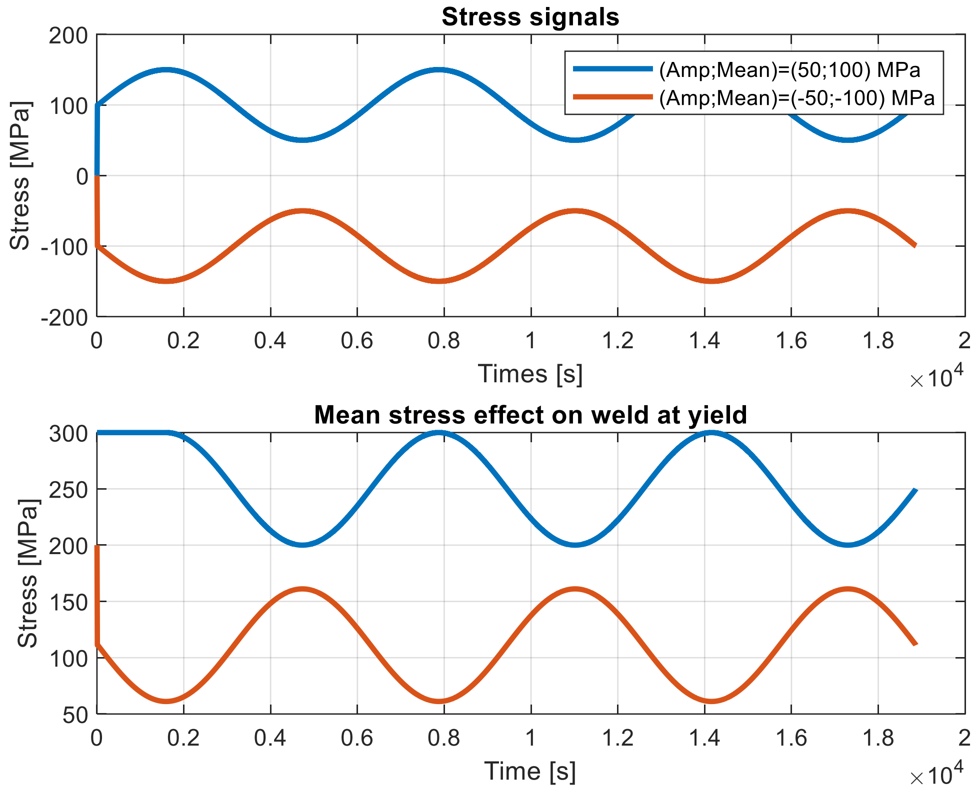


Figure 2: Stress response in a weld at residual stress equal to yield strength of 300 MPa for tensile and compressive mean nominal stresses

4.5.2. Calculation of stress range

- Use:
  - Nominal stress ranges for details in Tables 8.1 to 8.10
  - Modified nominal stress ranges where:
    - Abrupt changes of a section close to initiation site not included in Tables 8.1 to 8.10
  - Geometric (hot spot) stress ranges:
    - Where high stress gradients occur close to weld toe in joints covered by Table B.1
- Design value of stress range to be used:

$$\Delta\sigma_{Design} = \gamma_{Ff}\Delta\sigma_{E,2}$$

Where $\boldsymbol{\Delta\sigma_{E,2}}$ **corresponds to** $\boldsymbol{N_C = 2 \times 10^6}$ cycles on the relevant $\Delta\sigma_R - N$ curve and is:

$$\Delta\sigma_{E,2} = \frac{\Delta\sigma_{Design}}{\gamma_{Ff}}$$

$\Delta\sigma_{E,2}$ is the equivalent constant amplitude stress range related to 2×10^6 cycles [MPa]

γ_{Ff} is the partial factor for equivalent constant amplitude stress ranges $\Delta\sigma_E$ & $\Delta\tau_E$ and is in most cases $\boldsymbol{\gamma_{Ff} = 1.0}$ because the exact loads are used

4.5.2.1. Design value of nominal stress ranges

$$\gamma_{Ff}\Delta\sigma_{E,2} = \gamma_1\gamma_2\gamma_i \dots \gamma_n \times \Delta\sigma(\gamma_{Ff}Q_k)$$
$$\gamma_{Ff}\Delta\tau_{E,2} = \gamma_1\gamma_2\gamma_i \dots \gamma_n \times \Delta\tau(\gamma_{Ff}Q_k)$$

Where $\Delta\sigma(\gamma_{Ff}Q_k), \Delta\tau(\gamma_{Ff}Q_k)$ is the stress range caused by the fatigue loads specified in EN 1991.

γ_i are damage equivalent factors depending on the spectra as specified in EN 1993

Use Annex A where no appropriate data is available for γ_i

4.5.2.2. Design value of modified nominal stress range

$$\gamma_{Ff}\Delta\sigma_{E,2} = k_f\gamma_1\gamma_2\gamma_i \dots \gamma_n \times \Delta\sigma(\gamma_{Ff}Q_k)$$
$$\gamma_{Ff}\Delta\tau_{E,2} = k_f\gamma_1\gamma_2\gamma_i \dots \gamma_n \times \Delta\tau(\gamma_{Ff}Q_k)$$

Where:

k_f is the stress concentration factor to take account of local stress magnification in relation to detail geometry not included in the reference $\Delta\sigma_R - N$ curve. Use handbooks or FEA to determine.

4.5.2.3. Design value of stress range for **geometrical (hot spot) stress**

$$\gamma_{Ff}\Delta\sigma_{E,2} = k_f(\gamma_{Ff}\Delta\sigma^*_{E,2})$$

Where

k_f is stress concentration factor

4.5.2.4. Design value of stress range for welded joints of **hollow sections**

$$\gamma_{Ff}\Delta\sigma_{E,2} = k_1(\gamma_{Ff}\Delta\sigma^*_{E,2})$$

Where:

$\gamma_{Ff}\Delta\sigma^*_{E,2}$ is the design value of stress range calculated with simplified truss model with pinned joints

k_1 is the magnification factor according to Table 4.1 or Table 4.2 – see below

For latticed girders made of hollow sections:

- Model based on simplified truss model with pinned connections.
    - Provided that stresses due to external loading applied to members between joints are taken into account.
- The effects from the secondary moments due to the stiffness of the connection can be allowed for by the use of k_1-factors.

Table 4.1: k_1-factors for circular hollow sections under in-plane loading

Type of joint		Chords	Verticals	Diagonals
Gap joints	K type	1,5	1,0	1,3
	N type / KT type	1,5	1,8	1,4
Overlap joints	K type	1,5	1,0	1,2
	N type / KT type	1,5	1,65	1,25

Reference: BS EN 1993-1-9, 2005:11

Table 4.2: k_1-factors for rectangular hollow sections under in-plane loading

Type of joint		Chords	Verticals	Diagonals
Gap joints	K type	1,5	1,0	1,5
	N type / KT type	1,5	2,2	1,6
Overlap joints	K type	1,5	1,0	1,3
	N type / KT type	1,5	2,0	1,4

Reference: BS EN 1993-1-9, 2005:12

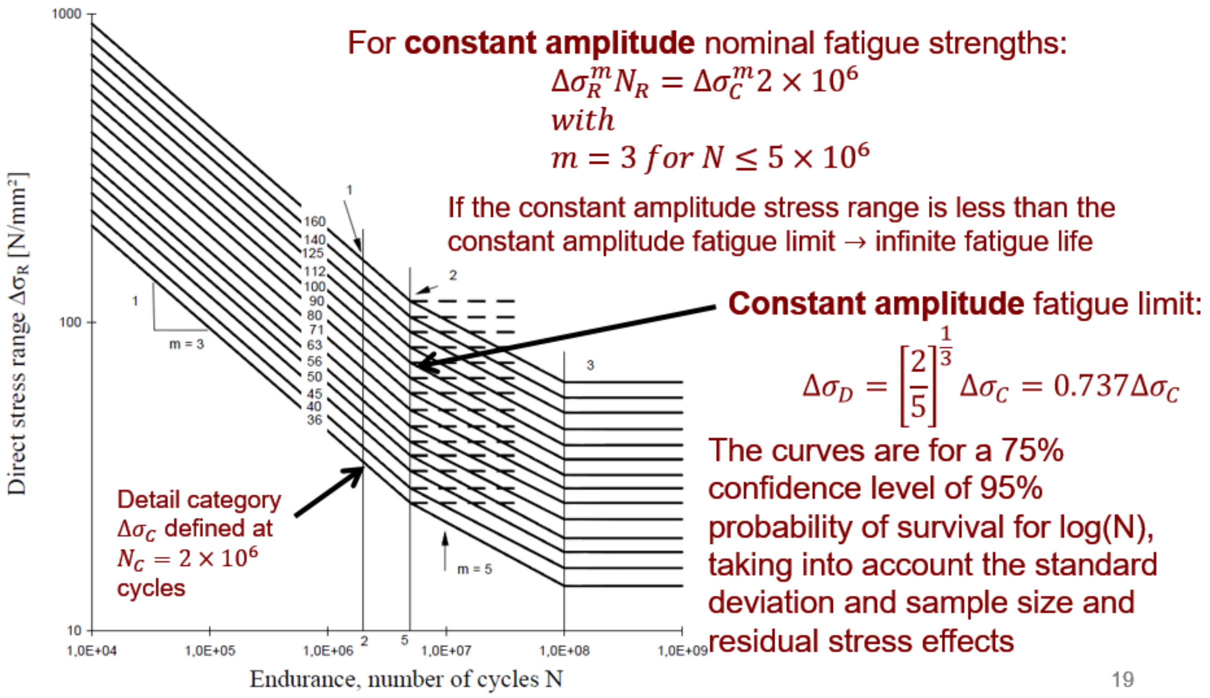
4.6. Damage fatigue factors for bridges

$$\lambda = \lambda_1\lambda_2\lambda_3\lambda_4 \le \lambda_{max}$$

- λ_1:
    - Takes damage effect of traffic into account
    - Depends on critical length of the influence line or area
- λ_2:
    - Takes spectrum of traffic frequency and weights into account
    - Fairly crude factor
- $\lambda_3 = \left(\frac{t_{Ld}}{100}\right)^{\frac{1}{5}}$:

- Takes into account the design life of the bridge where t_{Ld} is the design life in years
- λ_4:
  - Takes into account traffic on other lanes
  - Due to the ability of most bridges to transmit load transversely, detail will usually attract fatigue stress from vehicles passing in lanes remote from those directly above them

4.7. Fatigue strength: Constant amplitude stress



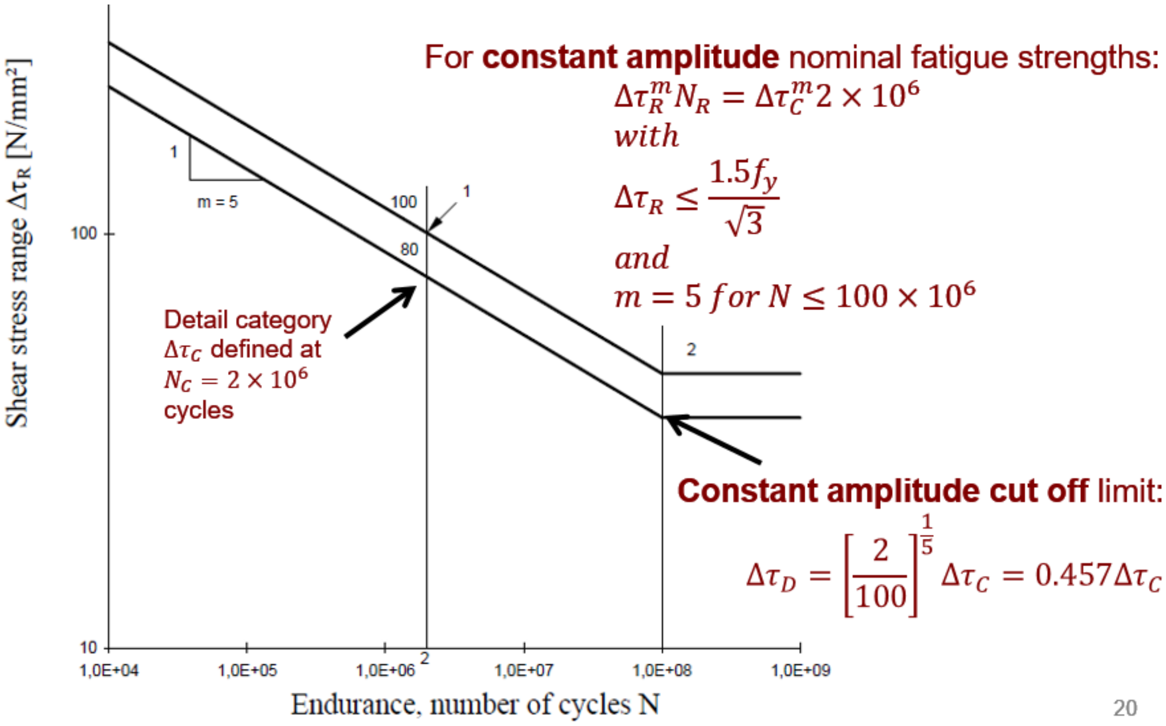


Figure 3: S_r-N curves: Constant amplitude fatigue limit

4.8. Fatigue strength: Variable amplitude loading

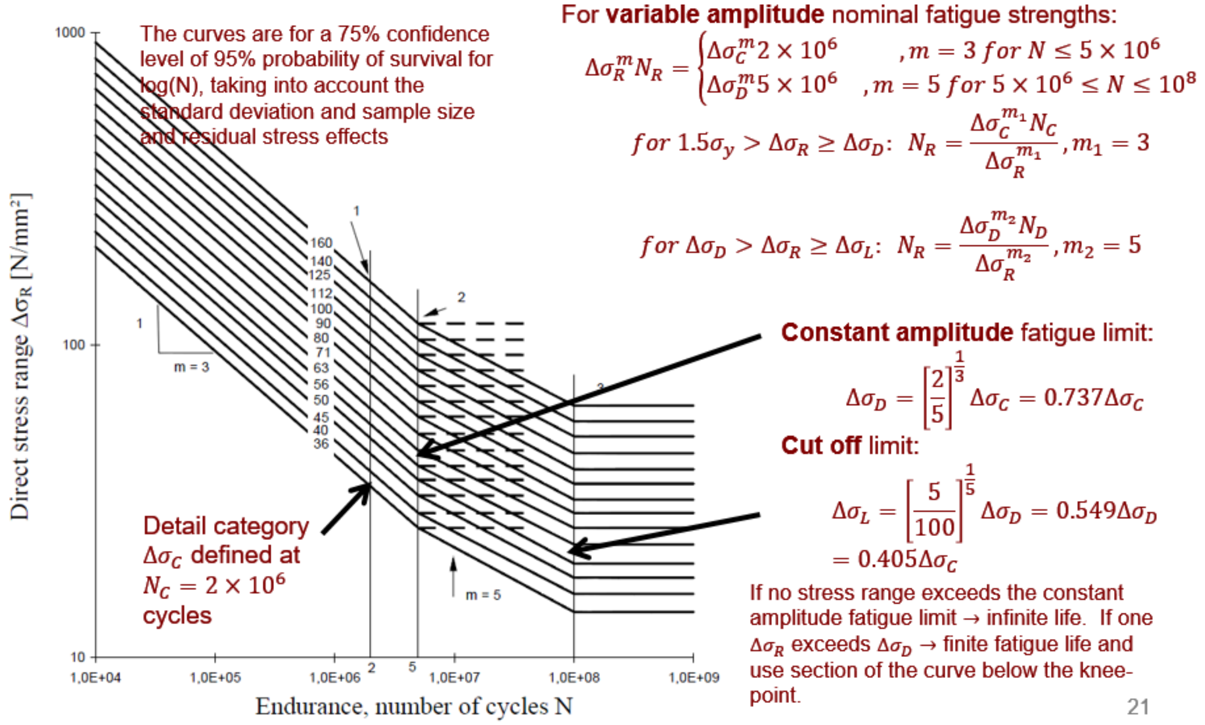


Figure 4: S_r-N curves: Variable amplitude loading

For **variable amplitude** nominal fatigue strengths:

$$\Delta\sigma_R^m N_R = \begin{cases} \Delta\sigma_C^{m_1}\cdot 2\times10^6 & for\ N_R\le 5\times10^6 \\ \Delta\sigma_D^{m_2}\cdot 5\times10^6 & for\ 5\times10^6\le N_R\le10^8 \end{cases}$$

