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Guide to fatigue design and assessment of steel products

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Contents

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Summary of pages

This document comprises a front cover, an inside front cover, pages i to vi, pages 1 to 142, an inside back cover and a back cover.

Foreword

Publishing information

This British Standard is published by BSI Standards Limited, under licence from The British Standards Institution, and came into effect on 31 March 2014. It was prepared by Technical Committee WEE/37, *Acceptance levels for flaws in welds*. A list of organizations represented on this committee can be obtained on request to its secretary.

Supersession

BS 7608:2014+A1:2015 supersedes BS 7608:2014, which is withdrawn.

Information about this document

Guidance on general fatigue design philosophy is given in Annex A, which also contains a brief description of the method of using this British Standard. A more general method for assessing welded joints using the hot-spot stress, only included previously for assessing tubular joints, is also included.

The relevant application standard or specification for the particular product being assessed specifies the following:

a) the loading to be assumed for design purposes, including its magnitude and frequency;

b) the required life of the structure;

c) the environmental conditions;

d) the required nominal probability of failure.

BS 7608:2014 was a full revision of the standard, and introduced the following principal changes [1]:

- Introduction of the hot-spot stress method with guidance on finite element stress analysis (FEA).
- New correction for both plate thickness and applied bending with allowance for welded joint proportions.
- Additional weld details; some have been reclassified.
- Weld quality requirements based on fitness for purpose.
- Revised sea water corrosion fatigue data.
- New rules for bolts.
- Design data to resist shear fatigue failure.
- Guidance on stress calculation for combined loading.
- Revised cumulative damage rules.
- Comprehensive guidance on use of weld toe improvement methods.
- New guidance on acceptance fatigue testing and statistical analysis of results.

European standards containing fatigue rules for steel structures and pressure vessels have been published since the 1993 edition of this British Standard. It is therefore not applicable to product areas covered by them. It is applicable to a wide range of other steel product areas that do not have specific fatigue rules.

Text introduced or altered by Amendment No. 1 is indicated in the text by tags A1 A1. Minor editorial changes are not tagged. The principal changes are to Table 4 to Table 10, Clause **14**, Clause **16**, Table 18, new Table 21, Annex C and Annex F.

Use of this document

As a guide, this British Standard takes the form of guidance and recommendations. It should not be quoted as if it were a specification or a code of practice and claims of compliance cannot be made to it.

Presentational conventions

The guidance in this standard is presented in roman (i.e. upright) type. Any recommendations are expressed in sentences in which the principal auxiliary verb is "should".

Commentary, explanation and general informative material is presented in smaller italic type, and does not constitute a normative element.

Contractual and legal considerations

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

Compliance with a British Standard cannot confer immunity from legal obligations.

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1 Scope

1.1 General

This British Standard gives methods for assessing the fatigue life of parts of steel products that are subject to repeated fluctuations of stress. It is applicable to all areas of industrial application that are not covered by other British Standards containing fatigue assessment rules.

NOTE Some British Standards have specific product acceptance tests for fatigue life, but do not have assessment rules. In such cases the guidance in this British Standard might be applicable for product development purposes.

1.2 Applications not covered

This British Standard is not applicable to the following application areas;

a) lighting columns (see BS EN 40);

b) concrete building and civil engineering structures (see BS EN 1992);

c) steel building and civil engineering structures [see BS EN 1993 (all parts)];

d) composite steel and concrete building and civil engineering structures [see BS EN 1994 (all parts)];

e) unfired pressure vessels (see BS EN 13445); and

f) fixed offshore structures (see BS EN ISO 19902).

1.3 Materials

This British Standard covers:

a) wrought steel material products;

b) welds in fully machined areas of steel casting;

c) ferritic alloy and low alloy steels;

d) austenitic and duplex stainless steels;

e) unprotected weathering steels; and

f) threaded fasteners.

It is applicable to yield strengths in the range 200 N/mm^2 to 960 N/mm^2 and ultimate tensile strengths in the range 360 to 1 200 N/mm^2 for material thicknesses 3 mm and greater.

This British Standard is not applicable to the following:

1) proprietary fasteners;

2) steel castings;

3) cold drawn products;

4) wire ropes; and

5) steel for reinforcement in concrete.

1.4 Manufacturing processes

This British Standard is applicable to machined products with the following exceptions:

a) rough sawn surfaces;

b) surfaces requiring high quality surface finish (e.g. lapping, polishing, honing, fine grinding); and

c) machined details with sharp corners (e.g. key ways, un-radiused shoulders).

The following manufacturing processes are also covered:

1) cold formed wrought products;
2) weld toe improvement methods;
3) arc welded joints, with the exclusion of joints between rectangular and square hollow sections;
4) in-line butt welds made by power beam [A1⟩ or ⟨A1] friction welding;
5) tensioned and un-tensioned bolted joints and hot-driven riveted lap joints loaded in shear; and
6) thermal cutting.

This British Standard is not applicable to the following manufacturing processes:

i) resistance welding processes and brazing;
ii) contact joints under pressure where fretting occurs;
iii) adhesively bonded joints;
iv) shearing and punching; and
v) surface hardening.

1.5 Environment

The fatigue design data in this British Standard are applicable to internal and external air environments. They are applicable to structural steel products exposed to sea water.

They are not applicable to unprotected stainless or weathering steel products in sea water or aggressive corroding environments (e.g. chloride, sulphide, strong acid or alkali).

The data are applicable to products operating at temperatures below the creep range of the steel.

This British Standard is not applicable to products operating in the creep regime.

2 Normative references

The following documents, in whole or in part, are normatively referenced in this document and are indispensable for its application. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

BS 3643-2, *ISO metric screw threads – Part 2: Specification for selected limits of size*

BS 3692, *ISO metric precision hexagon bolts, screws and nuts – Specification*

BS 4190, *ISO metric black hexagon bolts, screws and nuts – Specification*

BS 4395 (all parts), *Specification for high strength friction grip bolts and associated nuts and washers for structural engineering*

BS 7910, *Guide to methods for assessing the acceptability of flaws in metallic structures*

BS EN 1011-1, *Welding – Recommendations for welding of metallic materials – Part 1: General guidance for arc welding*

BS EN 1011-2, *Welding – Recommendations for welding of metallic materials – Part 2: Arc welding of ferritic steels*

BS EN 1011-3, *Welding – Recommendations for welding of metallic materials – Part 3: Arc welding of stainless steels*

BS EN 10163 (all parts), *Delivery requirements for surface condition of hot-rolled steel plates, wide flats and sections*

BS EN 1993-1-8, *Eurocode 3 – Design of steel structures – Design of joint*

BS EN ISO 3506, *Mechanical properties of corrosion-resistant stainless steel fasteners – Bolts, screws*

BS EN ISO 4014, *Hexagon head bolts – Product grades A and B*

BS EN ISO 4017, *Hexagon head screws – Product grades A and B*

BS EN ISO 4762, *Hexagon socket head cap screws*

BS EN ISO 9013, *Thermal cutting – Classification of thermal cuts – Geometrical product specification and quality tolerance*

BS ISO 12108, *Metallic materials – Fatigue testing – Fatigue crack growth method*

3 Terms and definitions

For the purposes of this British Standard, the following terms and definitions apply.

3.1 cycle counting method

method of counting the numbers of stress cycles of different magnitudes which occur in a service stress history

NOTE The loads applied to the structure, considered in sequence, generate a particular stress history at each detail of interest. This stress history can be broken down into equivalent stress ranges by the operation of cycle counting.

3.2 detail class

rating given to a particular structural detail to indicate which of the fatigue strength (*S-N*) curves should be used in the fatigue assessment

NOTE 1 Also known as joint class.

NOTE 2 The class is denoted by one of the following letters: A, B, C, D, E, F, F2, G, G2, S_1, S_2, TJ, W1 or X. The categorization takes into account the stress being used in the assessment (e.g. nominal, hot-spot or shear stress), the local stress concentration at the detail, the size and shape of the maximum acceptable discontinuity, the stress direction, metallurgical effects, residual stresses and a post-weld improvement method.

3.3 design life

period within which there is a defined nominal probability that failure by fatigue cracking is unlikely to occur

NOTE This can be longer or shorter than the service life (see Annex A).

3.4 design spectrum

tabulation of the number of occurrences of all the stress ranges, S_r, of different magnitudes produced by the load spectrum in the design life of the structure or component, to be used in the fatigue assessment

NOTE 1 Also known as stress spectrum.

NOTE 2 Different components of a product can have different design spectra.

3.5 fatigue
damage of a structural part by the initiation and gradual propagation of a crack or cracks caused by repeated applications of stress

3.6 fatigue failure
achievement of a through-section fatigue crack or a sufficiently large fatigue crack to cause static failure or excessive deformation

NOTE See Annex A.

3.7 fatigue life, *N*
number of stress cycles that produce a given probability of fatigue failure

3.8 fatigue loading
loading on a structure which is liable to cause fatigue cracking

*NOTE It can be composed of several different types and magnitudes of loading events (see Clause **7**).*

3.9 fatigue strength
constant amplitude stress range, S_r, causing failure in a specified number of cycles (N)

3.10 hot-spot stress, S_H
structural stress at a weld toe or weld end

3.11 initial non-propagating stress range, S_{oc}
constant amplitude stress range below which (in the absence of any previous loading) a crack is assumed not to propagate

NOTE 1 Also known as constant amplitude fatigue limit, CAFL.

NOTE 2 Its magnitude depends on the structural detail being assessed. For parts in air or adequately protected against corrosion, it is assumed to be the stress range corresponding to a life of 10^7 cycles on the design S-N curve for all detail classes except S_1 and S_2 , for which it corresponds to 10^8 cycles. For unprotected joints in a corrosive environment it should be assumed that $S_{oc} = 0$ for all classes.

3.12 load spectrum
tabulation showing the relative number of occurrences of all the loading events of different types and magnitudes expected to be experienced by the structure in its design life

3.13 loading event
defined loading sequence on the structure

NOTE 1 This can be characterized by its relative frequency of occurrence as well as its magnitude and geometrical arrangement.

NOTE 2 For example, in the case of handling equipment this could involve the lifting, movement and depositing of a load. For design purposes each loading event is assumed to repeat a given number of times in the design life of the structure.

3.14 Miner's summation
linear cumulative damage summation based on the rule devised by Palmgren and Miner

3.15 nominal stress
structural stress that would exist in the absence of the structural discontinuity being considered

NOTE Nominal stress is a reference stress that can be calculated using elementary theory of structures. It excludes the effects of structural discontinuities (e.g. welds, openings, thickness changes) and secondary bending due to a local weld detail.

3.16 *S-N* curve
quantitative relationship between the fatigue strength *S* and the number of cycles *N* corresponding to a specific probability of failure for a detail, derived from test data

3.16.1 basic *S-N* curve
S-N curve, for the required probability of failure, for a detail of basic thickness (see **16.3.2**) operating in air without the application of any fatigue strength improvement method apart from those given in Table 3 to Table 10

3.16.2 design *S-N* curve
S-N curve adopted for design purposes for the detail being assessed

*NOTE It is derived from the relevant basic S-N curve modified, if necessary, to allow for the influence of material thickness, bending, environment, fatigue strength improvement techniques (additional to those given in Table 3 to Table 10), stress relief (see **16.3.6**) or workmanship (see Clause **14**).*

3.16.3 standard basic *S-N* curve
basic *S-N* curve for 97.7% probability of survival [two standard deviations of log*N* below the mean (d = 2)] assuming that the test data are represented by a normal distribution of log life

3.17 service life
period in which a structure or component is required to perform safely with an acceptable probability (see **A.2**) that it is unlikely to require repair or withdrawal from service as a result of fatigue cracking

3.18 slope transition point
point on the *S-N* curve beyond which it is extrapolated at a shallower slope for use in cumulative damage calculations of fatigue under variable amplitude loading

3.19 stress cycle
pattern of variation of stress at a point defined by the cycle counting method and consisting of a change in stress between defined minimum (trough) and maximum (peak) values and back again

NOTE 1 Also known as cycle of stress.

NOTE 2 One loading event can produce one or more stress cycles at any particular point.

3.20 stress range S_r
algebraic difference between the two extremes (reversals) of a stress cycle

NOTE 1 Also known as range of stress.

*NOTE 2 See **15.2** for parent metal.*

*NOTE 3 See **15.3** for weld metal.*

3.21 structural stress
surface value of the linearly distributed stress across the section thickness arising from applied loads (forces, moments, pressure, etc.) and the corresponding reaction forces on the particular structural part

NOTE The linear stress distribution includes the effects of gross structural discontinuities (e.g. presence of an attachment, aperture, change of cross-section, misalignment, intersection of members) and distortion-induced bending moments. However, it excludes the notch effects of local structural discontinuities (e.g. weld toe, weld end) which give rise to non-linear stress distributions across the section thickness.

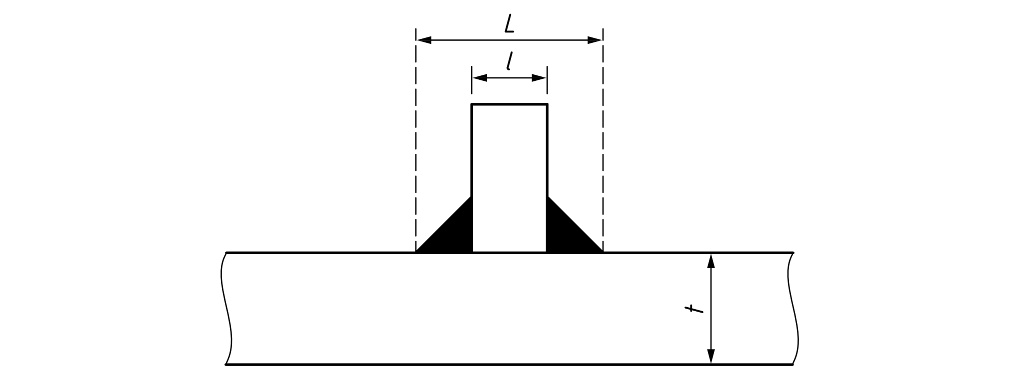
4 Symbols and units

Unless otherwise indicated, for the purposes of this British Standard, the following symbols and units apply.

A	Net area of cross section (in mm²)
a	Weld throat thickness (in mm)
b	Exponent in correction term for thickness and bending
C_o	Parameter defining the mean line S_r-N relationship
C_2	Parameter defining the S_r-N relationship for two standard deviations of log N below the mean line
C_d	Parameter defining the S_r-N relationship for d standard deviations of log N from the mean line
D	Miner's summation $\Sigma \frac{n}{N}$
d	Number of standard deviations of log N from the mean S_r-N curve
A1 E_B	Reference elastic modulus at temperatures $T \leq 150$ °C (in N/mm²) A1
E_T	Elastic modulus at temperature T (in N/mm²)
e	Axial misalignment (eccentricity or centre-line mismatch) (in mm)
F	Fatigue test factor
g	Gap between welds in intermittently welded joint (in mm)
h	Weld length (in mm)
K_b	Bolt stiffness (in N/mm)
K_c	Summation of clamped components stiffness (in N/mm)
K_f	Stress concentration factor under fatigue loading
k_m	Stress magnification factor due to misalignment
k_{tb}	Correction factor for plate thickness and bending
L	Overall attachment length, including welds (see Figure 1) (in mm)
l	Attachment length parallel to direction of loading considered (in mm)
M, M^1	Applied bending moments (in N·mm)
m	Inverse slope of log S_r-log N curve (i.e. $S_r^m \cdot N$ = constant)
N_1, N_2 . . .	Number of cycles to failure under constant amplitude loading with stress ranges S_{r1}, S_{r2} . . ., etc., corresponding to n_1, n_2 etc. (in cycles)
N_{oc}	Constant amplitude endurance corresponding to S_{oc} (in cycles)
N_{ov}	Endurance at slope transition point, S_{ov}, on S-N curve (in cycles)
n	Number of fatigue test results

n_1, n_2 . . .	Number of cycles of damaging stress ranges S_{r1}, S_{r2} . . ., etc. in a design spectrum
P, P^1, P_N, P_L	Applied axial forces (in N)
P_r	Applied force range (in N)
R	Stress ratio (ratio of minimum to maximum algebraic value of applied stress) or chord radius in mm (see Annex G)
r	Radius (in mm)
S	Fatigue strength of the structural detail under consideration, including any required thickness correction (in N/mm²)
S_B	Fatigue strength obtained from the basic S_r-N curve (in N/mm²)
S_H	Hot-spot stress (in N/mm²)
S_{Hr}	Hot-spot stress range (in N/mm²)
S_{oc}	Constant amplitude initial non-propagating stress range (in air S_o = S_r at $N = 10^7$ cycles for all design classes except S_1 and S_2 when it coincides with N = 10^8 cycles) (in N/mm²) *NOTE Also known as constant amplitude fatigue limit (CAFL).*
S_{ov}	Stress range at slope transition point (in N/mm²)
S_r	Stress range in any one cycle (in N/mm²)
S_{r1}, S_{r2} . . .	Individual stress ranges (S_r) in a design spectrum (in N/mm²)
S_w	Resultant stress range on weld throat (in N/mm²)
SD	Standard deviation of log N
W	Plate width or thickness of longitudinal attachment (in mm)
T	Applied torque (in N.mm)
t	Plate thickness of the member under consideration (in mm)
t_B	Thickness relevant to the basic S_r-N curve for the detail (in mm)
t_c	Cover plate thickness (in mm)
t_{eff}	Effective plate thickness (in mm)
w	Combined size of effective weld throats (in mm
Z	Section modulus (in mm³)
σ_x, σ_y	Direct stresses acting in the x- and y-directions (in N/mm²)
$\Delta\sigma_b$	Applied bending stress range (in N/mm²)
$\Delta\sigma_m$	Applied membrane stress range (in N/mm²)
$\Delta\sigma_\perp$	Direct stress range on weld throat (in N/mm²)
$\Delta\sigma_w$	Engineering shear stress range on weld throat (in N/mm²)
σ_Y	Nominal tensile yield strength (in N/mm²)
τ	Shear stress (in N/mm²)
$\Delta\tau$	Shear stress range (in N/mm²)
$\Delta\tau_\perp$	Transverse shear stress range on weld throat (in N/mm²)
$\Delta\tau_{//}$	Longitudinal shear stress range on weld throat (in N/mm²)
Ω	Degree of bending ($\Delta\sigma_b/(\Delta\sigma_m + \Delta\sigma_b)$

Figure 1 **Definition of length *L* for use in thickness-bending correction**



5 Fatigue assessment procedure

A stressed element can contain a number of potential fatigue crack initiation sites. All of these should be checked (see **12.1**). The regions subjected to the highest stress fluctuations and/or containing the severest stress concentrations should normally be given the highest priority. The design procedure in this British Standard involves calculation of the fatigue damage accumulation during the design life or comparison of the maximum applied stress range with the relevant constant amplitude fatigue limit. It is primarily intended for use in "safe life" design (see **A.3.1**). It might also be suitable for "damage-tolerant" design (see **A.3.2**).

The steps that should be followed are:

a) establish the required design life of the product (see Clause **6**);

b) establish a conservative estimate of the loading expected in the life of the product (see Clause **7**);

c) estimate the resulting stress history at the detail under consideration (see Clause **15**);

d) reduce the stress history to an equivalent number of cycles n_i of different stress ranges *S*ri using a cycle counting technique (see **15.9**);

e) classify the detail in accordance with Table 1 to Table 10;

f) use this classification to define the basic design S_r-N curve (see **16.2**);

g) calculate the resulting fatigue life on the basis of comparison of the maximum applied stress range and the relevant CAFL (see **16.6**) or a cumulative damage calculation (see **16.7**);

h) if all applied stress ranges are below the relevant CAFL (see **16.6**), the detail can be assumed to have a life exceeding the specified design life in a) (see Clause **6**);

i) if some of the applied stress ranges exceed the relevant CAFL and that the life calculated using the full design stress spectrum and the cumulative damage method (see **16.7**) exceeds the specified design life in a), the requirement for safe life design is met; and

j) if the procedure in i) results in a calculated life less than the specified design life in a), the requirement for safe life design is not met. The following measures should be taken:

   1) adjust the detail design so that a higher S_r-N curve can be used;

   2) if 1) is not adequate, increase the cross-section at the potential fatigue crack initiation site to reduce the stress ranges.

3) if measures 1) and 2) result in severe economic consequences, use of a damage-tolerant approach involving periodic in-service non-destructive testing (NDT) for the detection of fatigue cracking may be employed to ensure that the overall probability of failure without warning during the design life is no less than that assumed for safe life design. For further guidance see **A.3.2**.

6 Design life

The design life of a product is usually pre-determined by factors such as market expectations, planned service life or obsolescence, contract or warranty requirements and uncontrollable deterioration by mechanisms other than fatigue, such as wear and tear or corrosion. Where a design life has not already been specified for a product in which fatigue is a potential failure mode, it should be selected on the basis of the period of service over which the probability of failure by fatigue is required to be low. This is achieved in design by the use of lower-bound fatigue strength data (see Clause **16**) and upper-bound loading data (see Clause **7**).

7 Fatigue loading

When assessing fatigue performance a realistic estimate of the fatigue loading is crucial to the calculation of life, and all types of cyclic loading should be taken into account. Cyclic loading from different sources might be significant at different phases of the life of a structure, e.g. manufacture, transport, storage, installation service, and can involve different loading modes and frequencies.

Uncertainties exist in assessing both the stresses resulting from applied loads and the response of a particular joint, which control fatigue performance. The basis of the fatigue analysis should be the use of an upper bound estimate of these stresses, recording the uncertainties involved, combined with *S-N* curves derived from experimental data. In this way, uncertainties associated with the life of a particular joint, e.g. size, weld detail, local environment, can be separated from those associated with applied stress.

Uncertainties can also exist in the number of applications of the load expected to occur during the product's design life. The design load spectrum should be selected on the basis that it is an upper bound estimate of the accumulated service conditions, including both loading and number of cycles, over the full design life of the product. The adoption of mean plus two standard deviations data for applied loads levels or an upper bound estimate based on knowledge of the actual or predicted loading environment and applied numbers of cycles, when used with the design (i.e. mean minus two standard deviations of log *N*) S_r-*N* data in Clause **16**, usually results in an acceptably low probability of failure during the design life, commensurate with safe-life design principles.

Because of the sensitivity of calculated life to the accuracy of estimates of stress, stress ranges should not be underestimated. Account should be taken of all likely operational and environmental loads arising from the foreseeable usage of the product during that period. Use of this approach compensates for the use of design *S-N* curves that correspond to a finite probability of failure and means that load factors are not required.

The following are some important sources of cyclic loading that should be taken into account, any or all of which can be relevant in particular applications:

a) fluctuating loads;

b) acceleration forces in moving structures;

c) pressure changes;

d) temperature fluctuations;

e) mechanical vibrations; and

f) environmental loading (wind, currents and waves, especially when vortex shedding is induced, e.g. on slender members).

It is particularly important to assess dynamic magnification effects where loading frequencies are close to one of the natural frequencies of the component or structure. In some instances the loading to be assumed for fatigue design purposes is specified in the design specification. Where such information is not available, assumptions as to the loading to be expected in service should be made, and it might be useful to obtain data from existing products subjected to similar effects. In particular, in assessing an existing product, it might be possible to compile a design spectrum from strain readings or loading records obtained from continuous monitoring.

In all cases, the objective is to define the spectrum in terms of the numbers of cycles of each of the individual stress ranges expected in the life of the product. If it is required to convert the spectrum into a series of constant amplitude blocks, the resulting simplified spectrum should be equivalent in terms of fatigue damage to the actual spectrum. To achieve this, a sufficient number of intervals of stress should be selected to avoid discretion errors due to insufficient resolution in the stress spectrum.

8 Environmental considerations

The fatigue assessment should take into account the environmental conditions that the product is exposed to during all phases of its anticipated service life. For example, products designed to operate in seawater might be constructed and transported in an air environment, installed and commissioned in a freely corroding marine environment, and operated in seawater with cathodic protection. This British Standard provides *S-N* curves for three environmental conditions and the most appropriate curve should be used for each segment of the anticipated service life. Periods of free corrosion should be avoided, for example during temporary storage of products awaiting installation in seawater or following commissioning hydro-tests of pressurized components, as this can cause pitting and an associated reduction in the expected fatigue life.

It might be necessary to take the operating temperature into account (see **16.3.3**). The fatigue strength of steel products in air varies with temperature, in accordance with the corresponding change in elastic modulus. Therefore it is improved at subzero temperatures. However, in the case of ferritic steels, as the fracture toughness is reduced, it is possible that overall fatigue life would be reduced if the product suffers premature failure by brittle fracture from a small fatigue crack. Consequently, no allowance should be made for the beneficial effect of sub-zero temperature on fatigue strength but, where such operation is expected, for consistency with the fatigue design data provided in this British Standard, the steel used should be capable of tolerating the presence of through-section fatigue cracking at the minimum anticipated temperature. This is achieved with steels that have minimum low temperature impact properties specified in the material standard and are operated at or above the stated temperature.

There are no particular recommendations for austenitic stainless steels operating at low temperature, as they are not susceptible to brittle fracture. For elevated temperatures, no effect is assumed for temperatures up to 150 °C, but at temperatures higher than this fatigue strength decreases in line with the decrease in elastic modulus for temperatures below the creep range of the steel concerned. No guidance is provided for combined creep and fatigue.

9 Factors on fatigue life

The reliability of a product's fatigue life is dependent on the following factors:

a) selection of a safe level of fatigue loading (see Clause **7**);

b) correct calculation of stress ranges (see Clause **15**);

c) correct detail classification (see Clause **12**);

d) application of appropriate controls during manufacture (see Clause **14**);

e) in some cases, fatigue testing might be appropriate (see Annex E).

The fatigue design data provided in this British Standard are in the form of mean *S-N* curves and the corresponding standard deviations of log N to enable different probabilities of survival to be adopted. The standard basic *S-N* curves (see **16.2**) represent 97.7% probability of survival, as they are based on the mean minus at least two standard deviations of log*N* curves for relevant experimental data. Their use therefore indicates a finite probability of failure (up to 2.3%) for the calculated life. In some circumstances, for example in a failure investigation of a part that has experienced fatigue cracking in service, it might be more appropriate to assume a higher probability of failure and make use of an *S-N* curve less than two standard deviations of log N below the mean. Conversely, it might be appropriate to use *S-N* curves based on the mean minus more than two standard deviations of log N for components with inadequate structural redundancy or difficult access for inspection. In selecting the number of standard deviations to be used to define the design *S-N* curve, account should be taken of the accessibility of the joint and the proposed degree of manufacture and in-service inspections, as well as the consequences of failure. As a crack grows in one part of a structure, the load might be shed to other members and lead to further fatigue cracks in those members.

10 Features influencing fatigue behaviour

For both welded and bolted steel products the fatigue life is normally governed by the fatigue behaviour of the joints, including both main and secondary joints [2, 3]. Even fabrication or handling aids, such as welded brackets or lifting lugs, that remain in the completed product could provide sites for fatigue cracking and should therefore be assessed. Optimum fatigue behaviour is obtained when the product is detailed and constructed such that stress concentrations are kept to a minimum and, where possible, the elements are able to deform in their intended ways without introducing secondary deformations and stresses due to local restraints. Stresses can also be reduced by increasing the thickness of parent metal or the weld throat, depending on the potential failure mode. In the case of the former, allowance might need to be made for the fact that fatigue strength tends to decrease with increasing plate thickness for some types of joint when assessing the resulting benefit (see **16.3.2**).

Optimum joint performance is achieved by avoiding joint eccentricity and misalignment, welds at free edges, and by other controls over the quality of the joints.

In the specific case of pipe-to-pipe joints made from seamless pipe, the accumulation of maximum allowable manufacturing tolerances for thickness, ovality and diameter leads to a large potential for girth weld misalignment and associated penalty on fatigue life. Apart from the resulting introduction of secondary bending stresses, such misalignment can also intensify the stress concentration due to the geometry of the weld root bead in welds made from one side. It is therefore advantageous to record the actual range of tolerances achieved for each batch of pipe supplied and use this information to calculate the maximum potential misalignment. Performance is also adversely affected by concentrations of stress at holes, openings and re-entrant corners. Guidance in these aspects is given in Table 1 to Table 10, Annex B, Annex C and Annex G.

The magnitude and nature of stresses that cause propagation of a crack and therefore reduce the number of stress repetitions to cause failure are affected by the presence of residual stress, inherent flaws in welds and adjacent parent metal, surface flaws and any other stress raisers interfering with the flow of stress. These are taken into account in the classifications given in Table 1 to Table 10.

11 Fracture mechanics

In some situations the normal fatigue assessment procedures might be inappropriate, but fracture mechanics methods might be helpful. Guidance on the use of fracture mechanics is given in Annex D.

12 Classification of details

12.1 General

For the purposes of fatigue design, joints are divided into several classes, each with a corresponding design *S-N* curve (see **16.2**). This classification depends upon the following:

a) the type of stress being used to assess the detail (nominal or hot-spot stress);

b) the geometrical arrangement and proportions of the detail;

c) the direction of the fluctuating stress relative to the detail;

NOTE Any reference to transverse or longitudinal welds, welded joints or welded attachments refers to their orientation with respect to the direction of stressing.

d) the location of possible fatigue crack initiation at the detail; and

e) the methods of manufacture and inspection.

In any product or component that is liable to be subjected to repeated applications of stress, every welded joint, including non-structural attachments, lifting lugs and temporary fabrication aids left in place, or other forms of stress concentration, such as a bolt hole or cut edge, is potentially a source of fatigue cracking. Each part of every constructional detail should be assessed individually and should, where possible, be placed in its relevant joint class in accordance with Table 1 to Table 10. Where this is not possible, the detail should be classified in accordance with Clause **13**.

In bolted or riveted joints, fatigue cracks normally initiate from the bolt or rivet hole or, in the case of friction-grip bolted joints, in the plate by fretting or in the bolt itself. In plate with cut edges or welded details fatigue cracks can potentially initiate in the following places:

1) from any point on the plate edge or plate surface;

2) in the parent metal of either part joined adjacent to;
    i) the end of the weld;
    ii) a weld toe;
    iii) a change of direction of the weld;
3) in the weld metal starting from;
    i) the weld root;
    ii) the weld surface;
    iii) an internal flaw.

In the case of members or elements connected at their ends by fillet welds or partial penetration butt welds, crack initiation can occur in the parent metal or in the weld throat. Both possibilities should be assessed by taking into account the appropriate classification and stress range. Similarly, fatigue crack initiation can occur from a weld toe on either the outside or inside of a full-penetration girth weld between pipes or tubes. The most critical location depends on the relevant classification and stress range, both of which can differ between the outside and inside. For other details, the classifications given in Table 1 to Table 10 cover crack initiation at the location indicated. Notes on the potential modes of failure for each detail are given in Annex B.

12.2 Classification of details

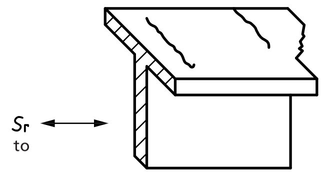
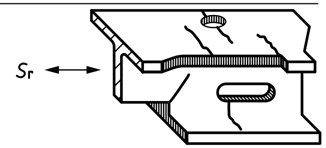
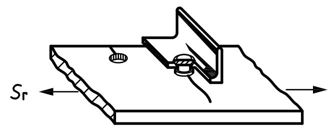
Table 1 to Table 10 correspond to the following basic types of details:

- plain material (Table 1);
- bolted or riveted connections (Table 2);
- continuous longitudinal butt welds and welded attachments (Table 3);
- other welded attachments (Table 4);
- transverse butt welds in plates (Table 5);
- transverse butt welds in sections, tubes and pipes (Table 6);
- load-carrying fillet and T-butt joints (Table 7);
- slotted connections and penetrations through stressed members (Table 8);
- details relating to tubular members (Table 9); and
- branch connections to vessels (Table 10).

Detail classifications are given for assessments based on applied nominal stresses and, where appropriate, hot-spot stresses. Where relevant, the need to apply a correction for thickness and bending (see **16.3.2**) and the applicability of a weld toe improvement technique (see **16.3.5**) are included. Each classified detail is illustrated and given a type number. Table 1 to Table 10 also give associated criteria and diagrams that illustrate the geometrical features and potential crack locations for the direction of loading shown which determine the class of each detail. They should be used to assist with initial selection of the appropriate type number.

A detail should only be designated a particular classification if it conforms to all criteria in Table 1 to Table 10 appropriate to its type number or if a suitable classification can be justified on the basis of relevant published fatigue data or the results of specific fatigue tests in accordance with Annex E. Class A is generally inappropriate for structural work and therefore no design data are provided. The practical difficulty of achieving the special inspection standards relevant to classes B and C might limit the feasibility of adopting these classifications in structural work.

Table 1 Classification of details: plain material free from welding

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special requirements	Design stress area	Class A)	Notes	Sketch
1.1	Rolled steel plates and sections	Away from all welds or connections	Member of constant or smoothly varying cross section with no holes or re-entrant corners	All surfaces fully machined and polished. No flame cutting			A	See **B.2.2**	
1.2				Edges as rolled or machined smooth. No flame cutting. The surface condition should be in accordance with BS EN 10163 as a minimum.			B B)		
1.3		At any external or internal edge	Member with or without apertures, re-entrant corners or other discontinuities	Any flame cut edges subsequently machined or ground smooth. The surface condition should be in accordance with BS EN 10163 as a minimum.	Flame-cut edges in steel sheets cold formed after cutting should be checked for cracking and any such cracks ground out to leave a smooth edge surface	Net cross section	B B)	All visible signs of drag lines should be removed from the flame cut edge by grinding or machining. See also note for type 1.4.	
1.4				Any cutting of edges by planning or machine flame cutting with controlled procedure Thermal cut quality in accordance with BS EN ISO 9013 with mean height of the profile range 2. No visible gouges greater than 0.5 mm deep. Any deburring or rectification by grinding should be longitudinal to the plate edge. No repair of gouges by welding.		Net cross section	C	The controlled flame cutting procedure should ensure that the resulting surface hardness is not sufficient to cause cracking. Types 1.3 and 1.4. The presence of an aperture, re-entrant corner or other discontinuity implies the existence of a stress concentration and the design stress should be the stress on the net section multiplied by the relevant stress concentration factor (see Annex C).	
1.5		At a small hole (may contain bolt for minor fixtures)		Hole drilled or reamed. Minimum distance between centre of hole and plate edge = 1.5 × hole diameter		Net cross section	D	This type may be deemed to include bolt holes for attaching light bracing members where there is negligible transference of stress from the main member in the direction S_r. The classification includes allowance for the stress concentration created by the hole.	
1.6						Gross cross section	B B)	The classification does not include allowance for the stress concentration created by the hole.	

A) One class lower in the case of unprotected weathering steels.
B) For unprotected steel exposed to sea water or other corrosive environments the class is reduced to C.

Table 2 **Classification of details: bolted and riveted connections**

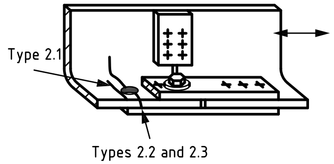
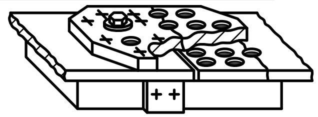
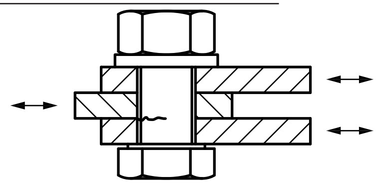
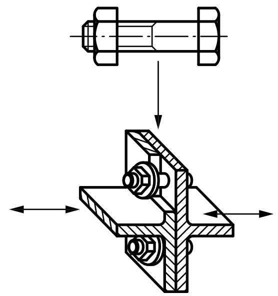
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special requirements	Design stress area	Class	Thickness and bending correction (see 16.3.2)	Notes	Sketch
2.1	Rolled steel plates and sections	At joint, in plate away from the hole	Double covered symmetrical joint made with high strength friction grip bolts	Holes drilled or reamed. Bolts should be tightened in accordance with BS EN 1993-1-8.		Gross section of plate	C	Not applicable	This covers connections designed for slip resistance at the ultimate limit state and where secondary out-of-plane bending of the joint is restrained or does not occur i.e. double-covered symmetrical joints. Failure initiates by fretting in front of the hole.	
2.2		At joint, in plate from the edge of a hole				Net section of plate	C		See **B.2.2.2** regarding the tightening of bolt groups. The classifications include allowance for the stress concentrations created by the bolt hole.	
2.3			Double covered symmetrical joint made with rivets or precision bolts				D			
2.4			Single covered joint made with high strength friction grip bolts				D			
2.5			Joint made with rivets	Hot-driven			D			
2.6			Joint made with precision bolts and close tolerance holes.	Torque tightened using a controlled procedure.	Faced under head and turned on shank in accordance with BS 4190		E			
2.7			Joint made with black bolts.				G			
2.8		Anywhere in shank for bolts loaded in shear	Any bolted lap joint but thread not in shear plane	Type 2.3 or 2.4 apply as long as bolt torque-tightened using controlled procedure that is designed to carry entire shear load in friction grip throughout its design life, taking into account any relaxation that might occur.		Cross-sectional area of shank	S_1	Not applicable	Shear failure of the bolt itself	
2.9	Threaded fasteners	At thread root.		Fastener in a butt joint with fastener axis parallel to S_r. Bolts conforming to BS 3692, BS 4395 (all parts). Screw threads conforming to BS 3643-2. BS EN ISO 4014, BS EN ISO 4017, BS EN ISO 4762. BS EN ISO 3506	See **14.3.3** This classification is not applicable to bolts that have been welded.	Tensile stress area (see **15.8**)	X	Applicable assuming $\Delta s_B = 0$ and $b = 0.25$.	This classification applies to failure at the thread root in normal commercial quality threaded fasteners. See **15.8** and **16.2.2** for type limitations and enhancements.	