Normally we have the stress range $\Delta\sigma_R$ and want to calculate the endurance, N_R,:

$$N_R = \begin{cases} \left(\frac{\Delta\sigma_C}{\Delta\sigma_R}\right)^{m_1} N_C \quad m_1 = 3 & for\ 1.5f_y > \Delta\sigma_R \ge \Delta\sigma_D \\ \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_2} N_D \quad m_2 = 5 & for\ \Delta\sigma_D > \Delta\sigma_R \ge \Delta\sigma_L \\ \infty & for\ \Delta\sigma_R < \Delta\sigma_L \end{cases}$$

If we have the endurance, N_R, and want to calculation the stress range $\Delta\sigma_R$:

$$\Delta\sigma_R = \begin{cases} \Delta\sigma_C\left(\frac{N_C}{N_R}\right)^{\frac{1}{m_1}} & for\ N_R\le N_D \\ \Delta\sigma_D\left(\frac{N_D}{N_R}\right)^{\frac{1}{m_2}} & for\ N_D < N_R \le N_L \\ \Delta\sigma_L & for\ N_R > N_L \end{cases}$$

Test data for some details do not exactly fit the fatigue curves.

These are marked with * to avoid non-conservative conditions and are located **one detail category lower** than their fatigue strength at 2×10^6 cycles would require.

An alternative assessment may **increase the classification of these * details by one category provided that the constant amplitude fatigue limit $\Delta\sigma_D$ is defined as the fatigue strength at 10^7 cycles and $m = 3$.**

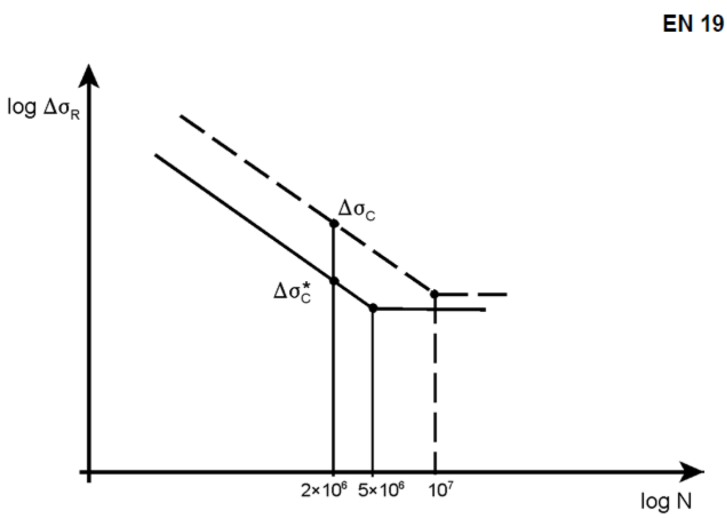


Figure 5: Increase detail category of * details

4.9. Reduction factors from IIW Bulletin 520

4.9.1. Corrosion

- Can reduce the fatigue class and reduce the position of the knee point
- For steel, except stainless steel, in marine environment:
  - For fatigue:
    - Reduce the S-N to 70% - that's just more than one detail category
    - No fatigue limit applies, that is, the knee point disappears
  - For Fracture Mechanics:
    - Increase the crack growth constant by 3 ($3 \times C_o$)
    - No threshold value applies

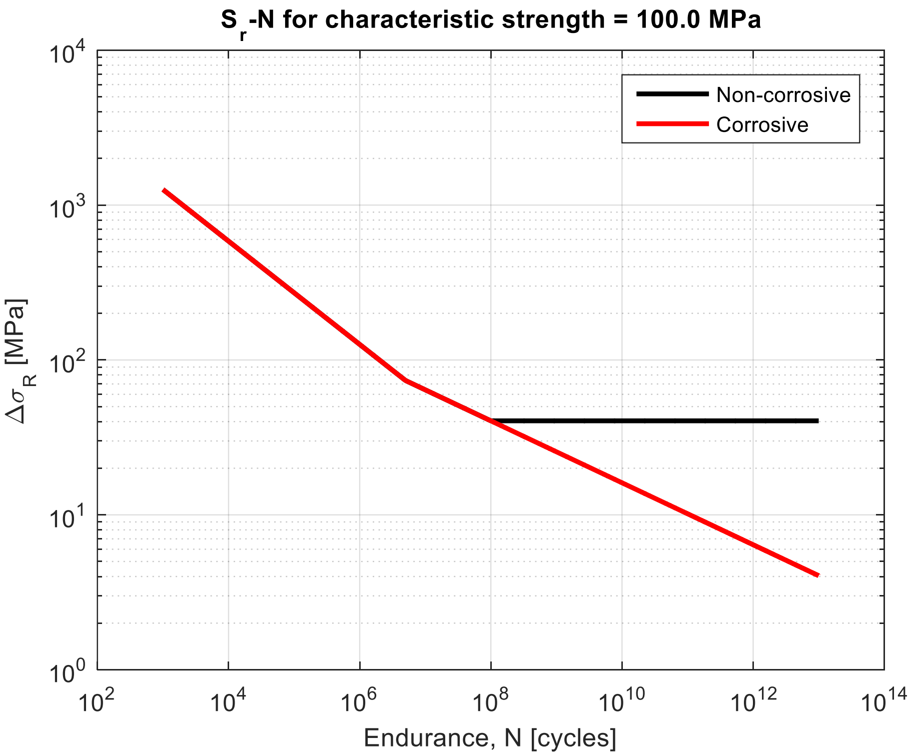


Figure 6: Effect of corrosion on the Sr-N curve

4.9.2. Temperature

Application of the effect of temperature on the BS EN 1993-1-9 curves implies:

$$\Delta_{C,HT} = \Delta\sigma_C \frac{E_{HT}}{E_{20°C}}$$

That is, the fatigue curve is scaled by the change in temperature dependent modulus of elasticity

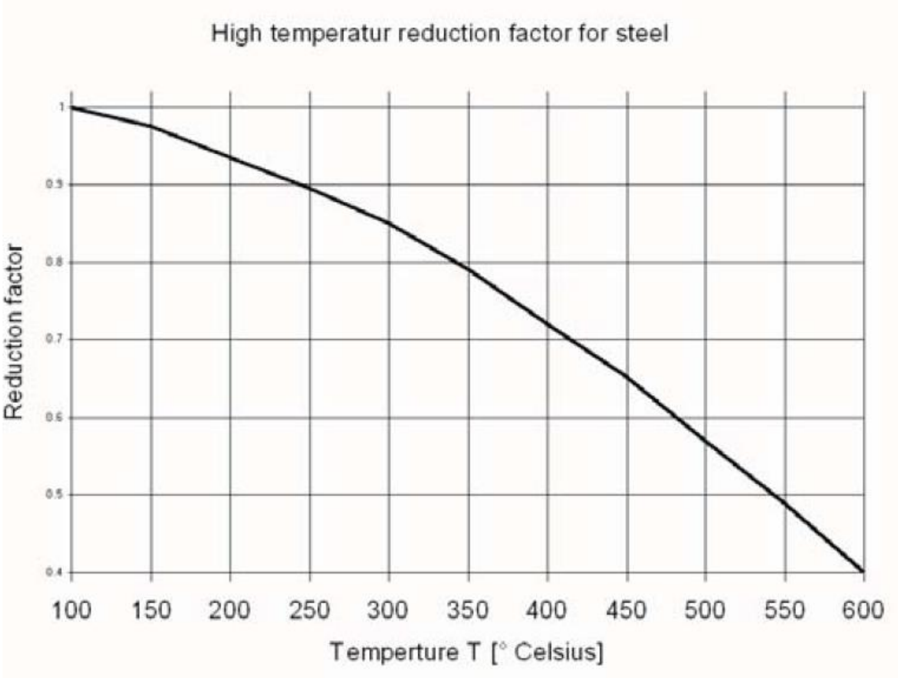


Figure 7: High temperature reduction factor for steel

4.9.3. Post weld improvement

In general

- PW improvement may raise the fatigue resistance
- How:
  - Improve weld profile and reduce stress concentration
    - Machining or grinding of weld seam flush to surface
    - Machining or grinding the weld transition at the toe
    - Remelting the weld toe by TIG-, plasma or laser dressing
  - Control residual stresses
    - Peening (hammer-, needle-, shot-, brush-peening or ultrasonic treatment)
    - Overstressing (proof testing)
    - Stress relieving thermal treatments
  - Improve environmental conditions
    - Painting
    - Resin coating
- Improvement techniques may be used to:
  - Increase the fatigue strength of new structures
  - Ensure sufficient life during repair or upgrading of existing structures
- Applicability
  - All arc welded steel or aluminium components subjected to fluctuating/cyclic stress
  - Structural steel for $f_y \leq 900\ MPa$
  - Weldable structural aluminium alloys commonly used in welded structures
  - Apply to welded joints in:
    - Plates

- Sections built up of plates
- Similar rolled or extruded shapes
- Hollow sections

– Thicknesses:
  - Steel: 5 to 150 mm
  - Aluminium: 4 to 50 mm

– Apply with the nominal stress or structural hot spot stress verification techniques
– Improvement techniques apply solely to the weld toe and hence to a potential fatigue crack growth from this point
– Benefit factors apply to the as-welded joint
– Techniques can be joined (grinding then peening), but, must be proofed by testing to confirm a higher benefit factor than the last improvement
– No benefit factors for joints operating under free corrosion

4.9.3.1. Grinding

- Weld toe fatigue cracks initiate at:
  – Undercuts
  – Cold laps
  – Sharp edge-like imperfections
- Aim:
  – Remove imperfections
  – Create smooth transition between weld and plate

  Therefore, remove stress concentrations
- Benefit factor:

$$\Delta\sigma_C = min\begin{cases}1.3 \times \Delta\sigma_C \\ 112\end{cases}$$

Tab.3.5-2a: FAT classes for use with nominal stress at joints improved by grinding

Area of application and maximum possible claim	Steel	Aluminium
Benefit at details classified in as-welded condition as FAT≤90 for steel or FAT≤32 for aluminium	1.3	1.3
Max possible FAT class after improvement	FAT 112	FAT 45

Tab.3.5-2b: FAT classes for use with structural hot-spot stress at joints improved by grinding

Material	Load-carrying fillet welds	Non-load-carrying fillet welds and butt welds
Mild steel. f_y<350 MPa	112	125
Higher strength steel f_y<350 MPa	112	125
Aluminium alloys	45	50

4.9.3.2. TIG Dressing

- Remelt the toe in order to:
  – Remove imperfections
  – Produce smooth transition from the weld to plate surface

- Apply to PJP and CJP
- Benefit factor:
  - For steels with $f_y \leq 900\ MPa$ and thickness $\geq 10\ mm$

$$\Delta\sigma_C = min\begin{cases} 1.3 \times \Delta\sigma_C \\ 112 \end{cases}$$

Tab.3.5-3a: FAT classes for use with nominal stress at joints improved by TIG dressing

Area of application and maximum possible claim	Steel	Aluminium
Benefit at details classified in as-welded condition as FAT≤90 for steel or FAT≤32 for aluminium	1.3	1.3
Max possible FAT class after improvement	FAT 112	FAT 45

Tab 3.5-3b: FAT classes for use with structural hot-spot stress at joints improved by TIG dressing

Material	Load-carrying fillet welds	Non-load-carrying fillet welds and butt welds
Mild steel. f_y<350 MPa	112	125
Higher strength steel f_y<350 MPa	112	125
Aluminium alloys	45	50

4.9.3.3. Hammer peening

- Plastic deformation at the weld toe:
  - Introduce compressive residual stresses
- Apply for thicknesses:
  - Steel: 10 to 50 mm
  - Aluminium: 5 to 25 mm
  - Arc welded fillet welds with minimum leg length $0.1t$ where t is the thickness of the stressed plate
- Special requirements
  - Maximum of nominal compressive stress including proof loading $< 0.25f_y$
  - Dependent on stress ratio:
    - $R < 0$ effective stress range = $\Delta\sigma$
    - $0 < R \leq 0.4$ effective stress range = maximum applied stress σ
    - $R > 0.4$ no benefit

Tab.3.5-4a: FAT classes for use with nominal stress at joints improved by hammer peening

Area of application and maximum possible claim	Mild steel f_y<355 MPa	Steel f_y ≥355 MPa	Aluminium
Benefit at details classified in as-welded condition as FAT≤90 for steel or FAT≤32 for aluminium	1.3	1.6	1.6
Max possible FAT after improvement	FAT 112	FAT 125	FAT 56

Tab.3.5-4b: FAT classes for use with structural hot-spot stress at joints improved by hammer

Material	Load-carrying fillet welds	Non-load-carrying fillet welds and butt welds
Mild steel. f_y<350 MPa	112	125
Higher strength steel f_y<350 MPa	125	160
Aluminium alloys	56	63

4.9.3.4. Needle peening

Plastic deformation at the weld toe:

- Introduce compressive residual stresses

Special requirements

- Maximum of nominal compressive stress including proof loading $< 0.25f_y$
- Dependent on stress ratio:
    - $R < 0$ effective stress range = $\text{benefit factor} \times \Delta\sigma$
    - $0 < R \leq 0.4$ effective stress range = benefit factor x maximum applied σ
    - $R > 0.4$ no benefit

Tab.3.5-5a: FAT classes with nominal stress at joints improved by needle peening

Area of application and maximum possible claim	Mild steel f_y<355 MPa	Steel f_y ≥355 MPa	Aluminium
Benefit at details classified in as-welded condition as FAT≤90 for steel or FAT≤32 for Aluminium	1.3	1.6	1.6
Max possible FAT after improvement	FAT 112	FAT 125	FAT 56

Tab3.5-5b: FAT classes for use with structural hot-spot stress at joints improved by needle peening

Material	Load-carrying fillet welds	Non-load-carrying fillet welds and butt welds
Mild steel. f_y<350 MPa	112	125
Higher strength steel f_y<350 MPa	125	160
Aluminium alloys	56	63

4.10. Using the ideas

Detail on how to use the ideas are given in the formula sheet, which will be made available during tests and exams. The process to follow if you have stress spectrum and S_r-N curve is summarised below.

$$\Delta\sigma_{C,mod} = \frac{\Delta\sigma_C}{\gamma_{Mf}} C_t C_T C_{PWT}$$

$$\Delta\sigma_D = \Delta\sigma_{C,mod}\left(\frac{N_C}{N_D}\right)^{\frac{1}{m_1}}$$

$$N_R = \begin{cases} \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_1} N_D & m_1 = 3 \quad 1.5f_y > \Delta\sigma_R \geq \Delta\sigma_D \\ \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_2} N_D & m_2 = 5 \quad \Delta\sigma_D > \Delta\sigma_R \geq \Delta\sigma_L \\ \infty & \Delta\sigma_R < \Delta\sigma_L \end{cases}$$

$$N_C = 2 \times 10^6 \ cycles$$
$$N_D = 5 \times 10^6 \ cycles$$
$$N_L = 100 \times 10^6 \ cycles$$

4.11. Problem 1

The flange of a welded steel girder is classified as Detail category 125 according to BS EN 1993-1-9. The component is subject to 500 000 cycles for stress range 200 MPa. Adopt a safe life strategy with low consequence of failure. The partial factor for equivalent constant amplitude stress range is $\gamma_{Ff} = 1.0$. Is this design acceptable?

4.12. Problem 2

The fatigue performance of a welded detail in a steel linkspan structure can be represented by a fatigue curve corresponding to BS EN 1993-1-9 Detail Category 36. The linkspan carries typical vehicles of weight 1, 2 and 5 ton. A linear elastic finite beam element analysis revealed the stress ranges in the welded detail as summarised in the table below with the proportion of vehicles carried by the ferry 70%, 28% and 2% respectively as summarised in the table. The linkspan is used twice per day. No more than one vehicle can occupy the linkspan at any one time. The design life required is equal to the service life of 40 years. Is this design sufficient if a damage tolerant with high consequence of failure strategy is implemented? A total of 50 vehicles are carried per day, and two stress cycles are caused to the linkspan per vehicle (on- and off loading).

Therefore, as an example, the total number of cycles will be:

$$50 \times 2 \times 365 \times 40 = 1\,460\,000$$

of which 70% will be the number of cycles for the 1 tonne vehicles which is $1\,470\,000 \times 0.7 = 1\,022\,000$ cycles. In the same way the number of cycles for the 2 and 5 tonne vehicles were calculated.

Table 2: Linkspan load spectrum

Frequency of vehicles			50	per day
Stress cycles per vehicle			2	
Design life			40	years
Vehicle mass [ton]	**Stress range [MPa]**	**Proportion [%]**	**Applied cycles n_i**	
1	20	70%	1022000	
2	40	28%	408800	
5	100	2%	29200	

4.12.1. Solution

The equation for the Sr-N curve for large scale manufactured components:

$$\Delta\sigma_{C,mod} = \frac{\Delta\sigma_C}{\gamma_{Mf}} C_t C_T C_{PWT}$$

$$\Delta\sigma_D = \Delta\sigma_{C,mod} \left(\frac{N_C}{N_D}\right)^{\frac{1}{m_1}}$$

$$N_R = \begin{cases} \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_1} N_D \quad m_1 = 3 \quad 1.5 f_y > \Delta\sigma_R \geq \Delta\sigma_D \\ \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_2} N_D \quad m_2 = 5 \quad \Delta\sigma_D > \Delta\sigma_R \geq \Delta\sigma_L \\ \infty \quad \Delta\sigma_R < \Delta\sigma_L \end{cases}$$

$$N_C = 2 \times 10^6 \ cycles$$
$$N_D = 5 \times 10^6 \ cycles$$
$$N_L = 100 \times 10^6 \ cycles$$

$$\Delta\sigma_{C,mod} = \frac{\Delta\sigma_C}{\gamma_{Mf}} C_t C_T C_{PWT}$$

4.12.1.1. Partial factor for fatigue

In this case $\gamma_{Mf} = 1.15$.

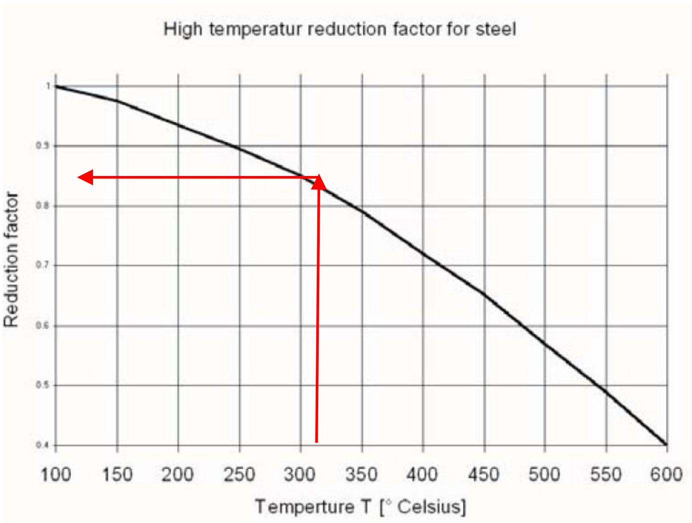
Assessment method	Consequence of failure	
	Low consequence	High consequence
Damage tolerant	1.00	1.15
Safe life	1.15	1.35
Source: BS EN 1993-1-9, 2005:11		

4.12.1.2. Thickness modification

Nothing is said about thickness, so $C_t = 1.0$.

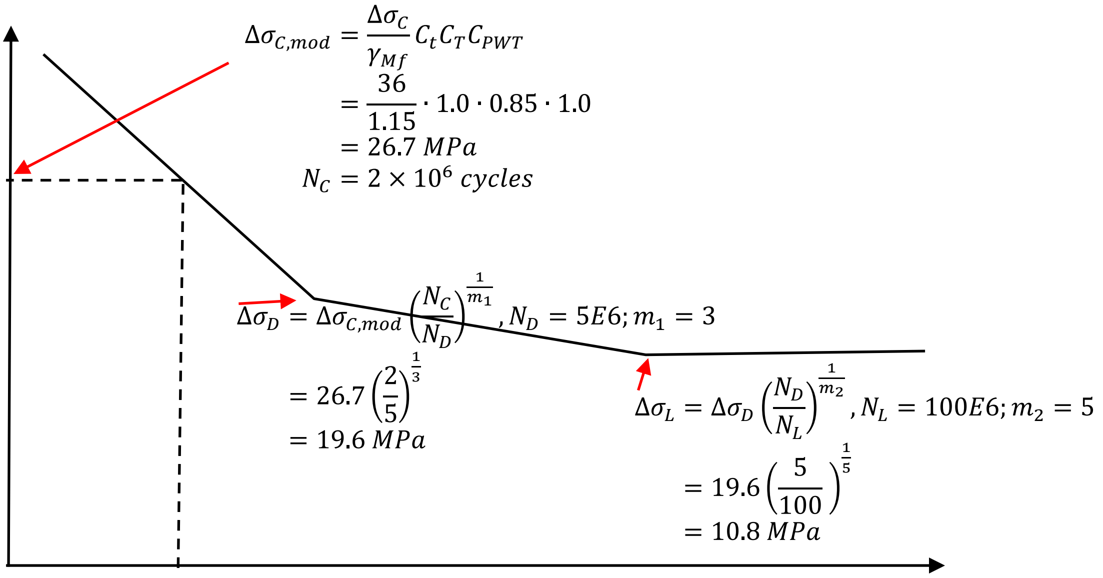
4.12.1.3. Temperature modification

At 300 ℃ $C_T = 0.85$.



4.12.1.4. Post-weld treatment

Nothing is mentioned, take $C_{PWT} = 1{,}0$.



4.12.2. Damage and fatigue

The equation from which the endurance at any stress range can be calculated is as follows:

$$N_R = \begin{cases} \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_1} N_D & m_1 = 3 \quad 1.5 f_y > \Delta\sigma_R \geq \Delta\sigma_D \\ \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_2} N_D & m_2 = 5 \quad \Delta\sigma_D > \Delta\sigma_R \geq \Delta\sigma_L \\ \infty & \Delta\sigma_R < \Delta\sigma_L \end{cases}$$

Frequency of vehicles			50	per day
Stress cycles per vehicle			2	
Design life			40	years
Vehicle mass [ton]	**Stress range [MPa]**	**Proportion [%]**	**Applied cycles n_i**	
1	20	70%	1022000	
2	30	28%	408800	
5	40	2%	29200	

For $\Delta\sigma_R = 20\ MPa$ applied for 1 022 000 cycles we have:

$$N_{R20} = \left(\frac{19.6}{20}\right)^3 \cdot 5 \times 10^6 = 4\,709\,890\ cycles$$

$$d_{20} = \frac{1\,022\,000}{4\,709\,890} = 0.217$$

For $\Delta\sigma_{30} = 30\ MPa$ applied for 408 800 cycles we have:

$$N_{R30} = N_{R20} = \left(\frac{19.6}{30}\right)^3 \cdot 5 \times 10^6 = 1\,395\,522\ cycles$$

$$d_{30} = \frac{408\,800}{1\,395\,522} = 0.293$$

For $\Delta\sigma_{40} = 40\ MPa$ and 29 200 cycles we have:

$$N_{R40} = \left(\frac{19.6}{40}\right)^3 \cdot 5 \times 10^6 = 588\,736\ cycles$$

$$d_{40} = \frac{29\,200}{588\,736} = 0.050$$

For $\Delta\sigma_{15} = 15\ MPa$ and 3 000 000 cycles we have:

$$N_{R15} = \left(\frac{19.6}{15}\right)^5 \cdot 5 \times 10^6 = 19\ 045\ 573\ cycles$$

$$d_{15} = \frac{3\ 000\ 000}{19\ 045\ 573} = 0.156$$

For $\Delta\sigma_7 = 7\ MPa$ and 100 000 000 cycles we have:

$$N_{R7} = \begin{cases} \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_1} N_D & m_1 = 3 \quad 1.5f_y > \Delta\sigma_R \geq \Delta\sigma_D \\ \left(\frac{\Delta\sigma_D}{\Delta\sigma_R}\right)^{m_2} N_D & m_2 = 5 \quad \Delta\sigma_D > \Delta\sigma_R \geq \Delta\sigma_L \\ \infty & \Delta\sigma_R < \Delta\sigma_L \end{cases}$$
$$= \infty$$
$$d_7 = 0$$

The total damage is:

$$\begin{aligned} D &= d_{20} + d_{30} + d_{40} + d_{15} + d_7 \\ &= 0.217 + 0.293 + 0.050 + 0.158 + 0 \\ &= 0.718 \end{aligned}$$

Repetitions to failure:

$$B_f = \frac{1}{D} = 1.4$$

The fatigue life:

$$\begin{aligned} L &= Period \times B_f \\ &= 40 \times 1.4 \\ &= 56\ years \end{aligned}$$

Therefore, the design is OK.

4.13. Fatigue verification where data for $\Delta\sigma_{E,2}$ or $\Delta\tau_{E,2}$ are available

Nominal, modified nominal or geometric stress ranges due to frequent loads $\psi_1 Q_k$ (EN 1990) should not exceed:

$$\Delta\sigma \leq 1.5 f_y \; for\ direct\ stress\ range$$

$$\Delta\tau \leq \frac{1.5 f_y}{\sqrt{3}} \; for\ shear\ stress\ range$$

Verify that under fatigue loading:

$$\frac{\gamma_{Ff}\Delta\sigma_{E,2}}{\frac{\Delta\sigma_C}{\gamma_{Mf}}} \leq 1.0$$

and

$$\frac{\gamma_{Ff}\Delta\tau_{E,2}}{\frac{\Delta\tau_C}{\gamma_{Mf}}} \leq 1.0$$

This design approach is used only for cyclic loading where loads are prescribed at 2 million cycles.

Note, BS EN 1993-1-9 Tables 8.1 to 8.9 require stress ranges to be based on principal stress for some details.

For **combined stress** ranges, except if BS EN 1993-1-9 Tables 8.1 to 8.9 indicate otherwise:

$$\frac{\gamma_{Ff}\Delta\sigma_{E,2}}{\frac{\Delta\sigma_C}{\gamma_{Mf}}} + \frac{\gamma_{Ff}\Delta\tau_{E,2}}{\frac{\Delta\tau_C}{\gamma_{Mf}}} \leq 1.0$$

This implies that damage due to shear and direct stresses at a point must be accumulated.