Table 3 **Classification of details: butt welds and continuous welded attachments essentially parallel[A)] to the direction of applied stress**

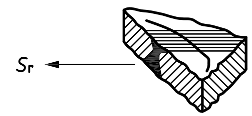
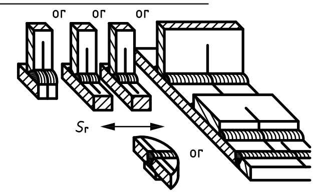
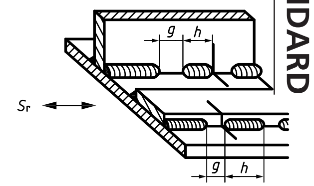
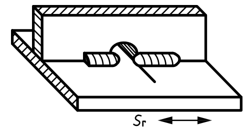
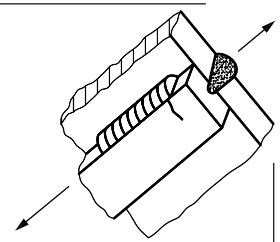
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
3.1	Rolled steel plates, sections and built-up members	Weld surface or embedded flaw	Butt weld with full penetration and no permanent backing strip.	Weld reinforcement dressed flush.	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**)	Minimum transverse cross section of member at location of potential crack initiation	B [B)]	Not applicable	Not applicable	Finish machining should be in the direction of S_r. In view of the difficulty of ensuring freedom from significant flaws (see **14.3.4**), this class is only recommended for use in exceptional circumstances (see **B.2.2.1** and **B.3.1**)	
3.2		Fillet weld root or ripples on weld surface away from the weld end.	Butt weld with full penetration or butt or fillet welded web or attachment (in the direction of S_r), including lap joints.	All welds continuous with no stop/starts. Butt welds with full penetration. If used, backing should be incorporated (e.g. joggle joint), continuous and either not attached or attached by continuous fillet welds.	Weld toe improvement techniques not relevant.		C	Not applicable	Not applicable	Accidental stop/starts are not uncommon even in automatic processes. Repair to the standard of a C classification should be the subject of specialist advice and inspection and as a result, the use of this type is not recommended.	
3.3		Stop / start position on weld surface away from the weld end.		As type 3.2 but with stop/starts.			D	Not applicable	Not applicable	For the situation at the ends of flange cover plates see joint type 4.5. Backing strips, if used, need to be continuous and either not attached or attached by continuous fillet welds. If the backing strip is attached by discontinuous fillet welds see type 3.6.	

Table 3 **Classification of details: butt welds and continuous welded attachments essentially parallel[A)] to the direction of applied stress**

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
3.4		Weld end at an intermediate gap in a longitudinal weld	Intermittent fillet weld with g/h ≤ 2.5		Weld toe improvement techniques not suitable.	Minimum transverse cross section of member at location of potential crack initiation	E	Not applicable	Not applicable	The limiting gap ratio *g*/*h* applies even though adjacent welds can be on opposite sides of a narrow attachment (as in the case of a longitudinal stiffener with staggered fillet welds). Long gaps between intermittent fillet welds are not recommended as they increase the risk of corrosion and, in the case of compression members, may cause local buckling. If intermediate gaps longer than 2.5*h* are required the class is reduced to F.	
3.5		At end of longitudinal weld in flange at cope hole	Web to flange joint at cope hole.	Cope hole no higher than 40% of web height Weld might or might not continue round plate ends.		Minimum transverse cross section of member at location of potential crack initiation. Based on normal stress range in flange S_r and shear stress range in web $\Delta\tau$ at weld ends, with design curve being used in conjunction with normal stress range	F if $\Delta\tau \leq 0.15S_r$, otherwise with class F allowable stress ranges multiplied by the greater of $1.74(\Delta\tau/S_r)$ or 0.5.	D	Applicable assuming t_{eff} = flange thickness and b = 0.25	The existence of the cope hole is allowed for in the nominal stress classification. It should not be regarded as an additional stress concentration in relation to cracking in the flange. With regard to a butt weld in the web, see types 6.2 and 6.3.	
3.6		At weld end in backing strip	Discontinuous fillet weld attaching backing strip	Backing strip to be continuous.		Minimum transverse cross section of member at location of potential crack initiation.	E	D	Not applicable	This type includes tack welds to the edges of continuous longitudinal backing strips irrespective of spacing, provided that the welds conform in all respects to the workmanship requirements for permanent welds and that any undercut on the backing strip is ground smooth. The effects of tack welds which are subsequently fully ground out or incorporated into the butt weld by fusion need not be considered. If the backing strip is either not attached or is attached by a continuous fillet weld, see type 3.3.	

A) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2**.

B) For unprotected steel exposed to sea water or other corrosive environments the class is reduced to C.

Table 4 **Classification of details: welded attachments on the surface or edge of a stressed member**[A)]

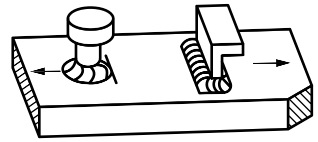
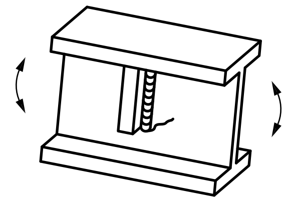
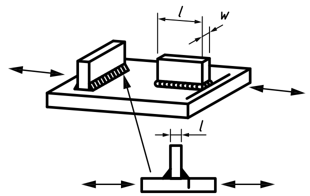
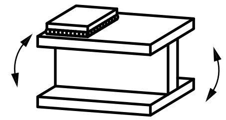
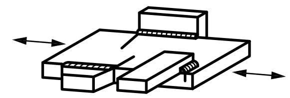
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
4.1	Rolled steel plates, sections, tubes, and built-up members	At weld toe	Fillet welded shear connector. Stud diameter or bracket thickness ≤ 15 mm. For larger values treat as type 4.3 equating diameter or thickness to l.		Weld toe improvement techniques applicable but whole toe should be treated	Minimum transverse cross section of member at location of potential crack initiation	E	D	Applicable assuming b = 0.25		
4.2		At weld toe or end	[A1⟩ Small transverse (thickness l ≤ 15 mm) or longitudinal (length l ≤ 15 mm) attachment ⟨A1]	Fillet or butt welds with welds continued around ends or not. Care needed to avoid undercutting plate edge if weld ends at or close to plate edge. Any such undercutting should be ground out.	Weld toe improvement techniques applicable but only if weld continued around ends in the case of longitudinal attachments. Weld toe improvement might lead to fatigue failure from weld root but the stated benefit still applies.		E	D	Applicable assuming b = 0.25	Butt welded joints require an additional reinforcing fillet so as to provide a toe profile transverse to S_r similar to that which would exist in a fillet welded joint (see **B.4.2.2**)	
4.3		At weld toe or end	[A1⟩ Transverse attachment with 15 mm < l ≤ 150 mm, or longitudinal attachment with 15 mm < l ≤ 150 mm and W ≤ 50 mm ⟨A1]				F	D		The decrease in nominal stress class with increasing attachment length is because more load is transferred into the longer gusset, giving an increase in stress concentration.	[A1⟩  ⟨A1]
4.4		At weld toe or end	Long, narrow attachment with l > 150 mm, W ≤50 mm				F2	D			
4.5		At weld toe or end	Long, wide attachment with l > 150 mm W > 50 mm t_c ≤ 32 mm				G	D	[A1⟩ Applicable assuming b = 0.25 ⟨A1]	The nominal stress classification may be deemed to include stress concentrations arising from normal eccentricities in the thickness direction.	
4.6			l > 150 mm W > 50 mm t_c > 32 mm				G2	D		This type includes parent metal adjacent to the ends of flange cover plates regardless of the shape of the ends. The use of cover plates wider than the flange, to which they are attached, is not recommended.	Cover plate length l, width W and thickness t_c

Table 4 Classification of details: welded attachments on the surface or edge of a stressed member[A)]

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
4.7	Rolled steel plates, sections, tubes, and built-up members	At weld toe or end	Any attachment fillet or butt welded to plate edge	Care needed to avoid undercut on plate corners or to grind it out to a smooth profile should it occur. In particular, weld returns across a corner should be avoided.	Weld toe improvement techniques only applicable with weld continued around ends of attachment.	Minimum transverse cross section of member at location of potential crack initiation	G	D	Not applicable	This type applies regardless of the shape of the end of the attachment. The classification applies to all sizes of attachment both in-plane and out-of-plane with respect to the loaded member. It would therefore include, for example, the junction of two flanges at right angles. In such situations a low fatigue classification can often be avoided by the use of a transition plate (see also joint type 5.6).	

A) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2**.

Table 5 **Classification of details: full penetration butt welds between co-planar plates**[A), B)]

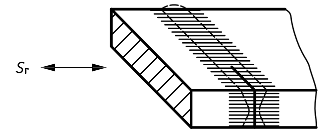
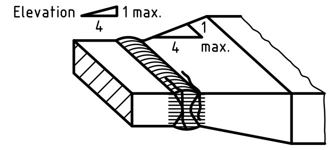
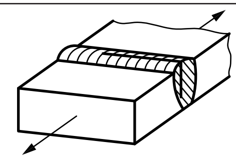
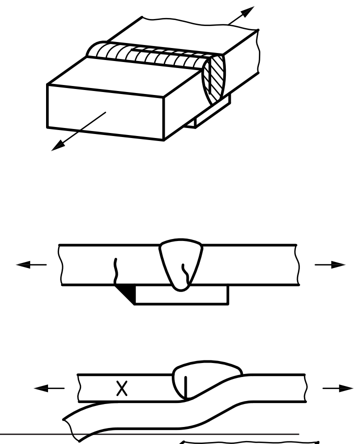
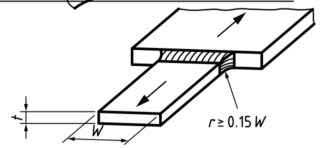
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
5.1	Rolled steel plates only	Weld surface or embedded flaw	Transverse butt weld between plates of equal width and thickness Longitudinal axes in line	Full penetration weld. Misalignment slope ≤1 in 4. Weld reinforcement dressed flush with all traces of the original weld toe removed.	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**)	Minimum transverse cross section of member at location of crack initiation. These classifications do not include allowance for any misalignment of the joint (see **B.5.2.1**)	C	Not applica-ble	Not applicable	In view of view of the difficulty of ensuring freedom from significant flaws (see **14.3.4**), this class should not normally be used in structural work (see **B.2.2.1** and **B.5.2.6**).	
5.2		At weld toe	Transverse butt welds made from both sides by any arc welding process in any position. Longitudinal axes in line Any width or thickness change ≤1 in 4 slope	Full penetration weld. Grind smooth any undercut.	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**) Weld toe improvement techniques applicable		D	D	A1⟩ Applicable assuming *b* = 0.2 ⟨A1		
5.3		At root or toe of weld root bead (Note: classification for weld cap toe same as that for welds made from both sides)	Transverse butt weld made from one side either with temporary non-fusible backing or without backing.	Plate surfaces in contact with temporary backing should be aligned within ±1 mm to justify Class E. Grind smooth undercut on weld face.	Proved free of all flaws which are likely to degrade the joint below its stated classification especially at the weld root where there should be full weld penetration but freedom from excess penetration and significant undercut (see **14.3.4**). Weld toe improvement techniques applicable with access to root.		E	E		Weld root condition assessed by appropriate NDT. Without direct access to root inspection by automated ultrasonic testing (AUT) is recommended (see **B.5.2.4**).	
5.4							F2	F2		Weld root condition not assessed directly by NDT	

Table 5 **Classification of details: full penetration butt welds between co-planar plates[A), B)]**

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
5.5		At weld root	Transverse butt weld made from one side onto permanent backing strip, whether fillet or tack welded in position or integral (e.g. joggle joint)	Full penetration weld.	Weld toe improvement techniques not applicable to weld root.	Minimum transverse cross section of member at location of crack initiation. These classifications do not include allowance for any misalignment of the joint (see **B.5.2.1**).	F	F	[A1] Applicable assuming b = 0.2; related to member X in joggle joint [/A1]		
5.6	Rolled steel plates only	At weld toe	Transverse butt weld between plates of different width. Longitudinal axes in line. Abrupt width change.	Full penetration weld. Corner welds should be built up to radius ≥ 0.15W and corner reinforcement ground flush for 2*t*.	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**). As-welded toes suitable for application of weld toe improvement technique.	Minimum transverse cross section of member at location of crack initiation	F2	D	[A1] Applicable assuming b = 0.2 [/A1]	The effect of the stress concentration at the corner of the joint between two individual plates of different widths in line is included in the nominal stress classification. Stress concentrations due to abrupt changes of width can often be avoided by tapering the wider plate (see type 5.2) Where the end of one plate is butt welded to the surface of another, refer types 7.1 and 7.2.	

A) The joints covered by this table might also fail from internal weld flaws if they are more severe than the external geometrical discontinuity. Weld quality is therefore pertinent to the various classifications (see **14.3.4**).

B) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2**.

Table 6 Classification of details: transverse butt welds in sections, tubes and pipes A)

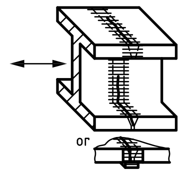
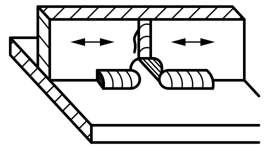
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
6.1	Rolled steel sections and built-up members	Embedded flaw in weld.	Butt weld joining two sections of similar profile end to end with longitudinal axes in line.	Full penetration weld. Misalignment slope ≤ 1 in 4. Dressing of the weld overfill advised to overcome poor profile resulting from the greater misalignments which can occur in the jointing of sections.	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**).	Minimum transverse cross section of member at location of crack initiation.	F2	Not applicable	Not applicable	This joint is prone to weld flaws, which are difficult to detect, in the region of the web/flange junction. A higher class may be justifiable in exceptional circumstances (see **B.6**).	
6.2		Web butt weld toe at cope hole		End of butt weld and reinforcement within a distance equal to cope hole radius to be ground flush (see Figure B.8).		Minimum transverse cross section of member at web butt weld toe. This classification does not include allowance for stress concentration due to cope hole.	D	D	[A1] Applicable assuming b = 0.2 [/A1]	Triangular shaped cope holes are not recommended.	
6.3				Weld not ground.			E	E			

Table 6 **Classification of details: transverse butt welds in sections, tubes and pipes** A)

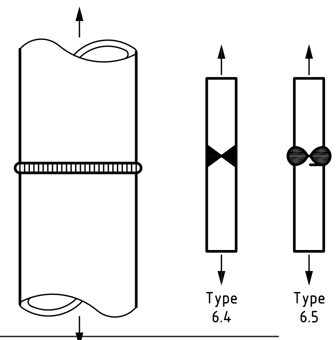
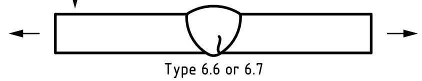
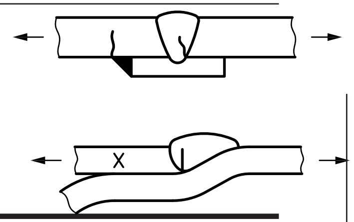
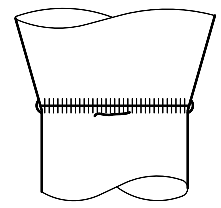
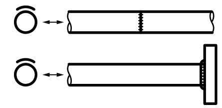
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
6.4	Steel pipes or tubes	Weld surface or embedded flaw	Circumferential butt weld in tube	Weld made from one or both sides with overfill on both outside and inside dressed flush	Proved free of all flaws which are likely to degrade the joint below its stated classification (see **14.3.4**)	Minimum transverse cross section of member at location of crack initiation. This classification does not include allowance for any misalignment of the joint or a thickness change (see **B.5.2.1**)	C	Not applicable	Not applicable	NDT technique capable of ensuring the detection of significant flaws should be selected (see **14.3.4** and **B.5.2.6**).	
6.5		Toe of weld cap or root bead, or toe of weld cap in welds made from one side		Weld made from one or both sides. Weld toe improvement techniques applicable.			D	D	[A1] Applicable assuming b = 0.2 [A1]		
6.6		At weld root bead toe on inside	Circumferential butt weld made from one side either with temporary non-fusible backing or without backing.	Full penetration weld. Pipe surfaces in contact with temporary backing should be aligned within ±1 mm to justify Class E.	As type 5.3 Weld toe improvement techniques only applicable if access to root.		E	E	[A1] Applicable assuming b = 0.2 [A1]	Weld root condition assessed by appropriate NDT (see **B.5.2.4**). Without direct access to root automated ultrasonic testing (AUT) is recommended. Alternatively, class can be validated by testing (see Annex E)	
6.7							F2	F2		Weld root condition not assessed directly by appropriate NDT	
6.8		At butt weld root or toe of fillet weld if used	Circumferential butt weld made from one side onto permanent backing strip whether fillet or tack welded in position or integral (e.g. joggle joint)	Full penetration weld.			F	F	[A1] Applicable assuming b = 0.2; related to member X in joggle joint [A1]		

Table 6 Classification of details: transverse butt welds in sections, tubes and pipes [A)]

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
6.9	Steel tubes	At weld toe on inside or outside of tube	Circumferential butt weld between tubes and conical sections.	See types 6.4 to 6.8, as appropriate.	See types 6.4 to 6.8 as appropriate	Minimum transverse cross-section of member at location of potential crack initiation.	C D E F2 F	C D E F2 F	See type 6.4 to 6.8, as appropriate	Class and stress should be those corresponding to the joint type as indicated in types 6.4 to 6.8, but the nominal stress has also to include the stress concentration factor due to overall form of the joint, including any thickness change. If a stiffener or diaphragm is situated adjacent to the joint, see also type 9.3.	
6.10	Solid or hollow tubular members	At weld toe in tube	Fillet or butt weld between solid or hollow tube and any member		Weld toe improvement techniques not applicable.	Minimum transverse cross-section of tubular member at weld toe.	S_1	Not applicable	Not applicable	Shear failure in tubular member under applied shear stress. For combined loading see **15.2** and Table 17.	
		At weld root				Effective weld throat area (see **15.3**)	S_2	Not applicable		Shear failure across weld throat under applied shear stress. For combined loading see **15.3** and Table 17.	

A) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2.**

Table 7 Classification of details: load carrying fillet and T-butt joints A) between plates B)

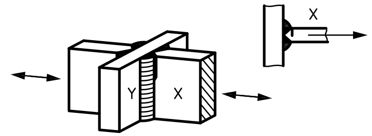
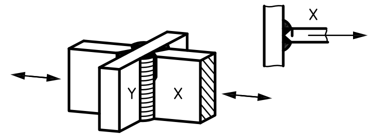
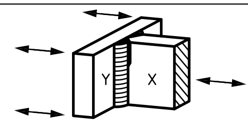
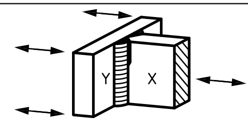
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
7.1	Rolled steel plates, sections and built-up members	At weld toe in member X	Weld joining two members end to end with third member transverse through joint. Member Y can be regarded as one with a non-load-carrying weld (see joint type 4.2 or 4.3).	Full penetration butt weld with longitudinal axes in line.	Any undercut should be ground smooth particularly on the corners of member X If width permits make weld continuous around the joint; otherwise grind ends flush with edge of member X. All regions of plate Y stressed in the through thickness direction to be free from lamellar defects and tears. Weld toe improvement techniques applicable but note need to assess partial penetration welded joints with respect to potential failure through the weld throat (see type 7.8)	Cross section of member X at weld toe. Allowance should be made for any misalignment of the joint exceeding that already allowed for in the classification (see **B.7.2** and **B.5.2.1**). In some circumstances (see **B.7.2**).it might be necessary to include a stress concentration factor in the nominal stress design calculation.	F	D	[A1] Applicable assuming b = 0.25; related to plate X [A1]	Weld metal failure does not govern with full penetration welds.	
7.2				Partial penetration butt or fillet weld with longitudinal axes in line.			F2	D		In this type of joint failure is likely to occur in the weld throat unless the weld is made sufficiently large. (See joint type 7.8).	
7.3			Weld joining the end of one member to the surface of another. Member Y can be regarded as one with a non-load-carrying weld (see joint type 4.2 or 4.3).	Full penetration butt weld made from one or both sides.		Cross section of member X at weld toe	F	D		See joint type 7.1.	
7.4				Partial penetration butt or fillet welds made from both sides			F2	D		See joint type 7.2.	

Table 7 Classification of details: load carrying fillet and T-butt joints A) between plates B)

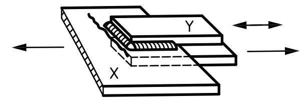
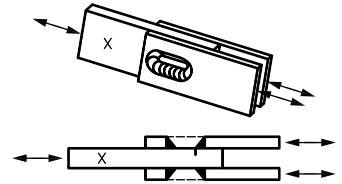
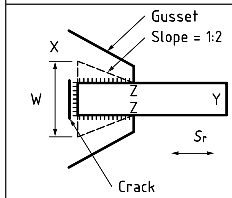
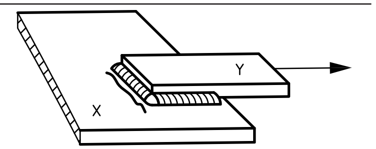
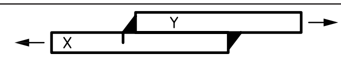
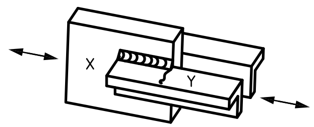
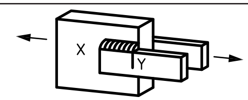
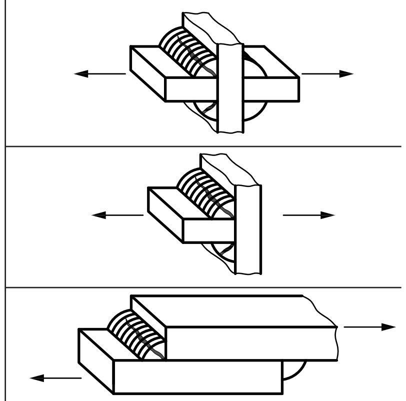
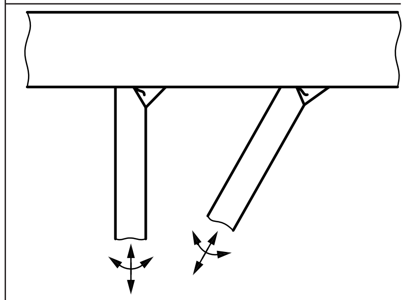
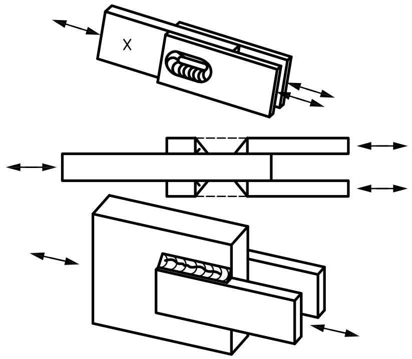
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
7.5	Rolled steel plates, sections and built-up members	At weld toe in member X	Transverse load carrying fillet weld in lap joint on both sides of plate symmetrically.	Care needed to avoid undercutting plate edge if weld ends at or close to edge. Any such undercutting should be ground out.	Weld toe improvement techniques applicable but note need to assess joints with respect to potential failure through the weld throat (see type 7.8 or 7.10)	Cross section of member X at weld toe.	F2	D	[A1] Applicable assuming b = 0.25; related to plate X [/A1]	The nominal stress classification may be deemed to include stress concentrations arising from normal eccentricities in the thickness direction. Where a narrow attachment, Y, is transferring the entire load out of a wide member, X, as in the case of a welded lap type connection between, for example, a cross brace and a gusset, the nominal stress in the gusset at the end of the cross brace varies substantially across the section. For assessing the nominal stress in the gusset X the effective width (W) should be taken as shown for type 7.6.	 Minimum weld length 150 mm 
7.6		At weld toe in member X	Transverse load carrying fillet weld in lap joint on one surface only			These classifications do not include allowance for bending due to misalignment of the applied forces (see **B.5.2.1**)	G	D		For failure in the cross brace at Z, the cross brace, Y, is the "member" and the gusset is the "attachment" (See type 7.7)  W= effective width for nominal stress calculation	 Minimum weld length 150 mm 
7.7		At weld end in member Y	Longitudinal load carrying fillet welds, with the weld end on a plate edge. Weld length ≥ cover plate width	Welds on both edges of member Y. Any undercutting of plate edge should be ground out.	Weld toe improvement techniques not suitable.	Cross section of member Y	G	D	Not applicable	This type applies regardless of the shape of the end of the attachment. Weld returns across a corner should be avoided and the use of cover plates wider than the flange, to which they are attached, is not recommended. For fatigue cracking in member X see types 7.5 or 7.6.	 

Table 7 **Classification of details: load carrying fillet and T-butt joints A) between plates B)**

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
7.8	Rolled steel plates, sections and built-up members	At weld root	Cruciform, lap or T-joint	Fillet or partial penetration weld made from both sides	Weld toe improvement techniques not applicable	Effective weld throat area (see **15.3**) Allowance to be made for any misalignment of cruciform joints exceeding that already allowed for in the classification and for that in a single lap joint (see **B.5.2.1**)	W1 if $\Delta\tau_{//} < 0.3\Delta\sigma_w$ (see **15.3**) S_2 if $\Delta\tau_{//} \geq 0.3\Delta\sigma_w$ (see **15.3**)	Not applicable	Not applicable	This includes joints in which a pulsating load may be carried in bearing, such as the connection of bearing stiffeners to flanges. In such examples the welds should be designed on the assumption that none of the load is carried in bearing.	
7.9		At weld root	Corner or T-joint	Fillet, partial or full penetration weld made from one side		Effective weld throat area (see **15.3**)	G	Not applicable	[A1] Applicable assuming t or t_{eff} = weld throat thickness and b = 0.25 [A1]		
7.10		At fillet weld root	Lap joints	Fillet welded		Effective weld throat area (see **15.3**)	S_2	-	Not applicable	Refers to fatigue failure by shear across weld throat	

A) Butt welded joints should be made with an additional reinforcing fillet so as to provide a similar toe profile to that which would exist in a fillet welded joint (see **B.4.2.2**).

B) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2**.

Table 8 **Classification of details: slotted connections and penetrations through stressed members**

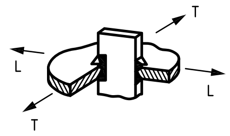
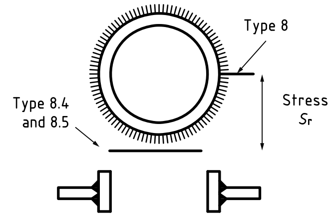
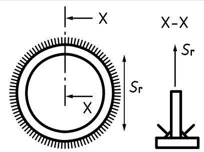
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class: Nominal stress	Class: Hot-spot stress	Thickness and bending correction (see 16.3.2)	Notes	Sketch
8.1	Steel plates, sections or tubes	At weld toe in stressed member	Full penetration butt weld connecting slotted through member	Length of slotted through member, parallel to S_r, ≤ 150 mm	Weld toe improvement techniques applicable	Minimum transverse cross-section of member at location of potential crack initiation	F	D	Applicable assuming b = 0.25	These classifications do not apply to fillet welded joints (see joint type 7.2). However they do apply to loading in either direction (L or T in the sketch).	
8.2				Length > 150 mm			F2	D			
8.3		Weld surface	Welded penetration, on a plane essentially perpendicular to S_r	Full penetration butt weld			D	Not applicable	Not applicable	In this situation the relevant nominal stress should include the stress concentration factor due to the overall geometry of the detail. Full penetration welds are normally required in this situation. If they are not used, the possibility of failure through the weld has to be considered (see type 8.6).	
8.4		At weld toe adjacent to penetration		Full penetration welds	Weld toe improvement techniques applicable but the possibility of failure through a partial penetration weld should be considered (see type 8.6).		F	D	[A1] Applicable assuming b = 0.25 [A1]	For failure at weld toe (see sketch).	
8.5				Fillet or partial penetration welds			F2	D			
8.6	Steel plates, sections or tubes	At weld root	Welded penetration, on a plane essentially perpendicular to S_r	Fillet or partial penetration welds	Weld toe improvement techniques not applicable	Effective weld throat area.(see **15.3**)	W1 if $\Delta\tau_{//} < 0.3\Delta\sigma_w$ (see **15.3**) S_2 if $\Delta\tau_{//} \geq 0.3\Delta\sigma_w$ (see **15.3**)	Not applicable	Not applicable	The stress in the weld should include an appropriate stress concentration factor to allow for the overall joint geometry. This type of failure could subsequently lead to failure of type 8.3. Weld metal failure does not govern with full penetration welds.	

Table 9 **Classification of details: circular tubular members** A)

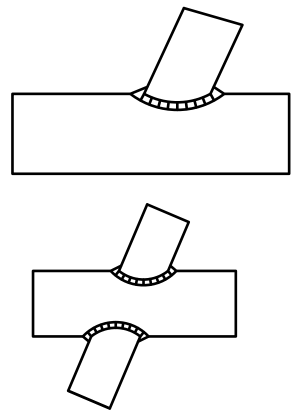
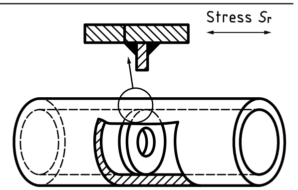
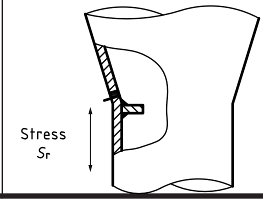
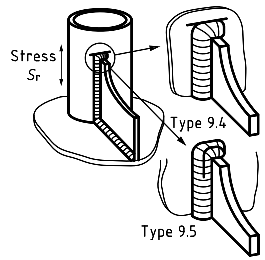
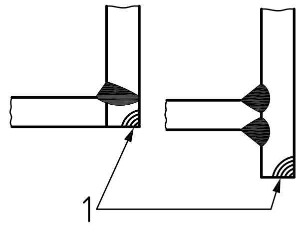
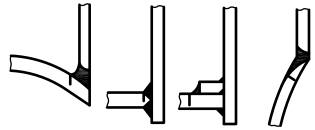
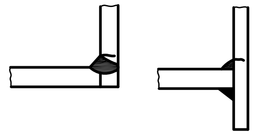
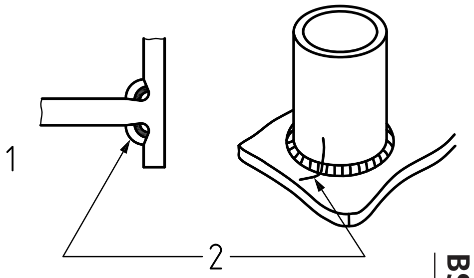
Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
9.1	Steel tubes	Weld toe in chord or brace	Tubular nodal joint	Full penetration welds	Weld toe improvement techniques applicable	Minimum transverse cross-section of tube at weld toe	-	TJ	[A1] Applicable assuming b = 0.25 and $\Delta\sigma_b$ = 0 [/A1]	This type of detail is not classified on the basis of nominal stress. Consequently, design should be based on the hot-spot stress range (see Annex G)	
9.2		At weld toe in tube	Weld attaching diaphragm or stiffener to tubular member				F	D	Applicable assuming b = 0.25	S_r might be intensified as a result of the presence of the attachment, particularly if it is large. Nominal stress should include the stress concentration factor due to overall shape of adjoining structure.	
9.3			Bevel butt or fillet welded attachment in a region of stress concentration				E, F or F2 depending on attachment size (see types 4.2 to 4.5)	D	Applicable assuming b = 0.25		

Table 9 Classification of details: circular tubular members [A)]

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Class		Thickness and bending correction (see **16.3.2**)	Notes	Sketch
							Nominal stress	Hot-spot stress			
9.4	Steel tubes	At weld toe in tube	Welded gusseted connections	Butt or fillet welded	Weld toe improvement techniques applicable but gusset shape should also be tapered to provide smooth transition from gusset edge to tube surface.	Minimum transverse cross-section of tube at weld toe	F	D	[A1⟩ Applicable assuming b = 0.25 ⟨A1]	The design stress has to include any local bending stress adjacent to the weld end.	
9.5		At weld root		Fillet welded	Weld toe improvement techniques not applicable	Effective weld throat area. (see **15.3**)	W1 if $\Delta\tau_{//} < 0.3\Delta\sigma_w$ (see **15.3**) S_2 if $\Delta\tau_{//} \geq 0.3\Delta\sigma_w$ (see **15.3**)	Not applicable	Not applicable	For failure in the weld throat of fillet welded joints.	

A) The classifications apply for stresses acting within ±15° of the stressing direction shown. For greater angles see **15.2**.

Table 10 Classification of details: branch connections in pressurized containers

Type no.	Product form	Location of potential crack initiation	Detail	Manufacturing requirements	Special treatments or requirements	Design stress area	Detail class		Thickness and bending correction (see 16.3.2)	Notes	Sketch
							Nominal stress	Hot-spot stress			
10.1	Steel plates	At crotch corner	Nozzle to shell joint		Fatigue strength improved with generous radius on nozzle corner	Nozzle section normal to plane of cracking	D	Not applicable	Not applicable	Nominal stress has to include the relevant stress concentration factor.	 1 = Cracks radiate from corner into plate – sketches show plane of crack
10.2		At weld toe in shell			Weld toe improvement techniques applicable	Minimum transverse cross-section of member at location of potential crack initiation	F	D	[A1] Applicable assuming *b* = 0.25 [/A1]	Nominal stress has to include the relevant stress concentration factor. Note that the possibility of failure through the weld throat under stresses acting normal to weld length should be assessed as for joint type 7.8	
10.3		At weld toe in branch					F	D			
10.4		At weld root		Full-penetration weld	Weld toe improvement techniques not applicable	Minimum transverse cross-section of weld	D	Not applicable	Not applicable		
				Fillet or partial penetration weld			F				1 = Plane of crack 2 = Cracks radiate from root

13 Unclassified details

13.1 General

Details that are not expressly classified should be treated as class G2, or class W1 for load-carrying weld metal, unless a higher classification can be justified either by reference to published experimental work or by carrying out special tests. Such tests should be sufficiently extensive to allow the basic or design *S-N* curve to be determined in the manner used for the standard classes (see Annex E).

13.2 Post-welding treatments

Where the classification of Table 4 to Table 10 does not give adequate fatigue resistance, the performance of weld details may be improved by post-welding treatments, as detailed in **16.3.5** and Annex F. The detail types that are suitable for application of the techniques described in Annex F are indicated in Table 4 to Table 10. When improved fatigue performance is required, and the proposed improvement method is not covered by Annex F, or the improvement stipulated therein is insufficient, the detail should be classified by testing (see **13.1**).

14 Workmanship and Inspection

14.1 General

The classifications for structural details in Table 1 to Table 10 refer to specific modes of fatigue failure. However, the details concerned can contain features that could provide alternative sites for fatigue crack initiation or increase the severity of those covered by the classification scheme, in both cases possibly resulting in lower fatigue performance. Therefore, in fatigue-loaded steel products there should be adequate control and inspection of manufacturing quality, together with acceptance limits related specifically to fatigue resistance. Where the classification of a detail is dependent on particular manufacturing or inspection requirements, the necessary standards of workmanship and inspection should be indicated on the relevant drawings.

In practice, the overall quality requirements for a product are determined by a number of performance requirements, depending on its application. These might or might not be sufficient to cover the quality requirements needed for fatigue performance.

Typical performance requirements other than fatigue can include one or more of the following:

a) static strength (resistance to fatigue or buckling);

b) resistance to deformation or deflection;

c) energy absorption (ductility);

d) geometrical precision (dimensional tolerances);

e) clearance to moving parts;

f) fit-up to adjacent parts;

g) resistance to corrosion;

h) lack of obstruction to fluid flow;

i) aesthetic appearance; and

j) hygiene (elimination of crevices).

Some of the recommendations in the sub-clause can impose very high requirements for specific aspects of quality.

The existing product specification requirements should therefore be adequate for fatigue performance purposes. If not, then the non-conforming aspects should be enhanced.

It should not be assumed that the quality requirements needed for fatigue performance alone adequately cover those required for other performance requirements. In some cases they might not.

Quality aspects which are detrimental to fatigue performance are described in **14.2**.

Guidance on quality levels needed to ensure that the various detail classes are achieved is given in **14.3**.

14.2 Quality aspects detrimental to fatigue

14.2.1 General

The presence of unspecified notches in a new product can have the same effect on the remaining fatigue life as a fatigue crack of the same size. Such notches can therefore significantly shorten the fatigue life of the product, by eliminating the period of early crack growth.

The most severe notches are those that exhibit the following characteristics:

a) planar with a sharp tip;

b) orientated normal to the direction of fluctuating stress;

c) located at or close to a surface; and

d) located at or close to one of the classified initiation sites.

Other quality aspects which can adversely affect fatigue life are those which result in an increase in cyclic stress at one of the classified initiation sites. These are usually caused by unspecified variations in product geometry.

The most common types of quality aspects important to fatigue are listed in **14.2.2** to **14.2.4**.

Some quality restrictions are given in Table 1 to Table 10. Guidance on others is given in **14.3**.

14.2.2 Parent materials

Particular material quality aspects that should be controlled for fatigue purposes are:

a) surface imperfections (rolling flaws and their repairs, corrosion pitting, accidental damage such as arc strike and weld spatter);

b) internal laminations and inclusions;

c) incorrect geometry (e.g. deviations in shape, flatness or thickness at the limits of tolerances, see Note).

d) cracks due to inadequate removal of hydrogen (e.g. heat treatment or plating processes, bolts of Grade 10.9 and above).

e) liquid metal assisted cracking (LMAC) in heavy galvanized products.

NOTE Design stresses calculated using nominal thickness are applicable to plates manufactured to the thickness tolerance requirements of BS EN 10051 and BS EN 10029:2010 Classes A, B and C.

14.2.3 Bolted joints

Particular quality aspects that should be controlled for fatigue purposes are:

a) correct alignment of holes (in un-tensioned lap joints in particular);

b) correct fit-up between faying surfaces prior to tightening (most important with pre-tensioned tensile joints);

c) correct bolt tightening procedure and sequence; and, in cases where a torque measure method is applied;

d) correct friction coefficients in threads and under washer faces and the correct lubrication required by the torque tightening specification.

14.2.4 Welded joints

Particular quality aspects that should be controlled for fatigue purposes are:

a) cracks (usually a sign of serious material or process problems);

b) lack of fusion or penetration (surface and embedded);

c) undercut at weld toes (transverse to stress direction);

d) volumetric flaws (e.g. porosity, slag);

e) excess weld cap or root profiles;

f) loss of weld throat (butt and fillet welds); and

g) linear or angular misalignment in cruciform or in-line butt welds.

No unauthorized attachments, introduced as fabrication aids or later in service, should be welded to the product in regions where such attachments would reduce the classification assumed for design. Unless allowance is made for them at the design stage, they should be removed and the area ground out and checked by NDT to ensure that no unacceptable flaws remain.

14.3 Control of quality for fatigue

14.3.1 General

Table 1 to Table 10 provide limited information on manufacturing requirements for certain type numbers. In general, most of the necessary controls on fatigue-related quality aspects described in **14.2** are not provided in the tables. Guidance on control of these quality aspects is given in **14.3.2** to **14.3.4**.

The detail classes given in Table 1 to Table 10 denote the highest recommended *S-N* curves for each detail type. Any manufacturing recommendations in the Table 1 to Table 10 assume that the detail is to be stressed up to the limit allowed by that *S-N* without a shortfall in fatigue life. In practice there might be a number of details in a product that are stressed to similar levels, but have different detail classes. On the assumption that all details have been assessed to have a satisfactory fatigue life, details with the higher classes are not stressed to the maximum limit permitted by the classification. For example if a component has a long wide welded attachment on its surface (detail type 4.5, nominal stress class G) and there is a transverse double sided butt weld (detail type 5.3, class D) close by and stressed to the same level then the butt weld cannot be stressed to its maximum permissible limit (e.g. 90 N/mm^2 at 2×10^6) because the class G detail would be limited to 50 N/mm^2 at 2×10^6 cycles. Thus, in the case of the butt weld the required class for this product is G, i.e. 45% lower in stress terms than the maximum permitted detail class D according to the table. The required class is therefore a function of the stress spectrum only and is the lowest *S-N* curve which just provides an acceptable fatigue life.

[A1] It is therefore possible to select the fatigue quality control requirements for a particular product on a fitness-for-purpose basis, using the assessment methods in BS 7910. [A1] For products with medium or low cyclic stressing this is likely to result in significant quality control savings compared to those required for meeting the maximum permitted class for all the details in the product. The required class at any location in the product can be readily determined by repeating the calculation procedures in Clause **15** and Clause **16** for various *S-N* curves until the lowest acceptable curve is found for that location. The required class, as in the case of the detail class in Table 1 to Table 10, depends on the direction of stress fluctuation and is likely to be different in the two orthogonal directions.

14.3.2 Parent material

14.3.2.1 Material quality

Where material is to be welded, selection of a suitable carbon equivalent value (CEV) is important for prevention of cracks.

Where material is to be used for transverse elements in welded cruciform or T-joints (see element "Y" in Table 7, types 7.1 to 7.4) and there is the risk of lamellar tearing, the following testing should be carried out:

a) through-thickness deformation (Z) testing; and

b) ultrasonic inspection to ensure that it is free from laminations and inclusion cluster.

14.3.2.2 Surface condition

The following should be carried out to ensure the surface is in an acceptable condition.

a) Visual inspection should be carried out to ensure that the surfaces are free from damage for required classes D and above (corrosion pits, arc strikes and spatter, mechanical damage).

b) NDT of any areas where temporary welding has been carried out for required classes F and above (repair of rolling flaws, removed temporary attachments) should be undertaken to ensure freedom from cracks.

c) Excessive irregularities (see Type 1.4, Table 1) in thermally cut edges should be ground smooth for required classes G and above.

d) Cold formed radiused edges should be checked for cracks by NDT where the class requirement is G or above in the transverse direction and E or above in the longitudinal direction.

e) Bolts of Grade 10.9 and above which are to be coated or galvanized should preferably not be subject to acid or electro-deposition processes. If this is unavoidable, heat treatment to remove hydrogen should be carried out effectively (otherwise stress corrosion cracking might occur).

14.3.3 Bolted joints

Groups of holes in lap joints, (e.g. detail types 2.6 and 2.7), with bolts carrying load in bearing should be jig, Computer Numerically Controlled (CNC) or match drilled to ensure uniform bearing between all bolts in the group. Appropriate tolerances on diameter, position and alignment of parts should be specified.

When the fatigue design of a bolted joint is based on bolts being pre-tensioned to a nominated or minimum value, it should be ensured that the required pre-tension is achieved. Bolt tightening, (and therefore bolt pre-tension), can be obtained using torque, angle or bolt-stretch measurements. The bolt pre-loading method and hence pre-load accuracy should be selected after assessing the criticality of the joint. For all pre-loading methods, the correct measurements, torque sequences and torque-checking data should be defined. Calibration tests should be carried out to confirm that the required pre-tension is achieved. Established data on similar fasteners may be used if allowance is made for variation between batches and products. A system of monitoring and recording should be implemented to ensure that procedures are followed correctly.

Fit-up between parts with flat contact surfaces should be designed to ensure uniform contact at the initial hand-tight stage prior to final tensioning.

The required bolt pre-load should be obtained by the following procedure.

a) Using a torque control method apply, in a defined sequence, sufficient torque to bring the components into full face to face contact.

b) Using a torque measurement, bolt or nut part-turn or bolt-stretch method, that is appropriate for the criticality of the joint, apply the final pre-load.

This procedure does not preclude the use of a final stage of tightening using other methods.

Where a hollow jack might be used to tension long studs or bolts, tests should be carried out to measure the pre-load loss, due to the applied dynamic loads. This should be taken into account when determining the minimum pre-tension to be used as the basis of design.

14.3.4 Welded joints

Welding is a process where notch-like imperfections and deviations in geometry are difficult to eliminate entirely, particularly in arc welds. The degree of severity depends primarily on the level of control of the process, but also on other factors such as the type of process, the materials, the joint geometry and access.

The size of imperfection that can be tolerated without compromising the fatigue life based on the details classified in Table 3 to Table 10 becomes smaller as the required class (defined in **14.3.1**) increases and the choice of viable details becomes more limited. For example, if the required class is C and a transverse butt weld in a plate is required, only detail type 5.1 is acceptable. This detail involves removal of all local geometrical stress raisers such as weld toes, welding access from both sides and a high level of NDT. If the component cross-section is a more complex shape than a flat plate then the access for welding is reduced and the likelihood of producing larger imperfections and also not detecting them by NDT increases significantly. For this reason the maximum detail class for a butt weld in this case is reduced to F2 (see detail type 6.1), a reduction of four classes.

In contrast, if the required class is only W1 or less, a very wide choice of joints is available. This includes transverse load-carrying fillet welded joints where the design allows major areas of the joint to remain unfused with severe notches at the weld roots (see detail types 7.2 and 7.4). Whilst these cruciform and T-joints can be stressed in the parent metal to the level of nominal stress class F2 as far as failure from the toe is concerned, failure from the root through the weld metal is limited to W1 (detail type 7.8).

Direction of stress fluctuation with respect to weld axis is a further important factor. Weld imperfections with more notch-like features (cracks, lack of fusion, lack of penetration, undercut, slag inclusions) are generally orientated parallel to the weld axis and are therefore much more severe in terms of fatigue performance when the stress direction is transverse to the weld axis. Local stress raisers such as weld toes are similarly more unfavourably orientated with respect to transverse stresses. This is reflected in the detail classifications where, for example, a continuously fillet welded T-joint stressed longitudinally is acceptable for a required class of C (or D if it contains stop-starts, see detail types 3.2 and 3.3 respectively); whereas if stressed transversely through the base of the Tee it would be restricted typically to nominal stress class E or F (detail type 4.2 or 4.3 respectively, within certain dimensional limits). If stressed transversely through the stalk of the Tee it would be restricted to W1 with respect to weld throat stresses.

The control of welding in production is generally achieved by use of the following measures:

a) relevant documented weld procedures qualified by test;

b) welders appropriately qualified by test for the range of procedures used on the product;

c) appropriate welding quality management system;

d) appropriate inspection and testing procedures and acceptance criteria for the production welds; and

e) good practice for storage and handling of welding consumables to prevent dampness and contamination.

Where fatigue performance is an important requirement for the product (i.e. the requirement is class W1 or higher) all these control measures should be adopted in accordance with BS EN 1011-1, BS EN 1011-2 (for ferritic steels) and BS EN 1011-3 (for stainless steels).

For small mass-produced initial products it might be practicable to inspect the quality on a random sampling basis by destructive tests (e.g. sectioning of welds). Routine performance testing (by loading) is usually impracticable for verifying fatigue life because of the time it takes. In the majority of cases verification that the required quality has been achieved in production is by NDT. If high fatigue classes are required the limitations of the various testing techniques in detection, characterization and sizing of the various welding imperfections should be taken into account.

In some instances it might be current practice to use the same acceptance criteria in production as used for procedure or weld qualification (see BS EN ISO 15609-1 and BS EN 287-1) or those in BS EN ISO 5817. Such specifications are based on arbitrary quality criteria and might have aspects that are not adequate for higher fatigue class requirements. They are also likely to contain requirements that incur additional cost for lower fatigue class requirements. For these reasons it is preferable to specify quality requirements that are based more on fitness-for-purpose principles. Guidance is given [A1⟩ in Table 11 ⟨A1] on appropriate limits of those imperfection types which need to be controlled for fatigue purposes in addition to those specified in Table 3 to Table 10. This guidance is expressed in terms of the "required class", described in **14.3.1**, so that an economic level of quality can be specified for any given product.

Where an existing production weld quality specification exists for the product, it should be checked to ensure that it covers the types and limiting sizes of imperfection recommended below for controlling fatigue performance. Where it does not fully cover the fatigue quality requirements the specification should be amended accordingly and it should be made clear that if any fatigue imperfection limit is exceeded in production it is cause for rejection. The scope of inspection might need to be enhanced in accordance with the recommendations that follow.

If no production weld quality specification already exists for the product the specification should be based on the recommendations in this clause, provided that any other relevant performance requirements (see **14.1**) do not require enhancement above those required for fatigue.

Where the fatigue stressing requirements in a product are moderate, e.g. a maximum class requirement no higher than F2 anywhere, a general fitness-for-purpose requirement for the product quality should be based on the maximum class required, subject to any enhancement for other performance requirements (see **14.1**). If only a small minority of specific welds in the product experience high fatigue loading and have a higher class requirement than F2, it might be more economical to identify these by their class requirement so that extra inspection and higher quality requirements can be targeted to those welds that are more critical. The welds in question should be indicated on the drawings with their class requirement as defined in **14.3.1** (not their detail class as given in Table 3 to Table 10).

Table 12 gives guidance on the fatigue acceptance of volumetric embedded imperfections to meet Class C or lower (Class B is excluded because no embedded imperfections are permitted).

Table 13 gives acceptance levels for undercut at transverse weld toes. These limits might not be small enough for other performance requirements (see **14.1**) and might be larger than in some general quality control specifications.

The welds should not be smaller than the specified size. This applies particularly to transversely loaded partial penetration butt welds and both the leg length and throat dimension of transversely loaded fillet welds, where the fatigue life is significantly reduced by small shortfalls in these dimensions.

With regard to deviations from the nominal geometry, particular attention is drawn to joint misalignment, as the acceptance levels in many quality control-based specifications are too large for fatigue-loaded joints. It is usually necessary to allow for the effect of even nominally acceptable misalignment as a source of additional stress. This effect can be expressed in terms of the stress magnification factor, k_m, which is used either to increase the estimated applied stress range or to reduce the fatigue strength obtained from the relevant *S-N* curve (see **B.5.2.1**). Assessment of any misalignment is required for transverse butt welds as Class D refers to perfectly aligned joints. This is also true for any assessment of potential weld toe failure using the hot-spot stress, although that effect could be included in the calculation of the hot-spot stress if the relevant type and extent of misalignment was included in the finite element model or the hot-spot stress was obtained by surface stress extrapolation from strains measured on the actual welded detail concerned. However, linear misalignment e/t up to 0.05, corresponding to k_m= 1.15, is included in the data behind Classes F and F2 for cruciform joints. Therefore, on a fitness-for-purpose basis, only misalignment above this level needs to be considered when assessing cruciform joints. Table 14 to Table 16 illustrate the effect of misalignment on fatigue strength for these three situations.

Table 11 **Guidance on non-destructive testing of planar imperfections in welds**

Required class	Accessible surface breaking imperfections								Buried or inaccessible surface breaking imperfections										
	NDT capabilities			Transverse welds			Longitudinal welds		NDT capabilities			Transverse welds				Longitudinal welds			
		SIZE			ACC					SIZE		Butt welds		Fillet welds		Butt welds		Fillet welds	
	DET	L	H	EXT	Toe	Cap, root	EXT	ACC	DET	L	H	EXT	ACC	EXT	ACC	EXT	ACC	EXT	ACC
B	(M)	X	X	-	-	-	C	N	(U)	X	X	-	-	-	-	C	N	-	-
C	M	X	X	C	N	N	C	N	(U)	X	X	C	N	-	-	C	N	P	N
D	M	M	X	C	N	N	P+S	N/F	U	(U)	(U)	C	N	-	-	P	N	P	N
E	M,P	M,P	(A)	C	N	N/F	P+S	N/F	U	U	(U)	C	N	-	-	P	N/F	O	N/F
F	M,P,E	M,P	A	P	N	N/F	P	N/F	U	U	U	C	N/F	-	-	O	N/F	O	N/F
F2	M,P,E	M,P	A	P	N/F	N/F	P	N/F	U	U	U	P	N/F	-	-	O	N/F	X	X
G	M,P,E	M,P	A	P	N/F	N/F	O	N/F	U	U	U	P	N/F	P	N/F	O	N/F	X	X
G2	M,P,E	M,P	A	O	N/F	N/F	O	N/F	U	U	U	P	N/F	P	N/F	X	X	X	X
W1	M,P,E	M,P	A	O	N/F	N/F	O	N/F	U	U	U	P	N/F	P	N/F	X	X	X	X

Key

DET Techniques with adequate detection capability: M = magnetic particle; P = penetrant; E = eddy current or alternating current field measurement (ACFM); A = alternating current potential drop (only needed for fitness-for purpose assessment); U = ultrasonics; ( ) = borderline - special techniques needed. Radiography not suitable for planar imperfections.

SIZE Sizing capability adequate for typical dimensions of acceptable imperfection: L = length transverse to weld direction; H = height in through thickness direction; X = none available.

EXT Extent of inspection in terms of proportion of short welds tested or proportion of length of long welds: C = complete (100%); P = partial (50-10%); O = occasional (≤ 5%); S = start/stop (including repairs) only; X = not normally required. If chronic planar imperfections are detected during ‘P’ or ‘O’ inspections then ‘C’ should be implemented until the cause has been eliminated

ACC Acceptability of imperfection: N = not acceptable (grind out, repair by welding where necessary); /F option of assessing on a fitness-for-purpose basis using L and H measurements and procedures in BS 7910; X = accept on basis of surface inspection only.

Table 12 **Fatigue based acceptance levels for embedded non-planar imperfections in butt welds** A)

Required class	Maximum length of solid inclusion, mm		Maximum % projected area of porosity on radiograph B) C)	Individual pore diameter, mm C)
	As-welded	Stress relieved by PWHT		
C	1.6 D)E)	5 D)	3	0.25t, ≤ 3 mm maximum
D	2.4 D)E)	19	3	0.25t, 6 mm maximum
E	4 D)	58	3	
F	10	No limit	5	
F2	35		5	
G or lower	No limit		5	

A) Applicable to imperfections which are not located within 5 mm of the surface or an adjacent imperfection (or within a distance equal to the maximum permitted length, whichever is the smaller). [A1⟩ Imperfections at or within 5 mm of a surface could be assessed as planar flaws in accordance with BS 7910. ⟨A1]

B) For assessing porosity, the area of radiograph used should be the length of the weld affected by porosity multiplied by the maximum width of weld.

C) Any porosity level which obstructs an ultrasonic inspection should be rejected.

D) Only measured by radiography.

E) Accurate measurement difficult by NDT.

Table 13 **Fatigue based acceptance levels for undercut in transversely stressed welds** A) B)

Required class	Depth of undercut C)	
	Butt welds	Fillet welds
C	Not permitted	—
D	0.025t, ≤1 mm	0.01t, ≤ 0.5 mm
E	0.05t, ≤1 mm	0.025t, ≤ 1 mm
F	0.075t, ≤ 1 mm	0.05t, ≤ 1 mm
F2	0.10t, ≤ 1 mm	0.075t, ≤ 1 mm
[A1⟩ G or lower ⟨A1]	0.10t, ≤ 1 mm	0.10t, < 1 mm

A) Applicable only to undercut with a clearly visible root radius. Sharp undercut should not be accepted as the depth cannot be measured and can conceal cracks or lack of sidewall fusion.

B) Applicable for 10 ≤ t ≤ 40 mm; undercut in thinner or thicker material should be assessed as a planar flaw in accordance with BS 7910.

C) Undercut in continuous welds stressed in the longitudinal direction only does not affect the fatigue life of a joint. Therefore, there are no limits to their sizes from the fitness-for-purpose viewpoint.

Table 14 **Effect of misalignment on the fatigue strength of transverse butt welded joints**

Class	k_m A)	Corresponding linear misalignment, e/t B)
D	1	0
E	1.14	0.05
F	1.34	0.11
F2	1.52	0.17
G	1.84	0.28
G2	2.17	0.39

A) k_m is the stress magnification factor due to any form of misalignment (linear, angular or a combination) that can be assumed to be included if the butt weld is designed on the basis of the nominal applied stress range in conjunction with the *S-N* curve for the indicated class.

B) The corresponding values for the extent of angular misalignment depend on too many factors to tabulate them (see **B.5.2.1**).

Table 15 **Effect of misalignment on the fatigue strength of cruciform welded joints**

Class	Full-penetration butt welds		Fillet or partial-penetration welds	
	Effective k_m A)	Corresponding linear misalignment, e/t B)	Effective k_m A)	Corresponding linear misalignment, e/t B)
F	1.0	0.05	—	—
F2	1.12	0.09	1.0	0.05
G	1.36	0.17	1.20	0.12
G2	1.63	0.26	1.43	0.19

A) Effective values of k_m allow for the fact that the nominal stress class refers to joints in which e/t can be up to 0.05. However, they refer to the extent of any form of misalignment (linear, angular or a combination) that can be assumed to be included if the cruciform joint is designed on the basis of the nominal applied stress range used in conjunction with the *S-N* curve for the indicated class.

B) The corresponding values for the extent of angular misalignment depend on too many factors to tabulate them (see **B.5.2.1**).

Table 16 **Effect of misalignment on the fatigue strength of transverse butt or cruciform welded joints being assessed in terms of hot-spot stress**

Class	k_m A)	Corresponding linear misalignment, e/t B)
D	1	0
E	1.14	0.05
F	1.34	0.11
F2	1.52	0.17
G	1.84	0.28
G2	2.17	0.39

A) k_m is the stress magnification factor due to any form of misalignment (linear, angular or a combination) that can be assumed to be included if the weld joint is designed on the basis of the hot-spot stress range calculated assuming perfect alignment used in conjunction with the *S-N* curve for the indicated class.

B) The corresponding values for the extent of angular misalignment depend on too many factors to tabulate them (see **B.5.2.1**).

15 Stress calculations

15.1 General

The procedure for the fatigue analysis for welded structures, including flame-cut edges, is based on the assumption that it is only necessary to consider ranges of cyclic stress in determining the fatigue life, i.e. mean stresses are not taken into account. However, for non-welded details subjected to stress cycles which are partly or wholly in compression the effective stress range might be lower (see **15.4**).

The stress used to assess the structural details included in Table 1 to Table 10 depends on the class and the mode of potential fatigue cracking considered. In most cases the latter is by direct (or normal) stress fatigue failure under crack opening mode I (see Annex D) and the fatigue assessments are performed using either nominal or hot-spot structural principal stress ranges. The corresponding approaches are referred to as the nominal stress-based (see **15.6**) or hot-spot stress-based fatigue assessment method (see **15.7**). However, classes S_1 and S_2 refer to potential shear fatigue failure under crack opening modes II or III (see Annex D). In such cases the stress used in the fatigue assessment depends on the location of cracking, the ratio of applied shear to direct stress and whether combined applied shear and direct stresses act in-phase (i.e. principal stress directions remain constant during load cycling) or not, as summarized in Table 17. As indicated, in some cases a direct stress-based class might be appropriate. Further details are provided in **15.2, 15.3** and **15.4**.

Table 17 **Stresses used in fatigue assessments involving applied shear stresses**

Case	Fatigue cracking mode	Loading	Class	Stress
Parent metal failure (from weld toe or bolt surface)	Shear modes II or III	Pure shear	S_1	Nominal shear stress range
	Normal mode I	Combined in-phase direct and shear stresses	D, E, F, F2, G or G2, as appropriate	Maximum nominal or hot-spot structural principal stress range
	Combined modes I and II or III	Combined out-of-phase direct and shear stresses	S_1 but endurances halved	
Weld throat failure (from root of directly loaded fillet or partial penetration weld)	Shear modes II or III	Pure shear	S_2	Nominal shear stress range
	Normal mode I	Combined in-phase direct and shear stresses with $\tau/\sigma \leq 0.3$	W1	Nominal resultant stress range
	Combined modes I and II or III	Combined in-phase direct and shear stresses with $\tau/\sigma > 0.3$	S_2	
	Combined modes I and II or III	Combined out-of-phase direct and shear stresses	S_2 but endurances halved	

Nominal stresses should be used with nominal stress *S-N* curves. The nominal stress should include the effects of local stress concentrations if the detail being assessed is not fully represented by the description and figures in Table 1 to Table 10. Alternatively, in assessments of potential failure from a weld toe, hot-spot stresses should be used (see **15.7**). Stress ranges for post-weld heat-treated joints should be assumed to be the same as for as-welded joints (but see **16.3**).

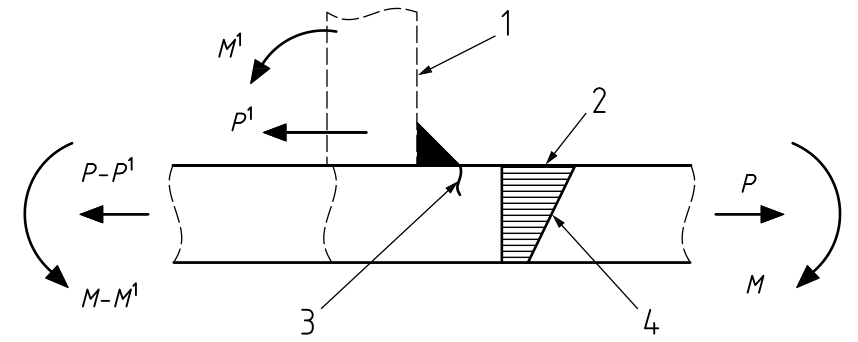
15.2 Stress range in parent material

In most cases the potential fatigue crack is located in parent material adjacent to some form of stress concentration, e.g. at a weld toe or bolt hole. Provided that the direction of the principal stress does not change significantly in the course of a stress cycle (< 20°), the relevant cyclic stress for fatigue assessment should be taken as the maximum range through which any principal stress passes in the parent metal adjacent to the potential crack location, as shown in Figure 2a). Tension stresses are considered positive and compression stresses negative. In practice, the through-thickness component of stress is rarely relevant and may usually be ignored.

NOTE For tubular nodal joints see Annex G.

In estimating the maximum principal stress, shear stresses less than 15% of a coexistent direct stress may be neglected. In the case of pure shear, the relevant stress is the nominal shear stress range on the section of potential fatigue failure, as shown for applied torsion of a tube in Figure 2b).

Figure 2 **Reference stress in parent metal**



a) Applied direct stress

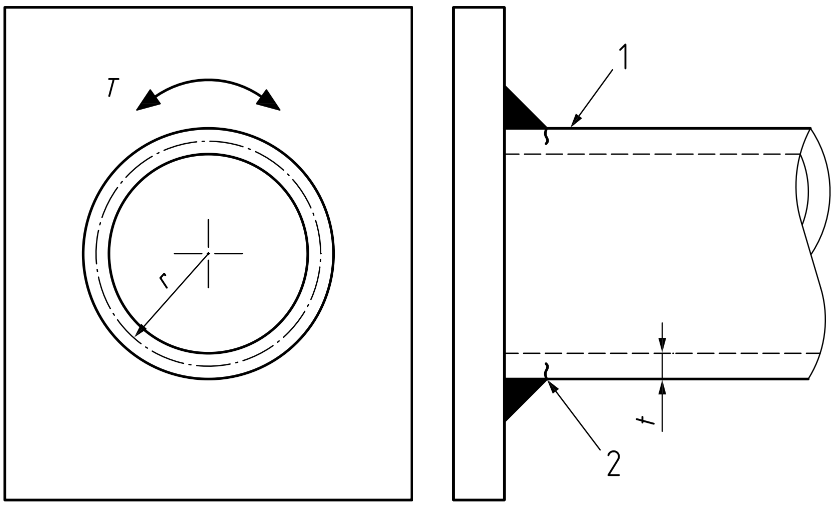
Key

1 Welded attachment

2 Nominal stress = ($P/A + M/Z$)

3 Potential crack location

4 Stress distribution



b) Applied shear stress

Key

1 Nominal stress = T/Ar

2 Potential crack location

If the direction of principal stress changes during the stress cycle, e.g. as a result of a phase difference between the two fluctuating load sources, the cyclic stress range should be derived from the principal stresses calculated at the two extremes, i.e. the peak and trough, of the combined loading cycle.

The peak and trough values of principal stress should be those on principal planes which are not more than 45° apart. Therefore, if σ_x, σ_y and τ are the coexistent values (with appropriate signs) of the two orthogonal direct stresses and the shear stresses at the point under consideration, the relevant principal stress should be selected if either:

a) $\sigma_x - \sigma_y$ is at least double the corresponding shear stress τ at both peak and trough; or

b) the signs of $\sigma_x - \sigma_y$ and τ both reverse or both remain the same at the peak and the trough.

In either a) or b), provided that $\sigma_x^2 \geq \sigma_y^2$ at both peak and trough, the required stress range is the algebraic difference between the numerically greater peak principal stress and the numerically greater trough principal stress.

Where cycling is of such a complex nature that it is not clear which two load conditions would result in the greatest value of principal stress range, they should be established by calculating principal stresses for all pairs of load conditions. Alternatively, it may be assumed that the required stress range is the difference between the algebraically greatest and smallest principal stresses occurring during the whole loading cycle regardless of their directions.

The classification for welded joints Table 3 to Table 10 depends on the direction of loading and the classification scheme is generally arranged on the basis that the welded joint under consideration is either parallel (longitudinal joints) or normal (transverse joints) to the stress direction indicated. Applied stresses that are not acting in either of these directions might be effectively less damaging than assumed, but there might be the risk of fatigue failure from a different location where the classification might be lower. For example, the class of a type 3.2 detail drops from C to E or F with respect to any stress acting normal to the weld toe. Therefore, for stress directions that are greater than ±15° from that shown in the figures the welded joint should be assessed with respect to every potential fatigue crack initiation site. The assessment should be based on the maximum principal stress range if it acts within ±45° of normal to the potential fatigue crack while for greater angles it can be based on the normal stress range acting on the plane of potential fatigue cracking considered, as determined from the principal stress ranges.

Unless otherwise stated in Table 1 to Table 10, the stress should be based on the net section. Where appropriate, allowance should be made for geometrical stress concentrations (see **15.6.4**).

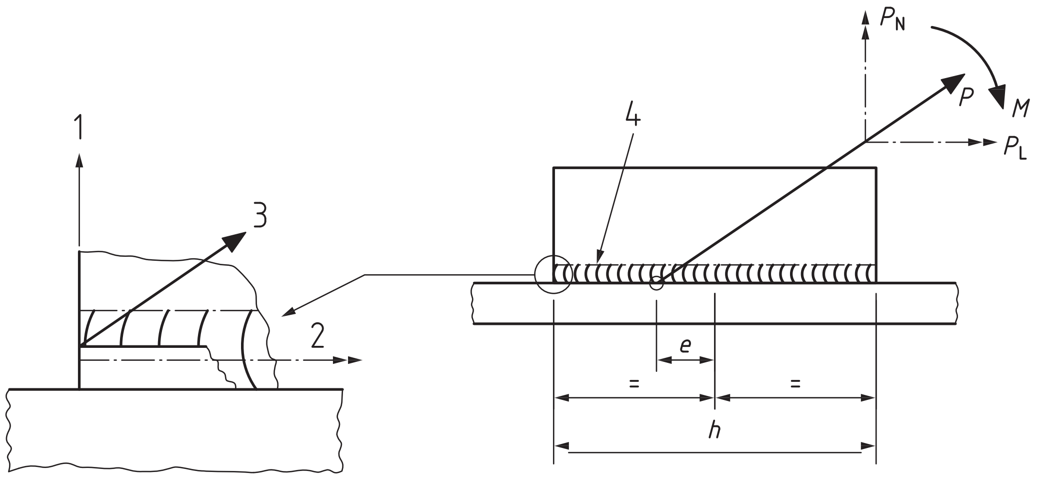
15.3 Stress range in fillet welds

In load-carrying partial penetration or fillet-welded joints, where cracking could occur in the weld throat, the relevant reference stress S_w is the resultant stress range in the weld metal. Provided that the direction of the principal stress does not change significantly in the course of a stress cycle (<20°), this should be taken, for each stress cycle, as the vector difference between the greatest and least vector sum of the applied normal and shear stresses, based upon the effective dimensions of the weld throat, as detailed in Figure 3. It should be assumed that none of the load is carried in bearing between parent materials. The τ/σ value that influences the choice of detail class (see Table 17) is the ratio longitudinal shear stress/engineering shear stress or $\Delta\tau_{//}$ / $\Delta\sigma_w$ in Figure 3. However, the longitudinal shear stress range $\Delta\tau_{//}$ may be neglected if $\Delta\tau_{//}/\Delta\sigma_w \leq 0.15$.

Where stress cycling is due to more than one load source but the directions of the stresses remain fixed, the resultant stress S_w is based on the maximum ranges of the loads (e.g. P_N and P_T in Figure 3) on the weld. However, if the direction of the stress vector on the weld throat changes by more than 20° during a cycle between two extreme load conditions, the resultant stress range is the magnitude of the vector difference between the two stress vectors. Where cycling is of such a complex nature that it is not clear which two load conditions would result in the greatest value of S_w the vector difference should be found for all pairs of extreme load conditions. Alternatively, it may be assumed that:

$$S_w = \left\{\left(\sigma_{\perp max} - \sigma_{\perp min}\right)^2 + \left(\tau_{\perp max} - \tau_{\perp min}\right)^2 + \left(\tau_{/\!/ max} - \tau_{/\!/ min}\right)^2\right\}^{0.5} \tag{1}$$

Figure 3 **Reference stress on weld throat**



Key

1 Transverse direct stress

$$\Delta\sigma_w = \frac{P_N}{wh} + \frac{P_N \, . \, e + M}{wh^2/6}$$

NOTE Also known as engineering shear stress range on weld throat

2 Longitudinal shear stress

$$\Delta\tau_{/\!/} = \frac{P_L}{wh}$$

3 Resultant stress range

$$S_w = \left(\Delta\sigma_W^2 + \Delta\tau_{/\!/}^2\right)^{0.5}$$

4 w = combined size of effective weld throats

15.4 Effective stress range for details in unwelded members in which the whole or part of the stress is compressive

In unwelded details where the stress range, allowing for dead load and residual stresses (due to fabrication), is entirely compressive, the effects of fatigue loading may be ignored. However, when the resultant stress range involves stress reversals through zero, the effective stress range to be used in the fatigue assessment should be obtained by adding 60% of the range from zero stress to maximum compressive stress to that part of the range from zero stress to maximum tensile stress.

15.5 Calculation of stresses

Stresses should be calculated using elastic theory and taking account of all axial, bending and shearing stresses occurring under the design loading. No redistribution of loads or stresses, (e.g. as might be allowed for checking static strength at ultimate limit state, including implicit allowance for redistribution in simplified elastic design rules, or for plastic design procedures), should be made.

15.6 Calculation of nominal stresses

15.6.1 General

The nominal stress, S_N, at any location is the summation of the membrane and bending stresses at that location. Examples of nominal stresses in parent material and a fillet weld throat are shown in Figure 2 and Figure 3 respectively.

15.6.2 Effects to be ignored

The effects of the following should not be included in nominal stress calculations:

a) residual stresses in welded details [see **15.6.3**f)];

b) stress concentrations due to:

   1) a weld detail itself;

   2) bolt, rivet or small drilled holes (but excluding types 1.3 and 1.4 in Table 1).

15.6.3 Effects to be included

All stresses arising from sources other than those specifically excluded in **15.6.2** should be included in the analysis. The following stresses should be included together with any others that influence the stress applied to the joint:

a) stress concentrations due to the overall joint geometry (see **15.6.4**);

b) eccentricities or misalignment occurring in a joint detail, except where otherwise indicated in this code;

c) shear lag, restrained torsion and distortion, transverse stresses and flange curvature;

d) stress distribution in wide plates;

e) stresses in triangulated skeletal structures due to load applications away from joints, member eccentricities at joints and rigidity of joints;

f) residual stresses (but only for non-welded details or stress relieved welds under, nominally, fully or partly compressive loading; see **15.4** and **16.3.6**); and

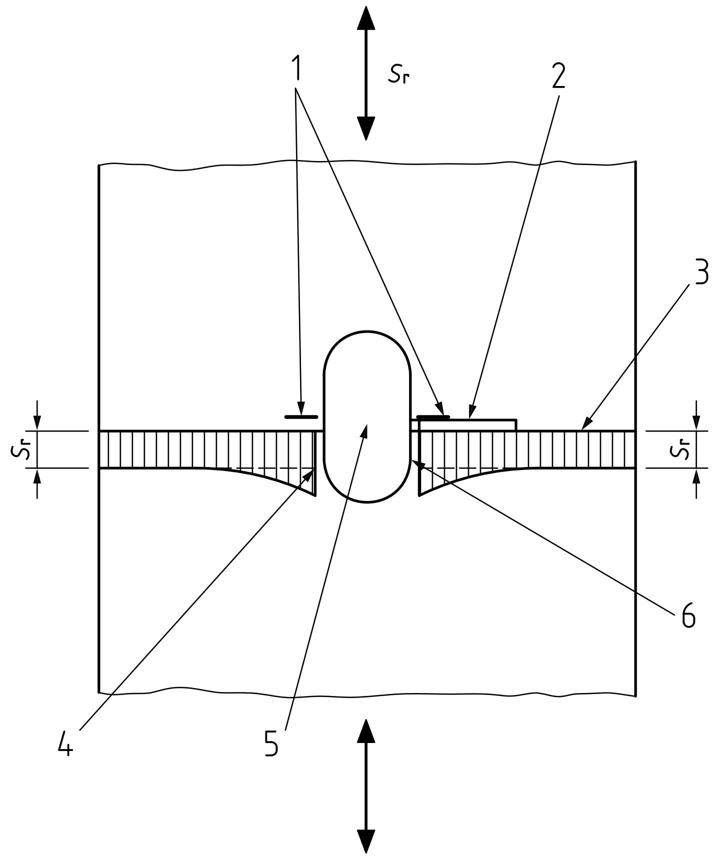
g) fabrication tolerances, except as covered in **B.7.2** for misalignment.

15.6.4 Geometrical stress concentrations

Unless otherwise indicated in Table 1 to Table 10 the stress concentrations inherent in the make-up of the structural detail concerned (e.g. arising from the general geometry of a welded joint and the weld shape) have been taken into account in the classification of the details. However, where there is an additional geometrical discontinuity, such as an aperture or a change of cross section (see Figure 4 and Figure 5), which is not a natural characteristic of the standard detail category itself, the resulting stress concentration relevant to fatigue design should be determined either by specific analysis or, where appropriate, by the use of predefined stress concentration factors [4, 5, 6], such as those given in Figure 6.

Figure 3 shows a typical example of a member which could be analysed using a stress concentration factor, such as given in Figure 6(a), and the resulting stress used in conjunction with the nominal stress-based assessment method.

Figure 4 **Typical example of stress concentrations due to geometrical discontinuity**

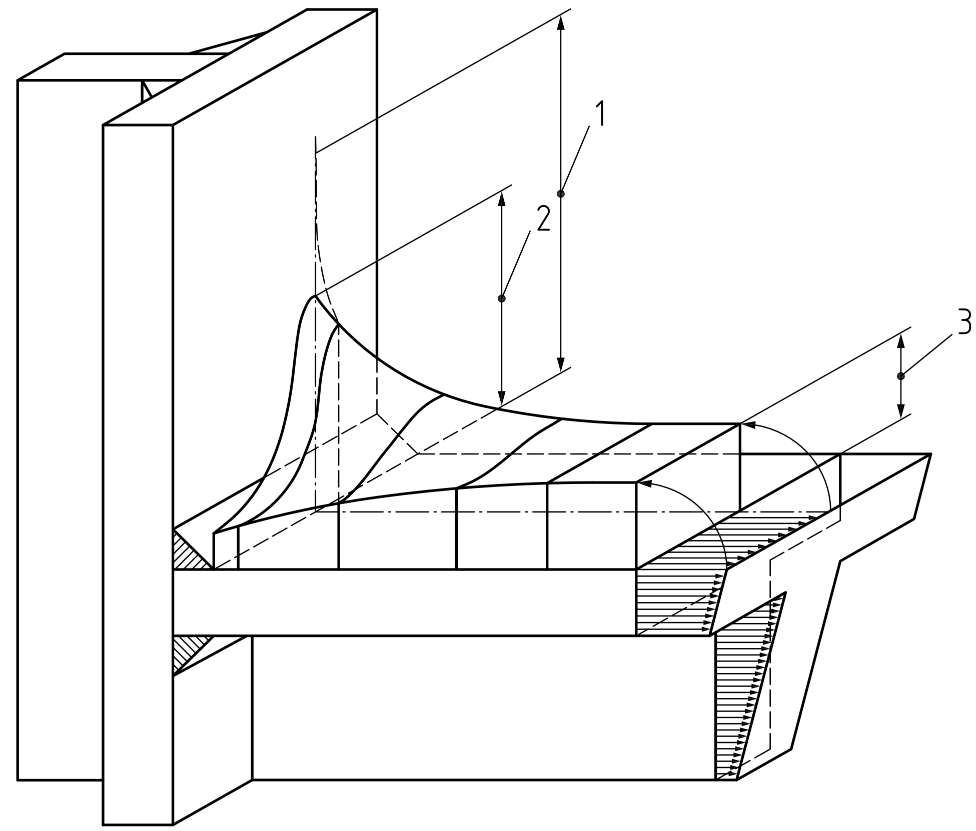


Key

1 Potential crack locations
2 Welded attachment
3 Typical stress distribution
4 The design stress is applied to the appropriate plain material classification
5 Manhole or re-entrant corner. The design stress for location 4 or 6 should be taken as the stress on the net section multiplied by the fatigue stress concentration factor (= $K_f.S_r$)
6 At the attachment the design stress is applied to the appropriate nominal stress joint classification

In contrast, Figure 5 shows an example of a welded joint which would require special analysis because the geometrical layout of the joint obviously creates a hard spot and hence a non-uniform stress distribution. In general form the joint is type 7.4.