4.14. Fatigue verification where no data for for $\Delta\sigma_{E,2}$ or $\Delta\tau_{E,2}$ are available

- Loading events
  - Loading sequences that represent credible estimated upper bound of all service load events expected during that fatigue design life
- Stress history
  - Determine from loading events at the structural detail
    - Take into account type and shape of relevant influence lines to be considered and effects of dynamic magnification of structural response
  - Determine stress histories from measurements or dynamic/transient calculations of structural response (finite element modelling or manual)
- Cycle counting
  - Rainflow method
  - Reservoir method

  The result is:

  1. Stress ranges and their number of cycles
  2. Mean stresses, where the mean stress influence needs to be taken into account
- Stress range spectrum
  1. Present stress ranges and associated number of cycles in descending order
  2. May be modified neglecting peak values of stress ranges representing less than 1% of total damage and stress ranges below the cut off limit
  3. May be standardized according to their shape e.g. with the coordinates $\overline{\Delta\sigma} = 1.0\ and\ \overline{\sum n} = 1.0$

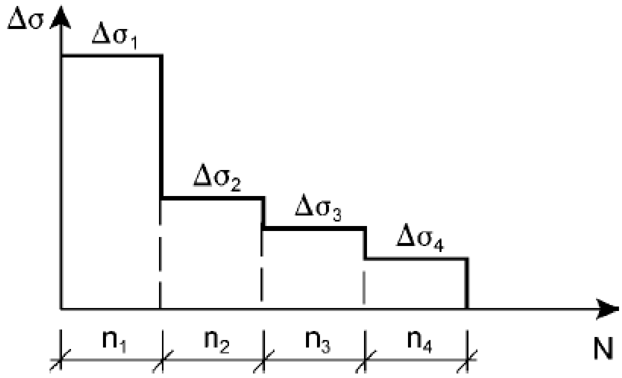


Figure 8: Stress range spectrum

4.15. Fatigue design using applied stresses

- Damage calculation with applied stress ranges $\Delta\sigma_i$:
  - Applied stress ranges shall be factored to obtain stress range to use on the $\frac{\Delta\sigma_C}{\gamma_{Mf}} - N_R$ curve as follows:

$$\Delta\sigma_{Ri} = \Delta\sigma_i \times \gamma_{Ff} where\ \gamma_{Ff} = 1.0\ for\ most\ applications$$

  - Use the $\frac{\Delta\sigma_C}{\gamma_{Mf}} - N_R$ curve to find the **endurance value N_{Ri} at each $\Delta\sigma_{Ri}$**
  - Damage is then:

$$D_d = \sum_{i=1}^{n} \frac{n_{Ei}}{N_{Ri}} \leq 1.0$$

Where

n_{Ei} = number of cycles associated with stress $\gamma_{Ff}\Delta\sigma_i$

N_{Ri} = endurance (in cycles) form the factored $\frac{\Delta\sigma_C}{\gamma_{Mf}} - N_R$ curve for stress range $\Delta\sigma_{Ri} = \gamma_{Ff}\Delta\sigma_i$

From this it is clear that the fatigue curve is dropped in strength from $\Delta\sigma_C - N_R$ to the $\frac{\Delta\sigma_C}{\gamma_{Mf}} - N_R$ curve, and the actual stress is used on this curve. Where applicable, the curve is further dropped by other factors (size, temperature, etc.).

- Verification
  - Based on damage accumulation: $D_d \leq 1.0$
  - Based on stress range at 2 million cycles:

$$\gamma_{Ff}\Delta\sigma_{E,2} \leq \sqrt[m]{D_d}\frac{\Delta\sigma_C}{\gamma_{Mf}},\ where\ m = 3$$

Figure 9 shows the sensitivity for $\Delta\sigma_{E,2}$ for the allowable damage that this stress range may cause for a detail category $\Delta\sigma_C = 100\ MPa$. E.g., if the constant amplitude stress range at 2 x 10^6 cycles may only cause damage of 0.6, the design stress range shall be ~20 MPa (or 1/5 the original strength of $\Delta\sigma_C = 100\ MPa$ in this case).

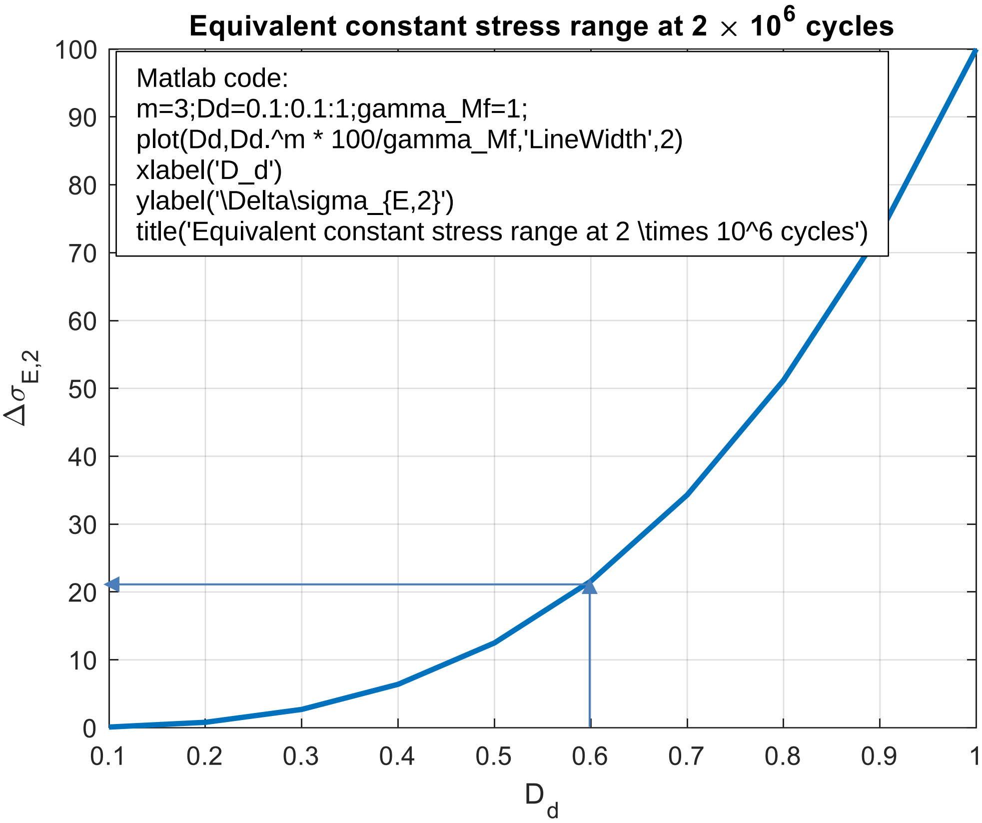


Figure 9: Constant amplitude stress range at 2 x 10⁶ cycles for different damage values

- Conversion of damage by any signal into that by a constant amplitude at any number of cycles
  - Use equivalence of D_d
  - Calculate the fatigue equivalent load Q_e associated with the cycle number $n_{max} = \sum n_i$ or $Q_{E,2}$ associated with cycle number $N_C = 2 \times 10^6$

4.15.1. Stress range at any number of cycles to reconstruct the damage of a spectrum

The following process can be used:

1. Calculate the damage that was caused by the spectrum.
2. Calculate the damage equivalent stress range at 2 x 10^6 cycles:

$$\Delta\sigma_{E,2} = D_d^{\frac{1}{m}} \left(\frac{\Delta\sigma_C}{\gamma_{Mf}}\right)$$

3. Calculate the equivalent stress range at any number of cycles:

$$\Delta\sigma_E = \Delta\sigma_{E,2} \left(\frac{2 \times 10^6}{N_E}\right)^{\frac{1}{m}}$$

4. $m = 3$ because it will not make sense to test below the constant amplitude fatigue limit.

4.15.2. Using the principles

Using the principles: What is the allowable design stress at 100 000 cycles that may be applied if the allowed damage is 70%? Use the characteristic strength as reference.

Theory:

Step 1: Equivalent stress range $\Delta\sigma_2$ on the $\Delta\sigma_R - N_R$ curve at N_2 cycles:

$$\Delta\sigma_2 = \frac{\Delta\sigma_C}{\gamma_{Mf}} \cdot \left(\frac{N_C}{N_2}\right)^{\frac{1}{m}}$$

Remember, there may be a knee-point between the reversals. Calculate accordingly.

Step 2: Calculate the stress range that represents the damage D_d at the reversals required:

$$\Delta\sigma_d = \Delta\sigma_2 D_d^{\frac{1}{m}}$$
$$= \frac{\Delta\sigma_C}{\gamma_{Mf}} \cdot \left(\frac{N_C}{N_2}\right)^{\frac{1}{m}} D_d^{\frac{1}{m}}$$

Step 3: Substitute values

$$\Delta\sigma_{E,10^5} = \frac{\Delta\sigma_C}{\gamma_{Mf}} \left(\frac{N_C}{10^5}\right)^{\frac{1}{3}} D_d^{\frac{1}{3}}$$

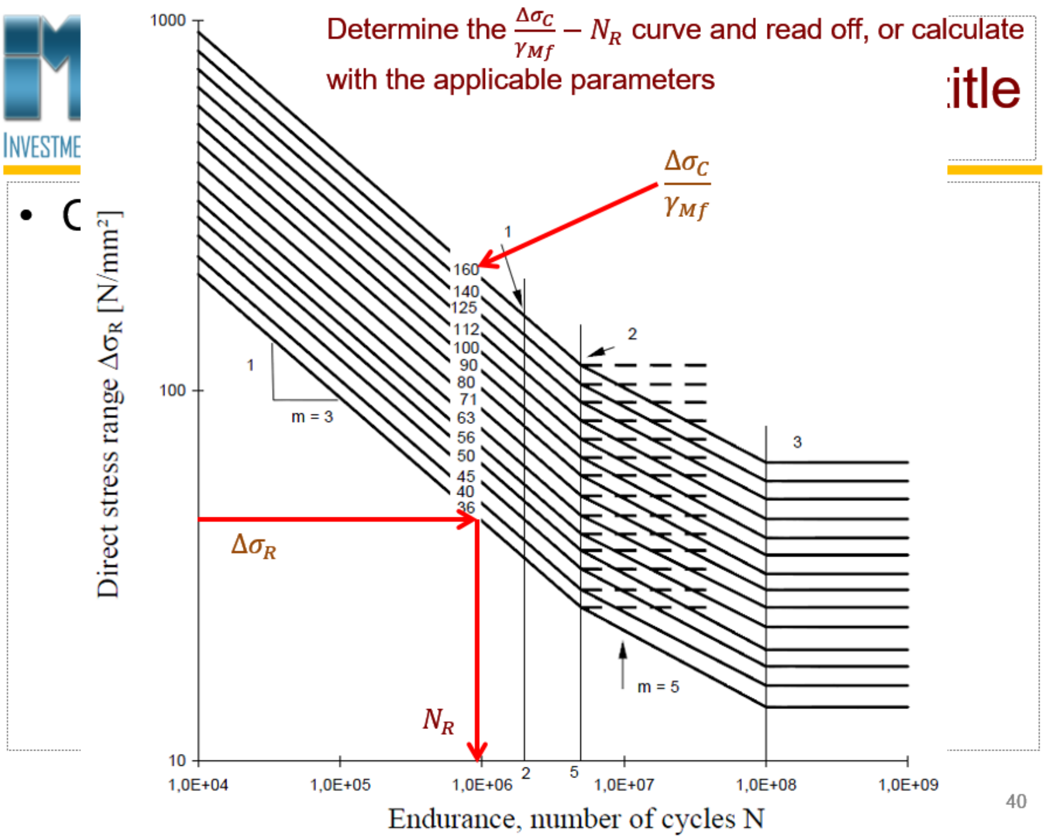


Table 8.2: Welded built-up sections

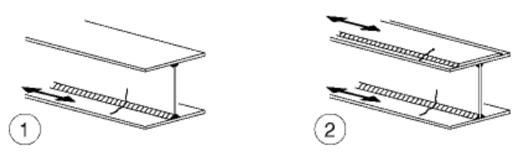
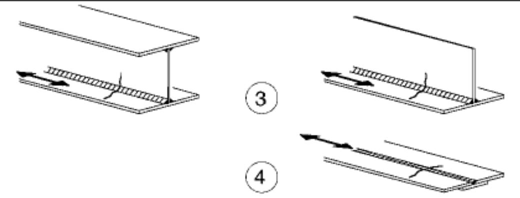
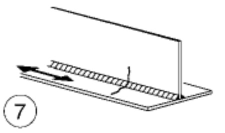
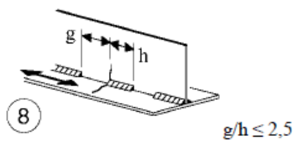
Detail category	Constructional detail	Description	Requirements
125		<u>Continuous longitudinal welds:</u> 1) Automatic butt welds carried out from both sides. 2) Automatic fillet welds. Cover plate ends to be checked using detail 6) or 7) in Table 8.5.	<u>Details 1) and 2):</u> No stop/start position is permitted except when the repair is performed by a specialist and inspection is carried out to verify the proper execution of the repair.
112		3) Automatic fillet or butt weld carried out from both sides but containing stop/start positions. 4) Automatic butt welds made from one side only, with a continuous backing bar, but without stop/start positions.	4) When this detail contains stop/start positions category 100 to be used.
100		5) Manual fillet or butt weld. 6) Manual or automatic butt welds carried out from one side only, particularly for box girders	5), 6) A very good fit between the flange and web plates is essential. The web edge to be prepared such that the root face is adequate for the achievement of regular root penetration without break-out.
100		7) Repaired automatic or manual fillet or butt welds for categories 1) to 6).	7) Improvement by grinding performed by specialist to remove all visible signs and adequate verification can restore the original category.
80		8) Intermittent longitudinal fillet welds.	8) $\Delta\sigma$ based on direct stress in flange.