Figure 5 **Typical example of stress concentration caused by a geometric hard spot**



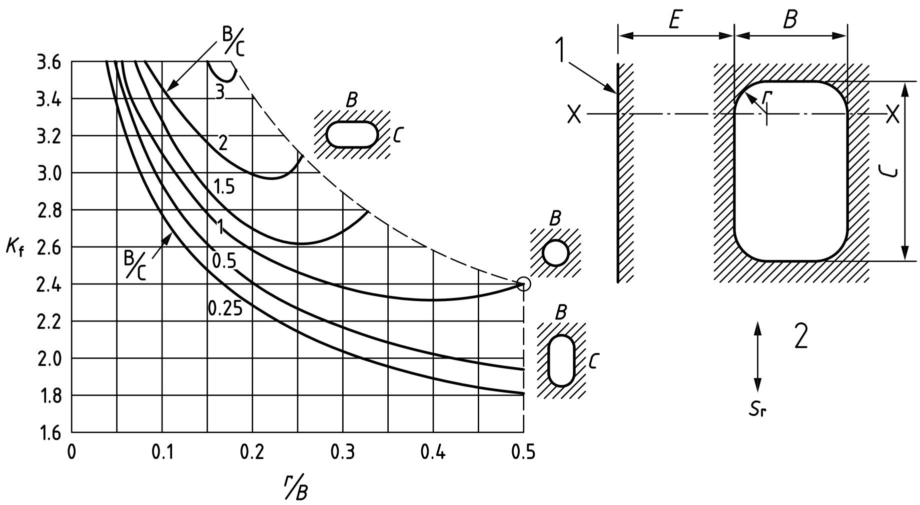
Key

1 S_z at weld toe taking into account SCF due to weld toe

2 S_z at weld toe (by extrapolation) = Design stress = (effectively) $K_f.S_z$ (nominal)

3 S_z (nominal)

Figure 6 **Fatigue stress concentration factors**



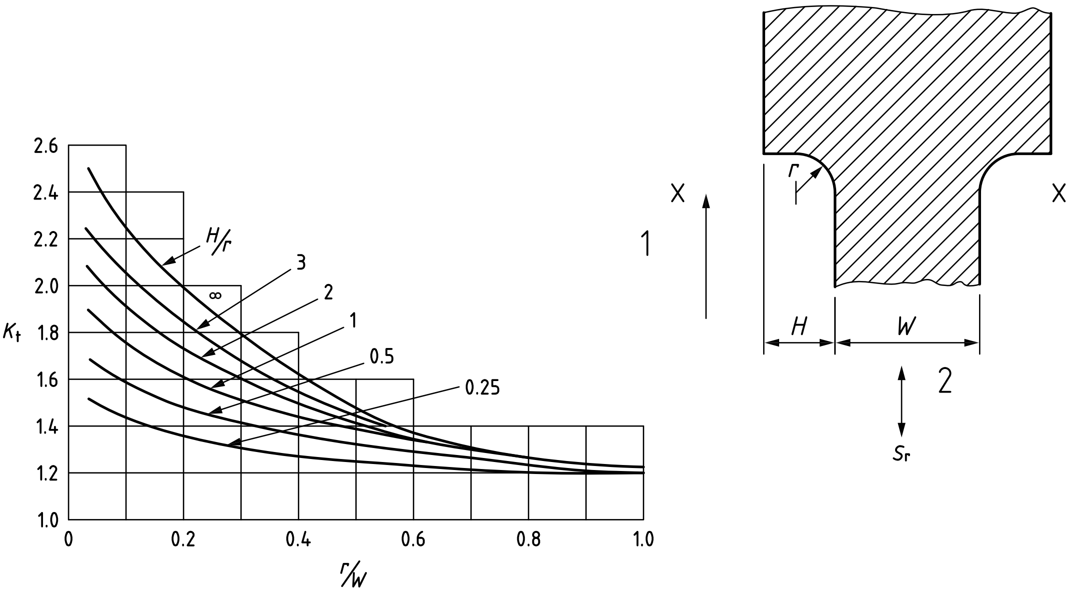
NOTE K_f values are conservative when E < B/2

a) Fatigue stress concentration factor for unreinforced apertures K_f (based on net stress at X-X)

Key

1 Free edge

2 Stress fluctuation



b) Fatigue stress concentration factor for re-entrant corners K_f (based on net stress at X-X)

Key

1 Length of straight ≥ 2*r*

2 Stress fluctuation

Figure B.10 illustrates a case where flexibility produces a non-uniform stress distribution. In all cases the corresponding applied stress for design purposes should be the structural stress at the weld toe. Other typical examples are tubular nodal joints (see Annex G) and the joints shown in Figure B.3, Figure B.5, Figure B.6, Figure B.8 and Figure B.9.

For such cases and those for which standard stress concentration factors are not applicable, the alternative hot-spot stress-based fatigue assessment method should be adopted with stresses determined in accordance with **15.7**.

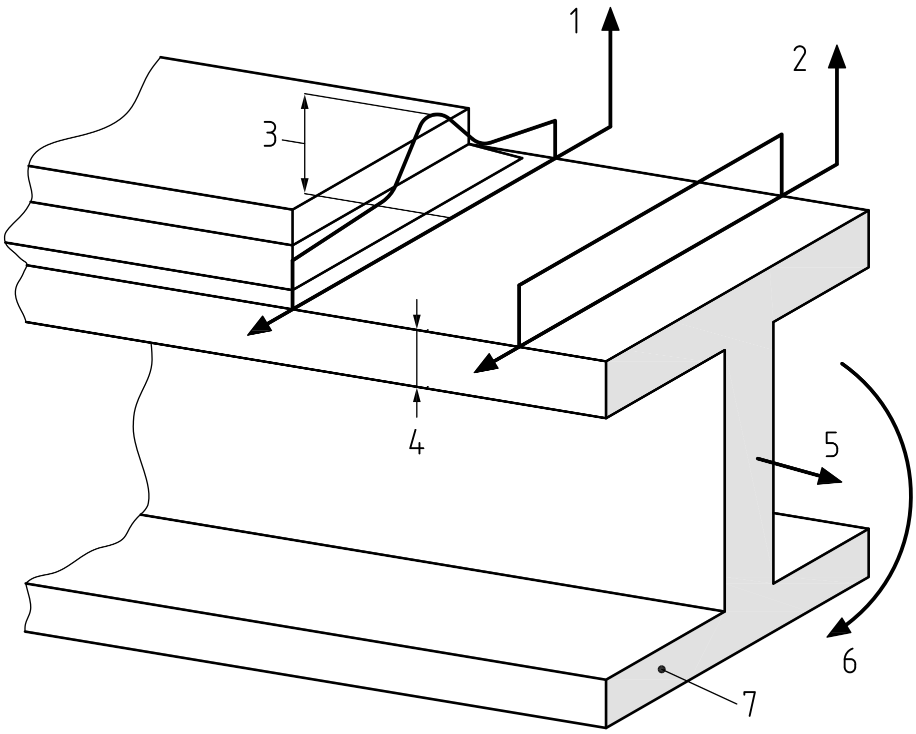
15.7 Calculation of hot-spot stresses

15.7.1 General

The hot-spot stress S_H is the structural stress at the weld toe. This might vary along a weld toe, as illustrated in Figure 7, in which case the largest value is generally used for design.

Three methods for calculating the hot-spot stress are described in Annex C while Annex G provides rules for the specific case of tubular joints.

Figure 7 **Comparison of nominal, structural and hot-spot stresses in a beam with a welded cover plate**



Key

1	Structural stress, defined over flange thickness, t	5	Force, P
2	Nominal stress = M/Z+F/A	6	Moment, M
3	Hot-spot stress	7	Section modulus, Z and area, A
4	Flange thickness, t		

15.7.2 Effects to be ignored

The effects of the following should not be included in hot-spot stress calculations:

a) residual stresses in welded details [for exceptions, see **15.7.3**f)];

b) stress concentration due to the weld toe.

15.7.3 Effects to be included

All stresses arising from sources other than those specifically excluded in **15.7.2** should be included in the analysis. The following stresses should be included together with any others that influence the stress applied to the joint:

a) stress concentrations due to the overall joint geometry;

b) eccentricities or misalignment occurring in a joint detail, except where otherwise indicated in this code;

c) shear lag, restrained torsion and distortion, transverse stresses and flange curvature;

d) stress distribution in wide plates;

e) stresses in triangulated skeletal structures due to load applications away from joints, member eccentricities at joints and rigidity of joints;

f) residual stresses in stress-relieved welded joints in which the whole or part of the applied stress component is compressive, see **15.4** and **16.3.6**;

g) fabrication tolerances.

15.8 Axial stresses in bolts

This sub-clause applies only to ISO metric threads (BS 3643-2, BS EN IS0 4014, BS EN ISO 4017, BS EN ISO 4762). It is applicable to hexagon bolts, screws and nuts (BS 3692) high strength friction grip bolts (BS 4395-2) and stainless steel bolts (BS EN ISO 3506). For black bolts (see BS 4190) it should only be used for fluctuating loads if the bolt head underside has a machined face. This sub-clause is not applicable to welded bolts.

The stress range should be calculated on the tensile stress area, BS 3692, Table 10, of the bolt and should include the effects of axial and bending loads, including any effect of prying. It should take into account the pre-load in the bolt and the compressibility and specified fit-up of the connected parts. Where the fatigue design of a joint relies on the pre-load of the bolts to limit the stress range in the bolts, the pre-load should be at least 1.5 times the calculated applied bolt load.

The compressibility effect in a bolted joint is determined from the relative stiffness of the bolt and the clamped components.

The portion of the applied load that is carried by the bolt can be determined from the relative stiffness of the bolt and clamped components.

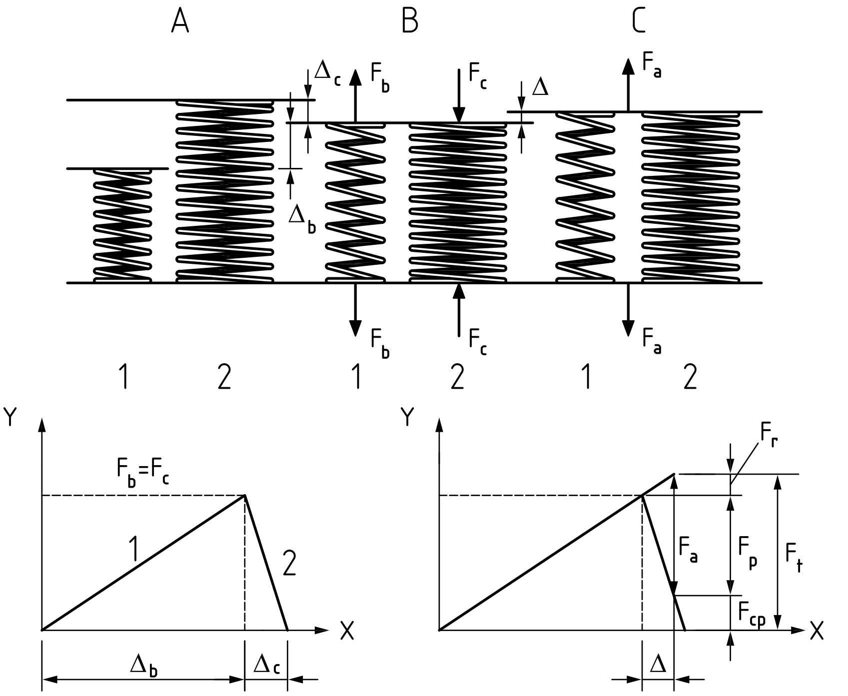
Figure 8 illustrates the relative stiffness effects on the fluctuating load on the bolt.

Where the applied load includes both a permanent tension component and a fluctuating tension component, the total applied load should be assumed to be a fluctuating load unless a full joint calculation is carried out which accounts for both the permanent and fluctuating components of applied load.

*NOTE 1 Permissible stresses in bolts are covered in **16.2.2**.*

NOTE 2 VDI 2230:2003 [7] gives detailed instructions on the calculation procedures for bolted joints.

Figure 8 **Relative stiffness effects on the fluctuating load in a bolt in a concentrically clamped and concentrically loaded bolted joint**



Key

Δ_b =	displacement of bolt due to bolt tightening	F_r =	fluctuating load in bolt
Δ_c =	displacement of joint interface due to bolt tightening	F_p =	proportion of F_a that relieves joint clamping effect
F_a =	load applied to joint	F_{cp} =	residual clamping load
F_b =	bolt pre-load	F_t =	residual bearing load under bolt head
F_c =	clamping action at joint interface	$\triangle$ =	displacement due to F_r
X =	Dispacement	Y =	Load
1	Bolt	2	Clamped plates

Bolt fluctuating load,

$$F_r = \frac{K_b}{(K_b+K_c)} \times F_a$$

where: K_b = bolt stiffness and K_c = summation of clamped components stiffness.

A	Initial state	B	Assembled state
C	Loaded state		

15.9 Derivation of stress spectrum

In situations in which the loading spectrum is not specified, e.g. in a relevant application standard, the expected spectrum should be derived.

In general the total loading on a structure is composed of several different loading events, each with different magnitudes, geometrical arrangements and frequencies of occurrence. To derive the design spectrum for any particular detail, a mix of loading events that is representative of the loading to be expected in a given time interval should be established. It should be applied to the relevant influence line(s) in order to obtain the pattern of stress fluctuations to which the detail is to be subjected in that time interval. Account should be taken of the possibility of two or more loading events occurring simultaneously, or following each other in particular orders, such that higher stress ranges might be caused. In selecting load cycles for inclusion in the spectrum low stress ranges, below 5 N/mm², and occasional high stress ranges that contribute less than 1% of the total damage may be ignored (but also see **16.1**).

This pattern should be broken down into a convenient spectrum of cycles, expressed in terms of stress ranges S_{ri} and numbers of applications n_i, by a suitable cycle counting method (see Annex H). The numbers of cycles counted should be combined with the appropriate total numbers of occurrences of the various loading events in the design life of the structure to compile the overall design spectrum.

In most computer programs used for the derivation of cycle counts the results are accumulated into a relatively small number of stress range intervals (sometimes called bins). In most instances about 40 intervals is sufficient.

16 Allowable fatigue stresses

16.1 Tensile stress limitations

The procedures given in this British Standard for deriving fatigue stresses should only be deemed to be valid if the calculated maximum fibre stress on the net area of a member, remote from geometric stress concentrations and excluding self-regulating stresses (such as residual or thermal stresses), does not exceed 60% of yield stress under normal operating conditions and 80% of yield stress under extreme loading conditions. In this context the expected number of cycles exceeding normal operating conditions in the anticipated life of the structure should be not greater than 100.

16.2 *S-N* curves

16.2.1 Plain material and welded, riveted or bolted joints

For each design class the relationship between the applied stress range, S_r, and the number of cycles to failure, N, under constant amplitude loading conditions is of the following form:

$$\log N = \log C_o - d\,SD - m \log S_r \qquad (2)$$

The values of these terms for joints in air are shown in Table 18, and the mean line relationships are plotted in Figure 9. They are applicable to all steels covered by this British Standard, using the equation:

$$\log C_d = \log C_o - d\,SD \qquad (3)$$

Then Equation 2 can be written as:

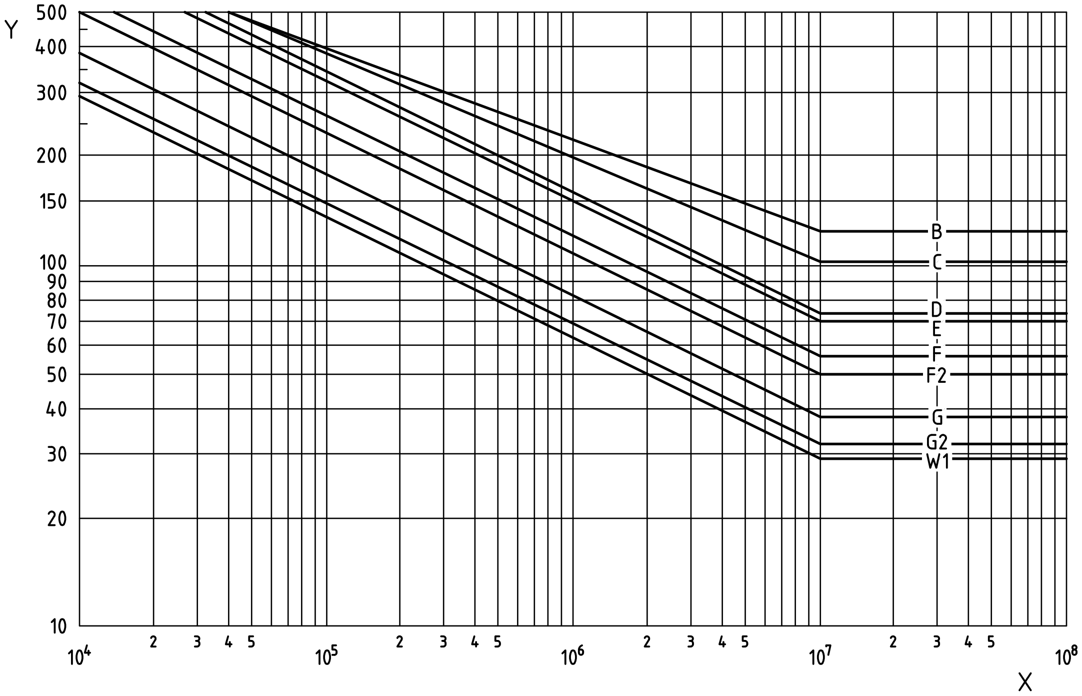
$$S_r^m N = C_d \qquad (4)$$

Thus, from Equation 2 to Equation 4, the required basic *S-N* curve can be derived for any desired value of *d*. The nominal probability of failure corresponding to various possible values of d, based upon an assumed normal distribution, is shown in Table 19. The standard basic S_r-*N* curves should be taken to represent two standard deviations below the mean lines i.e. $d = 2$. They are shown in graphical form in Figure 10, and the corresponding values of C_2 are included in Table 18. If a curve for a welded joint crosses the class B curve for plain steel, the class B curve governs [see Figure 10c) and **16.5**].

All the basic S_r-*N* curves are applicable, for the relevant value of *d*, to details in air or other non-corrosive environments, or for details with adequate corrosion protection in the form of paint or other coating.

For design purposes, however, the curves might have to be modified in order to allow for the factors given in **16.3** and **16.4**.

Figure 9 **Mean S_r-*N* curves**



These curves should not be used for calculation purposes (see Table 18).

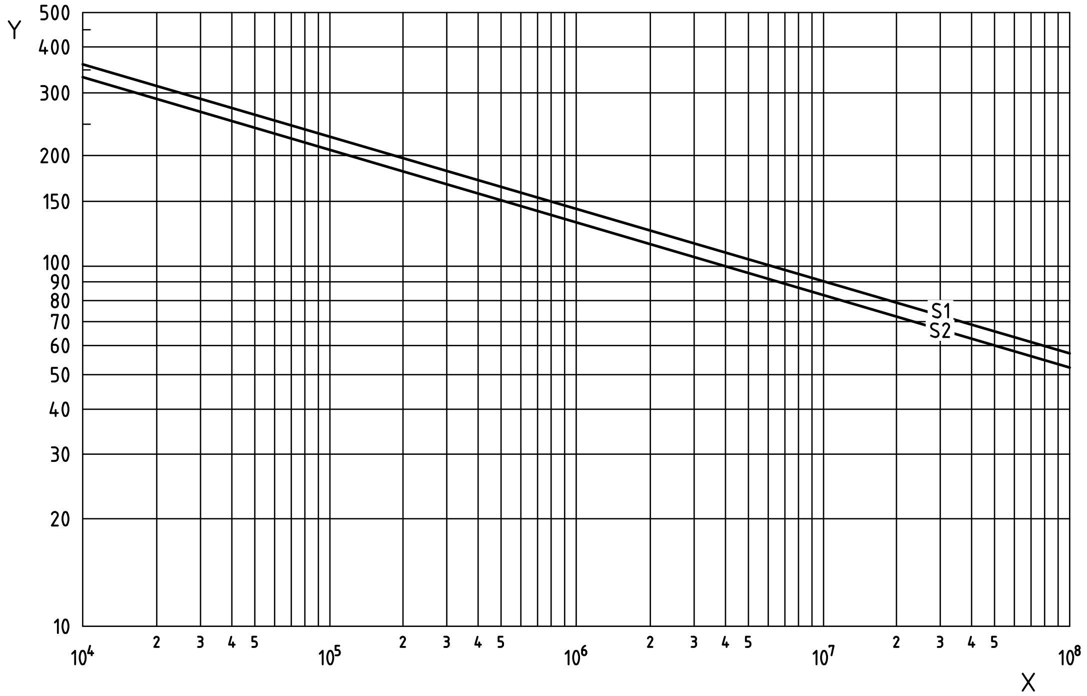
Key

X Endurance *N*, cycles

Y Stress range S_r, N/mm²

a) Mean S_r-*N* curves for direct stress failure

Figure 9 **Mean S_r-N curves**

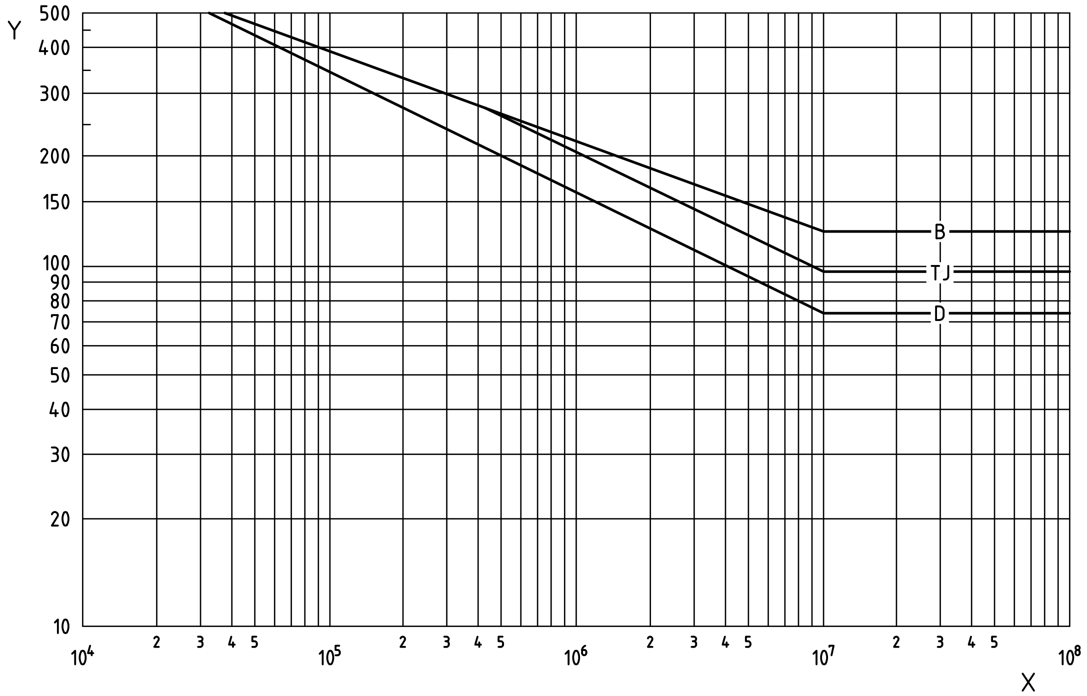


These curves should not be used for calculation purposes (see Table 18).

Key

X Endurance *N*, cycles Y Shear stress range, S_r, N/mm²

b) Mean S_r-N curves for shear stress failure



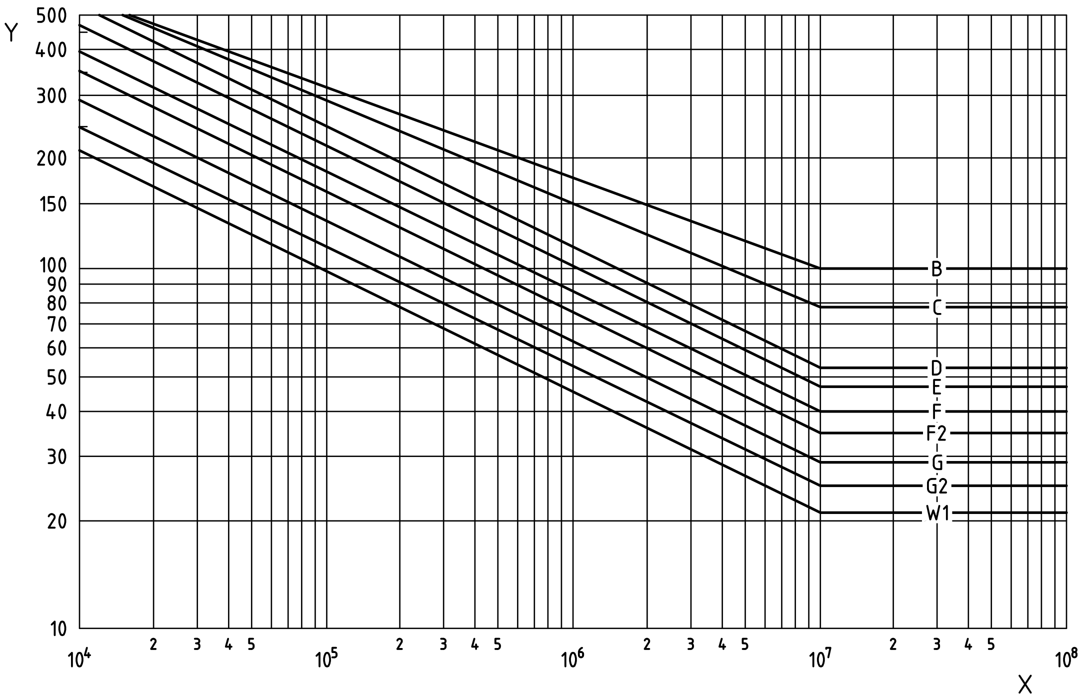
These curves should not be used for calculation purposes (see Table 18).

Key

X Endurance *N*, cycles Y Hot-spot stress range, S_{Hr}, N/mm²

c) Mean S_r-N curves for use with hot-spot stress

Figure 10 **Standard basic design S_r-N curves**



These curves should not be used for calculation purposes (see Table 18).

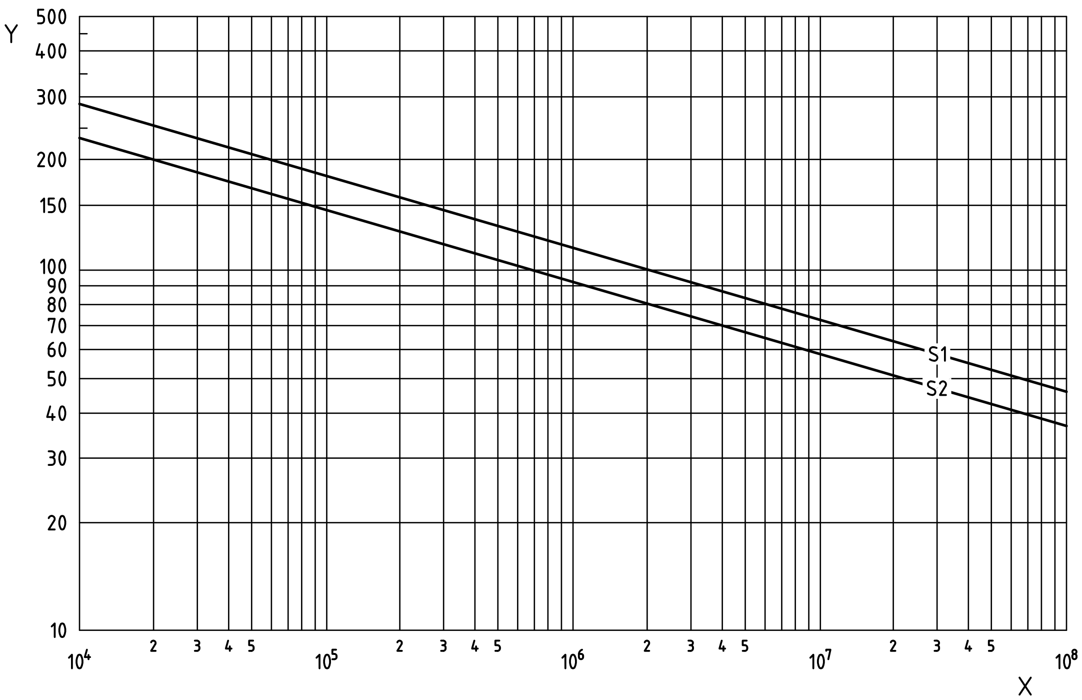
Adjustments should be made, where appropriate in accordance with **16.3** and **16.4**.

Key

X Endurance *N*, cycles

Y Stress range S_r, N/mm²

a) Standard basic design S_r-N curves (mean minus two standard deviations of log N) for direct stress failure



These curves should not be used for calculation purposes (see Table 18).

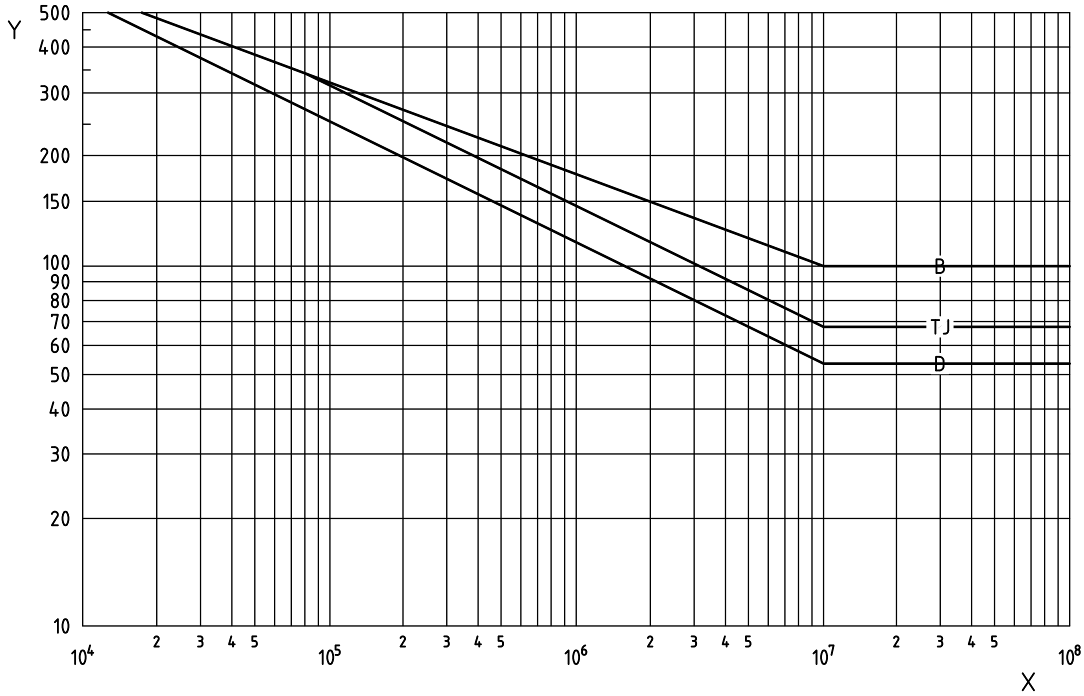
Key

X Endurance *N*, cycles

Y Shear stress range S_r, N/mm²

b) Standard basic design S_r -N curves (mean minus two standard deviations of log N) for shear stress failure

Figure 10 **Standard basic design S_r-N curves**



These curves should not be used for calculation purposes (see Table 18).

Adjustments should be made, where appropriate in accordance with **16.3** and **16.4**.

Key

X Endurance *N*, cycles

Y Hot-spot stress range, S_{Hr}, N/mm²

c) Standard basic design S_r-N curves (mean minus two standard deviations of log N) for use with hot-spot stress

16.2.2 Axially loaded bolts

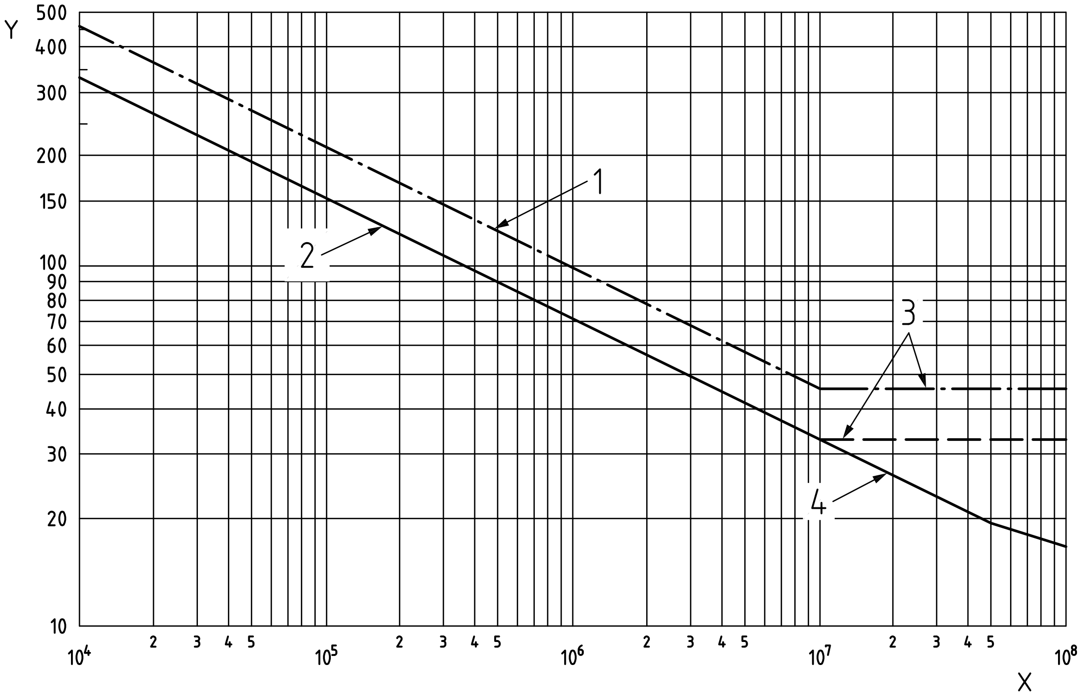
The mean and design (mean – 2*SD*) *S-N* curves for axially loaded bolts up to 25 mm diameter (class X) are shown in Figure 11. These are expressed in terms of the stress range on the tensile stress area, calculated in accordance with **15.8**. For bolts with diameter >25 mm, the fatigue strength defined by the class X curve should be reduced in accordance with **16.3.2**. In addition, the following conditions apply.

a) Class X is directly applicable to bolts with cut or ground threads and to rolled threads that have then been heat treated.

b) If thread rolling is carried out after any heat treatment the fatigue strength defined by class X may be increased by 25%.

c) If the bolt is electro plated the fatigue strength defined by class X should be reduced by 20%.

Figure 11 ***S_r-N* curves for bolts with threads under direct loading (class X)**



These curves should not be used for calculation purposes (see Table 18).

Key

X	Endurance *N*, cycles	Y	Stress range S_r, N/mm^2
1	Class X, mean	3	Constant amplitude
2	Class X, design	4	Variable amplitude

Table 18 **Details of basic *S-N* curves**

Class	C_o	$Log_{10}C_o$	*m*	Standard deviation of $log_{10}N$, *SD*	C_2	S_{oc} (N=10^7 cycles) N/mm²	S_{ov} (N=5×10^7 cycles) N/mm²
B	2.343×10^{15}	**15.3697**	**4.0**	**0.1821**	1.01×10^{15}	100	67
C	1.082×10^{14}	A1⟩ **14.0344** ⟨A1	**3.5**	**0.2041**	4.23×10^{13}	78	A1⟩ 49 ⟨A1
D	3.988×10^{12}	A1⟩ **12.6008** ⟨A1	**3.0**	**0.2095**	1.52×10^{12}	53	31
E	3.289×10^{12}	A1⟩ **12.5171** ⟨A1	**3.0**	**0.2509**	1.04×10^{12}	47	A1⟩ 27 ⟨A1
F	1.726×10^{12}	A1⟩ **12.2371** ⟨A1	**3.0**	**0.2183**	A1⟩ 6.32×10^{11} ⟨A1	40	23
F2	1.231×10^{12}	A1⟩ **12.0902** ⟨A1	**3.0**	**0.2279**	A1⟩ 4.31×10^{11} ⟨A1	35	21
G	5.656×10^{11}	A1⟩ **11.7526** ⟨A1	**3.0**	**0.1793**	A1⟩ 2.48×10^{11} ⟨A1	29	17
G2	3.907×10^{11}	**11.5918**	**3.0**	**0.1952**	A1⟩ 1.59×10^{11} ⟨A1	25	A1⟩ 15 ⟨A1
W1	2.500×10^{11}	**11.3979**	**3.0**	**0.2140**	9.33×10^{10}	21	12
X	9.298×10^{11}	**11.9684**	**3.0**	**0.2134**	A1⟩ 3.48×10^{11} ⟨A1	33	19
S_1	5.902×10^{16}	**16.7710**	**5.0**	**0.2350**	2.00×10^{16}	46 (at 10^8 cycles) B)	46 (at 10^8 cycles) B)
S_2	3.949×10^{16}	**16.5965**	**5.0**	**0.3900**	6.55×10^{15}	37 (at 10^8 cycles) B)	37 (at 10^8 cycles) B)
TJ	8.750×10^{12}	A1⟩ **12.9420** ⟨A1	**3.0**	**0.2330**	A1⟩ 2.99×10^{12} ⟨A1	67 A)	39 A)

For example the TJ curve is $\log N = 12.942 - 0.233d - 3\log S_r$

A) Idealized hot-spot stress.