Table 8.3: Transverse butt welds

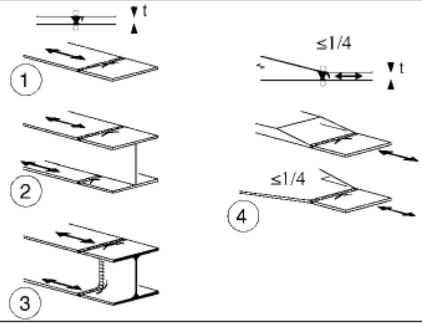
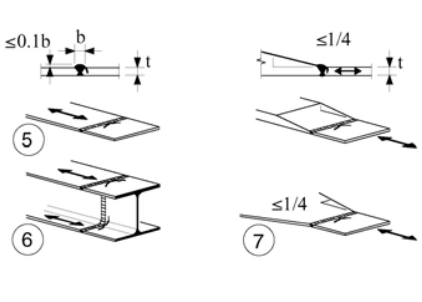
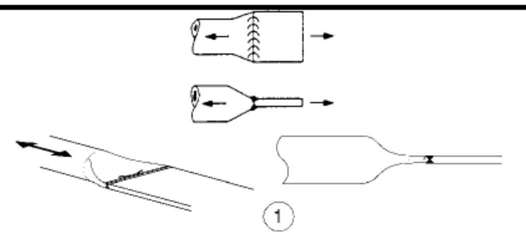
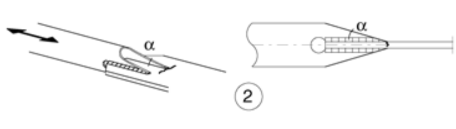
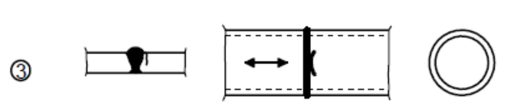
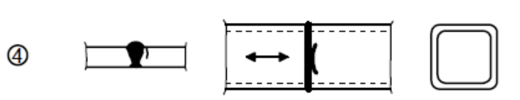
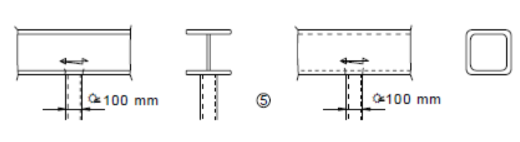
Detail category		Constructional detail	Description	Requirements
112	size effect for t>25mm: $k_s=(25/t)^{0,2}$		<u>Without backing bar:</u> 1) Transverse splices in plates and flats. 2) Flange and web splices in plate girders before assembly. 3) Full cross-section butt welds of rolled sections without cope holes. 4) Transverse splices in plates or flats tapered in width or in thickness, with a slope ≤ ¼.	- All welds ground flush to plate surface parallel to direction of the arrow. - Weld run-on and run-off pieces to be used and subsequently removed, plate edges to be ground flush in direction of stress. - Welded from both sides; checked by NDT. <u>Detail 3):</u> Applies only to joints of rolled sections, cut and rewelded.
90	size effect for t>25mm: $k_s=(25/t)^{0,2}$		5) Transverse splices in plates or flats. 6) Full cross-section butt welds of rolled sections without cope holes. 7) Transverse splices in plates or flats tapered in width or in thickness with a slope ≤ ¼. Translation of welds to be machined notch free.	- The height of the weld convexity to be not greater than 10% of the weld width, with smooth transition to the plate surface. - Weld run-on and run-off pieces to be used and subsequently removed, plate edges to be ground flush in direction of stress. - Welded from both sides; checked by NDT. <u>Details 5 and 7:</u> Welds made in flat position.

Table 8.6: Hollow sections (t ≤ 12,5 mm)

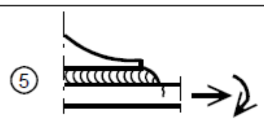
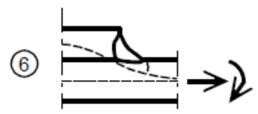
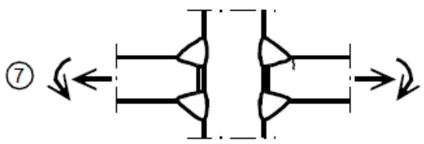
Detail category	Constructional detail	Description	Requirements
71		1) Tube-plate joint, tubes flatted, butt weld (X-groove)	1) Δσ computed in tube. Only valid for tube diameter less than 200 mm.
71	α≤45° 	2) Tube-plate joint, tube slitted and welded to plate. Holes at end of slit.	2) Δσ computed in tube. Shear cracking in the weld should be verified using Table 8.5, detail 8).
63	α>45°		
71		Transverse butt welds: 3) Butt-welded end-to-end connections between circular structural hollow sections.	Details 3) and 4): - Weld convexity ≤ 10% of weld width, with smooth transitions. - Welded in flat position, inspected and found free from defects outside the tolerances EN 1090. - Classify 2 detail categories higher if t > 8 mm.
56		4) Butt-welded end-to-end connections between rectangular structural hollow sections.	
71		Welded attachments: 5) Circular or rectangular structural hollow section, fillet-welded to another section.	5) - Non load-carrying welds. - Width parallel to stress direction ℓ ≤ 100 mm. - Other cases see Table 8.4.

The rest of the tables are available in the prescribed standard.

4.16. Geometric (hot spot) stress method

Applicable for cracks initiating from:

- Toes of butt welds
- Toes of fillet welded attachments
- Toes of fillet welds in cruciform joints

100		5) Bracket ends, ends of longitudinal stiffeners.	5) - Weld toe angle ≤60°. - See also NOTE 2.
100		6) Cover plate ends and similar joints.	6) - Weld toe angle ≤60°. - See also NOTE 2.
90		7) Cruciform joints with load-carrying fillet welds.	7) - Weld toe angle ≤60°. - For misalignment see NOTE 1. - See also NOTE 2.

4.16.1. Hot-spot stress according to BS 7608 from SSE

Note:

- Structural stress range = hot-spot stress range

$$S_H = \begin{cases} Maximum\ principal\ stress\ at\ weld\ toe & Direction \leq 45° \ from\ normal \\ \max \begin{cases} Minimum\ principal\ stress \\ Stress\ normal\ to\ weld\ toe \end{cases} & Direction > 45° \ from\ normal \end{cases}$$

For Type "a" hot-spot stress:

The relationship that was used to determine the hot-spot stresses in the fatigue test database used to validate the choice as Class D as the hot-spot stress-based design curve is as follows for surface stress at distances 0.4t and 1.0t from the weld toe:

$$S_H = 1.67\sigma_{0.4t} - 0.67\sigma_{1.0t}$$

Type "b" hot-spot or structural stress:

For a FEA of relatively fine mesh model, quadratic extrapolation is done from stresses on the plate edge at distances 4 mm, 8 mm and 12 mm from the weld toe as follows:

$$S_H = 3\sigma_{4\ mm} - 3\sigma_{8\ mm} + \sigma_{12\ mm}$$

FEA Model needs a node at 4 mm. Use either 2 mm linear or 4 mm quadratic elements.

For a FEA with a relatively coarse 10 mm x 10 mm quadratic elements, linear extrapolation from surface stress at distances 5 mm and 15 mm from the weld toe can be used:

$$S_H = 1.5\sigma_{5\ mm} - 0.5\sigma_{15\ mm}$$

4.16.2. Hot-spot stress according to BS 7608 from TTI & NF

The membrane stress is:

$$\sigma_m = \frac{\sum\{\sigma[n(i)] + \sigma[n(i+1)]\}(y_{i+1} - y_i)}{2t}$$

The bending stress:

$$\frac{\sigma_b t^2}{6} = \sum f(\sigma[n(i)], \sigma[n(i+1)], y_i, y_{i+1}) - \frac{\sigma_m t^2}{2}$$

$$f(\sigma[n(i)], \sigma[n(i+1)], y_i, y_{i+1}) = \frac{1}{6}[\sigma[n(i)](-2y_i^2 + y_i y_{i+1} + y_{i+1}^2) + \sigma[n(i+1)](-y_i^2 - y_i y_{i+1} + 2y_{i+1}^2)]$$

The hot-spot stress:

$$\sigma_H = \sigma_m + \sigma_b$$

4.17. Fatigue strength modifications

4.17.1. Stress relieving

For non-welded or stress-relived welded details in compression:

$$\Delta\sigma = \sigma_{max} - 0.6\sigma_{min}$$

Therefore, a smaller stress range result.

Can be easily implemented by multiplying signal parts < 0 by 0.6.

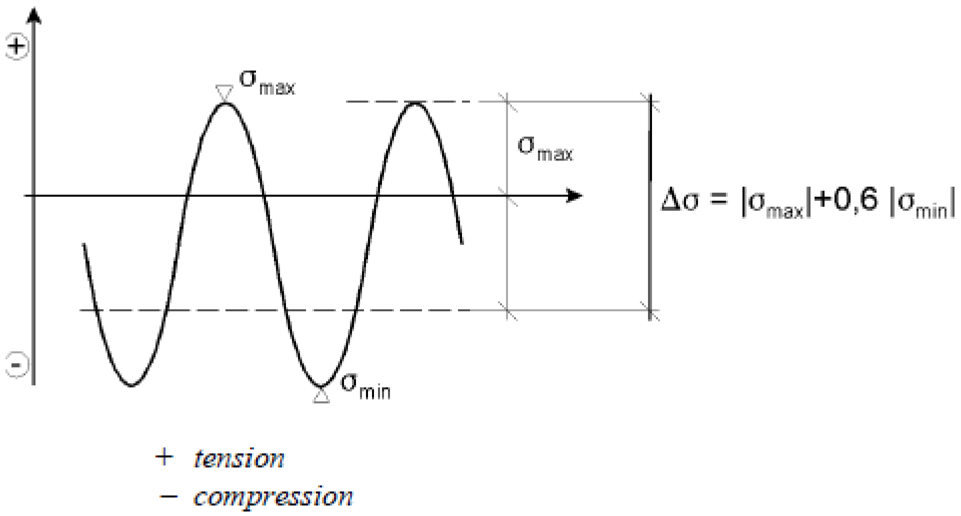
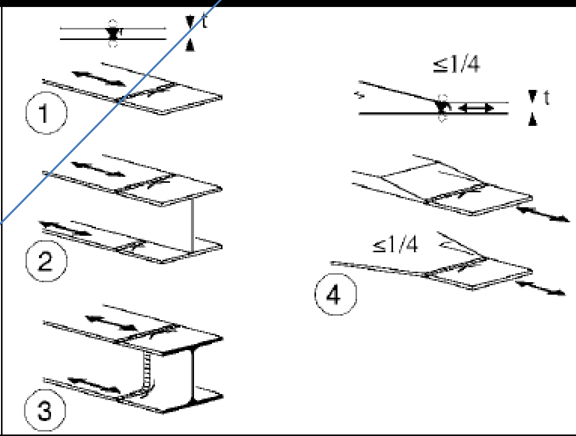


Figure 10: Compressive mean stress effect on some weld detail

4.17.2. Size

Reduce the fatigue curve further by the size reduction factor as follows:

$$\Delta C_{red} = k_s \Delta_C$$

Detail category	Constructional detail		Description
112	size effect for t>25mm: $k_s=(25/t)^{0,2}$		Without backing bar: 1) Transverse splices in plates and flats. 2) Flange and web splices in plate girders before assembly. 3) Full cross-section butt welds of rolled sections without cope holes. 4) Transverse splices in plates or flats tapered in width or in thickness, with a slope ≤ ¼.

Source: (BS EN 1993-1-9, 2005, p. 22)

Figure 11: Size effect on a transverse butt weld

4.18. Example: Allowable stress

A complete joint penetration butt weld made from both sides will be used in a joint between two 16 mm structural steel plates. Weld run-on and run-off pieces were used and subsequently removed; plate edges were then ground flush in the direction of the stress. All welds were ground flush to the plate surface perpendicular to the direction of the weld toe. No indications were found in the weld by NDT. The operating temperature is 350 °C. The design philosophy is infinite life and severe consequence of failure. Assume that the stress is mainly cyclic of nature. What is the allowable stress?

Solution

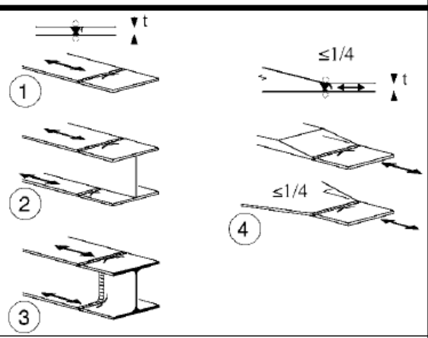
Step 1: Find the partial factor for fatigue

For safe life and high consequence of failure, that partial factor for fatigue strength is $\gamma_{Mf} = 1.35$

Assessment method	Consequence of failure	
	Low consequence	High consequence
Damage tolerant	1.00	1.15
Safe life	1.15	1.35
Source: BS EN 1993-1-9, 2005:11		

Step 2: Find the detail category

The characteristic fatigue strength is $\Delta\sigma_c$= 112 for detail category 112, and, no thickness effect need to be considered because the allowable thickness is 25 mm. Note, the fatigue strength below already makes provision for the grinding.

Detail category	Constructional detail		Description	Requirements
112	size effect for t>25mm: $k_s=(25/t)^{0,2}$		Without backing bar: 1) Transverse splices in plates and flats. 2) Flange and web splices in plate girders before assembly. 3) Full cross-section butt welds of rolled sections without cope holes. 4) Transverse splices in plates or flats tapered in width or in thickness, with a slope ≤ ¼.	- All welds ground flush to plate surface parallel to direction of the arrow. - Weld run-on and run-off pieces to be used and subsequently removed, plate edges to be ground flush in direction of stress. - Welded from both sides; checked by NDT. Detail 3): Applies only to joints of rolled sections, cut and rewelded.