B) Curve extrapolated to N=10^8 cycles without slope change.

NOTE The figures in ***bold*** *are to be taken as exact and definitive; figures that are not bold are calculated exactly from the exact definitive figures and then rounded to the displayed number of significant figures.*

Table 19 **Nominal probability factors**

Nominal probability of failure %	***d***
50	0 A)
31	0.5
16	1.0
2.3	2.0 B)
0.14	3.0

A) Mean-line curve.

B) The standard design curve.

16.3 Modifications to basic *S-N* curves

16.3.1 General

In order to derive the design S_r-N curves, the basic S_r-N curves should be modified in accordance with Clause **9** and, as appropriate, to allow for the factors given in **16.3.2** to **16.3.5**.

16.3.2 Effect of material thickness and bending

In the case of potential fatigue failure from the toe or end of a weld or from a thread root in a bolt, the fatigue strength is to some extent dependent on material thickness, (strength decreases with increasing thickness). The basic S_r-N curves relate to the following thicknesses and bolt diameters:

a) tubular nodal joints (class TJ): up to t_B = 16 mm;

b) non-nodal joints (classes B to G2): up to t_B = 25 mm;

c) bolts (class X): up to t_B = 25 mm diameter.

Similarly, in the case of potential fatigue failure from either the toe or end of a weld through the thickness of the loaded member or from the root through the throat of a transversely-loaded fillet or butt weld, the fatigue strength depends on the degree of through-thickness bending. The fatigue strength increases with increasing bending component for a decreasing stress range gradient through the thickness. However, the design S_r-N curves relate to applied loading conditions that produce predominantly membrane stresses. The potentially detrimental effect of increased thickness but beneficial effect from applied bending are combined by the application of the correction factor k_{tb} on stress ranges obtained from the relevant S_r-N curve, such that:

$$S = k_{tb} S_B \tag{5}$$

where,

for t > 25 mm:

$$k_{tb} = (t_B / t_{eff})^b \left[1 + 0.18 \Omega^{1.4}\right] \tag{6}$$

for 4 mm ≤ t ≤ 25 mm:

[A1⟩

$$k_{tb} = \left\{1 + \Omega\left[\left(\frac{t_B}{t}\right)^b - 1\right]\right\} \times \left(1 + 0.18\Omega^{1.4}\right) \tag{7}$$

⟨A1]

and

b is 0.25 or 0.2 as indicated in Table 4 to Table 10 for as-welded joints or **16.3.5** for improved weld toes or 0.25 for bolts;

t_{eff} is the greater of t_B or the actual thickness t in mm of the member or bolt diameter under consideration. However, if in a nominal stress-based assessment the weld detail can be described in terms of the dimensions L and t (see Figure 1) and $L/t \le 2$, then t_{eff} is the greater of $0.5L$ or t_B.

For hot-spot stress-based assessments (see **15.7**), $t_{eff} = t$.

For cases of increasing through-thickness stress range gradient, Ω should be assumed to be zero.

If the bending stress component is due to misalignment, Ω should be assumed to be zero.

For assessments of tubular node joints, Ω should be assumed to be zero as the class already includes allowance for bending.

NOTE The increase in fatigue strength for thicknesses less than 25 mm resulting from the use of Equation 7 only arises in the presence of bending.

16.3.3 Effect of temperature

There are no restrictions on the use of the fatigue design curves for components or structures that operate at sub-zero temperatures, provided that the steel through which a fatigue crack might propagate is shown to be sufficiently tough to ensure that fracture does not initiate from a fatigue crack.

[A1⟩ With regard to elevated temperature, the design curves are directly applicable for temperatures up to 150 °C. For higher temperature T, but below the creep range for the steel concerned, the fatigue strength obtained from the S-N curve should be multiplied by E_T/E_B, where E_T is the elastic modulus of the steel in N/mm² at temperature T and E_B is the reference value for $T \le 150$ °C, assumed to be 2.09×10^5 N/mm² for structural steels or 2.00×10^5 N/mm² for austenitic steels.

NOTE The following modification of Equation 4 to include the allowances for thickness, bending and temperature might assist in calculations:

$$S_r^m N = C_d \left(k_{tb} \times \frac{E_T}{E_B} \right)^m$$

⟨A1]

16.3.4 Effect of sea water

The fatigue strength of steel welded joints might be reduced in the presence of sea water, even if cathodic protection is applied. Corrosion protection in the form of paint or other coatings is effective; this should remain intact throughout the service life of the joint.

Cathodic protection producing voltages in the range –850 mV to –1 100 mV offers some resistance to corrosion fatigue, but only in the high-cycle regime (transition at constant amplitude endurance of [A1⟩ between 10^5 and 2×10^6 ⟨A1] cycles, depending on the class). In particular, a penalty of 2.5 on life, or 2.0 in the case of tubular joints, is applied before the transition, after which the penalty reduces until in-air performance is achieved for $N \ge 10^7$ cycles, as illustrated in Figure 13. The resulting design S_r-N curves are detailed in Table 20.

Table 20 **Details of design *S-N* curves for steel in sea water**

Class	In sea water with cathodic protection (–850 to –1 100 mV)											Unprotected joints freely corroding in sea water	
	$S_r \geq S_{rt}$			$S_{rt} > S_r \geq S_{oc}$ ($N \leq 10^7$ cycles)		$S_{oc} \geq S_r > S_{ov}$ ($10^7 \leq N < 5 \times 10^7$ cycles)			Variable amplitude loading $S_r \leq S_{ov}$, ($N \geq 5 \times 10^7$ cycles)			Any endurance	
	S_{rt} N/mm²	C_2	m	C_2	m	S_{oc} N/mm²	C_2	m	S_{ov} N/mm²	C_2	m	C_2	m
B	A1⟩ 251 ⟨A1	4.05×10^{14}	**4.0**	1.02×10^{17}	**5.0**	100	A1⟩ 1.01×10^{15} ⟨A1	**4.0**	67	A1⟩ 6.79×10^{16} ⟨A1	**5.0**	1.41×10^{13}	**3.5**
C	144	1.69×10^{13}	**3.5**	A1⟩ 2.92×10^{16} ⟨A1	**5.0**	78	4.23×10^{13}	**3.5**	49	A1⟩ 1.47×10^{16} ⟨A1	**5.0**		
D	A1⟩ 84 ⟨A1	6.08×10^{11}	**3.0**	A1⟩ 4.33×10^{15} ⟨A1	**5.0**	53	1.52×10^{12}	**3.0**	31	1.48×10^{15}	**5.0**	5.07×10^{11}	**3.0**
E	A1⟩ 74 ⟨A1	4.14×10^{11}	**3.0**	2.28×10^{15}	**5.0**	47	1.04×10^{12}	**3.0**	27	7.81×10^{14}	**5.0**	A1⟩ 3.45×10^{11} ⟨A1	**3.0**
F	63	2.53×10^{11}	**3.0**	A1⟩ 1.00×10^{15} ⟨A1	**5.0**	40	A1⟩ 6.32×10^{11} ⟨A1	**3.0**	23	3.43×10^{14}	**5.0**	A1⟩ 2.11×10^{11} ⟨A1	**3.0**
F2	A1⟩ 55 ⟨A1	1.72×10^{11}	**3.0**	A1⟩ 5.30×10^{14} ⟨A1	**5.0**	35	A1⟩ 4.31×10^{11} ⟨A1	**3.0**	20	1.81×10^{14}	**5.0**	A1⟩ 1.44×10^{11} ⟨A1	**3.0**
G	46	A1⟩ 9.91×10^{10} ⟨A1	**3.0**	A1⟩ 2.11×10^{14} ⟨A1	**5.0**	29	A1⟩ 2.48×10^{11} ⟨A1	**3.0**	17	7.20×10^{13}	**5.0**	A1⟩ 8.26×10^{10} ⟨A1	**3.0**
G2	A1⟩ 40 ⟨A1	A1⟩ 6.36×10^{10} ⟨A1	**3.0**	A1⟩ 1.01×10^{14} ⟨A1	**5.0**	25	A1⟩ 1.59×10^{11} ⟨A1	**3.0**	14	A1⟩ 3.44×10^{13} ⟨A1	**5.0**	A1⟩ 5.30×10^{10} ⟨A1	**3.0**
W1	33	3.73×10^{10}	**3.0**	A1⟩ 4.14×10^{13} ⟨A1	**5.0**	21	9.33×10^{10}	**3.0**	12	A1⟩ 1.41×10^{13} ⟨A1	**5.0**	3.11×10^{10}	**3.0**
X	52	A1⟩ 1.39×10^{11} ⟨A1	**3.0**	A1⟩ 3.71×10^{14} ⟨A1	**5.0**	33	A1⟩ 3.48×10^{11} ⟨A1	**3.0**	19	A1⟩ 1.27×10^{14} ⟨A1	**5.0**	A1⟩ 1.16×10^{11} ⟨A1	**3.0**
TJ	A1⟩ 95 ⟨A1	1.50×10^{12}	**3.0**	A1⟩ 1.34×10^{16} ⟨A1	**5.0**	67	A1⟩ 2.99×10^{12} ⟨A1	**3.0**	39	A1⟩ 4.58×10^{15} ⟨A1	**5.0**	9.97×10^{11}	**3.0**

NOTE 1 Class W1 is only relevant if sea water can gain access to the fully-embedded root of the fillet welded joint

*NOTE 2 These design curves have been produced by applying the fatigue life reduction factors quoted in **16.3.4** to the design curves for operation in air (Table 18). The database that provided these factors was insufficient to establish reliable estimates of the standard deviation of log N. However, it is considered to be reasonable to assume that the values quoted in Table 18 are equally applicable for sea water corrosion-fatigue conditions.*

*NOTE 3 The figures in **bold** are to be taken as exact and definitive; figures in normal weight are calculated exactly from the exact definitive numbers in this table and Table 18 and then rounded to the displayed number of significant figures.*

For unprotected joints exposed to sea water the basic S_r-N curves should be reduced by a factor of 3 on life. In the case of class B for plain steel, the part of the component or structure concerned is first down-graded to class C to allow for surface roughening or pitting from corrosion (see Table 1, Table 3 and Table 5). For all classes the correction relating to the number of small stress cycles (see **16.4**) is not applicable. Details of the resulting design S_r-N curves are included in Table 19 and an example is included in Figure 14.

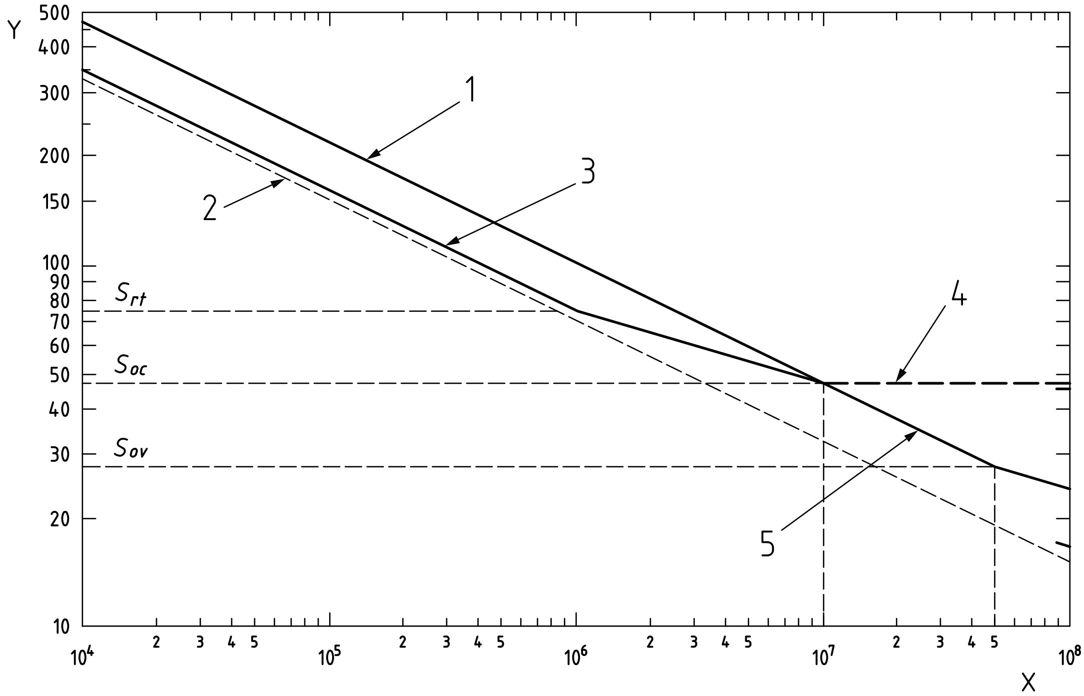
NOTE The recommendations in this sub-clause are based largely on studies of corrosion fatigue under the conditions experienced by offshore structures or vessels operating in the North Sea (wave loading frequency in the range 0.15 to 0.5Hz and sea water temperature in the range 5 to 20° C). The detrimental effect of sea water, with or without cathodic protection, could be even greater at lower frequencies and higher temperatures.

For high strength steels (σ_Y > 700 N/mm²) these penalties might not be adequate because of their greater susceptibility to cracking from hydrogen embrittlement and the use of such materials under corrosion fatigue conditions should be approached with caution.

No guidance is provided on the corrosion fatigue strength of austenitic or duplex stainless steels.

No guidance is currently available for considering corrosion fatigue failure by shear (crack opening modes II or III).

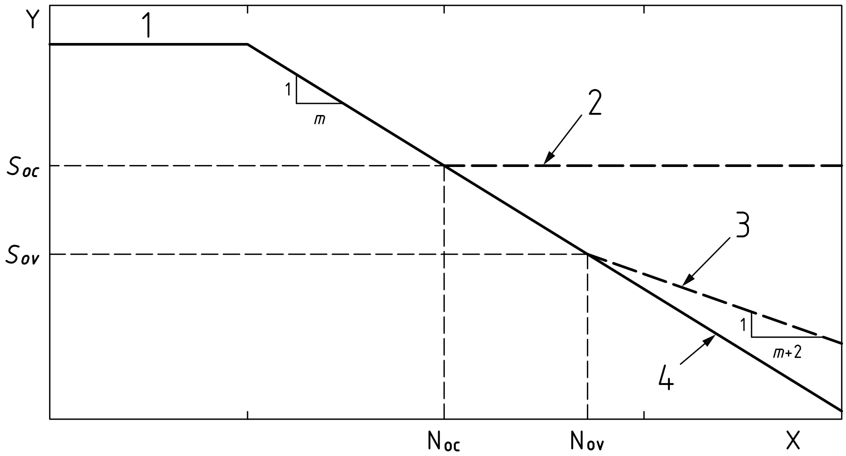
Figure 12 **Modifications made to S_r-N curves for welded joints in sea water**



Key

X	Endurance N, cycles	Y	Stress range S_r, N/mm²
1	In air	4	Constant amplitude
2	Freely corroding in sea water	5	Variable amplitude
3	In sea water, with cathodic protection		

Figure 13 **Typical S_r-N relationship**



NOTE Only the portion of this figure shown as a full line is based on experimental data.

Key

X	Endurance *N*, cycles – log scale	Y	Stress range S_r, log scale
1	Static limitations	3	Effective curve used for calculations involving variable amplitude loading, equivalent to changing slope of S_r-N curve above N_{ov}
2	Constant amplitude loading in clean air or, with details protected against corrosion, in sea water	4	Unprotected details in sea water.

16.3.5 Effect of weld toe improvement

For welded joints involving potential fatigue cracking from the weld toe, the fatigue strength can be increased by the application of a weld toe improvement technique. Techniques based on improvement of the weld toe geometry to reduce the local stress concentration by grinding or re-melting or on the introduction of beneficial compressive residual stress by peening are described in Annex F. Provided that the chosen technique is applied in accordance with Annex F and account is taken of potential alternative sites for fatigue crack initiation, the improvement in fatigue strength specified in Annex F may be assumed. In general this amounts to a 50% increase in fatigue strength at $N = 10^7$ cycles and rotation of the *S-N* curve to a slope of $m = 3.5$, but the benefit might be less under some loading conditions in the case of the peening techniques (e.g. after tensile pre-loading or for operation under high tensile mean stress). [A1⟩ Details of the resulting design *S-N* curves are given in Table 21. ⟨A1] Higher improvements might be achievable but these should be confirmed by special fatigue testing in accordance with Annex E.

In the assessment of improved weld toe with respect to potential fatigue failure from the toe the corrections for thickness and bending, Equation 6 and Equation 7, should be applied assuming $b = 0.20$ for dressed weld toes or 0.25 for peened weld toes.

NOTE 1 The benefit of dressing may be claimed only for welded joints which are adequately protected from corrosion, as the presence of a corrosive environment can cause pitting in the dressed region.

NOTE 2 In the case of partial penetration and fillet welds, where failure can occur from the weld root, improvement of the weld toe cannot be relied upon to give an increase in fatigue strength.

Table 21 **Design *S-N* curves for weld toe improved welded joints**

Class	Design data															
	As-welded			Toe dressed (see F.3.3 for limitations)			Toe peened (see F.4.3 for limitations)									
							$R < 0$			$0 \le R \le 0.28$, $S_{max} \le 80\%$ yield			$0.28 < R \le 0.4$, $S_{max} \le 80\%$ yield			$R > 0.4$ or $S_{max} > 80\%$ yield
	S_{oc}	C_2	m	S_{oc}	C_2	m	S_{oc}	C_2 A)	m	S_{oc}	C_2	m	S_{oc}	C_2	m	No benefit; treat as as-welded
D	53	1.52×10^{12}	**3**	80	4.59×10^{13}	**3.5**	80	$f \times 4.59 \times 10^{13}$	**3.5**	80	4.59×10^{13}	**3.5**	61	2.26×10^{12}	**3**	
E	47	1.04×10^{12}	**3**	70	2.93×10^{13}	**3.5**	70	$f \times 2.93 \times 10^{13}$	**3.5**	70	2.93×10^{13}	**3.5**	54	1.58×10^{12}	**3**	
F	40	6.33×10^{11}	**3**	60	1.65×10^{13}	**3.5**	60	$f \times 1.65 \times 10^{13}$	**3.5**	60	1.65×10^{13}	**3.5**	46	9.73×10^{11}	**3**	
F2	35	4.32×10^{11}	**3**	53	1.05×10^{13}	**3.5**	53	$f \times 1.05 \times 10^{13}$	**3.5**	53	1.05×10^{13}	**3.5**	40	6.52×10^{11}	**3**	
G	29	2.50×10^{11}	**3**	44	5.53×10^{12}	**3.5**	44	$f \times 5.53 \times 10^{12}$	**3.5**	44	5.53×10^{12}	**3.5**	33	3.71×10^{11}	**3**	
G2	25	1.48×10^{11}	**3**	38	3.30×10^{12}	**3.5**	38	$f \times 3.30 \times 10^{12}$	**3.5**	38	3.30×10^{12}	**3.5**	29	2.38×10^{11}	**3**	

*NOTE The figures in **bold** are to be taken as exact and definitive; figures in normal weight are calculated exactly from the exact definitive numbers in this table and Table 18 and then rounded to the displayed number of significant figures.*

A) Treated as stress relieved, in line with **16.3.6**, i.e. effective stress range assumed to be equal to tensile stress component plus 60% of compressive stress component. Corresponding C_2 values can be expressed as a function of R, as follows:
For $R < 0$, S_{max} = tensile component of applied stress while S_{min} = compressive component. Therefore, effective stress range $S_{r\ eff} = S_{max} + 0.6S_{min}$. But, since $R = S_{min}/S_{max}$, this can also be written $S_{r\ eff} = (1 - 0.6R)S_{max}$, where R is negative. Similarly, it can be expressed in terms of the applied stress range S_r, since $S_r = (1 - R)S_{max}$, giving $S_{reff} = (1-0.6R) \times \frac{S_r}{(1-R)} = \frac{(1-0.6R)}{(1-R)} \times S_r$, where again R is negative.

Thus, for negative R values, the design curve for the improved joint $S_{r\ eff}^{3.5} \times N = C_2$ can be written $S_r^{3.5} \times N = fC_2$, where $f = \left(\frac{1-0.6R}{1-R}\right)^{-3.5}$, or $f = \left(\frac{1-R}{1-0.6R}\right)^{3.5}$.

16.3.6 Effect of stress relief for welded details

If the applied stress range is fully tensile, stress relief has no significant effect on fatigue strength. However, if it can be demonstrated that a compressive component of the combined applied and residual stress exists, and can be quantified, it may be assumed that the relevant value of S_r is the sum of the tensile component and 60% of the compressive component.

*NOTE 1 As this unlikely to be satisfied very often this benefit is generally ignored (see **15.6.3**).*

In assessing the potential influence of stress relief, account should be taken of the following:

a) the extent to which stress relief actually reduces the residual stress in large complex joints is largely unknown; and

b) if joints are locally stress-relieved and then welded into a complete structure long range tensile residual stresses still exist.

NOTE 2 Such long range stresses can be as high, and have the same effect, as the residual stresses induced by welding.

16.4 Treatment of low stress cycles

Under fluctuating constant amplitude stresses there is a stress range, which varies both with the environment and with the size of any initial flaws, below which an indefinitely large number of cycles can be sustained. This is referred to in this British Standard as the non-propagating stress range but a commonly used alternative is CAFL. In air and sea water with adequate protection against corrosion, it is assumed that this non-propagating stress range, S_{oc}, (given the suffix “oc” to denote constant amplitude loading), is the stress corresponding to $N = 10^7$ cycles to failure (or 10^8 cycles in the case of classes S_1 and S_2) as obtained from the design *S-N* curve (relevant values of S_{oc} for the standard basic curves are shown in Table 18 and Table 20). In all cases S_{oc} is the stress derived after allowing for any required modifications to the *S-N* curves (see **16.3**). For unprotected joints in sea water it should be assumed that $S_{oc} = 0$.

When the applied fluctuating stress has varying amplitude, so that some of the stress ranges are greater than and some less than S_{oc}, the larger stresses cause growth of the flaw, thereby reducing the value of the non-propagating stress range below S_{oc}. Therefore, as time goes on, an increasing number of stress ranges below S_{oc} can themselves contribute to crack growth. The final result is an earlier fatigue failure than would be calculated by assuming that all stress ranges below S_{oc} are ineffective. Therefore, when considering variable amplitude loading account should be taken of the fatigue damage due to stress ranges below S_{oc}, but exceeding 5 N/mm² (see **15.9**), using a modified curve. This is produced by assuming that the *S-N* curve is extrapolated beyond $N = 10^7$ cycles until it has a change of inverse slope from m to $(m + 2)$ at N_{ov} (see Figure 14), where $N_{ov} = 5 \times 10^7$ cycles for classes B to W1 and X or 10^8 cycles for classes S_1 and S_2. The stress range at this slope transition point is S_{ov}, suffix “ov” denoting variable amplitude loading; values are included in Table 18 and Table 20. This procedure is equivalent to assuming that the number of repetitions of each stress range S_r less than S_{ov} is reduced in the proportion $(S_r/S_{ov})^2$.

NOTE 1 This correction does not apply in the case of unprotected joints in sea water.

NOTE 2 The following might assist in calculations:

for $S_r \geq S_{OV}$

$$\frac{n}{N} = \frac{nS_r^m}{C_2} = \frac{n}{5x10^7}\left(\frac{S_r}{S_{ov}}\right)^m$$

for $S_r \le S_{OV}$

$$\frac{n}{N} = \frac{nS_r^{(m+2)}}{C_2 S_{ov}^2} = \frac{n}{5x10^7}\left(\frac{S_r}{S_{ov}}\right)^{m+2}$$

NOTE 3 This approach might be too conservative for some stress spectra, including those that consist predominantly of stress ranges below S_{oc} *with only occasional stress ranges above it. A different procedure could be justified on the basis of specific tests in accordance with Annex E. Alternatively, the damaging effect of stress ranges below* S_{oc} *can be calculated using fracture mechanics (see Annex D).*

16.5 Treatment of high stress cycles

Where local bending or other structural stress concentrating features are involved, and the relevant stress range includes allowance for the stress concentration (including by the use of the hot-spot stress), the class B to G2, W1 and TJ design S_r-N curves for welded joints may be extrapolated back linearly (on the basis of log S_R versus log N) up to a limit of a stress range equal to twice the material nominal tensile yield stress ($2\sigma_Y$).

However, for joints in a region of simple membrane stress the design S_r-N curves should only be extrapolated back to a stress range given by twice the applicable tensile stress limitation given in **16.1**.

As all the *S-N* curves for welded joints cross the class B curve for plain steel in the high stress/low endurance regime to give higher fatigue lives for a given stress range, the class B, or class C for unprotected joints in a corrosive environment, curve should be used if it indicates a lower fatigue strength than the welded joint curve.

For the class S_1 and S_2 curves, extrapolation may be made back as for the other design classes but up to a limit of stress range defined by half the values given in this sub-clause (i.e. with reference to shear instead of tensile stress).

16.6 Joints subjected to a single stress range

For a joint subjected to a number of repetitions of a single stress range (i.e. constant amplitude loading), the range should be not greater than that defined by Equation 2 (see **16.2.1**) and Table 18 or Table 20 for the relevant environment, joint class, number of cycles and required probability of failure. Apart from the case of unprotected joints in sea water, if the stress range is lower than S_{oc} (see **16.4**) it may be assumed that fatigue failure will not occur. This might also be the case for some variable amplitude loading (see **16.7**).

16.7 Joints subjected to a stress spectrum

For a joint subjected to a number of repetitions n_i of each of several stress ranges, S_{ri}, (i.e. variable amplitude loading) the value of n_i corresponding to each S_{ri} should be determined from standard loading rules (if applicable), from stress spectra measured on a similar structural member, or by making reasonable assumptions as to the expected service history, as appropriate (see Clause **7** and **15.9**). The number of cycles to failure, N_i, at each stress range, S_{ri}, should then be determined from the basic *S-N* curve, modified as necessary in accordance with **16.3**, for the relevant joint class at the selected probability of failure. The fatigue damage sum D due to the stress spectrum to be endured for the required life of the joint is then calculated using Miner's linear cumulative damage rule, as follows:

$$D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + ..etc = \sum \frac{n_i}{N_i} \qquad (8)$$

If all values of S_{ri} are less than S_{oc} it may be assumed that fatigue failure will not occur. Experience (e.g. [8,9]) indicates that this can be assumed to be the case with a stationary narrow band variable amplitude loading spectrum, for which the histogram of stress cycles (including all increases due to misalignment, thickness correction etc.) follows a Rayleigh distribution, if the stress range corresponding to 9 times the root mean square of the stress amplitude for the stress-time history is no greater than the initial non-propagating stress range S_{oc}, (see **16.4**).

However, if any values of S_{ri} exceed S_{oc}, all the stress ranges, including those below S_{oc}, should be included in the summation. This is because the higher stresses in the spectrum are capable of propagating cracks which might then be propagated further by the lower stresses. Some of the lower stresses in the spectrum may be assumed to be less damaging than indicated by the basic *S-N* curve (see **16.4**).

In general, the required fatigue life is achieved if D ≤ 1.0. Fatigue testing (e.g. [10, 11, 12]) has confirmed that this assumption is safe if the loading produces a spectrum with gradual variation in stress range, notably narrow-band random loading conforming to a Rayleigh distribution of peaks. However, under some stress spectra, including those involving fully-tensile stress cycling about a high tensile mean stress or where there is little variation in the maximum applied tensile stress, tests have shown that fatigue failure can occur when D < 1, typically when D ~ 0.5 but sometimes even lower [12]. Therefore, if there is any uncertainty about the nature of the service stress spectrum, or for particularly critical cases, it is advisable to limit D to 0.5. Alternatively, D can be established for the particular stress spectrum and welded joint type concerned by reference to relevant published data or by special testing (see Annex E).

If the damage sum D obtained from Equation 8 is greater than the limiting value selected for design (e.g. 0.5), measures that can be taken to improve the fatigue strength of the joint include strengthening the detail to reduce the applied stresses, re-design of the detail to a higher class or the application of a weld toe improvement technique (see Annex F).

Annex A (normative) Fatigue design

A.1 S-*N* relationships

The S_r-*N* relationships for the various structural detail classes have been based on statistical analyses of available experimental data obtained under tensile or shear loading. The analyses involved linear regression analyses of log S_r and log *N* with the slopes of the curves predetermined. In addition some minor empirical adjustments were made to ensure compatibility of results between the various classes [1-3].

The change of slope in the curves from *m* to (*m* + 2) (see Figure 14) is a mathematical device to avoid difficulties in cumulative damage calculations using Miner's rule. The bent *S*-*N* curve should not be assumed to represent the results that would be obtained in tests under constant amplitude loading.

As far as welded joints are concerned, it has been shown experimentally that, when high tensile residual stresses are present, fatigue strength is a function of stress range alone; mean stress and stress ratio have no significant effect. In general it is impossible to predict what residual stresses might be present in any particular structure and therefore the design rules are based upon the assumption that high tensile residual stresses are likely to be present. For simplicity the same assumption has been made with regard to non-welded details. This is a reasonable assumption to make for both welded (as-welded or stress relieved) and non-welded details if long range tensile residual stresses are introduced, for example as a result of imperfect fit-up during subsequent assembly involving the parts concerned.

Although the fatigue test data used to determine the design S_r-*N* relationships were obtained from arc welded joints it has been found that they are also suitable for similar joints made with either power beam (electron beam or laser) or friction welding [13].

A.2 Fatigue life for various failure probabilities

The standard basic S_r-*N* curves in Figure 10 are based on two standard deviations below the mean line assuming a log normal distribution, with a nominal probability of failure of 2.3%. In certain cases, a higher probability of failure could be acceptable, for example where fatigue cracking would not have serious consequences or where a crack could be easily located and repaired. In situations where, for example, there is no structural redundancy or where the joint is uninspectable it might be desirable to design against a lower probability of failure.

The nominal probabilities of failure for a known stress spectrum associated with various numbers of standard deviations below the mean curve are given in Table 19. The *Sr*-*N* curves appropriate to other numbers of standard deviations below the mean curve can be derived from Equation 2 (see **16.2**).

*NOTE The overall probability of failure during the design life of a typical product is substantially lower than the values in Table 19 (which are only applicable to the fatigue strength distribution) when the upper bound values of loading are assumed (see Clause **7**).*

A.3 Fatigue design philosophy

A.3.1 Safe life design

The procedures and design data provided in this British Standard are primarily intended for use in safe life design. The safe life design approach is based on the use of standard lower-bound fatigue endurance data and an upper bound estimate of the fatigue loading. It therefore provides a conservative estimate of fatigue strength and does not depend on in-service inspection for fatigue damage.

A.3.2 Damage-tolerant design

The damage-tolerant design method should only be used if the product, or the relevant part of it, can be safely and economically inspected by appropriate NDT, and any cracks detected and repaired before they reach a length that could cause failure under static loading. The following should be taken into account and evaluated at the design stage when deciding to take a damage-tolerant approach:

a) the strength of the structure;

b) the consequences of failure; and

c) the need for inspection and the feasibility of repair.

NOTE Damage-tolerant design might be suitable for application where a safe life assessment shows that fatigue has a significant effect on design economy or where a higher risk of fatigue cracking during the design life may be justified than is permitted using safe life design principles.

Damage-tolerant design should ensure that when fatigue cracking occurs in service the remaining intact material can sustain the maximum working load without failure until the damage is detected. Therefore, a prescribed inspection and maintenance programme for detecting and correcting any fatigue should be put in place and followed throughout the life

The following design features should be used to help achieve damage tolerance:

1) selection of materials and stress levels to provide low rates of crack propagation and long critical crack lengths;

2) provision of multiple load paths;

3) provision of crack-arresting details; and

4) provision of readily inspectable details.

Damage tolerance depends on the level of inspection to be applied to the product and is not automatically ensured by replaceable components. Inspection should be planned to ensure adequate detection and monitoring of damage and to allow repair or replacement of components. The following factors should be taken into account:

i) location and mode of failure;

ii) remaining structural strength;

iii) detectability and associated inspection technique, which should be based on the largest flaw not likely to be detected rather than the smallest it is possible to find;

iv) inspection frequency;

v) expected propagation rate allowing for stress redistribution; and

vi) critical crack length before repair or replacement is required.

The calculated fatigue lives enable critical parts to be ranked in terms of fatigue sensitivity. When in-service inspection is an option, this can be used, in conjunction with an assessment of the consequences of failure of specific members, to establish a priority basis for developing a selective inspection programme to be followed during the service life of the product. For non-structurally redundant parts where the consequence of failure is high, or for which in-service inspection would be difficult to achieve, it is often preferable to re-design critical parts to a higher fatigue classification to reduce the level of future inspection required.

Annex B (normative) Explanatory notes on detail classification

B.1 General

This annex gives background information on the detail classifications given in Table 1 to Table 10, including the potential modes of failure, important factors influencing the class of each detail type and some guidance on selection for design.

B.2 Non-welded details (see Table 1 and Table 2)

B.2.1 Potential modes of failure

In unwelded steel, fatigue cracks normally initiate at surface irregularities, corners of the cross sections or holes and re-entrant corners. In steel which is connected with rivets or bolts, failure generally initiates at the edge of the hole and propagates across the net section. However, in double covered joints made with high strength friction grip bolts this mode of failure is eliminated by the pre-tensioning, providing joint slip is avoided, and failure initiates on the surface near the boundary of the compression ring due to fretting under repeated strain.

B.2.2 General comments

B.2.2.1 Classes A to C

In welded construction, fatigue failure rarely initiates in regions of unwelded material as the fatigue strength of the welded joints is usually much lower.

Class A requires special manufacturing procedures which generally render it inappropriate for structural work. Hence assessment of fatigue strength for this class is not included in this British Standard.

Classes B and C should be applied with caution in cases where the high class is dependent on surface finish as this might degrade in service, for example from corrosion or abrasion. However, with due attention to the detrimental features mentioned, there might be scope for adopting a higher design curve than those specified. Such alternatives should be established in accordance with Annex E.

B.2.2.2 Fit-up and pre-tensioning of bolted connections

The specified fit-up of bolted connections should be achieved in practice. If not, the stress ranges applied to the bolts might be much higher than those assumed in design, which could lead to premature failure.

After a group of bolts has been tightened, the torque on all the bolts should be checked. This is because it is possible for some pre-tension to be lost as later ones are tightened, even if they were originally tightened using tension or torque control. If not, bending could be introduced with the result that stress ranges applied to the bolts would be higher than those assumed in design.

B.2.2.3 Fasteners

In threaded fasteners fatigue cracks normally initiate at the root of the thread, particularly at the first load-carrying thread in the joint. Alternatively, failure is sometimes located immediately under the head of the bolt, particularly in bolts with rolled threads and in joints subjected to bending loads.

B.3 Continuous butt welds and welded attachments essentially parallel to the direction of stress (see Table 3)

B.3.1 Dressing of butt welds (type 3.1)

Fatigue tests on butt welds with the weld overfill dressed flush have shown that class C is a realistic design classification provided that the weld is free of flaws. However, this classification is reliant on the detection of surface and embedded flaws that could reduce the fatigue strength below class C. This has been shown to be possible [14] but it does require the provision of NDT techniques and operators capable of both detecting and sizing flaws, a requirement that is generally beyond the scope of routine workmanship inspection. Guidance is provided in Clause **14** and BS 7910. Acceptance levels for welding flaws to meet class C are given in **14.3.4**, or they can be determined using fracture mechanics (see Annex D).

B.3.2 Potential modes of failure

With the reinforcement dressed flush (type 3.1), failure in longitudinal butt welds tends to be associated with embedded flaws. In continuous butt welds or in butt or fillet welded continuous attachments (types 3.2 and 3.3), away from weld ends, fatigue cracks normally initiate at ripples or lumps at stop/start positions on the weld surface. However, in the case of discontinuous welds (types 3.4 and 3.5) fatigue cracks occur in the parent metal at the weld ends.

B.3.3 General comments

B.3.3.1 Welds near plate edges (see Figure B.1)

Although welding attachments to plate edges can result in a very low classification (e.g. types 4.7 and 7.7) there is no downgrading for welds close to a plate edge or ones that accidentally overlap an edge. However, as with edge attachment welds, in such cases the possibility of local stress concentrations occurring at unwelded corners as a result of, for example, undercut, weld spatter and excessive leg length at stop-start positions or accidental overweave in manual welding should be limited. Similarly, the unwelded corners of, for example, cover plates or box members [see Figure B.1a) and Figure B.1b)] should not be undercut. If this does occur, it should subsequently be ground out to a smooth profile.

Figure B.1 **Welds at plate edges**

Key

1 Avoid or grind out to a smooth profile any undercut to these exposed corners

B.3.3.2 Attachment of permanent backing strips

If a permanent backing strip is used in making a longitudinal butt welded joint it should be continuous or made continuous by welding. Any welds in the backing strip, and those attaching it, should also conform to the relevant class requirements. The classification might reduce to class E or lower (type 5.3 or 5.4) at any transverse butt welds in the backing strip that have not been dressed flush or nominal stress class E at any permanent tack weld (see type 3.6).

B.3.3.3 Tack welds

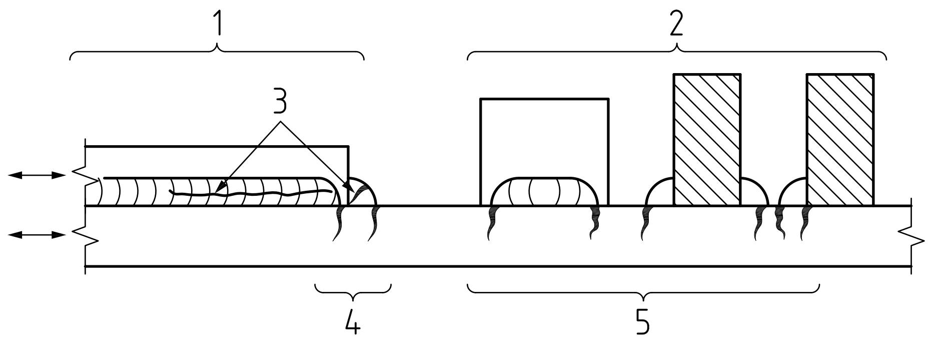
Tack welds, unless carefully ground out or buried in a subsequent run, provide potential crack locations similar to any other weld end. Their use in the fabrication process should be strictly controlled.

B.4 Welded attachments on the surface or edge of a stressed member (see Table 4)

B.4.1 Potential modes of failure (see Figure B.2)

When the weld is parallel to the direction of the applied stress fatigue cracks normally initiate at the weld ends, but when it is transverse to the direction of stressing they usually initiate at the weld toe; for attachments involving a single fillet weld, as opposed to a double, weld cracks might also initiate at the weld root. In each case the cracks then propagate into the stressed member.

Figure B.2 **Failure modes at weld ends and weld toes of welded attachments**



For classification purposes, an "attachment" should be taken as the adjacent structural element connected by welding to the stressed element under consideration. Apart from the particular dimensional requirements given for each type in Table 4 the relative size of the "stressed element" and the "attachment" influences the correction for thickness and bending (see **16.3.2**).

Key

1 Long attachment
2 Short attachments
3 Weld failure cracks (type, 7.8 and 7.10)
4 Joint types 4.2 to 4.6 (or 4.7 at edge)
5 Joint types 4.1 and 4.2 (or 4.7 at edge)

B.4.2 General comments

B.4.2.1 Stress concentrations

Stress concentrations are increased, and the fatigue strength or joint classification is therefore reduced, where the following apply:

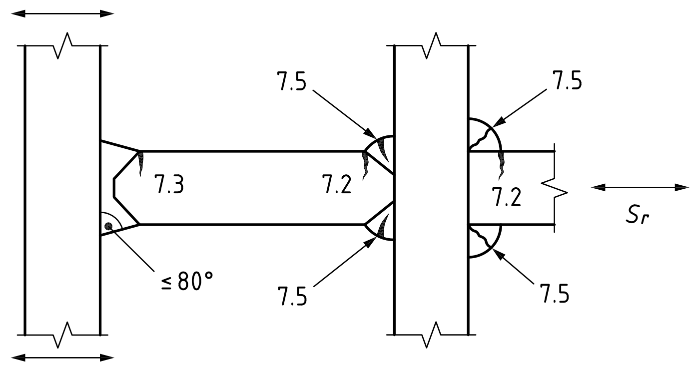
a) the weld ends or toes are on an unwelded corner of the element; and

b) the attachment is long in the direction of stressing and, as a result, transfer of a part of the load in the element to and from the attachment is likely to occur through welds adjacent to its ends. This effect is further intensified with thick attachments.

B.4.2.2 Weld forms

Full or partial penetration butt welded joints of T form, including cruciform joints, (e.g. those that would connect attachments to the surface of a stressed element) should be completed by fillet welds of sufficient leg length to produce nominal weld toe angles no greater than 80° (see Figure B.3). The fillets exclude the possibility of an increase in stress concentration arising at an acute re-entrant angle between the element surface and the toe of the weld, and it is therefore unimportant whether the attachment is fillet or butt welded to the surface when assessing the effects on the stressed element, as a similar toe profile results in both cases.

Figure B.3 **Failure modes in cruciform and T-joints for joint types indicated**



NOTE T-joint shown with minimum weld angle to be achieved when adding fillet weld to full or partial penetration butt welded joint.

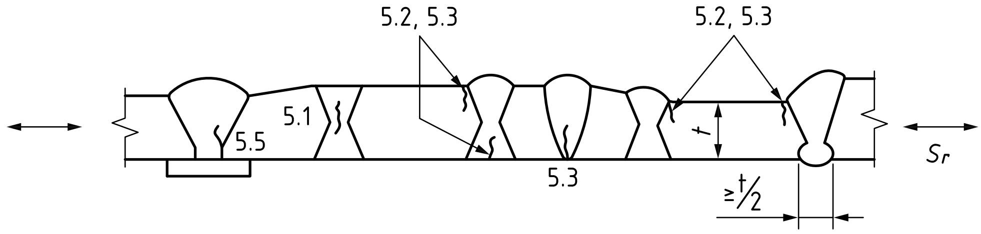
B.5 Transverse butt welds (see Table 5 and Table 6)

B.5.1 Potential modes of failure (see Figure B.4)

With the ends of butt welds machined flush with the plate edges, fatigue cracks normally initiate at the weld toe and propagate into the parent metal, so that the fatigue strength depends largely upon the toe profile of the weld. If the reinforcement of a butt weld is dressed flush (see type 5.1), failure is more likely to occur in the weld material from embedded flaws or from minor weld flaws which become exposed on the surface, e.g. surface porosity in the dressing area.

In the case of butt welds made on a permanent backing strip, fatigue cracks initiate at the weld metal backing strip junction and then propagate into the weld metal.

Figure B.4 **Failure modes in transverse butt welds for joint types indicated**



B.5.2 General

B.5.2.1 Misalignment

Butt welded joints between plates or tubes are susceptible to misalignment (see **C.6**) and therefore transverse joints might experience secondary bending under applied axial loading. No allowance is made for this in the classifications. For elements where out-of-plane bending is resisted by contiguous construction, e.g. beam flanges supported by webs, wide plates supported by effectively continuous stiffeners, eccentricities due to axial misalignments in the thickness direction may be neglected.

However, where such support is not provided, e.g. tension links, the design stress should include an allowance for the bending effects of any misalignment [1)], i.e. the nominal distance between the centres of thickness of the two abutting components, *e*. For components tapered in thickness, the mid-plane of the untapered section should be used. The nominal stress should be multiplied by the following stress magnification factor:

$$k_m = 1 + 6\frac{e}{t_1} \times \frac{t_1^3}{t_1^3 + t_2^3} \qquad (B.1)$$

where

t_1 is the thickness of the thinner plate

t_2 is the thickness of the thicker plate

Thus, when $t_1 = t_2$, the stress concentration factor becomes $1+3\frac{e}{t}$.

For other cases, including angular misalignment, see BS 7910. For additional solutions specific to girth welded joints in pipes, reference should be made to DNV-RP-C203.

B.5.2.2 Element edges

Fatigue failures in butt welded plates tend to be associated with plate edges and undercut at the weld toes on the corners of the cross section of the stressed element (or on the edge at the toes of any return welds) should be avoided. If it does occur, any undercut should be ground out to a smooth profile.

B.5.2.3 Part width welds

Butt type welds might also occur within the length of a member or individual plate, e.g. in the case of:

a) a plug weld to fill a small hole;

b) a weld closing a temporary access hole with an infill plate.

Although such geometries have not been given specific categories, the relevant type in Table 5 or Table 6, may be deemed to cover plug and infill plate welds.

[1)] This includes unintentional misalignment to the extent of the acceptance limit specified by the manufacturer or fabrication standard being used. Tightening that acceptance limit (see **14.3**) would be beneficial from the fatigue viewpoint.

B.5.2.4 Joints butt welded from one side only

Butt welds should be produced with full-penetration welds. The most reliable way to achieve this in joints welded from one side is by the use of permanent backing (types 5.5 and 6.8). However, the design class is relatively low (class F) and the alternative use of temporary non-fusible backing (types 5.3, 5.4, 6.6 and 6.7) allows some improvement [15]. In this case, inspection of the weld root to assess whether or not there is full penetration is feasible and should be carried out to justify class E. In the case of butt welds in pipes (types 6.6 and 6.7) the surfaces of the pipe and backing should be aligned to within ±1 mm to avoid the seepage of weld metal through the gap and the subsequent formation of a cold lap on the inside of the pipe [16]. In the absence of any backing (types 5.3, 5.4, 6.6 and 6.7) the main requirement is full penetration and to justify class E, this should be assessed by inspection. If this is only possible from one side of the joint, as in pipe girth welds, specialist NDT such as automated ultrasonic testing (AUT) should be carried out, further guidance is given in BS 7910. Experience indicates that the techniques used are capable of detecting root flaws due to incomplete penetration down to approximately 0.6 mm in depth [17].

B.5.2.5 Penetration of butt welds

Butt welds transmitting stress between plates, sections or built-up members connected end-to-end should be full penetration welds.

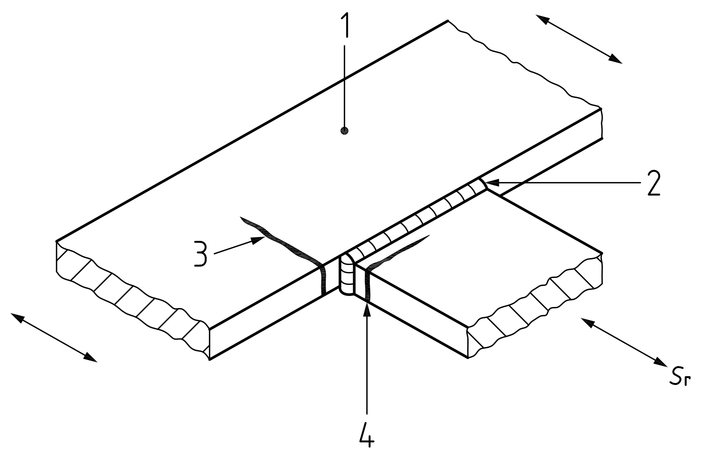
B.5.2.6 Dressing of butt welded types 5.1 and 6.4

The information given in **B.3.1** for longitudinal butt welds is equally applicable to transverse welds, with the additional recommendation that the dressing is sufficient to ensure that all the traces of the weld toe are removed.

B.5.3 Cruciform and T-joints between plates in the same plane (type 5.6)

Often, the load is transmitted from a member to a transverse member primarily via flange plates in the same plane (see Figure B.5). This can occur in the case of a junction between cross girders and main girders, diagonals and truss chords, or in Vierendeel frames.

Figure B.5 **T-junction of two flange plates**



Key

1 Transverse member

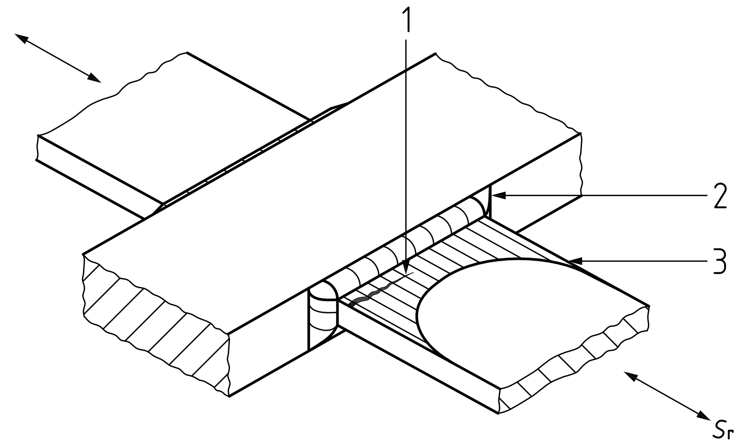
2 Full penetration butt weld

3 Type 4.7

4 Type 5.6 (with stress concentration factor)

If a full penetration butt weld is used and the joint geometry is in accordance with the requirements of type 5.6, and if in addition the joint is either of the cruciform type (see Figure B.6) or if the transverse member is relatively stiff, i.e. its width is at least three times the width of the stressed member, the nominal stress-based classification should be as given for type 5.6 with a stress concentration factor of unity.

Figure B.6 **Cruciform junction between flange plates**



Key

1 Type 5.6 (with stress concentration factor)

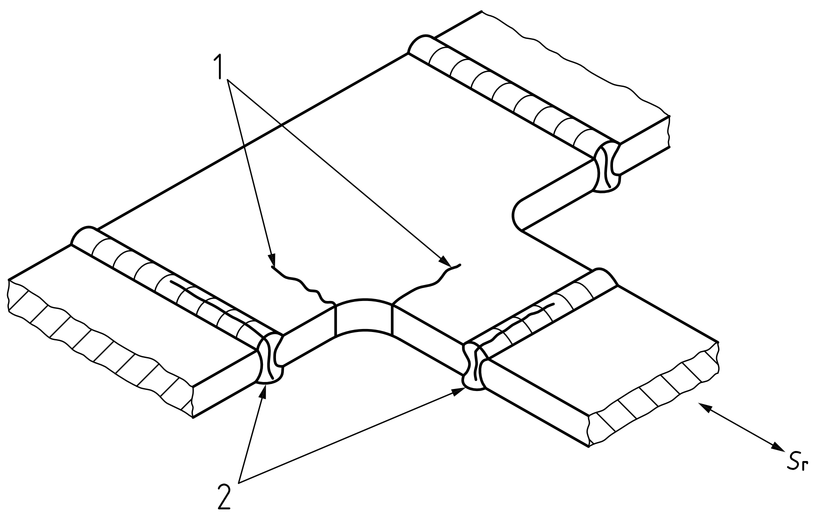
2 Full penetration butt weld

3 Typical stress pattern

Otherwise the classification should be assumed to be that of type 5.2 with the appropriate stress concentration factor.

In the case of trusses, secondary stresses due to joint fixity should be taken into account. The fatigue strength of both flange plates can be improved by the insertion of a smoothly radiused gusset plate in the transverse member so that all butt welds are further from re-entrant corners (see Figure B.7).

Figure B.7 **Alternative method for joining two flange plates**



Key

1 Types 1.3 or 1.4 (with stress concentration factor)

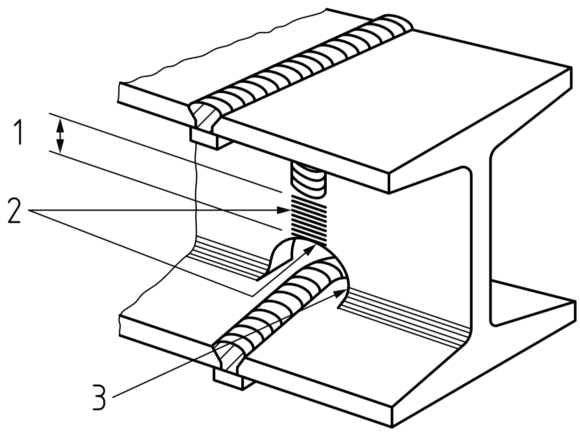
2 Types 5.2, 5.5 or 6.1

B.6 Transverse butt welds in sections (see Table 6)

Butt welds between rolled or built up sections (type 6.1) are prone to weld flaws in the region of the web/flange junction. The low classification reflects the fact that these are very difficult to detect. The normal classifications for transverse butt welds in Table 5 should only be adopted if special preparations, procedures and inspection have been undertaken that show the weld is free from significant flaws.

The same types of joints are frequently made using cope holes to provide access for making the weld in the flange. The end of the web butt weld at the cope hole (type 6.2) is equivalent to class D if the end of the butt weld, and the weld reinforcement within a distance equal to the radius of the cope hole, are ground flush (see Figure B.8). Otherwise, class E is applicable (type 6.3).The relevant stress should include allowance for the stress concentration effect of the of the cope hole (see DNV-RP-C203). Mitred cope holes of triangular shape should not be used.

Figure B.8 **Local grinding adjacent to cope hole in type 6.2 joint**



Key

1	Distance = *r*	3	Radius *r*
2	Grind		

B.7 Load-carrying fillet and T-butt joints (see Table 7)

B.7.1 Potential modes of failure (see Figure B.3)

Failure in cruciform or T-joints with full penetration welds normally initiates at the weld toe, but in joints made with load-carrying fillet or partial penetration butt welds cracking might initiate either at the weld toe and propagate into the plate or at the weld root and propagate through the weld. In welds parallel to the direction of the applied stress, however, weld failure is uncommon; cracks normally initiate at the weld end and propagate into the plate perpendicular to the direction of applied stress. Nevertheless, provision is made for possible shear failure through the weld throat (see type 7.10).

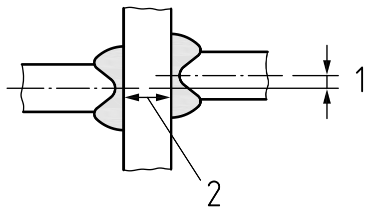
B.7.2 Cruciform joints (types 7.1, 7.2 and 7.8)

Cruciform joints between plates are susceptible to misalignment [see Figure B.9 a)] and thus secondary bending stresses are likely to be introduced under applied axial loading. However, the classifications allow for the effects of any accidental axial or centreline misalignment up to the lesser of 0.05 times the thickness of the thinner part or 2 mm. The secondary bending stress due to any misalignment that exceeds these limits should be allowed for in the calculation of the design stress. Equation B.1 is applicable for axial misalignment but other formulae for calculating the relevant k_m for assessing potential fatigue failure from the weld toe or weld root are given in BS 7910.

Where the third member is a plate it can be assumed that plane sections remain plane in the main members and that the axial and bending stress distributions in the S_r direction are unaffected. Where the third member is an open shape, for example, an I section or a hollow tube, particularly if different in width, a discontinuity in the main member stress pattern is likely to occur. In this case the stress parameter should include the stress concentration at the joint, for example by using the hot-spot stress. In the absence of published data on a particular joint configuration, the stress concentration factor for use with the nominal stress might need to be determined by finite element or model analysis.

Plane sections may be assumed to remain plane where the main member stress can be continued through the transverse member by additional continuity plating of comparable cross-sectional area, which is in line with the main member components [see Figure B.9 b)]. In this type of connection the joint regions of the third member should be checked before welding for lamellar rolling flaws and after welding for lamellar tears.

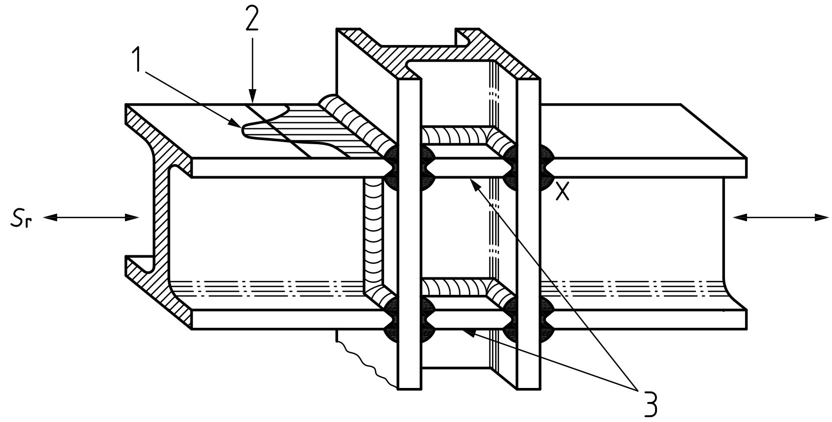
Figure B.9 **Use of continuity plating to reduce stress concentrations in type 7.1 and 7.2 joints**



a) Cruciform joint between plates

Key

1 Axial misalignment or eccentricity, *e* 2 Through-wall direction



b) Cruciform joint between sections

Key

1 Typical stress pattern with plating

2 Typical stress pattern without plating

3 Continuity plating

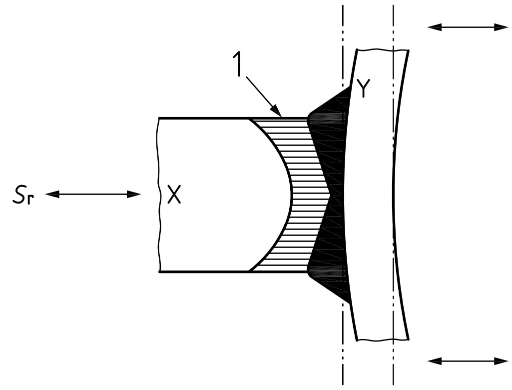
4 Detail X show in Figure B.9a)

B.7.3 T-joints (types 7.3 and 7.4)

T-joints (Figure B.10) are distinguished from cruciform joints by the absence of a similar member in line on the far side of the joint. In this case an axial load in member X would induce a bending moment and curvature in member Y. Unless member Y is very stiff in bending an uneven stress distribution results. Members with bolted end connections via transversely welded end plates are particularly susceptible to local increase of stress (see Figure B.10). If the transverse member is an open or hollow section, local bending increases the peak stress further (as in the case of types 7.1 and 7.2). These effects are included in the hot-spot stress but appropriate stress concentration factors should be applied to the nominal stress.

As far as fatigue failure of the transverse member Y is concerned, member X is treated as an attachment (types 4.1 to 4.6) and the stress parameter is the stress in the transverse member without the application of a stress concentration factor. In hollow or open transverse members this is often magnified by local bending of the walls.

Figure B.10 **Example of type 7.3 or 7.4 T-joint**



Key

1 Stress distribution due to flexibility of member Y

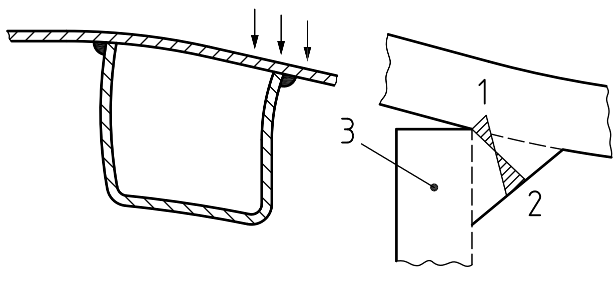
B.7.4 Load-carrying fillet welds failing in the weld (types 7.8, 7.10 and 9.5)

Provision is made for assessing load-carrying fillet welds with respect to potential failure from applied direct and shear stresses. Depending on the proportion of shear, either class W1 or class S_2 is applicable (see **15.3** and Table 17). Apart from cases of pure shear, class W1 is the most likely relevant class in practice. Class W1 is primarily intended to apply to all fillet or partial penetration butt weld joints where bending action across the throat does not occur. Where lapped joints are welded on two or more sides, or T- or cruciform joints are welded from both sides, such bending action is normally prevented. However, if significant bending does occur and it can be quantified, it can be included in the calculation of the resultant stress range and hence used in conjunction with class W1.

B.7.5 T-joints failing by bending in the weld (type 7.9)

In certain cases difficulty of access might only allow welding of T-joints to be done on one side of the joint. This applies particularly to small hollow members with welded corners which, if subject to loading that distorts the cross section, might cause failure of the corner weld in bending (see Figure B.11 and type 7.9). Where axial stress is also present, the stress range at the face of the weld might be different from that at the root. Failure from ripples or stop-start positions on the face can give a higher strength than class G, but expert advice should be sought if a higher strength is required. In most cases, failure from stress fluctuation in the root is critical and should be classified as class G with application of the thickness and bending correction k_{tb}.

Figure B.11 **Single fillet corner weld in bending (type 7.9)**



Key

1	Bending stress at root	3	Corner detail
2	Bending stress on weld face		

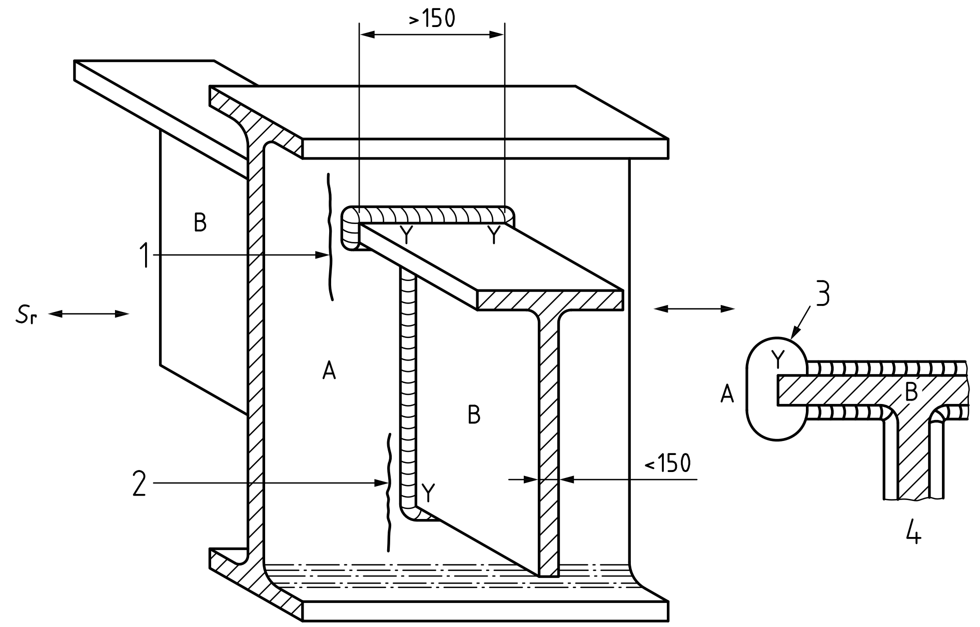
B.8 Slotted connections and penetrations through stressed members (see Table 8)

Slotted connections exist where a narrow member is slotted through a single main member away from an end connection (see also Figure B.12). In this case, the narrow member should be assumed to transmit the stress which the parent material would have carried before the slot was cut. The part of the narrow member projecting out of the plane of the stressed plate then becomes, effectively, a welded attachment, so that the classification becomes the same as for types 4.1 to 4.6. This detail should be avoided where possible, as slots are difficult to cut accurately and fit-up for welding is often poor. Where member B is called upon to carry high tensile stress, a slot in member A avoids any risk from lamellar tears.

However, with respect to stress fluctuation in member B, the detail shown in Figure B.12 is type 4.7 (nominal stress class G) at point Y. If member B is critical and member A is not, circular cut-outs at the corners of member B improve the class to that of type 4.3 (nominal stress class F).

Particular attention is drawn to the making of this type of joint with fillet welds instead of full penetration welds. In that case, the joint becomes type 7.2 and type 7.8.

Figure B.12 **Example of a third member slotted through a main member**



NOTE 1 Dimensions are in mm.

NOTE 2 Main member A is slotted while narrow member B is continuous and is welded all around the slot in A.

NOTE 3 Classes refer to nominal stress-based design; hot-spot stress class is D in both cases.

Key

1 Class F2 (type 8.2) 2 Class F (type 8.1)

3 Cut out 4 Alternative detail if member “B” is critical

B.9 Branch connections (see Table 10)

B.9.1 Potential modes of failure

There are four main sites for fatigue cracking in branch connections, the weld toes in the shell and branch, the weld root, leading to cracking through the weld throat, and the crotch corner. In every case account should be taken of the stress concentration in the region of potential fatigue cracking due to the gross structural discontinuity introduced by the nozzle.

B.9.2 Stress concentrations adjacent to branch connections

Four possible stress concentrations due to structural discontinuities in nozzles should be taken into account when calculating S_r.

a) *Crotch corner*. The class D fatigue design curve is used, based on nominal hoop stress range multiplied by K_f at the crotch corner where K_f is as defined in **15.6.4**.

b) *Weld toe in shell*. The appropriate fatigue design curve is normally used with nominal hoop stress range multiplied by K_f at the weld toe, where K_f is as defined in **15.6.4**. Alternatively, the hot-spot stress (see **15.7**) could be used directly. The possibility of stresses arising in the shell as a result of mechanical loading on the nozzle as well as pressure loading should be taken into account.

c) *Weld toe in branch*. This region should be treated as in item b). The possibility of mechanical as well as pressure loading should be taken into account.

d) *Weld root.* The class D or F fatigue design curve is used, based on the nominal hoop stress range on the minimum transverse cross-section of the weld multiplied by K_f at the weld root, where K_f is as defined in **15.6.4**. The possibility of mechanical as well as pressure loading should be taken into account.

*NOTE See **B.7.2** for potential failure from weld root under stresses acting normal to weld length.*

Annex C (normative) Guidance on stress analysis

C.1 General

The stress to be used with the S_r-N curves in Figure 9 and Figure 10 depends on the choice of method. In all the cases included in Table 1 to Table 10 one option is the nominal stress range at the potential cracking site shown in the sketches in the tables. It is the stress that would be calculated by conventional engineering methods, which does not take into account the effect of the local shape of the detail or the weld on the stress field. This stress is easily calculated in the case of simple axially loaded members or simple beams in bending. In more complex details the stress should be calculated adjacent to the detail which is analogous to the stress in the simple member. This means calculating the stress which is developed by reason of the shape of the structure, but without the perturbation in the stress field caused by the detailed shape of the weld itself. This annex provides guidance (**C.2** and **C.3**) on the determination of the resulting increase in nominal stress by means of experimental or numerical stress analysis or the use of appropriate stress concentration factors.

In the case of details in which the site of potential fatigue cracking is the weld toe (Table 3 to Table 10) an alternative approach is to use the specified S_r-N curve, usually class D for plate structures or class TJ for tubular joints, in conjunction with the hot-spot stress range. This is intended to include any stress concentration effect arising from the nature of the structure containing the weld detail as well as that due to the weld detail itself, but to exclude the stress concentration due to the weld toe. More detail is given in **C.4**.

C.2 Two dimensional stress systems

An example of a simple two dimensional stress system is a butt weld in a plate in a direction at right angles to the direction of stressing, such as a transverse weld in the flange of a box girder. In this situation the nominal stress can be calculated from simple bending theory. If a hole is made in the plate on the line of the weld the stress at the ends of the transverse diameter of the hole is magnified so that the weld at those points sustains a higher stress than at a distance from the hole. Figure 4 shows a typical stress distribution in such a case; further examples are shown in Figure B.5 to Figure B.10. It is this magnified stress which should be used as the input to the S_r-N curve for that particular detail. The effect on the fatigue life is quite marked because life is proportional to the third or the fifth power of the stress depending on the position on the S_r-N curve.

More complex shapes can be dealt with in the same way using published solutions [4, 5, 6] for most of the regular shapes of cut-out; two common details are shown in Figure 6. For shapes not included in the standard texts, stresses can be obtained from measured strains on a model or the full scale item. In such cases, the load cycles applied should be of sufficient magnitude and quantity to ensure that a shaken down state is reached by the strain gauge.

The designs of the models and the positions at which strains are measured should recognize the basis of the S_r-N curves to ensure that the relevant stress is used. This is more difficult in regions of steep stress gradients than in areas of low stress gradient.

Alternatively, the stresses can be calculated using the finite element stress analysis (FEA) method. This requires careful mesh generation to model the joint so that the relevant location for stress computation is selected. Too coarse a mesh in relation to the local stress gradient gives results which are insufficiently accurate. Too fine a mesh picks up the influence of the weld shape, if that is modelled, or the false influence of the idealized joint shape. Convergence checks provide a guide to the optimum mesh size.

Some other methods of stress measurement and visualization exist but tend to be more suited to experimental work rather than the immediate needs of the designer. These include techniques such as thermal imaging and X-rays.

C.3 Three dimensional stress systems

For most of the welded joint details shown in Table 3 to Table 10 the stress required to calculate the fatigue life is at the surface of the members. The stress analysis methods described in **C.2** give solutions where the stress field is two dimensional or axisymmetric. In the case of more complex stress fields, such as those produced by out-of-plane bending moments in plates (Figure B.10) or by brace loads on the walls of hollow section chords (Figure G.1), the standard analytically based solutions are not sufficient. In these cases, three dimensional methods such as photo-elasticity and FEA should be used for a full understanding of the stress pattern. Strain gauges give the surface stresses but if there is a need to know the stresses through the thickness of the material, (e.g. to separate the membrane and bending stress components), they are not suitable.

Finite element methods can be used to calculate the stresses in all types of joint. The selection of element types and sizes should reflect the geometry and the stress gradient so that a sufficiently reliable estimate of the stress at the weld detail is obtained. The modelling of the joint should reflect the overall geometry and the stiffness without introducing the details of the weld profile.

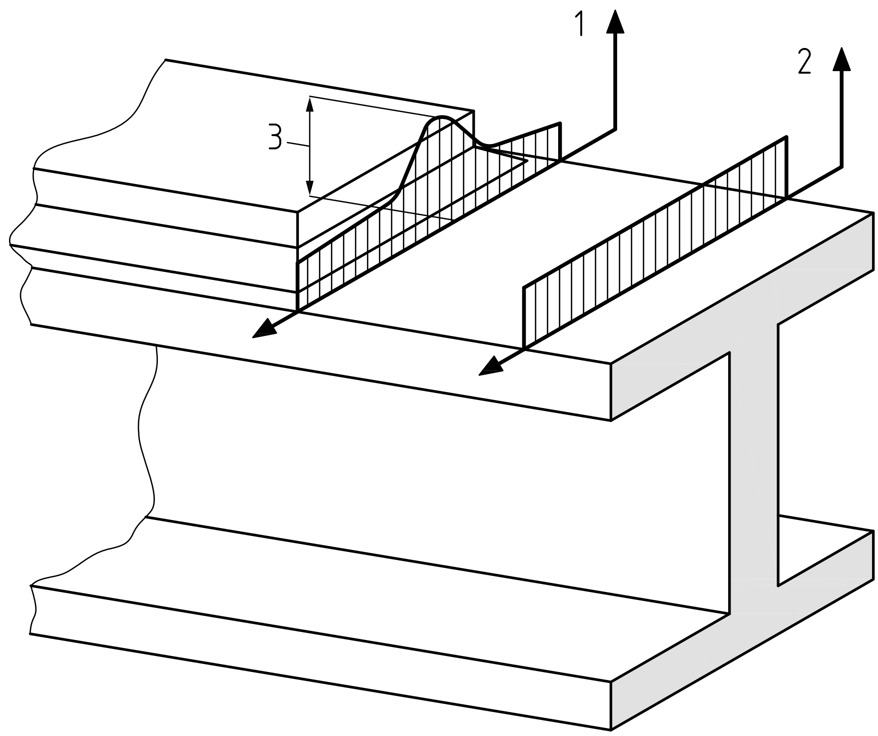
C.4 Calculation of hot-spot stress

C.4.1 Introduction

The hot-spot stress is widely used for the fatigue design of tubular nodal joints. The same approach offers many advantages for application more generally in the fatigue design of welded joints between plates and other structural sections when considering potential fatigue failure from a weld toe or end.

An example of the difference between the nominal and structural stress is shown for the simple case of an I-beam with a welded cover plate in Figure C.1. This detail is classified in **12.2** as class G for assessment based on nominal stresses. The class G design *S-N* curve was obtained from fatigue test results obtained from similar beams with cover plates and so the stress concentration effect of the cover plate welded to the flange is included in the *S-N* curve. Therefore, the relevant nominal stress that should be used with the class G *S-N* curve is that at the weld toe position based on the beam section, applied force and bending moment at that location, excluding the stress concentration effect of the cover plate, i.e. it is the structural stress at the weld toe that would be calculated using standard formulae for the beam in the absence of the cover plate.

Figure C.1 **I beam with cover plate showing distribution of structural stress and definition of hot-spot stress**



Key
1 Structural stress
2 Nominal stress
3 Hot-spot stress

In contrast, the actual structural stress at the weld toe, as would be obtained from stress analysis of the complete beam with a cover plate, is higher due to the stress concentrating effect of the attached cover plate and the extra stress concentration where the weld crosses over the web of the beam. This can be seen in the stress distributions across the width of the beam and approaching the cover plate (Figure C.1).

The structural stress is useful for assessment of potential fatigue cracking from a weld toe or the end of a short or discontinuous longitudinal weld. In such cases, the assessment is based on the maximum structural stress range at the weld toe, which is also referred to as the hot-spot stress range (S_{Hr}).

Application of the hot-spot stress approach to tubular joints (see Annex G) was aided by the provision of parametric formulae relating the hot-spot stress concentration factor to tube form, dimensions, joint type and mode of loading for a wide range of joint configurations. However, such formulae do not exist for the much wider range of welded joints that need to be assessed in non-tubular products. Therefore, it is necessary to determine hot-spot stresses by detailed stress analysis, typically by FEA or from strain measurements. This annex presents details of procedures for calculating hot-spot stresses that are consistent with the design *S-N* curves in Clause **16**.

C.4.2 Hot-spot stress calculation from surface stresses

As the through-thickness stress distribution remote from a weld is likely to be linear, the stresses on free surfaces remote from the weld are equal to the structural stress. Therefore, the structural stress at the weld toe, the hot-spot stress, can be estimated by extrapolation from the surface stresses at locations near the weld. This technique is called surface stress extrapolation (SSE). Stresses are used directly to calculate the weld toe structural stress.

The hot-spot stress is the maximum principal stress at the weld toe if its direction is within ±45° of the normal to the weld toe or, outside this range, the larger of the minimum principal stress or the stress component acting normal to the weld toe. Strictly speaking, the extrapolation should be performed for each stress component at the extrapolation point locations and the extrapolated values at the weld toe used to calculate the corresponding principal stresses. However, in practice it is sufficient to use either a principal stress or the stress component acting normal to the weld toe, whichever is dominant in accordance with the criteria in this sub-clause.

Various extrapolation methods have been proposed and investigated [18], dependent on the location of the weld toe (weld toe on plate surface or edge, weld toes in tubular joints) and the nature of the stress analysis.

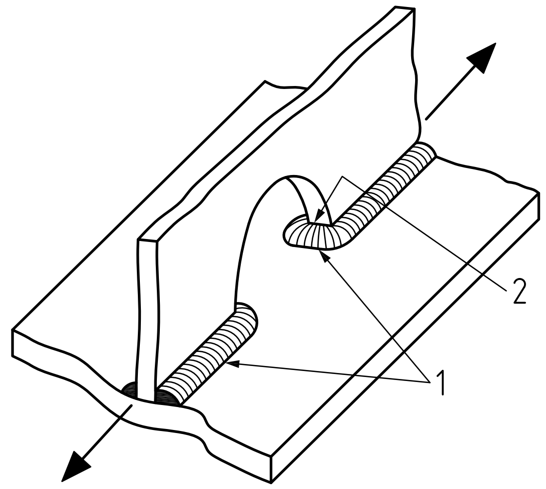
Where the stress distribution depends upon the plate thickness, type “a” hot-spot regions in Figure C.2, the most common SSE method is linear extrapolation from stresses on the surface at distances 0.4t and 1.0t from the weld toe. The corresponding hot-spot structural stress is given by:

$$S_H = 1.67\sigma_{0.4t} - 0.67\sigma_{1.0t} \tag{C.1}$$

This was the method used to determine hot-spot stresses in the fatigue test database used to validate the choice of class D as the hot-spot stress-based design curve [19]. Alternatives include linear extrapolation from stresses 0.5t and 1.5t from the toe and quadratic extrapolation from three locations, 0.4t, 0.9t and 1.4t from the weld toe.