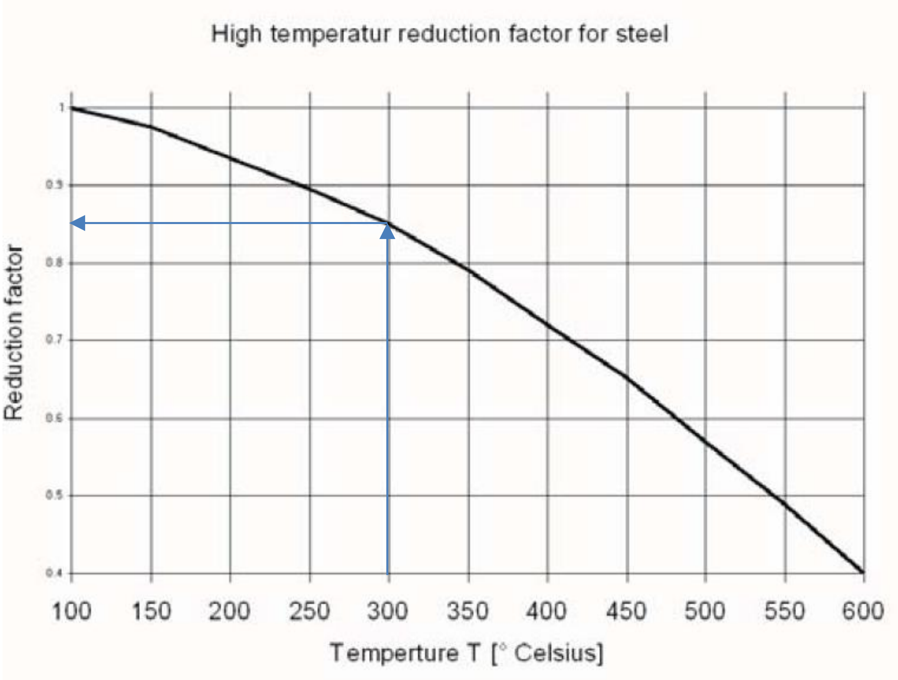
Step 3: Reduction factors

Temperature

Application of the effect of temperature on the BS EN 1993-1-9 curves implies:

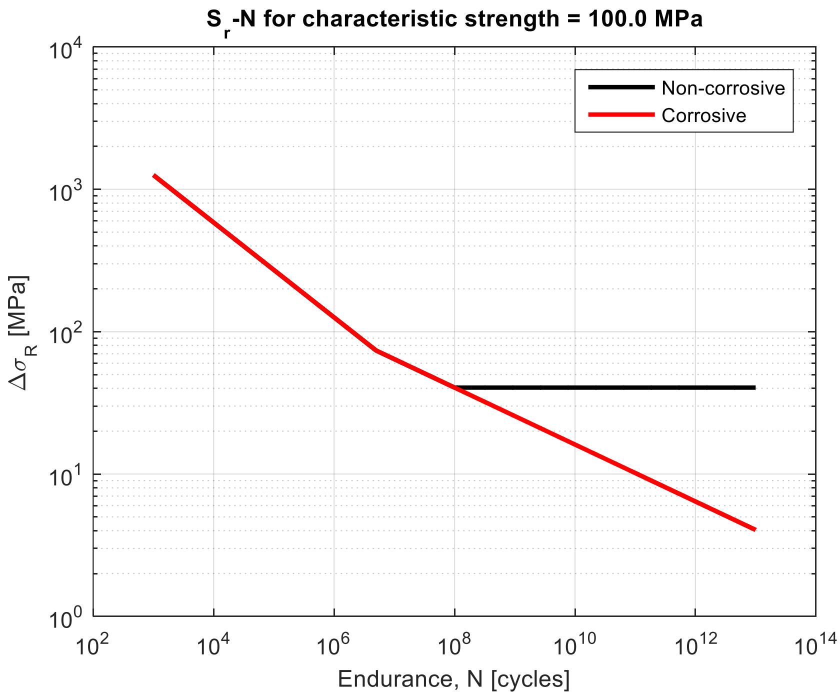
$$\Delta_{C,HT} = \Delta\sigma_C \frac{E_{HT}}{E_{20°C}}$$

$$k_T = 0.85$$



Corrosion

No indication was given on corrosion. It will be assumed that the component is surface protected. If there was no surface protection and the component is subject to a corrosive environment, the allowable stress range is 0 MPa.



Modification factor: Post Weld Treatment

The fatigue strength already makes provision for grinding and no additional factor is needed. If this was not the case, modify for grinding.

Step 4: Calculate allowable stress

No number of cycles was given in this example. With this as unknown, the allowable stress range will be constant amplitude fatigue limit.

The reduced characteristic fatigue strength at $N_C = 2 \times 10^6$ cycles is:

$$\Delta\sigma_{C,red} = \frac{\Delta\sigma_C}{\gamma_{Mf}} k_s k_T k_{PWT}$$
$$= \frac{112}{1.35} \times 0.85 \times 1 \times 1$$
$$= 70.5\ MPa$$

The constant amplitude fatigue limit, $\Delta\sigma_D$ is at endurance $N_D = 5 \times 10^6$ cycles:

$$\Delta\sigma_D = \Delta\sigma_{C,red} \left(\frac{N_C}{N_D}\right)^{\frac{1}{m_1}}$$
$$= 70.5 \left(\frac{2}{5}\right)^{\frac{1}{3}}$$
$$= 52\ MPa$$

Note, in a corrosive environment the endurance limit disappears.

4.19. Example 2

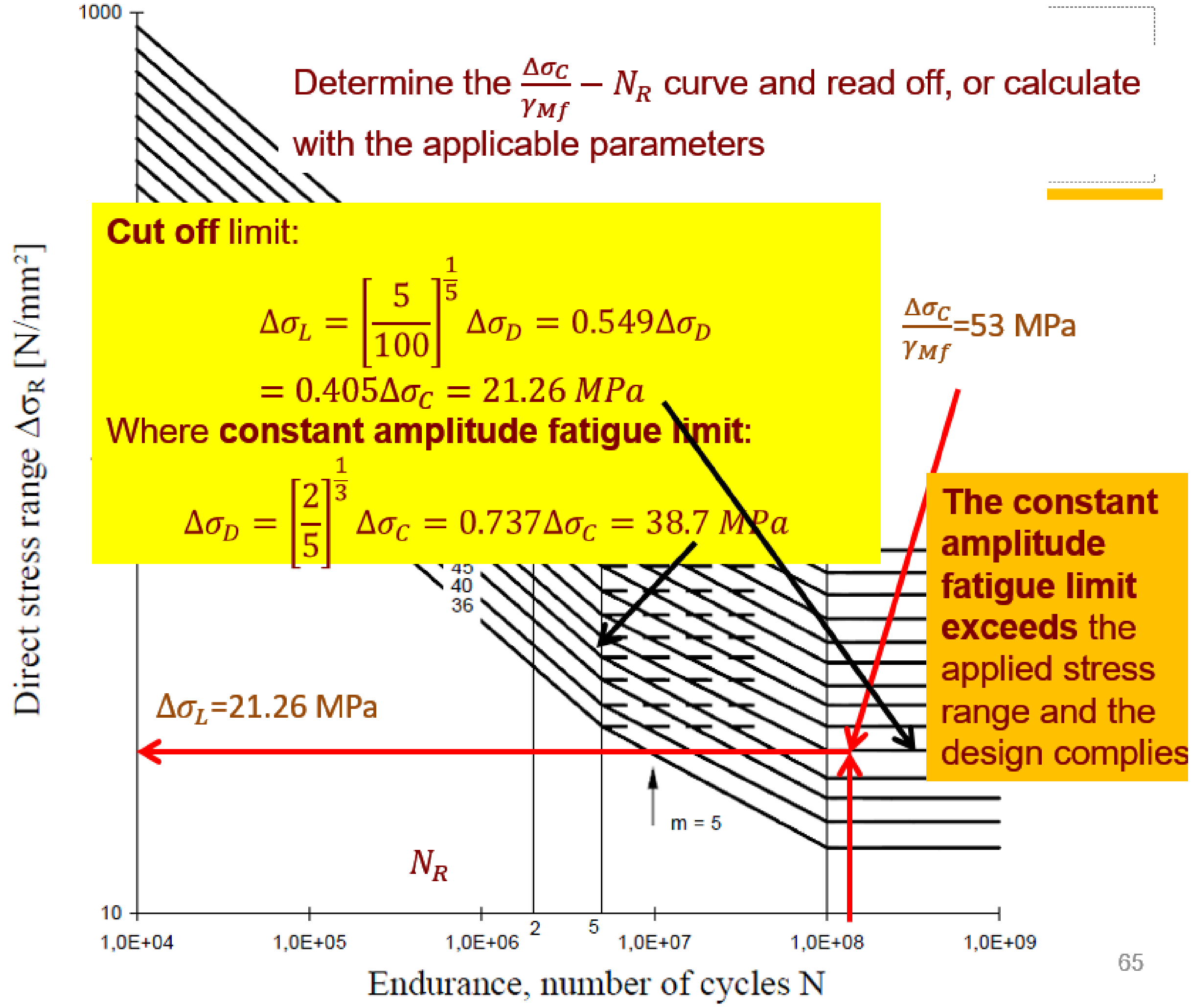
Problem statement

- If a design was made to have a maximum stress range in the structural detail shown below of 25 MPa (nominal stress calculated by FEA – include part geometry but not that of weld) for the following joints:
  - As-welded joints
    - CJP Butt welds made from one side, non-destructive examination tested to confirm absence of defects
    - CJP T-joins and C-joints from both sides
  - No post-weld improvements and treatment
  - Operating at room temperature
  - Surface is corrosion protected
- The joint will be exposed to 300 million cycles
- Use safe-life assessment method and high consequence of failure
- Do you expect crack initiation with confidence level of 75% of a 95% probability of survival?

Solution

- Use standard fatigue strength from Tables 8.1 to 8.10. Find the lowest fatigue curve from the detail mentioned above
  - Value found is $\Delta\sigma_c$=71
- The partial factor for fatigue strength is γ_{Mf}=1.35
- The S-N curve to use $\frac{\Delta\sigma_c}{\gamma_{Mf}}$ = 52.6 MPa

71	size effect for t>25mm: $k_s=(25/t)^{0.2}$		13) Butt welds made from one side only when full penetration checked by appropriate NDT.


Determine the $\frac{\Delta\sigma_C}{\gamma_{Mf}} - N_R$ curve and read off, or calculate with the applicable parameters
Cut off limit:
$\Delta\sigma_L = \left[\frac{5}{100}\right]^{\frac{1}{5}} \Delta\sigma_D = 0.549\Delta\sigma_D$
$= 0.405\Delta\sigma_C = 21.26\ MPa$
Where **constant amplitude fatigue limit**:
$\Delta\sigma_D = \left[\frac{2}{5}\right]^{\frac{1}{3}} \Delta\sigma_C = 0.737\Delta\sigma_C = 38.7\ MPa$
$\frac{\Delta\sigma_C}{\gamma_{Mf}}$=53 MPa
The constant amplitude fatigue limit exceeds the applied stress range and the design complies
$\Delta\sigma_L$=21.26 MPa
N_R
m = 5
45
40
36
1000
10
1,0E+04
1,0E+05
1,0E+06
1,0E+07
1,0E+08
1,0E+09
2
5
Direct stress range $\Delta\sigma_R$ [N/mm²]
Endurance, number of cycles N
65

4.20. Towers, masts and chimneys

Applicable standard: BS EN 1993-3-2:2006 Section 9 for Chimneys

Summary of fatigue on Chimneys:

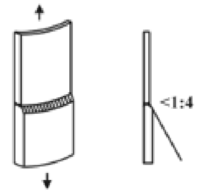
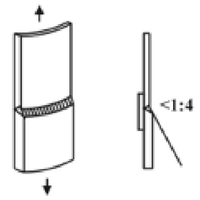
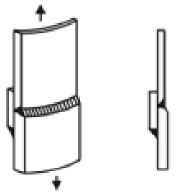
- General
  - Consider fatigue effects from stress ranges induced by in-line forces and cross wind forces
    - Fatigue from cross wind vortex vibrations normally governs design
  - Consider temperature induced damage with fatigue damage for chimneys made of heat resistant alloy steels and used at $T > 400\ °C$
  - Apply BS EN 1993-1-9 for fatigue
- Along-wind vibrations
  - Take gust effect into account (Gust effect factors are used to assess the effect of the structures' dynamic response at resonance for different damping factors by static design criteria, which is caused by transient speed changes in the wind speed)
  - Apply BS EN 1993-3-1 Paragraph 9.2.1
- Cross-wind vibrations
  - Determine stress ranges and number of cycles from BS EN 1991-1-4 Annex E Paragraphs 2.4 and 1.5.2.6
  - No fatigue verification needed for chimneys lower than 3 m in height
  - If the critical speed of the chimney for vortex excitation is $> 20\ m/s$ the correlation lengths below 16 m above ground need not be taken into account (EN 1991-1-4)
  - Consider higher modes where the critical wind speed for those modes is below the limiting value (EN 1991-1-14)
- High cycle fatigue resistance
  - Find detail categories according to BS EN 1993-1-9
  - If there is corrosion allowance for plate thickness instead of corrosion protection system, classify the detail one category lower than given in BS EN 1993-1-9 tables
- Safety assessment
  - Use $\Delta\sigma_{E,2} = \lambda\Delta\sigma_E$
  - λ is the equivalence factor to transfer $\Delta\sigma_E$ to $N_C = 2 \times 10^6$ cycles
  - $\Delta\sigma_E$ is the stress range associated with N cycles
  - Equation:

$$\lambda = \left(\frac{N}{2 \times 10^6}\right)^{\frac{1}{m}}$$

  - Use $\gamma_{Ff} = 1.0$ and γ_{Mf} according to the assessment method and consequence of failure

The BS EN 1993-3-2 standard allocates detail to BS EN 1993-1-9 detail categories for most weld detail, of which the section below is an extraction

Table C.1 Allocation of details to detail categories

Reference	Sketch of the detail	Description
EN 1993-1-9 Table 8.3 Detail 4 and 7		Transverse splices in shell. Butt weld carried out from both sides.
EN 1993-1-9 Table 8.3 Detail 14		Transverse splices in shell. Butt weld made from one side only.
EN 1993-1-9 Table 8.3 Detail 16 (<1:4)		Transverse splices in shell. Butt weld made on a permanent backing strip.

4.21. Steel bridges

Standard: BS EN 1993-2:2006 Section 9

Available for non-commercial purposes at: https://law.resource.org/pub/eur/ibr/en.1993.2.2006.pdf

For fatigue load models use: BS EN 1991-2:2003 Section 4.6

- The applicable section of the standard will be paged through to demonstrate the application of the method

4.22. Cranes

Table 2 — Groups of loads and dynamic factors to be considered as one characteristic crane action

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Load	Symbol	Clause	Groups of loads										
			Ultimate limit state								Test load		
			1	2	3	4	5	6	7	8	9		
Self-weight of crane	$Q_{c,k}$	5.6	ϕ_1	ϕ_1	1	ϕ_4	ϕ_4	ϕ_4	ϕ_4	1	ϕ_1		
Hoist load	$Q_{hl,k}$	5.6	ϕ_2	ϕ_3	–	ϕ_4	ϕ_4	ϕ_4	ϕ_4	–	–		
Part of hoist load	$\eta Q_{hl,k}$ [a]	5.6	–	–	–	–	–	–	–	1			
Acceleration of crane bridge	H_T, H_L	5.7	ϕ_5	ϕ_5	ϕ_5	ϕ_5	–	–	–	–	ϕ_5		
Skewing of crane bridge	H_S	5.7	–	–	–	–	1	–	–	–	–		
Acceleration or braking of crab or hoist block	$H_{T,3}$	5.7	–	–	–	–	–	1	–	–	–		
Misalignment of crane wheels or gantry rails	H_M	5.7	–	–	–	–	–	–	1	–	–		
Test load	Q_T	5.10	–	–	–	–	–	–	–	–	ϕ_6		
Buffer force	H_B	5.12	–	–	–	–	–	–	–	–	–		
Tilting force	H_{TA}	5.12	–	–	–	–	–	–	–	–	–		

[a] $\eta Q_{hl,k}$ is the part of the hoist load that remains when the payload is removed, but is not included in the self-weight of the crane.