In the case of weld toes on plate edges, type “b” hot-spot region in Figure C.2, the stress distribution approaching the weld toe does not depend on the plate thickness. Extrapolation methods have therefore been developed that use stresses located at absolute distances from the weld toe rather than proportions of plate thickness.

Figure C.2 **Types of hot-spot**



Key

1 Type “a” 2 Type “b”

For stress analysis based on measured strains or FEA of a relatively fine mesh model, quadratic extrapolation from stresses on the plate edge at distances 4 mm, 8 mm and 12 mm from the weld toe should be used. With regard to mesh size, in order to produce an accurate value of the stress at the 4 mm location the weld toe element should have a node at 4 mm. Either 2 mm linear or 4 mm quadratic elements should be used.

The corresponding hot-spot stress is given by:

$$S_H = 3\sigma_{4mm} - 3\sigma_{8mm} + \sigma_{12mm} \quad (C.2)$$

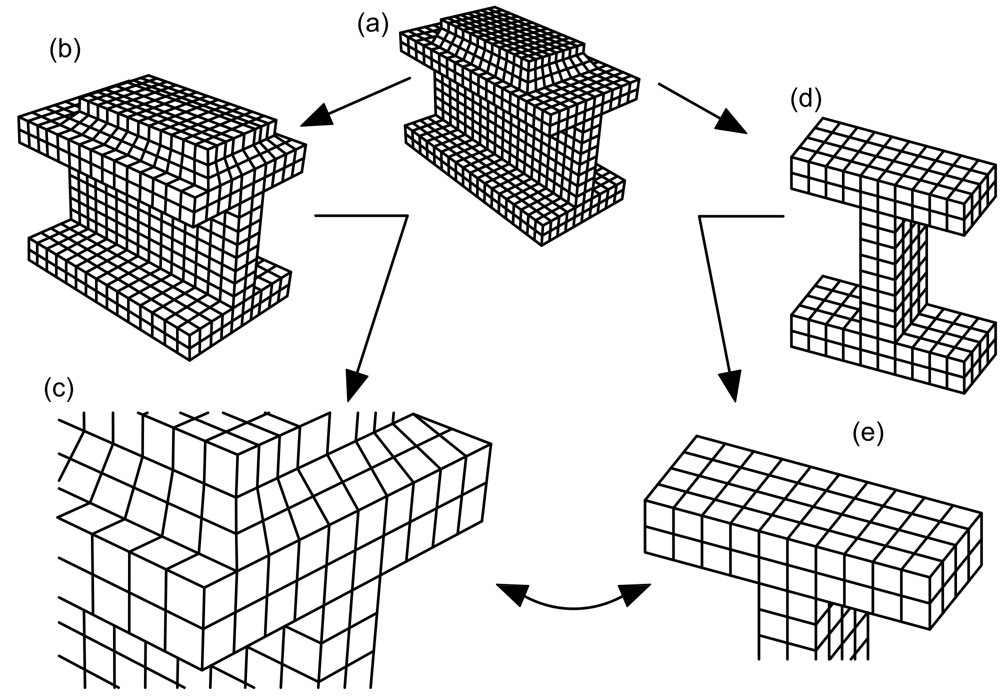
Alternatively, for stress analysis based on FEA with relatively coarse 10 mm × 10 mm quadratic elements, linear extrapolation from stresses on the plate edge at distances of 5 mm and 15 mm is suitable [18]. The corresponding hot-spot stress is given by:

$$S_\mathrm{H} = 1.5\sigma_{5\mathrm{mm}} - 0.5\sigma_{15\mathrm{mm}} \quad (C.3)$$

Guidance on the procedure for applying the SSE method on the basis of FEA is illustrated through a typical I-beam mesh of brick elements, as shown in Figure C.3. It shows a complete mesh and the two parts when it is separated along the plane through the weld. The two resulting separate mesh regions are shown in Figure C.4 and Figure C.5 where the nodes on the common face are identified. Additional nodes on the top face of the I-beam are shown in Figure C.5. The corresponding element numbers are shown in Figure C.6 and Figure C.7. These are suitable for the SSE calculations. The weld toe is represented by nodes n2 to n10. The SSE stress at node n2 is calculated from the stresses at nodes n2, n41, n52, n63 and n74 (Figure C.5). The stress distribution using the values at these nodes is shown in Figure C.8. The interpolated stresses at 0.4t and 1.0t from the weld toe are used in the extrapolation, as shown.

The stress at 0.4t from the weld toe should be determined from the stresses in the element adjacent to the weld toe (e.g. elements e71, e72 of Figure C.7) that are not in or under the weld. FEA software normally calculates the stress at node n2 as the average of the value at the node from all the elements it is attached to (e1, e2, e21 of Figure C.6 and e71, e72 of Figure C.8). The stresses in elements e1, e2 and e21 tend to be lower than those in e71, e71 because there is more material on that side of the weld toe. The stresses at node n2 should therefore only be taken from elements e71 and e72, where they are representative of those in the main plate.

Figure C.3 **Possible brick element model of an I beam with a cover plate**



Key

a) Weld region and adjacent elements

b) Elements underneath the weld

c) Close up of elements underneath the weld

d) Elements in beam adjacent to the weld

e) Close up of elements beyond the weld

Figure C.4 **Node numbers superimposed upon the weld mesh section in Figure C.3c)**

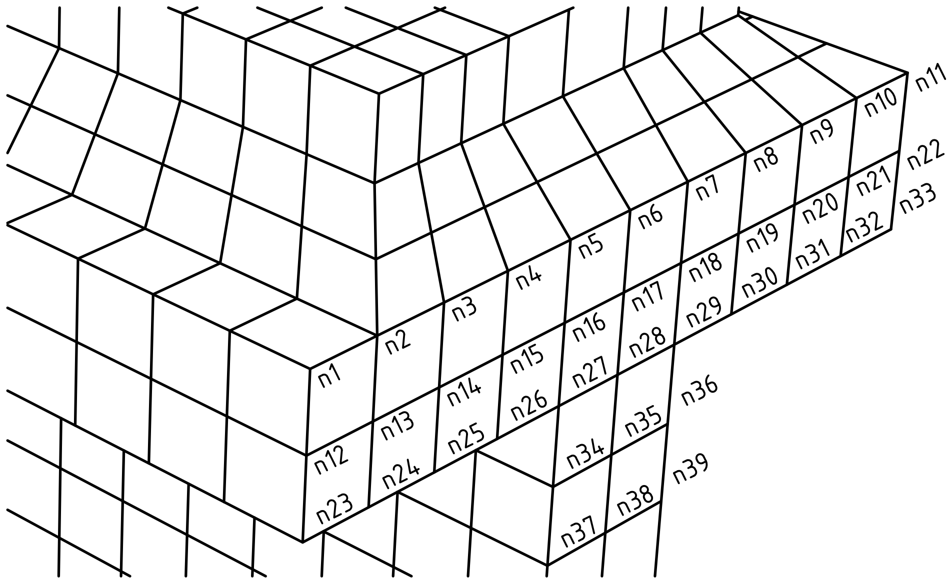
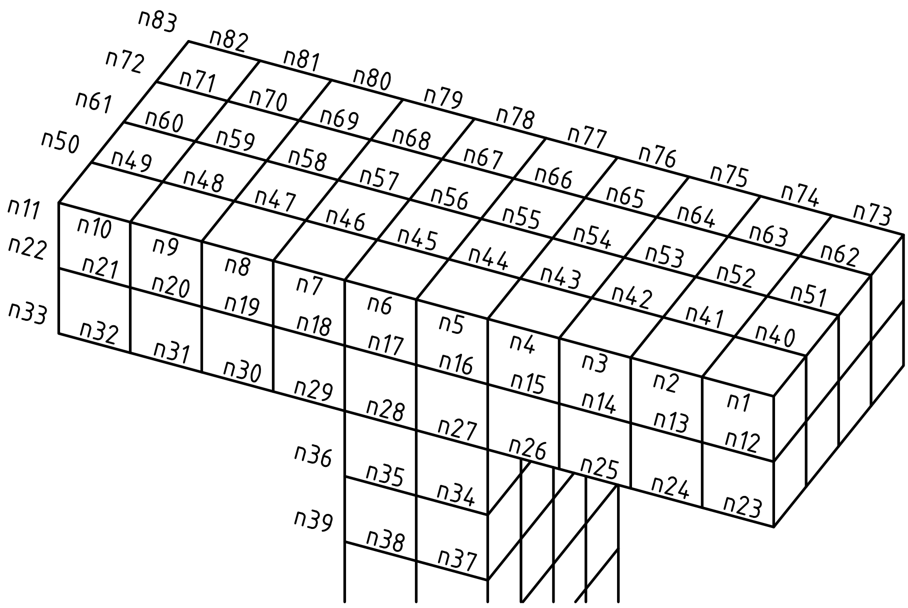
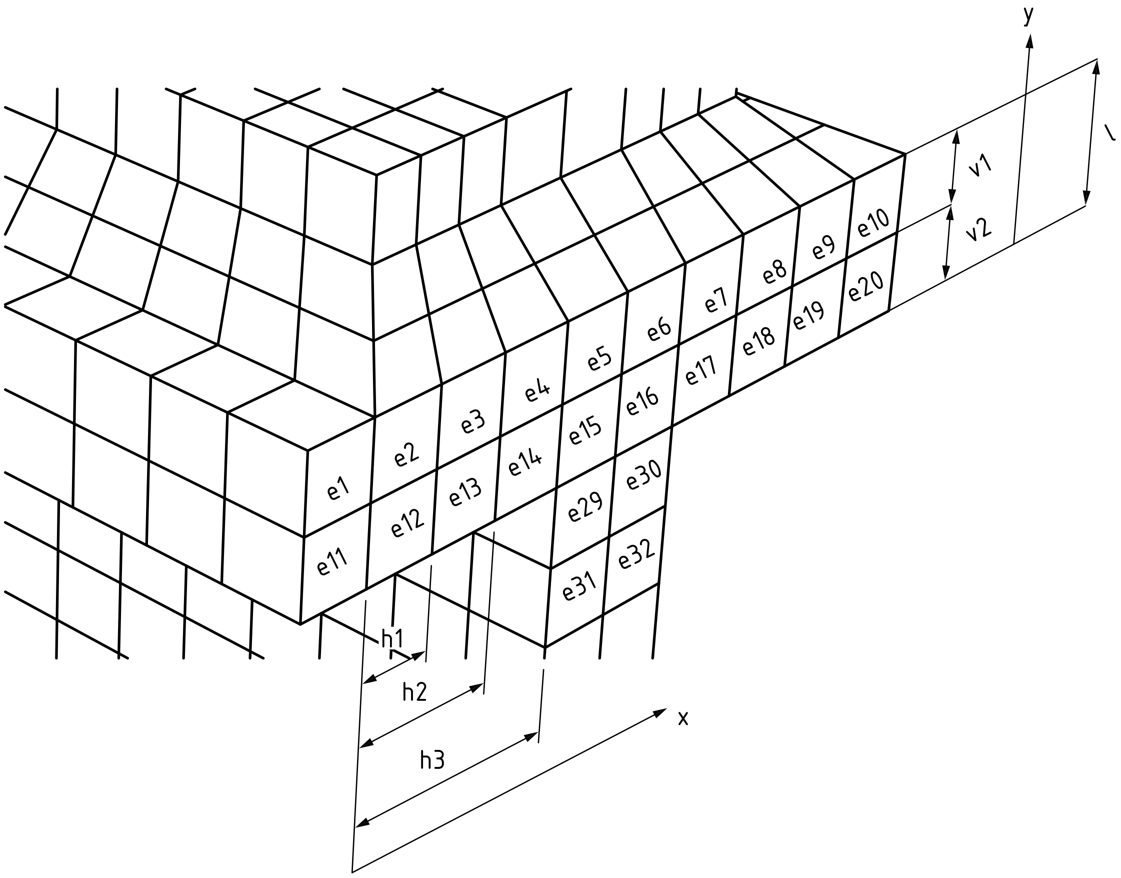


Figure C.5 **Node numbers superimposed upon the weld mesh section in Figure C.3e)**



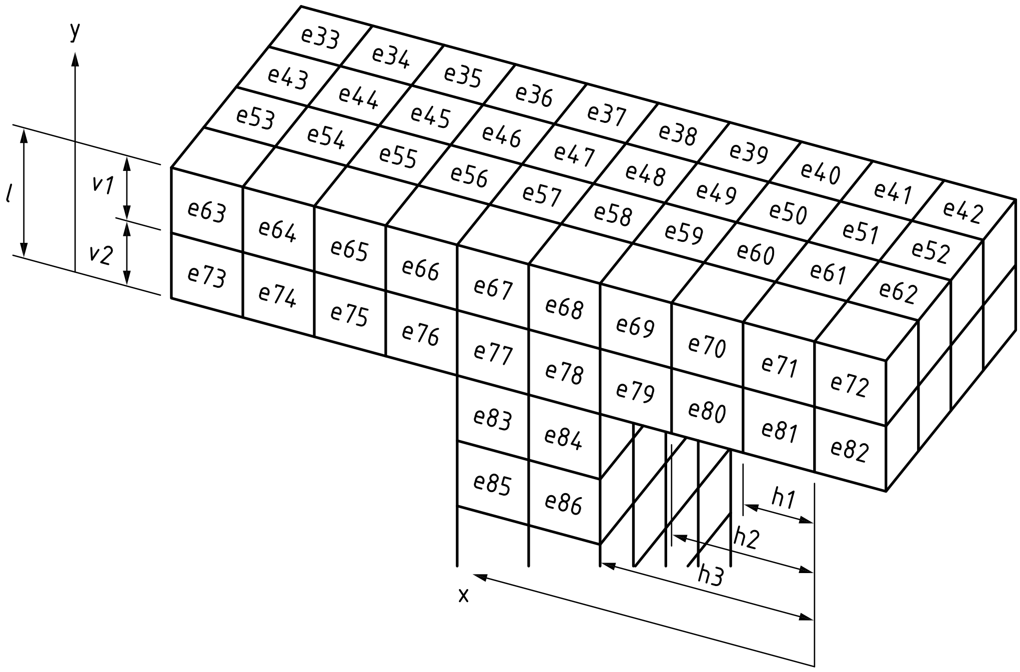
NOTE The nodes correspond to those on elements with the same face shown in Figure C.4.

Figure C.6 **Element numbers superimposed upon the weld mesh section in Figure C.3c)**



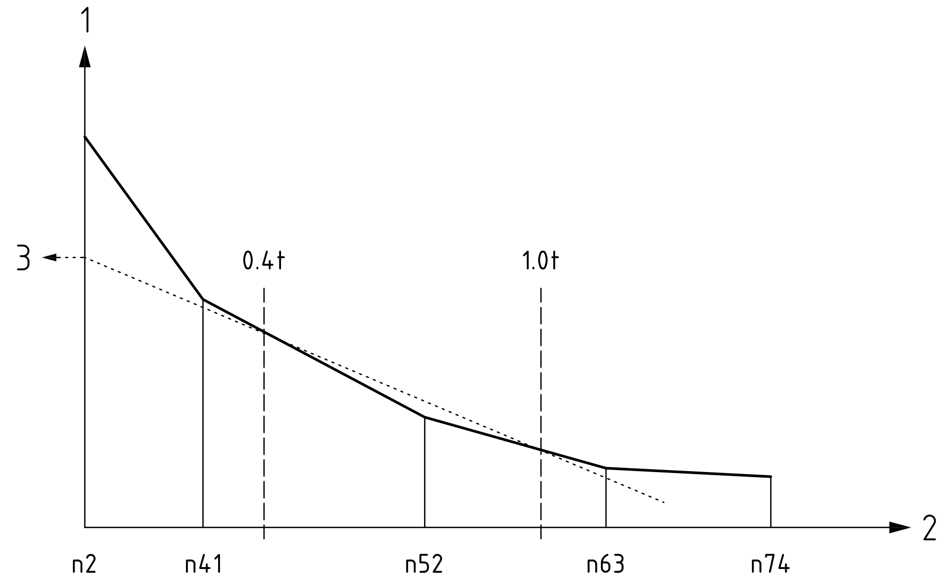
NOTE The nodes on this face correspond to those on elements with the same face shown in Figure C.7.

Figure C.7 **Element numbers superimposed upon the weld mesh section in Figure C.3e)**



NOTE The nodes on this face correspond to those on elements with the same face shown in Figure C.6

Figure C.8 **Calculation of the SSE stress at node n2 of the brick mesh shown in Figure C.3 to Figure C.5**



Key

1 Stress

2 Distance from weld toe on surface

3 Structural stress at node n2

A typical shell model of the same detail is shown in Figure C.9. This mesh illustrates the use of inclined weld elements. More details on the use of shell elements for calculations of structural stresses are given in **C.5**. The weld toe is represented by the node line n2 to n10 (Figures C.10 and C.11). The elements representing the weld metal and the material beneath the weld are shown in Figure C.12. Elements adjacent to these in the main section are given in Figure C.13.

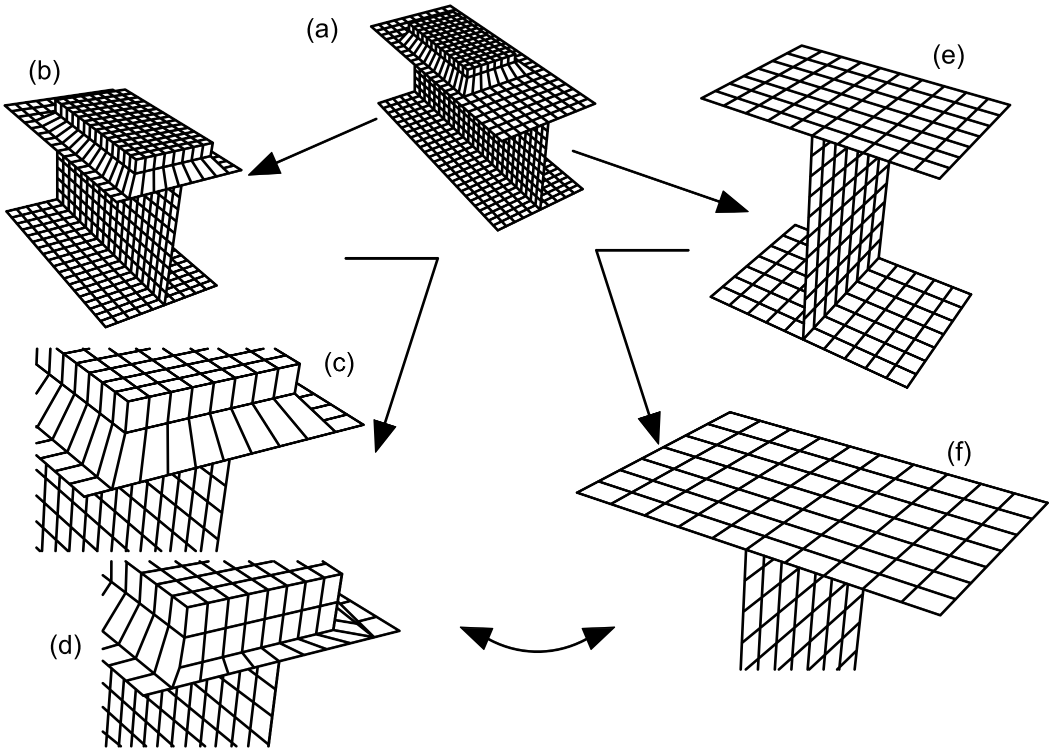
Surface stress extrapolation can be used to calculate the structural stress at node n2. The stress distribution along the line of n2 to n79 in Figure C.11 is used to estimate the stresses at 0.4t and 1.0t from the weld toe. The stress should be chosen consistently on the side of the shell that represents the top face of the beam section. The extrapolation from these two values should be made to the location of node n2 (Figure C.14).

The SSE method assumes that a stress at approximately 0.4t from the weld toe is not influenced by the stress concentration effect of the welded joint. A definition of thickness, t, for the I beam was made in Figure C.1. It should be assumed that the flange thickness is used where the cover plate is welded to the flange. A considerably lower SSE stress would be estimated if the whole beam depth was assumed for locations where the cover plate weld passes over the beam web (nodes n5, n6 and n7 of Figure C.4). The structural stress over the web is likely to be higher (Figure C.1); the flange thickness, t, should be used for SSE calculations here.

The structural stress on the side of the shell elements associated with the weld toe being assessed should be used for SSE.

If the surface stresses are obtained from strain measurements, using electrical resistance strain gauges, the gauges should ideally be located at the extrapolation positions. Alternatively, a series of strain measurements might be made covering the range that includes the extrapolation locations, for example using multi-element strip strain gauges. In such cases, sufficient gauges should be used to enable a curve to be fitted to the results to establish the required strains at the extrapolation positions by interpolation. Depending on the extrapolation procedure adopted, strains (or strain ranges if they are measured under cyclic loading) should be substituted for stresses in Equation C.1 to Equation C.3. If the stress state is close to uni-axial, the resulting structural strain can be converted to stress simply by multiplying by the elastic modulus of the material. However, for greater accuracy, account should be taken of any bi-axiality and principal stresses established using strain gauge rosettes.

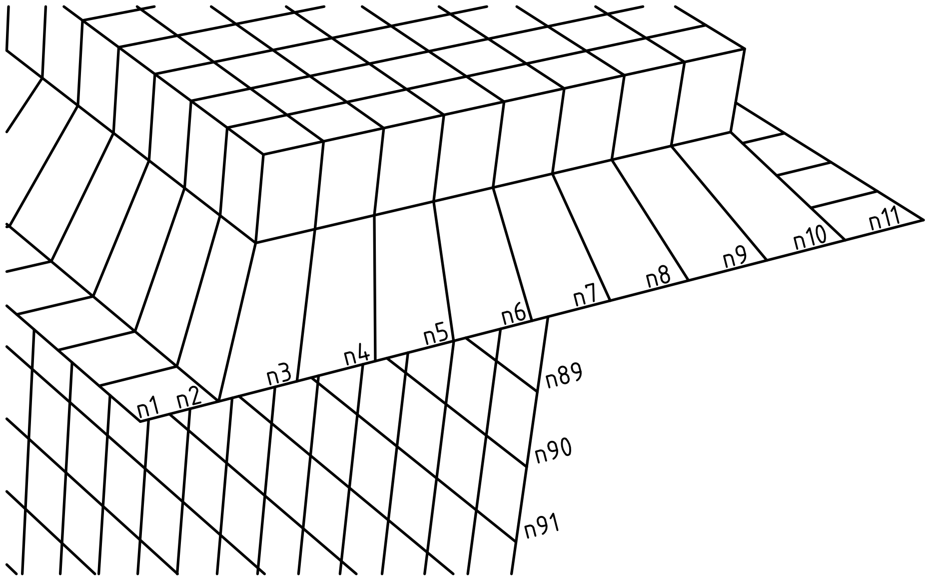
Figure C.9 **Possible shell element model of an I beam with a cover plate**



Key

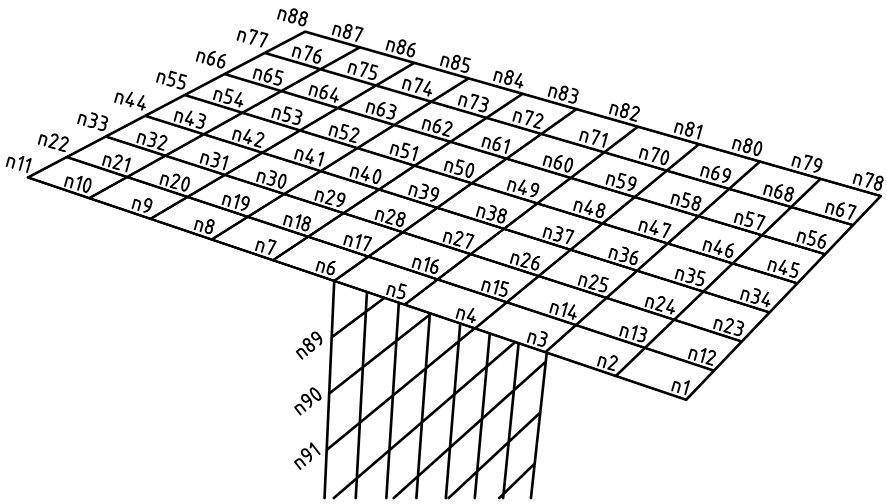
a)	Weld region and adjacent elements	d)	Close up of elements underneath the weld with some of the inclined "weld" elements removed
b)	Elements underneath the weld	e)	Elements beyond the weld
c)	Close up of elements underneath the weld	f)	Close up of elements beyond the weld

Figure C.10 **Node numbers superimposed upon the shell element weld mesh section in Figure C.9c)**



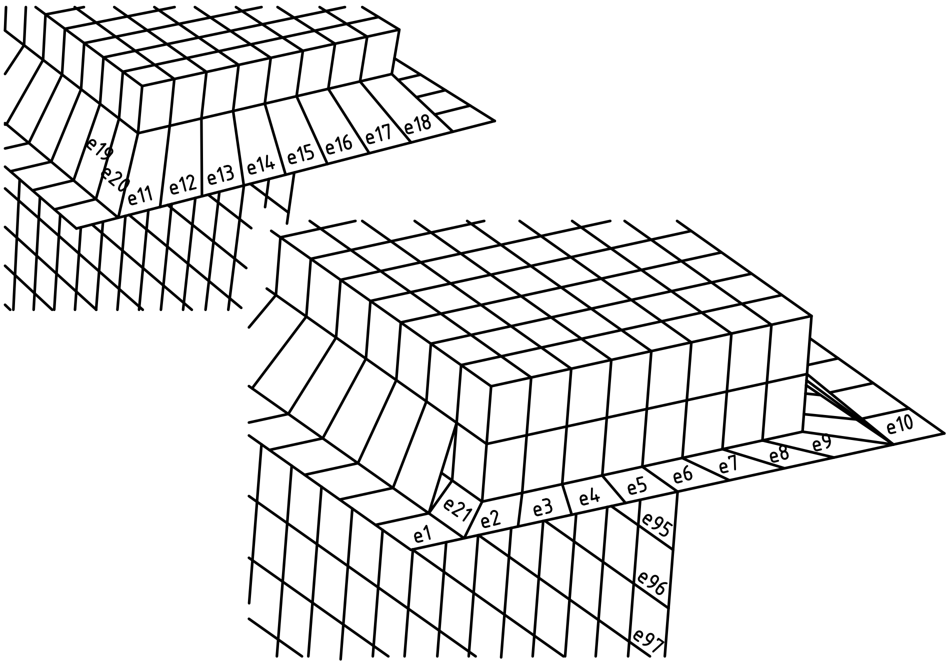
NOTE The nodes correspond to those on elements with the same face shown in Figure C.11.

Figure C.11 **Node numbers superimposed upon the shell element weld mesh section in Figure C.9f)**



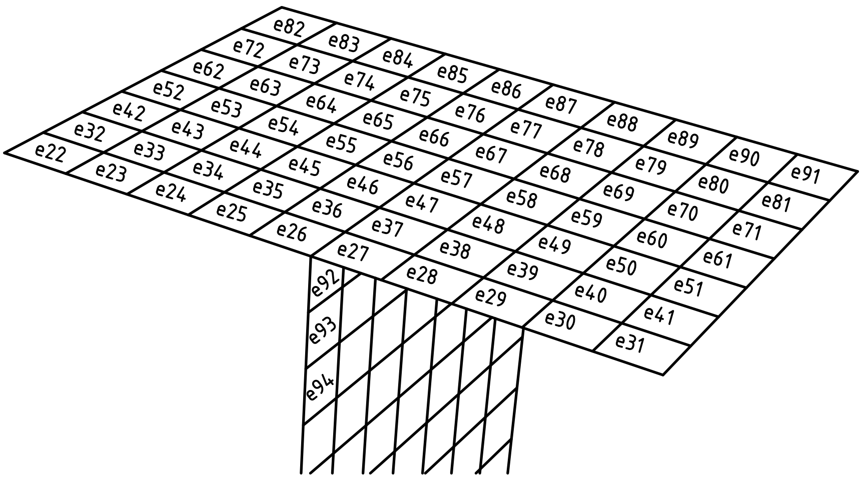
NOTE The nodes correspond to those on elements with the same face shown in Figure C.10.

Figure C.12 **Element numbers superimposed upon the shell element weld mesh section in Figure C.9c) and d)**



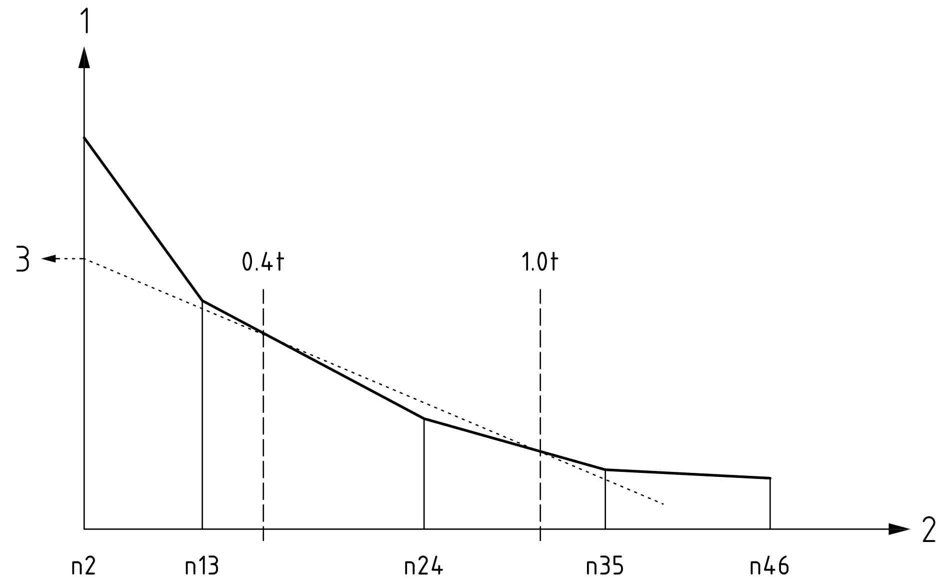
NOTE The nodes on this face correspond to those on elements with the same face shown in Figure C.13.

Figure C.13 **Element numbers superimposed upon the shell element weld mesh section in Figure C.9f)**



NOTE The nodes on this face correspond to those on elements with the same face shown in Figure C.12.

Figure C.14 **Calculation of the SSE stress at node n2 of the shell mesh shown in Figure C.9 to Figure C.12**



Key

1 Stress

2 Distance from weld toe through thickness

3 Structural stress at node n2

C.4.3 Hot-spot stress calculation using through-thickness stresses

C.4.3.1 General

Finite element analysis can be used to calculate the through-thickness distribution of stress as shown in Figure C.1. Therefore a force per unit length of weld and a moment per unit length of weld can be calculated. These values can be used with the section area per unit length and the section modulus per unit length to calculate the structural stress. This technique is called through thickness integration (TTI) because the stress distribution is integrated to calculate forces and moments.

Surface stress extrapolation and TTI both use stresses from FEA models to calculate the structural stresses. FEA also calculates forces. These are more fundamental quantities which offer an alternative method of calculating section forces and moments. Additional processing is needed to convert forces to loads per unit length of weld. The methods associated with this are described in this sub-clause and the technique is called nodal force (NF).

The TTI and NF methods estimate the structural stress from the distributions of forces and moments underneath the weld toe. For the NF method the nodes in the finite element model should lie on the plane normal to the surface. It is therefore important to decide the section thickness over which the calculation of force and moment is to be made. This decision is similar to the choice of thickness that is needed for SSE. Figure C.15 shows that the distribution of stress across the beam section is linear remote from the weld toe. The same structural stress is therefore calculated irrespective of the chosen region of integration. At the weld toe, there is a small region of stress concentration. This region is a significant proportion of the flange thickness. However, it is only a small proportion of the whole beam depth at the web. Integration across the whole web might not show a stress concentration because much of the stress distribution here is the background linear stress. The TTI and NF methods should therefore use an integration depth equal to the flange thickness even over the region of the web. For example the TTI or NF stress for node n5 (Figure C.4) should be calculated from nodal values at locations n5, n16 and n27 (the stress/nodal force for node n27 should be from elements in the flange (elements e68, e69 e78, e79) and not from element e84 which is in the web, Figure C.7).

A similar judgement should be made for other geometries, including the edge attachment shown in Figure C.16 (see **C.7**).

Figure C.15 **Stress distributions across sections of an I-beam with a cover plate**

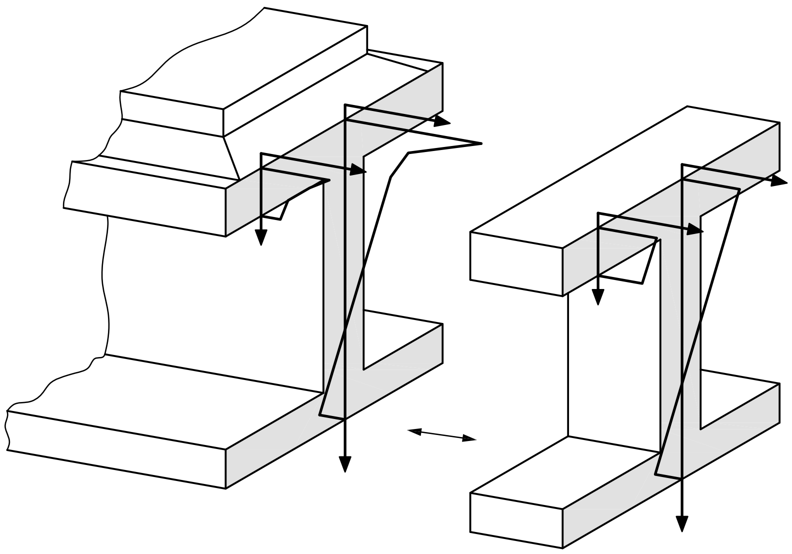
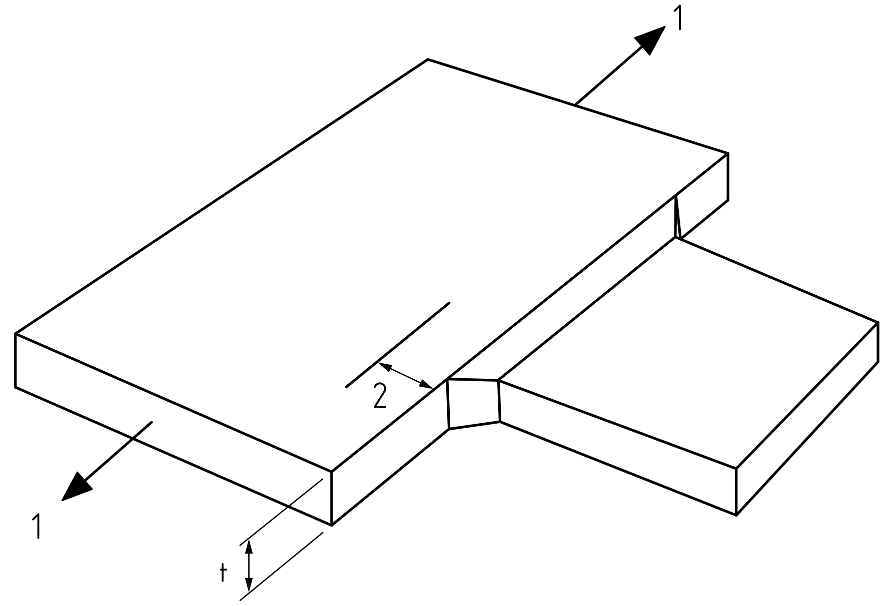


Figure C.16 **Region of TTI or NF integration for an edge attachment**

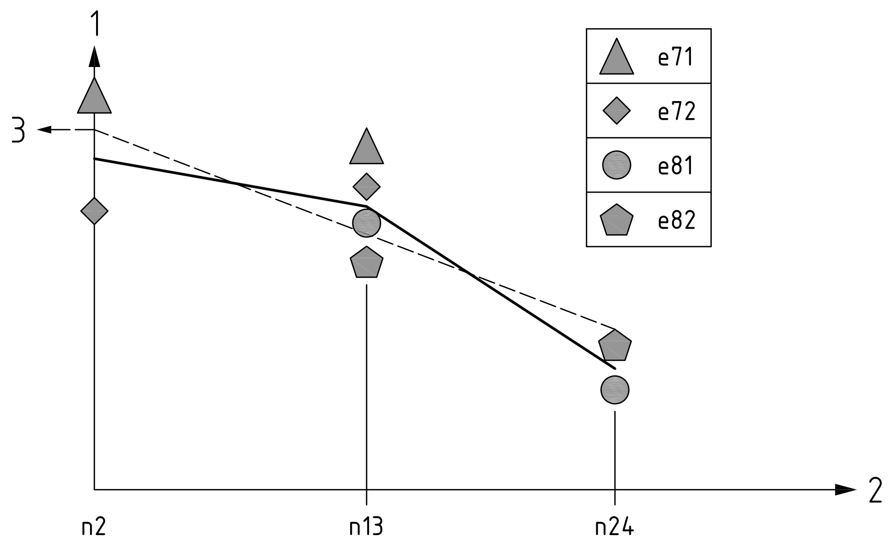


Key

1 Loading

2 TTI and NF region

Figure C.17 **Stress distribution through I beam flange underneath node n2 for solid mesh shown in Figure C.4, Figure C.5 and Figure C.7**



NOTE The legend shows which element the unaveraged stresses were from.