Cranes

Standard: SANS 10160-6, Section 5.11

Fatigue loads:

- Effects of fatigue on crane supporting structures shall be considered
  - Carried out for loading groups 1, 2, 3, 4, 6, 7 and 8
- Number of stress cycles for fatigue shall be determined in accordance with the intended use and design life of the structure

Process followed:

- Calculated stress response for load cases
- Determine stress range
- Apply assessment method and consequence of failure
- Do fatigue according to SANS 10162-1:2005 Section 26

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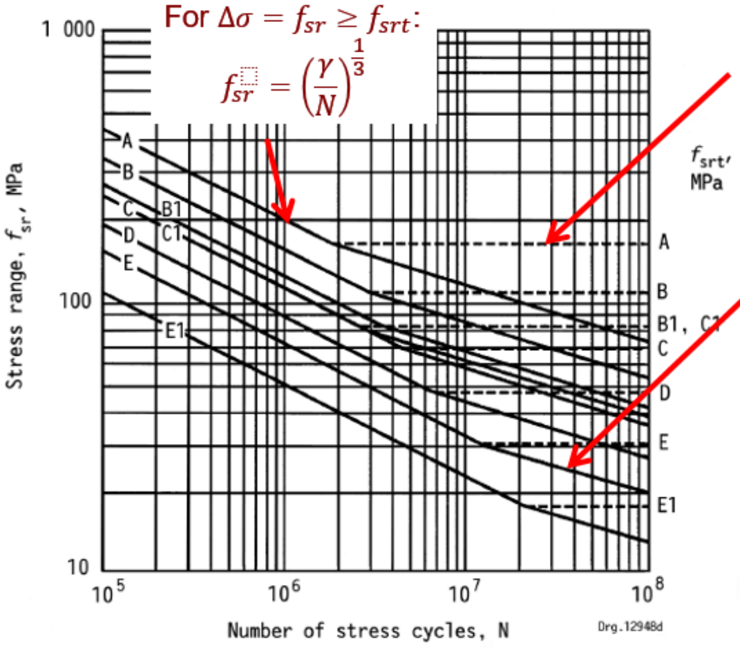
4.23. SANS 10162-1 fatigue curves

For $\Delta\sigma = f_{sr} \geq f_{srt}$:

$$f_{sr} = \left(\frac{\gamma}{N}\right)^{\frac{1}{3}}$$



NOTE Dotted lines show values of f_{srt}. (SANS 10162-1, 2005:86)

Constant amplitude threshold stress range, similar to the BS EN 1993-1-9 constant amplitude fatigue limit

For $\Delta\sigma = f_{sr} < f_{srt}$:

$$f'_{sr} = \left(\frac{\gamma'}{N}\right)^{\frac{1}{5}}$$

Obtain γ' by setting:

$$\left(\frac{\gamma}{N_{f_{srt}}}\right)^{\frac{1}{3}} = \left(\frac{\gamma'}{N_{f_{srt}}}\right)^{\frac{1}{5}}$$

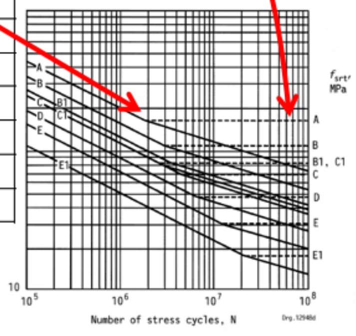
at $f_{sr} = f_{srt}$

70

Table 9 — Fatigue constant for various detail categories

1	2	3	4	5
Detail category	Fatigue life constant γ MPa	Constant amplitude threshold stress range f_{srt} MPa	Cycles $n{\cdot}N'$	Fatigue life constant γ', MPa
A	8 190 x 10^9	165	1,82 x 10^6	223 x 10^{15}
B	3 930 x 10^9	110	2,95 x 10^6	47,6 x 10^{15}
B1	2 000 x 10^9	83	3,50 x 10^6	13,8 x 10^{15}
C	1 440 x 10^9	69	4,38 x 10^6	6,86 x 10^{15}
C1	1 440 x 10^9	83	2,52 x 10^6	9,92 x 10^{15}
D	721 x 10^9	48	6,52 x 10^6	1,66 x 10^{15}
E	361 x 10^9	31	12,1 x 10^6	0,347 x 10^{15}
E1	128 x 10^9	18	22,0 x 10^6	0,0415 x 10^{15}

(SANS 10162-1, 2005:85)



NOTE Dotted lines show values of f_{srt}.

Slope of the curve is $m = 3$ for $\Delta\sigma = f_{sr} \geq f_{srt}$ and $m = 5$ for $\Delta\sigma = f_{sr} < f_{srt}$ 71

4.23.1. Fatigue according to SANS 10162-1

Standard: SANS 10162-1:2005 Section 26

General

- Members shall comply with STATIC conditions and FATIGUE
- Maximum loads are specified by the standard
  - Specified loads less than the maximum specified loads that occur for large number of cycles may govern failure and must be considered
- Design, detail and fabricate members and connections to minimize stress concentrations and abrupt changes in cross-section
- Take life as 50 years except if indicated otherwise
- Sizing of members for fatigue shall be done where loads are repetitive

<u>Live load induced fatigue</u>

- Use elastic analysis and principles of mechanics of materials to calculate stress range
- Only stress range due to live load need to be calculated
- Load-induced fatigue provisions need be applied only at locations that undergo a net applied tensile stress
  - That is, stress ranges that are completely in compression need not be investigated for fatigue
- Design criteria for load-induced fatigue:

$$f_{tr} \geq f_{sr} = \left(\frac{\gamma}{nN}\right)^{\frac{1}{3}} \geq f_{srt}$$

Where:

f_{tr} is the fatigue resistance

f_{sr} is the calculated stress range at the detail due to the variable load

γ is the fatigue constant

n is the number of stress ranges applied at a given detail

N is the number of applications of the load

f_{srt} is the constant amplitude threshold stress range

<u>Total damage:</u>

Apply Miner's rule:

$$D = \sum \frac{(nN)_i}{N_{s_i}} \leq 1.0$$

Where:

$(nN)_i$ is the number of expected stress ranges at $\Delta\sigma_i$

N_{si} is the endurance at $\Delta\sigma_i$

Limited number of cycles

Except for fatigue sensitive details with high $\Delta\sigma$, and compliance is achieved with STATIC requirements (where factored loads are used), fatigue life calculations are not required if:

$$nN < max\begin{cases} \frac{\gamma}{f_{sr}^3} \\ 20\,000 \end{cases}$$

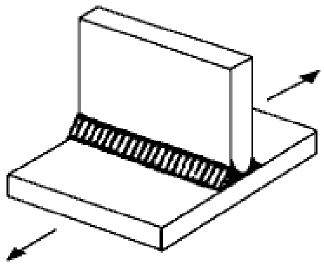
Detail categories: See SANS 10162-1:2005 pp. 85-94

4.24. Class problem

Problem statement

The 12 mm stiffener is welded to the 12 mm plate with 8 mm fillet welds as shown in the figure below. The nominal stress range in the plate is $\Delta\sigma = 100\ MPa$ with $R = -1$.

1. What is the fatigue life of the weld detail for a 95% probability of survival using the following methods:
    1. Nominal stress method
    2. Weld notch analysis method



Solution

Step 1: Joint classification

The joint is classified as FAT 80 joint, Type 511.

Step 2: Joint description

- Transverse non-load-carrying attachment, not thicker than the main plate
- Two sided fillet welds, as welded
- Note, that an angular misalignment corresponding to $k_m = 1.2$ is already covered

Step 3: Threshold stress range

The stress applied to the joint exceeds the FAT class and life below 2 million cycles is expected

Step 4: Stresses to use

Use nominal stress in the stressed plate

Step 5: Modification factors to consider

Grinding: Not applicable.

Hammer and needle peening: Not applicable.

TIG Dressing: Not applicable.

Thickness: Not required, plate is less than 25 mm thick

Corrosion: Not required.

Temperature: Not required.

Mean stress: Not required. No enhancement possible in this case

Safety factor: Not required.

Step 6: Partial factor for fatigue resistance

- Normally the characteristic stress range $\Delta\sigma_{R,k}$ is determined from the S-N curve at the number of cycles and modified to obtain the design stress range $\Delta\sigma_{S,d} \leq \frac{\Delta\sigma_{R,k}}{\gamma_M}$
- Partial safety factor obtained as shown in the table below

The design stress range was given in this case as 100 MPa

The characteristic stress range to use for life calculation is then:

$$\begin{aligned}\Delta\sigma_{R,k} &= 1.4 \times \Delta\sigma_{S,d} \\ &= 140\ MPa\end{aligned}$$

Tab.6.4-4: Possible example for partial safety factors $Y_M$for fatigue resistance

Partial safety factory Y_M – Consequence of failure	Fail safe and damage tolerant strategy	Safe life and infinite life strategy
Loss of secondary structural parts	1.0	1.15
Loss of the entire structure	1.15	1.30
Loss of human life	1.30	1.40

Step 7: Obtain the $\Delta\sigma - N$ curve parameters

The FAT class represents the stress range at a life of 2 million cycles

$$N = \frac{C}{\Delta\sigma^m}$$

$m = 3$ for this problem

Therefore, C is:

$$\begin{aligned}N &= \frac{C}{\Delta\sigma^m} \\ C &= N\Delta\sigma^m \\ &= 2 \times 10^6 \times 10^3 \\ &= 1.024 \times 10^{12}\end{aligned}$$

Step 8: Calculate endurance from the $\Delta\sigma - N$ curve

Stress range = 140 MPa

$$\begin{aligned}N &= \frac{C}{\Delta\sigma} \\ &= \frac{1.024 \times 10^{12}}{140^3} \\ &= 373{,}177\ cycles\end{aligned}$$

4.24.1. Solve using the Notch stress approach

Step 1: Joint fatigue resistance

- A finite element model of the joint was created modelling weld toe and root radius as 1 mm
- Stress was calculated in the stress concentrations and compared with the applicable S-N curve (FAT class) of 225 as shown in the tables below

Step 2: Calculate the maximum stress in the weld stress concentration

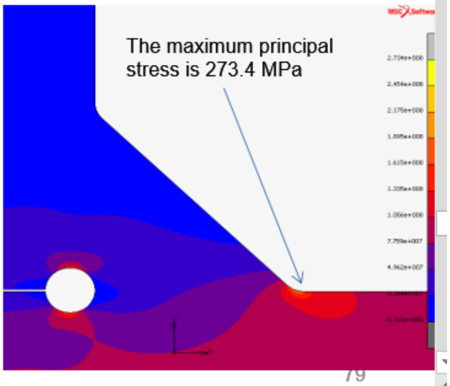
From the finite element analysis the stress was calculated as 273.4 MPa in the weld toe

Tab. {3.4}-1: Effective notch fatigue resistance for steel

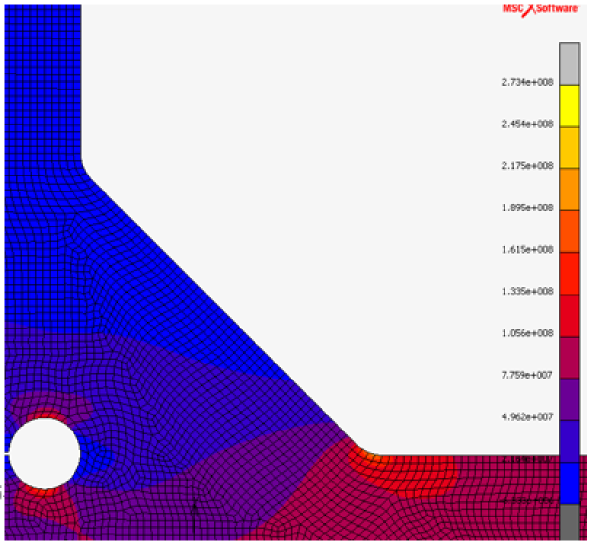
No.	Quality of weld notch	Description	FAT
1	Effective notch radius equalling **1 mm** replacing weld toe and weld root notch	Notch as-welded, normal welding quality m=3	225

Tab. {3.4}-2: Effective notch fatigue resistance for aluminium

No.	Quality of weld notch	Description	FAT
1	Effective notch radius equalling **1 mm** replacing weld toe and weld root notch	Notch as-welded, normal welding quality m=3	71



- Element size need to be small enough to obtain accurate stress distributions in the stress concentrations
- Note, in this case, crack initiation is expected to first occur at the weld toe
- Notch stress at the root lower



- The characteristic stress range:

$$\Delta\sigma_{R,k} = 1.4 \times \Delta\sigma_{S,d}$$
$$= 1.4 \times 273.4$$
$$= 382.2\ MPa$$

- Calculate the parameter C for $m = 3$:

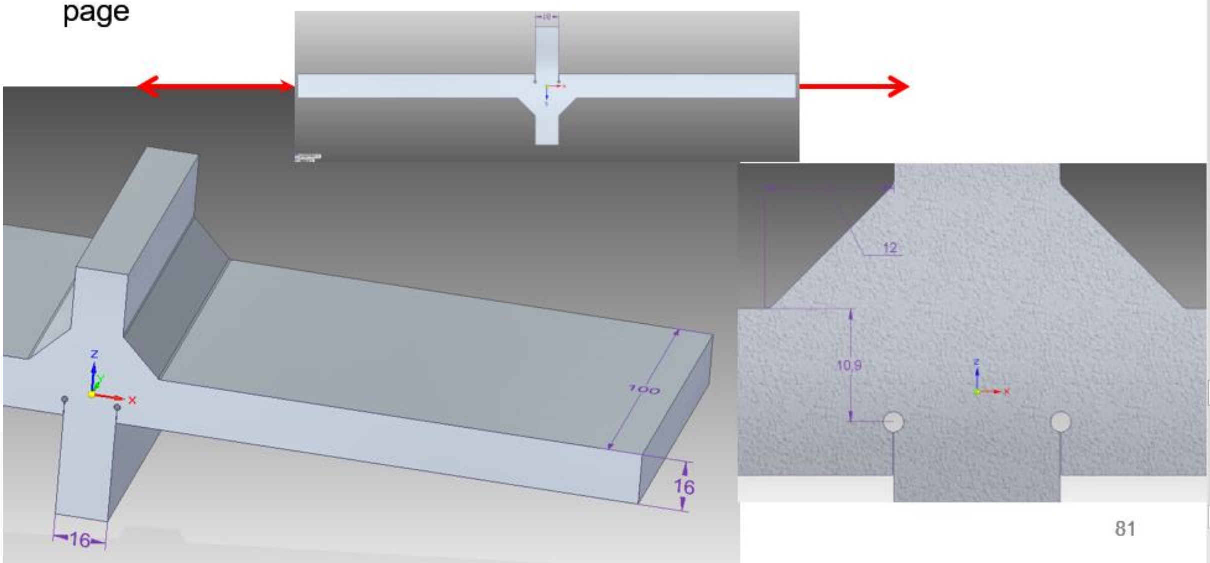
$$N = \frac{C}{\Delta\sigma^m}$$
$$C = N\Delta\sigma^m$$
$$= 2 \times 10^6 \times 225^3$$
$$= 2.278 \times 10^{13}$$

- Number of cycles to failure is then:

$$N = \frac{C}{\Delta\sigma}$$
$$= \frac{2.278 \times 10^{13}}{382.2^3}$$
$$= 408{,}042\ cycles$$

4.25. Misalignment in welded joints

The cruciform joint is made between plates with thickness 16 mm
The fillet weld is a 12 mm weld
The weld is loaded in the directions shown according to the table on the next page



Loading

From a strain measurement far away from the weld detail, the following nominal stress ranges and number of cycles were calculated

- $\Delta\sigma$ – The nominal stress range
- n_i - number of cycles at each stress range

$\Delta\sigma$	n_i
5	1 000 000
10	500 000
15	100 000
20	50 000
25	20 000
30	10 000

The stress spectrum above is applicable over a period of one year

The design shall make provision for a safe life and infinite life strategy because the possible consequence of failure is loss of human life
The component is used under room temperature conditions

Questions:

What is the fatigue life of the component in months according to:

Assessment using S-N curves
Weld notch stress analysis

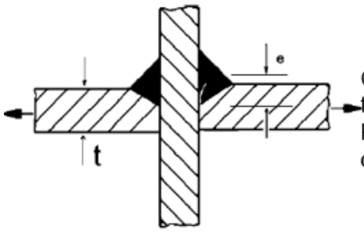
What weld improvements would you recommend and why?

Solution: Apply nominal stress and S-N curves

- In this case, the "design" stress is the stress measured on the component
- The characteristic stress to use on the S-N curves is:

$$\Delta\sigma_{R,k} = \Delta\sigma_{S,d} \times \gamma_M$$

Where, the partial safety factor was selected as 1.4



Crack initiation first to occur at the weld root
Remember to also check for crack initiation at weld toe

Joint classification

- The joint is classified as FAT 71 for steel and FAT 25 for Aluminium and is of Type 416

Where to calculate stress:

Analysis shall be based on axial and bending stress in the weld throat. The eccentricity e to be considered as follows to calculate stress at the weld root:

$$\Delta\sigma_{w,root} = \Delta\sigma_{w,nom} \times \left(1 + \frac{6e}{a}\right)$$

Where:
e is the eccentricity between midpoints plate
a is the weld throat, including the penetration, rotated into the vertical leg plane

Tab. {6.4}-4: Possible example for partial safety factors γ_M for fatigue resistance

Partial safety factor γ_M – Consequence of failure	Fail safe and damage tolerant strategy	Safe life and infinite life strategy
Loss of secondary structural parts	1.0	1.15
Loss of the entire structure	1.15	1.30
Loss of human life	1.30	1.40

Analyse geometry to determine a and e

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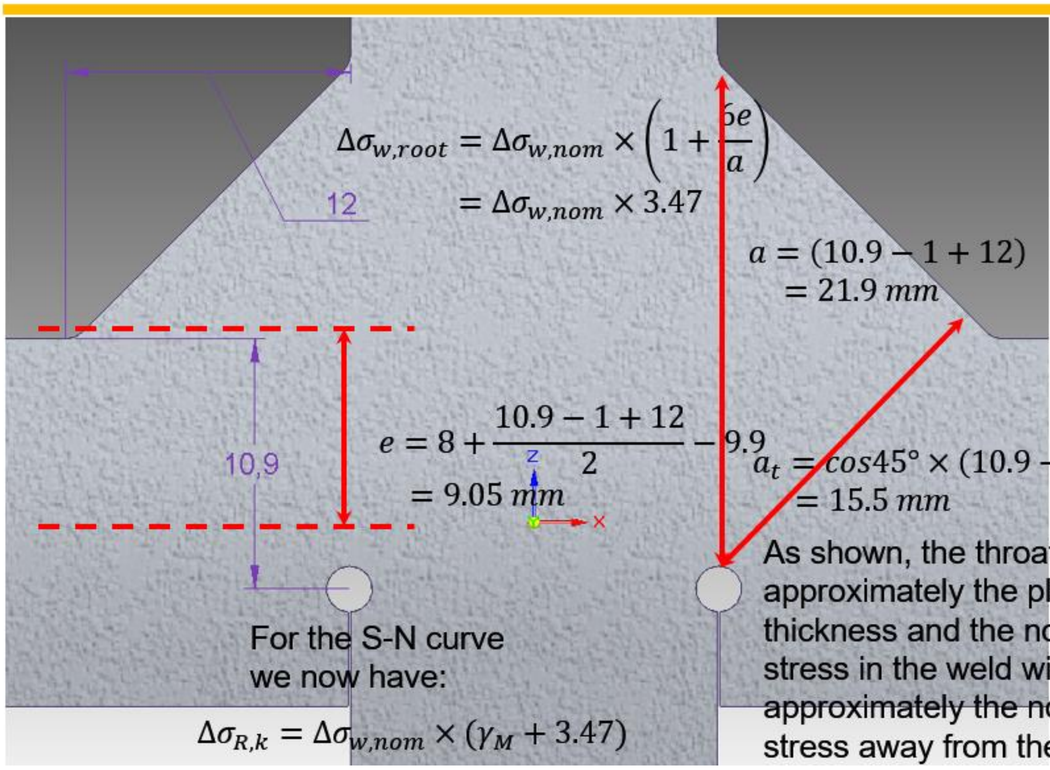

$$\Delta\sigma_{w,root} = \Delta\sigma_{w,nom} \times \left(1 + \frac{6e}{a}\right) = \Delta\sigma_{w,nom} \times 3.47$$

$$a = (10.9 - 1 + 12) = 21.9\ mm$$

$$e = 8 + \frac{10.9 - 1 + 12}{2} - 9.9 = 9.05\ mm$$

$$a_t = \cos 45° \times (10.9 - 1 + 12) = 15.5\ mm$$

As shown, the throat size is approximately the plate thickness and the nominal stress in the weld will be approximately the nominal stress away from the weld

For the S-N curve we now have:

$$\Delta\sigma_{R,k} = \Delta\sigma_{w,nom} \times (\gamma_M + 3.47)$$

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- The S-N curve applicable has a knee-point at 10 million cycles
  - Slope for stress ranges above the knee-point is $m = 3$

$$N = \frac{C}{\Delta\sigma^m}$$

  - Slope for stress ranges below the knee-point is $m_{knee} = 5$

$$N = N_{knee}\left[\frac{\Delta\sigma_{knee}}{\Delta\sigma}\right]^{m_{knee}}$$

  - This is because the of the variable amplitude that is applicable in this case

FAT	71	C =	7.15822E+11	m_knee	5
m	3	$\Delta\sigma_{R,k,knee}$	41.5	MPa	
$\Delta\sigma$	n_i	$\Delta\sigma_{R,k}$	N	D	
5	1 000 000	24	145 950 021	0.01	
10	500 000	49	6 243 563	0.08	
15	100 000	73	1 849 945	0.05	
20	50 000	97	780 445	0.06	
25	20 000	121	399 588	0.05	
30	10 000	146	231 243	0.04	
		Total Damage		0.30	
		Life in years		3.35	

Using principal stress at the weld Notch

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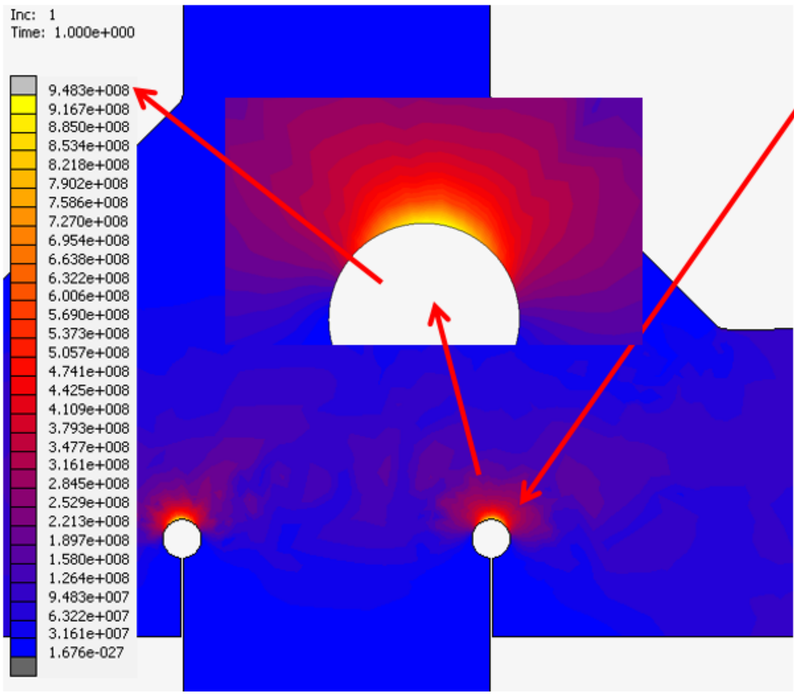
Solution: Apply the notch stress approach



For a 100 MPa load and linear elastic analysis, the stress in the stress concentration is 948 MPa

Stress concentration is at the weld root

This is where the crack will initiate and from where it will propagate through the specimen

The stress concentration factor on the nominal stress is then 9.48

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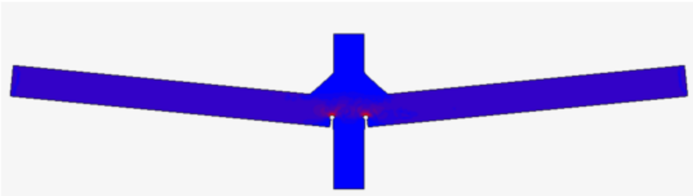
Based on the notch stress approach, the fatigue life is 5.26 years as calculated below:

FAT	225	C =	2.27813E+13	m_knee	5
m	3	$\Delta\sigma_{R,k,knee}$	131.6	MPa	
$\Delta\sigma$	n_i	$\Delta\sigma_{R,k}$	N	D	
5	1 000 000	66	306 500 154	0.00	
10	500 000	133	9 744 699	0.05	
15	100 000	199	2 887 318	0.03	
20	50 000	265	1 218 087	0.04	
25	20 000	332	623 661	0.03	
30	10 000	398	360 915	0.03	
		Total Damage		0.19	
		Life in years		5.26	



This life is slightly more than predicted using S-N curve. FEA analysis models bending moment effect accurately
Deformation of the joint also clarifies the reason for the high stress at the weld root notch
This deformation caused by the bending moment because of the off-centre weld
Note, the finite element model already makes provision for the bending moment

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4.26. Framework to select fatigue analysis method

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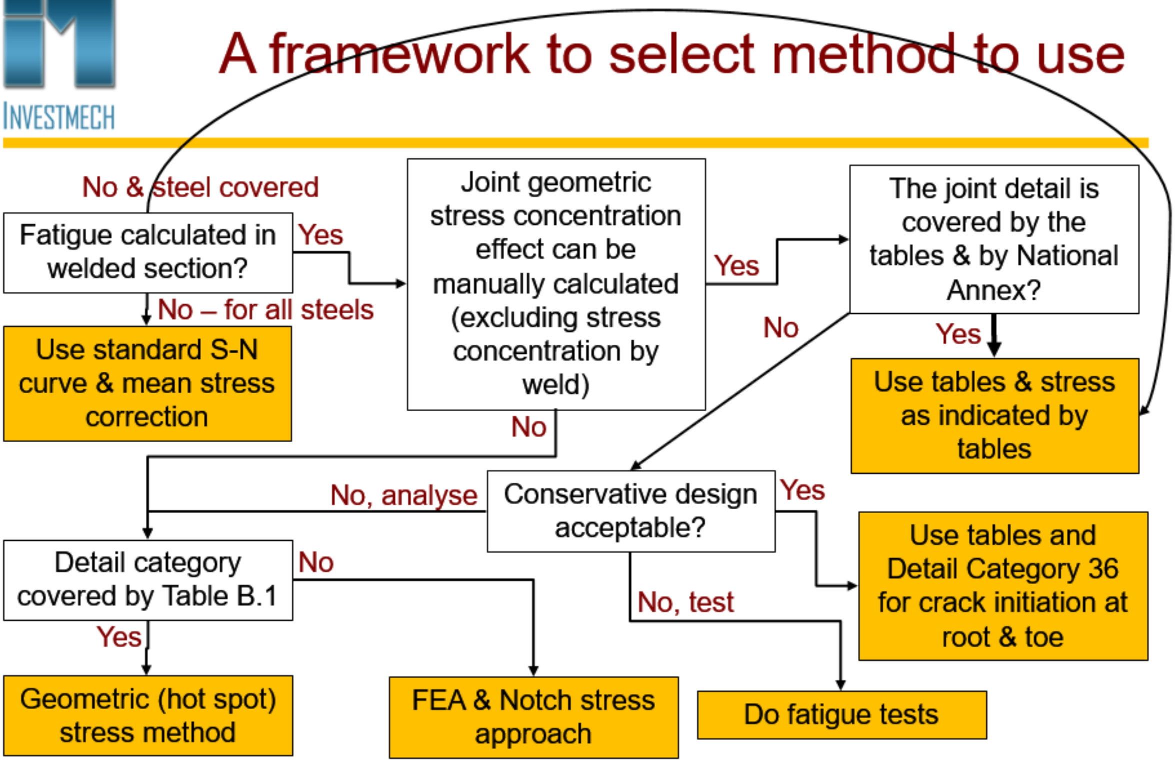
A framework to select method to use



The Notch stress approach can always be used. Just expensive for many problems that can be analysed by tables and nominal stress(es). 88

4.27. Class discussion

Discuss the case of a shaft joined with welding where the torque, bending and axial loads are transmitted through the weld only.

4.28. Finite element modelling and fatigue

Study the detail on surface stress extrapolation and through-thickness-integration in BS 7608. The topic will be briefly discussed in class.

4.29. Crack repair techniques

- Metal stitching or metalocking
  - See the following video: http://www.youtube.com/watch?v=Pq0wfU4ZaKk
  - http://www.locknstitch.com/AboutCSeries.htm
  - http://www.locknstitch.com/AboutLSeries.htm

4.30. References

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  - http://saisc.co.za/saisc/about_steel_products.htm
  - www.robor.co.za