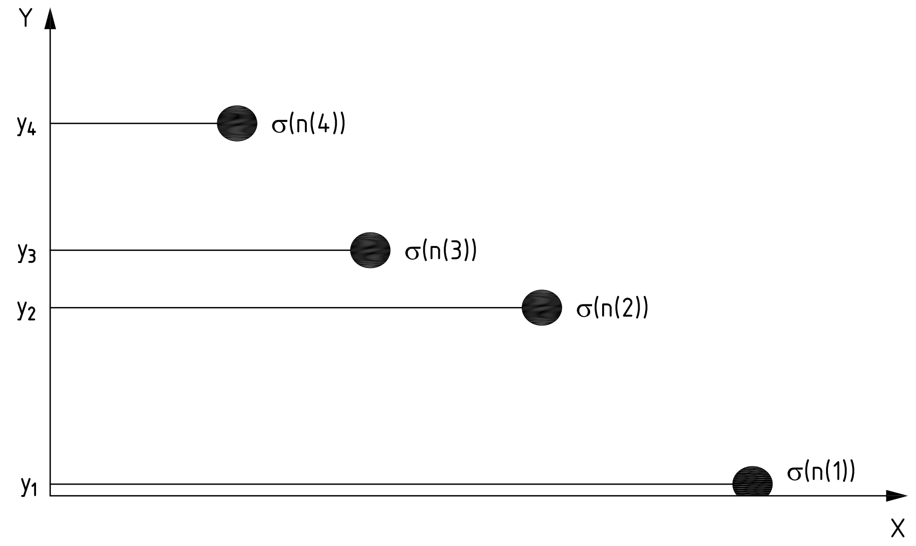
Key

1 Stress

2 Distance from the weld toe through thickness

3 Structural stress at node n2

Figure C.18 **Distribution of correctly averaged stresses plotted against distance y**



Distance y is given in Figures C.6 and C.7

Key

X Distance y

Y Stress

C.4.3.2 Hot-spot stress calculation using TTI

The TTI method is illustrated through a typical I beam mesh of brick elements as shown in Figure C.3. It shows a mesh and sections of mesh separated along the plane through the weld. The separation is shown in Figure C4 and Figure C5 where the nodes on the common face are identified. The weld toe is represented by nodes n2 to n10. The TTI stress at node n2 is calculated from the stresses at nodes n2, n13 and n24 (Figure C.5). The stress distribution from these nodes is shown in Figure C.17. There are multiple results because each node is connected to more than one element. The elements used for the TTI calculation are shown in the key. These are the elements in the flange at the weld toe but not underneath it (see Figure C.7). Elements under the weld experience less stress and should therefore be ignored.

Figure C.17 shows the calculated stress distribution using the correct averaging of stress at each node (continuous line). The equivalent linear distribution is also shown (dotted) line. The value of stress at node n2 from the linear distribution is equal to the TTI structural stress at this location.

The TTI structural stress has membrane and bending components. Using the nomenclature below and shown in Figure C.18, they can be calculated as follows:

- Correctly averaged stress at node n(i) is σ[n(i)].
- Height from the underside of the plate, *y* (Figure C.5, Figure C.6) of node n(i) is y_i.
- Thickness of the part at the location is *t*.
- Membrane stress σ_m is calculated using Equation C.4.

$$\sigma_m = \frac{\Sigma\{\sigma[n(i)] + \sigma[n(i+1)]\}\,.\,(y_{i+1} - y_i)}{2t} \qquad (C.4)$$

The bending stress σ_b is calculated using Equation C.5.

$$\frac{\sigma_b t^2}{6} = \sum \left(f(\sigma(n(i),\sigma(n(i+1)),y_i,y_{i+1})) - \frac{\sigma_m t^2}{2} \right. \qquad (C.5)$$

where

$$f(\sigma(\mathrm{n(i)},\sigma(\mathrm{n(i+1)}),\mathrm{y_i},\mathrm{y_{i+1}})$$

$$= \frac{1}{6}\left[\sigma(\mathrm{n(i)})\left(-2.\mathrm{y_i^2} + \mathrm{y_i y_{i+1}} + \mathrm{y_{i+1}^2}\right) + \sigma(\mathrm{n(i+1)})\left(-\mathrm{y_i^2} - \mathrm{y_i y_{i+1}} + 2.\mathrm{y_{i+1}^2}\right)\right] \qquad (C.6)$$

The TTI hot-spot structural stress is then given by Equation C.7.

$$S_H = (\sigma_b + \sigma_m) \qquad (C.7)$$

Shell elements generally provide structural stresses as output. The structural stress on the side of the shell associated with the weld toe should be used.

C.4.3.3 Hot-spot stress calculation using NF

The NF method is illustrated through a typical I beam mesh of brick elements as shown in Figure C.3. The diagram shows a mesh and sections of mesh separated along the plane through the weld. The separation is shown in Figure C.4 and Figure C.5 where the nodes on the common face are identified. The weld toe is represented by nodes n2 to n10. The NF stress at node n2 is calculated from the nodal forces at nodes n2, n13 and n24 (Figure C.5). The nodal force at each node is summed from elements that are at the weld toe but not underneath it. For example, the nodal force at node n2 should the summation of the nodal forces at that node from elements e71 and e72 (Figure C.7).

A manipulation of the forces over a length of weld is needed to convert the nodal forces into structural stresses. The nodal forces at each section (for example at x = h2, Figure C.7 and Figure C.8) are converted into a section force Fi and a section moment Mi at that section as follows (Figure C.19):

$$Fi = \Sigma NFi \qquad (C.8)$$

$$Mi = \Sigma NFi \times (y_i - (t/2)) \qquad (C.9)$$

Where the nodal force at node n(i) is NFi.

The section forces and section moments should then be converted to a distribution of line forces fi (or force per unit length of weld) and a distribution of line moments mi (or moment per unit length of weld). The membrane and bending stress can be calculated as follows:

$$\sigma_m = \left(\frac{fi}{t}\right) \qquad (C.10)$$

A1⟩

$$\sigma_\mathrm{b} = \left(\frac{6mi}{t^2}\right) \qquad (C.11)$$

⟨A1

The NF hot-spot structural stress is then calculated using Equation C.7.

The relation between section and line forces and moments is given in Equation C.12 for linear elements and in Equation C.13 for quadratic elements:

[A1⟩

$$\begin{bmatrix} F1 \\ F2 \end{bmatrix} = \begin{bmatrix} \frac{h1}{3} & \frac{h1}{6} \\ \frac{h1}{6} & \frac{h1}{3} \end{bmatrix} \begin{bmatrix} f1 \\ f2 \end{bmatrix} \quad \text{and} \quad \begin{bmatrix} M1 \\ M2 \end{bmatrix} = \begin{bmatrix} \frac{h1}{3} & \frac{h1}{6} \\ \frac{h1}{6} & \frac{h1}{3} \end{bmatrix} \begin{bmatrix} m1 \\ m2 \end{bmatrix} \tag{C.12}$$

⟨A1]

$$\begin{bmatrix} F1 \\ F2 \\ F3 \end{bmatrix} = \frac{1}{15} \begin{bmatrix} 2h1 & h1 & \frac{-h1}{2} \\ h1 & 8h1 & h1 \\ \frac{-h1}{2} & h1 & 2h1 \end{bmatrix} . \begin{bmatrix} f1 \\ f2 \\ f3 \end{bmatrix} \quad \text{and} \quad \begin{bmatrix} M1 \\ M2 \\ M3 \end{bmatrix} = \frac{1}{15} \begin{bmatrix} 2h1 & h1 & \frac{-h1}{2} \\ h1 & 8h1 & h1 \\ \frac{-h1}{2} & h1 & 2h1 \end{bmatrix} . \begin{bmatrix} m1 \\ m2 \\ m3 \end{bmatrix} \tag{C.13}$$

where

$F1$	is the section force at corner node 1
$M1$	is the section moment at corner node 1
$F2$	is the section force at [A1⟩ corner node 2 in (C.12) or mid-side node 2 in (C.13) ⟨A1]
$M2$	is the section moment at [A1⟩ corner node 2 in (C.12) or mid-side node 2 in (C.13) ⟨A1]
$F3$	is the section force at corner node 3
$M3$	is the section moment at corner node 3
$f1$	is the associated line force at corner node 1
$m1$	is the associated moment at corner node 1
$f2$	is the associated line force at [A1⟩ corner node 2 in (C.12) or mid-side node 2 in (C.13) ⟨A1]
$m2$	is the associated moment at [A1⟩ corner node 2 in (C.12) or mid-side node 2 in (C.13) ⟨A1]
$f3$	is the associated line force at corner node 3
$m3$	is the associated moment at corner node 3
$h1$	is the element side length between the corner nodes

Typical h values are shown in Figure C.6 and Figure C.7.

A similar set of matrix equations should be created for each element along the weld toe. A master matrix should then be assembled, the coefficients being added where two elements contribute to one node. The master matrix is then inverted to obtain the line forces and moments along the weld toe.

C.5 Accuracy and limitations of hot-spot stress analysis

C.5.1 Use of brick element models for hot-spot stress calculations

Continuum elements tend to have only linear degrees of freedom. Two-dimensional (2D) models and three-dimensional (3D) models have two and three degrees of freedom per node respectively. Three-dimensional models are constructed from volumetric elements that are generally called bricks.

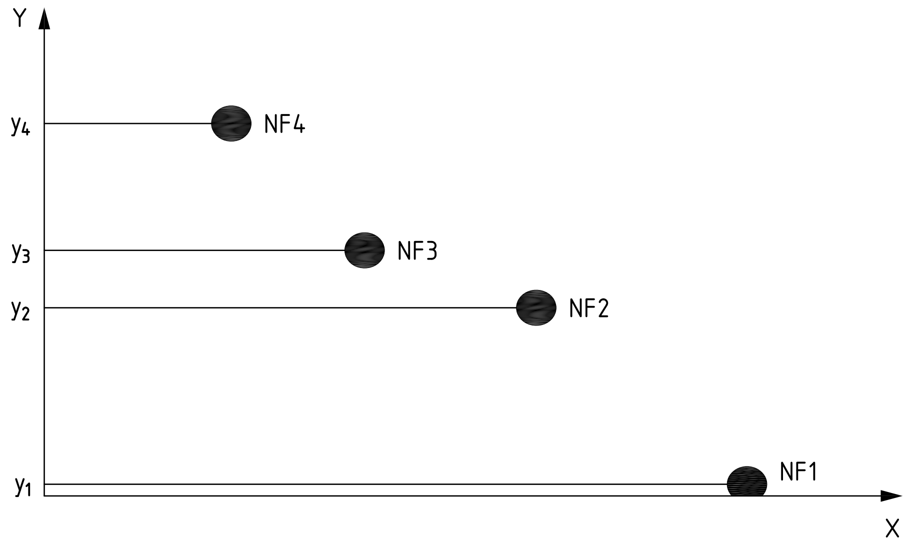
The assumed distributions of displacement in continuum elements are generally either linear or quadratic. Elements are therefore called linear or quadratic respectively. Bending modes of deformation tend to produce curved deformed shapes which quadratic elements are able to represent. However, linear elements are generally unable to represent bending accurately. The stress distributions at fatigue crack initiation sites often have a bending component, so linear elements might not be suitable for accurate hot-spot stress calculations (see **C.5.3**).

The use of quadratic 3D brick element models provides a relatively detailed indication of the distribution of stresses at structural details. Brick models can accurately represent the geometry of a detail like a welded joint, including the weld profile. This contrasts with shell models which represent the centre-plane of thin sections. As a result, accurate calculation of the stress distribution at a potential fatigue crack initiation site, like the weld toe, is possible using a very fine brick model. The need for fine meshes, however, means that brick models tend to be larger and more time consuming to prepare and run than equivalent shell element models (**C.5.2**).

The size of brick models can be reduced and the complexity of the assumed geometry simplified, but these changes affect the accuracy of the model. Modelling accuracy is discussed in greater detail in **C.5.3**. Some possible modelling techniques are described below. The dimension of the elements at the fatigue initiation site under consideration relative to the underlying plate thickness is important. Figure C.20 shows the dimension, *f,* of an element normal to a weld toe. The recommendations in **C.5.3** are based on the condition $f \leq 2t$.

The 3D brick model shown in Figure C.3 to Figure C.7 has elements that represent the overfill of the fillet weld around the cover plate. Figure C.21 shows an alternative mesh where the overfill has not been modelled. Instead, the cover plate is connected to the flange via a ring of elements around the cover plate perimeter as shown by the shading.

Figure C.19 **Distribution of correctly averaged nodal forces plotted against distance y**



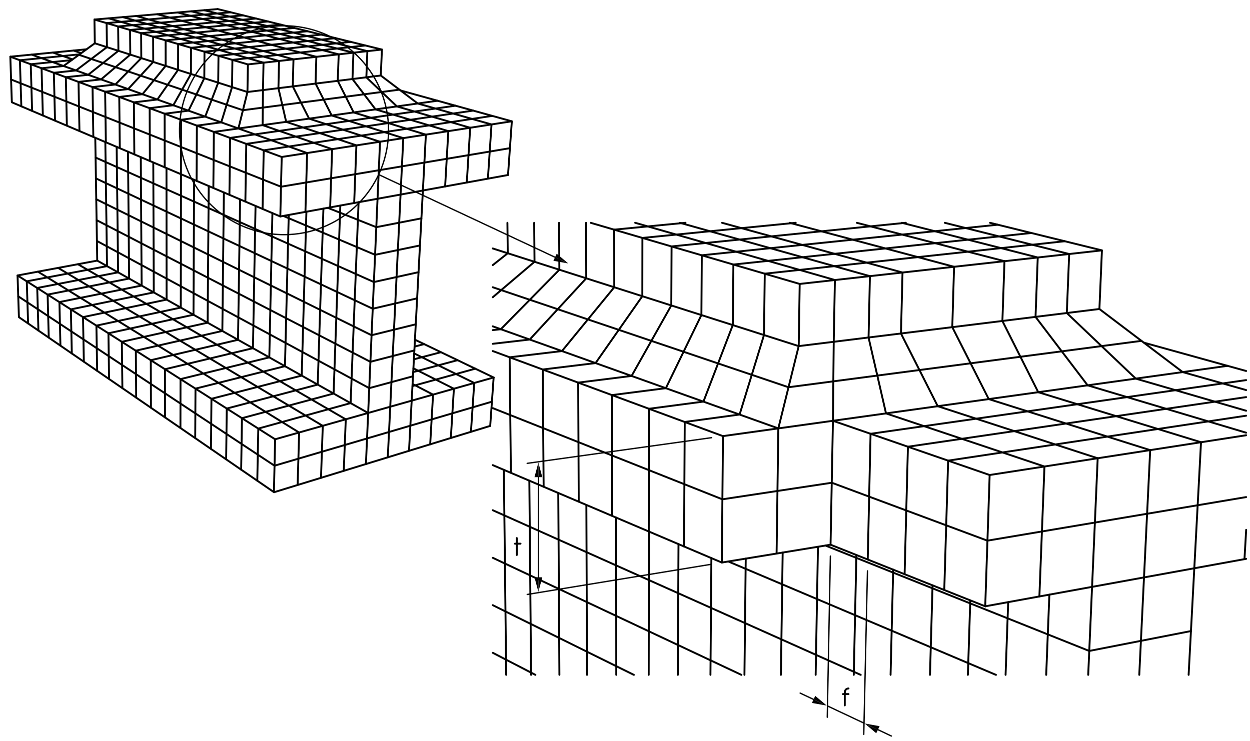
Distance y is given in Figures C.7 and C.8.

Key

Y Distance y

X Nodal force

Figure C.20 **Brick element mesh with definition of weld toe element size (f) and plate thickness (t)**



The definition is added to the inset where elements have been removed to illustrate the weld toe mesh.

Figure C.21 **Brick element model of an I beam showing the region of connection between the connectivity representing the joining surface (shaded) to a cover plate when the weld overfill is not modelled**

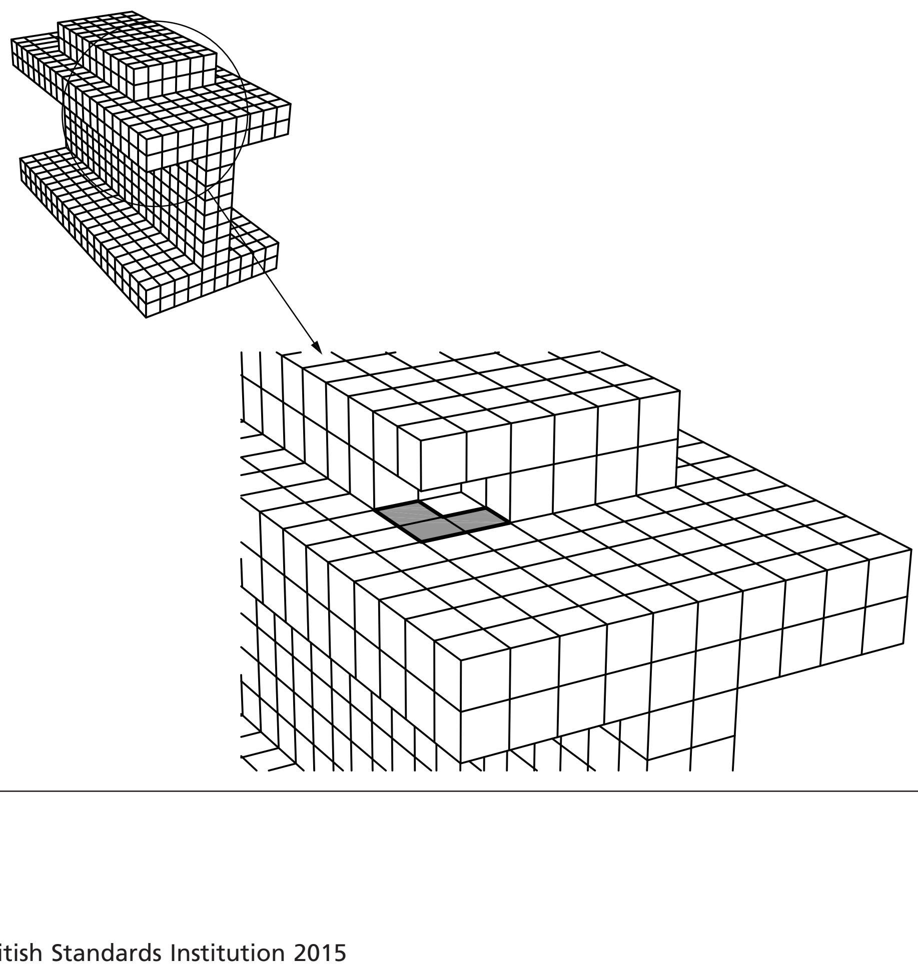
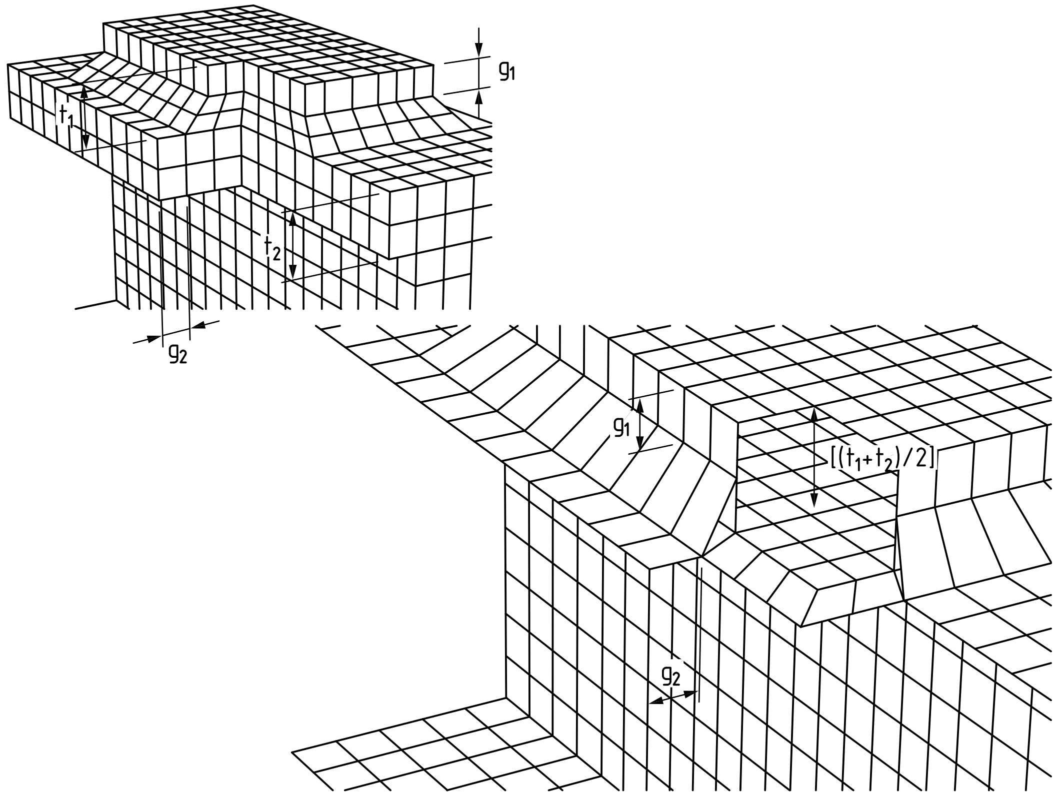
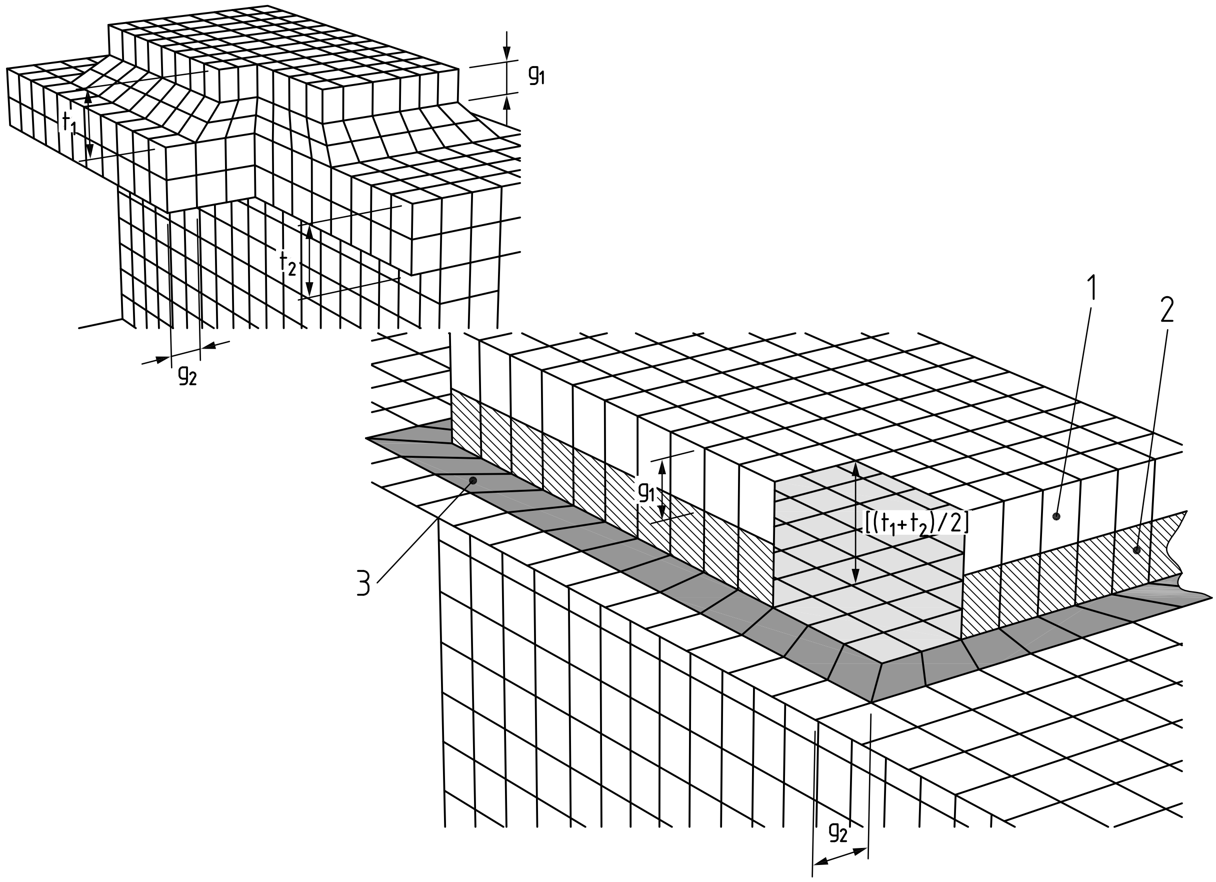


Figure C.22 **Dimensions used for inclined element representation of a fillet weld**



The inclined shell elements should have a thickness equal to the weld throat.

Figure C.23 **Dimensions used for thicker element representation of a fillet weld**



The thickness of the elements is based upon a weld throat thickness of w.

Key

1 Thickness = t_1

2 Thickness = (t_1 + w)

3 Thickness = (t_2 + w)

C.5.2 Use of shell element models for hot-spot stress calculations

Shell elements are part of a group of elements called structural elements. Meshing of structures with shell elements does not require an explicit representation of the thickness of the plates they represent. The thickness is a property which determines the stiffness of the shell. The through-thickness distribution of stresses is estimated on the basis of both linear and rotational degrees of freedom. The NF method uses the nodal moments of rotational degrees of freedom to estimate the distribution of the bending component of structural stresses. It is not possible to represent the geometry and stiffness of relatively short structural details such as welds explicitly using shell elements. The stiffness is that of a section of plate of the same size as the shell element.

Fillet weld details may be represented in two ways. One method is based upon inclined weld elements meant to represent the centre-plane of the fillet weld. Examples are shown in Figure C.10 and Figure C.12. Figure C.12 shows some of the elements representing the fillet (e.g. elements e11 to e20) removed so that the unfused land underneath the weld can be seen. Figure C.22 shows a suggested geometry to be used for this sort of shell model for the I beam with a cover plate.

Another possible representation of fillet welds is based upon a thickening of the elements in the plate underneath the fillet weld as shown in Figure C.23.

The accuracy of these methods has been assessed and is presented in **C.5.3** together with a plane mesh, without explicit representation of the weld. The plane mesh is similar to the geometry shown in Figure C.23, except that the section thicknesses are not modified to represent the stiffening of the weld.

The assumed distributions of displacement in structural elements in FEA are generally either linear or quadratic. Elements are therefore called linear or quadratic.

C.5.3 Accuracy of modelling and post-processing methods

The accuracy of the various methods for the calculation of hot-spot stresses has been determined through the analysis of some test cases [20]. These all considered weld toes or ends (referred to as type “a” in Figure C.2). The accuracy of each method was determined by comparison with the results obtained from benchmark reference cases in which the structural hot-spot stresses were determined using SSE from fine mesh brick models (four elements through the plate thickness). The results of this benchmark analysis led to the recommendations detailed in Table C.1. These are only valid for element sizes, f (Figure C.20), up to twice the underlying plate thickness, t. For brick models it is assumed that there are at least two elements through the plate thickness.

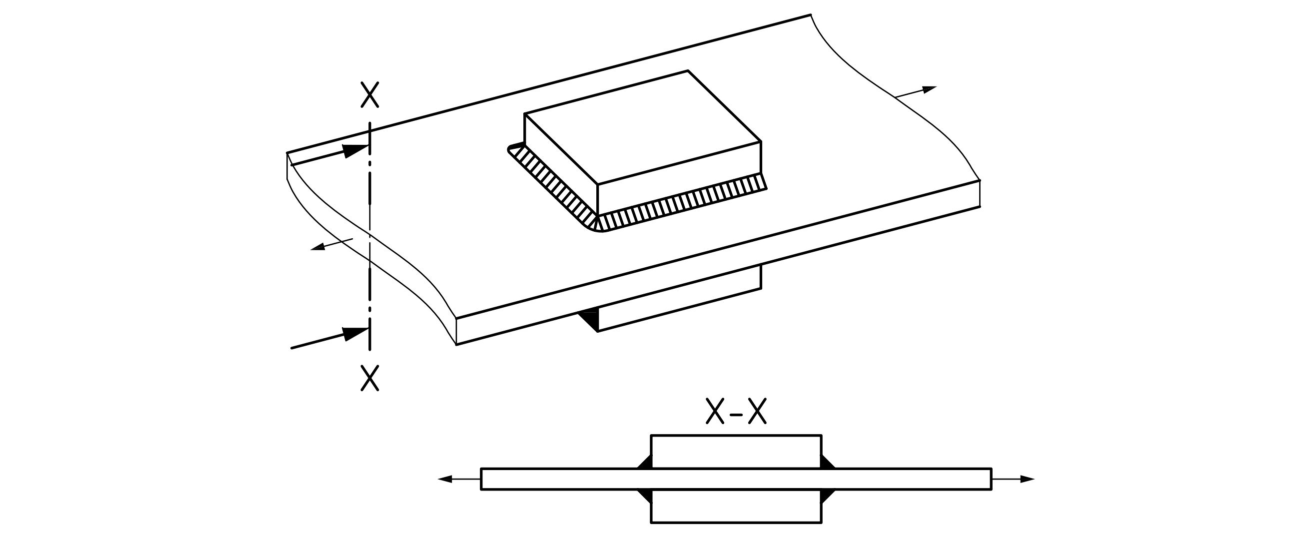
The benchmark analysis upon which the recommendations in Table C.1 are based did not cover every circumstance. Therefore a benchmark study should be conducted in any case where there is a need for additional confidence in the chosen method of calculating the hot-spot stress.

C.5.4 Limitations of the hot-spot stress calculation methods

Attention is drawn to the problem of using hot-spot stress calculation methods that are related to the plate thickness when considering the toe or end of an edge attachment weld (type “b” in Figure C.2). At this stage only SSE should be used for determining the hot-spot stress in such cases. As this is essentially a 2-dimensional problem, any of the FE modelling methods is suitable, provided the element sizes in the region being assessed meet the conditions described in **C.4.2**.

There are also some welded joint geometries for which the hot-spot stress determined by the methods described in this annex is under-estimated, such that it is virtually the same as the nominal stress. Consequently, use of the hot-spot stress design class (D) rather than the nominal stress class would be non-conservative. Particular problems arise with simple cruciform or T-joints between plates for which a cross-section normal to the weld toe being considered is symmetrical, as in Figure C.24. The hot-spot stress at the weld toe is known to increase with increase in attachment length, but this is not detected using any of the three calculation methods in this annex.

Figure C.24 **Example of symmetrical welded joint for which hot-spot stress is underestimated using methods in this annex**



A new method that has been shown to include the stress concentrating effect of the attachment size is to equate the hot-spot stress to the stress 1 mm below the weld toe [21]. A disadvantage of the method is the need for solid FE models with very small elements, although it is claimed that elements down to 1 mm in size are sufficient. An extensive database from a range of welded joints has been shown to be consistent with class D on the basis of the 1 mm stress [21].

An alternative approach is to assume that the plate thickness is effectively less than its actual value so that greater weight is given to stresses local to the weld toe. This approach is recommended by Dong et al [22] for assessing any symmetrical (as described above) joint assuming that the effective plate thickness is half the actual. Thus, SSE by linear extrapolation would use stresses located 0.2t and 0.5t from the weld toe, while calculation of the equivalent membrane and bending stresses for the TTI and NF methods would be based on half the plate thickness.

A similar problem can arise with shell element FE models if members that are large enough to attract loading cannot do this because they are modelled with two-dimensional elements.

Some of these deficiencies in the methods of calculating the hot-spot stress are the subject of research and improvements might appear in future. Meanwhile, unless the nominal stress-based approach can be used, expert advice should be sought for assistance in areas of doubt or concern. As a guide to the validity of a calculation of the hot-spot stress for a fillet weld, it should be at least 15% higher than the nominal stress at the weld toe.

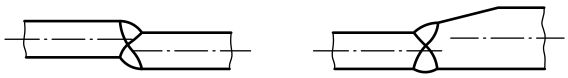
C.6 Misalignment and distortion

A source of stress concentration in all types of structures can be linear or angular misalignment at joints (see Figure C.25 and **B.5.2.1**). This can arise from poor attention to assembly or erection procedures or from welding distortion. Stresses can be set up by the local bending effects induced by this misalignment and these should be taken into account.

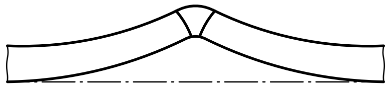
NOTE References [23–25] cover this in some detail.

Circular section pipes and vessels are particularly sensitive as offset and angular distortion in the longitudinal seams and ovality can individually or in combination magnify the nominal or hot-spot stress by an amount which can seriously degrade the fatigue performance under pulsating pressure. Similarly, some degree of linear and/or angular misalignment is virtually unavoidable in butt welded joints between co-planar plates and cruciform joints. Unless the extent of misalignment is already allowed for in the *S-N* curve, it should be included in the calculation of the hot-spot stress. It is rarely practical to include misalignment in a FE model. Therefore, if relevant, the calculated value of the hot-spot stress should be multiplied by k_m , the stress magnification factor due to misalignment (see **B.5.2.1**), to determine the actual value.

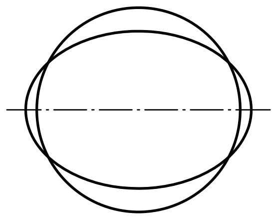
Figure C.25 **Types of misalignment and distortion**



a) Linear misalignment (plate centre-line offset)



b) Angular distortion



c) Ovality

Table C.1 **The performance of structural stress calculation procedures SSE, TTI and NF for assessing hot-spot type "a" weld toes or ends**[A)]

Element type	Weld modelling method	Hot-spot stress determination method	Comment
Quadratic brick	Weld modelled	SSE or NF	TTI if benchmark study
Quadratic brick	Weld not modelled	SSE or NF	TTI if benchmark study
Linear brick	Weld modelled	NF	
Linear brick	Weld not modelled	NF	Benchmark study required for assessing end of longitudinal weld
Quadratic shell	Weld modelled using inclined weld elements	SSE, TTI or NF,	Benchmark study recommended for assessing weld toes by TTI or NF
Quadratic shell	Weld modelled using thicker elements	SSE, TTI or NF	Benchmark study recommended for assessing weld toes by TTI or NF
Quadratic shell	Weld not modelled	SSE, TTI or NF	
Linear shell	Weld modelled using inclined weld elements	NF	Benchmark study recommended for assessing weld toes
Linear shell	Weld modelled using thicker elements	Not recommended	
Linear shell	Weld not modelled	NF	TTI if benchmark study

[A)] Applicable for elements sizes $f/t \leq 2$. Reference case was FE model with 0.25t quadratic brick mesh and weld included. Structural hot-spot stress was calculated using SSE.

Annex D (normative) Guidance on the use of fracture mechanics

D.1 Background

This annex provides guidance on the use of fracture mechanics methods for the fatigue assessment of structures or components subjected to high cycle fatigue conditions, in situations where the normal fatigue strength assessment methods in this British Standard might be unreliable or inappropriate.

In general, fracture mechanics is not suitable for calculating precise fatigue strengths or lives as the results are largely dependent upon the assumptions made, (e.g., the values of the constants in the crack growth equation, the size of the initial flaw(s) and the shape of the resulting fatigue crack, for example, for a crack at a weld toe, whether it is semi-elliptical or straight-fronted). Not all of this information is available at the design stage. Therefore, if the objective is to define a particular fatigue strength or life, assumptions made should be very conservative.

Fracture mechanics can, however, be a useful method for carrying out parametric studies, where the objective is to define the relative influence of a particular set of variables. In that situation all the variables, except the one under consideration, can be held constant and its influence can be evaluated.

The guidance in this annex does not replace the normal fatigue assessment procedures outlined in this British Standard when such procedures are applicable. For example, they are not intended to be used as a method to circumvent the normal requirements for good workmanship.

Some typical situations in which the normal procedures might be inappropriate and in which the use of fracture mechanics might be helpful are as follows.

a) When assessing the fitness for purpose of a structure known to contain flaws whose size, shape and distribution are outside normally acceptable limits (see **14.3.4**) but which would be difficult to repair.

NOTE In this case see BS 7910.

b) When the effects of relatively minor variations in the geometrical or stress parameters for a given detail are being studied.

c) When the joint detail under consideration is unusual and is not adequately represented by one of the standard joint classifications, or when a joint is subjected to the influence of another stress concentration.

d) When defining the frequency of in-service inspections.

e) When assessing the remaining fatigue life of a structure in which fatigue cracks already exist.

f) When determining the damaging effect of stress ranges below the constant amplitude fatigue limit in the calculation of fatigue life under spectrum loading.

In the case of item e) the structure would contain cracks whose sizes have to be determined by measurement and the sizes assumed have to allow for possible errors in such measurements. In other situations it has to be assumed that small, but unmeasurable, flaws exist at points of stress concentration (e.g. at weld toes) and that it is from them that fatigue cracking can originate. The size of such flaws should therefore be assumed.

In general this annex covers the application of fracture mechanics to fatigue cracking from a weld toe through the stressed member (Figure D.1) or from a weld root through the weld throat (Figure D.2). Reference should be made to BS 7910 for consideration of fatigue cracking from embedded flaws. The procedure recommended in **D.3** to **D.8** is based upon the principles of linear elastic fracture mechanics, similar to that in BS 7910, which also contains more detailed guidance and relevant design data.

D.2 Symbols and units

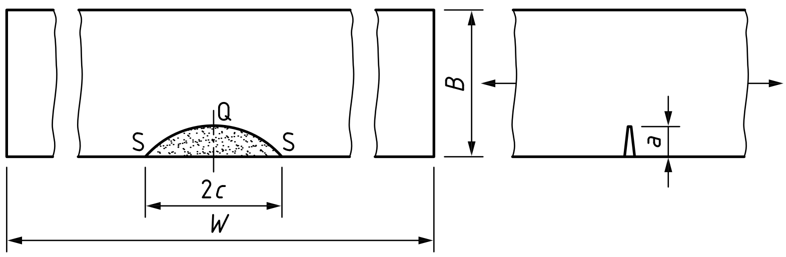
For the purposes of this annex only, the following symbols and units, (which are consistent with BS 7910), are used.

a	Measure of current crack size (length or depth) [2)] (in mm)
a_f	Final value of crack size[2)] (in mm)
a_i	Initial value of crack size[2)] (in mm)
A	Constant in the crack propagation equation
B	Plate thickness (in mm)
c	Half the surface length of a semi-elliptical surface crack of depth a (see Figure D.1) (in mm)
da/dN	Rate of fatigue crack propagation (mm/cycle)
F_m, F_b	Functions of crack size and shape and the proximity of the crack tip to free surfaces for membrane and bending stresses respectively
h	Weld leg length (see Figure D.2) (in mm)
L	Overall length from weld toe to weld toe of an attachment, measured in the direction of the applied stress (in mm)

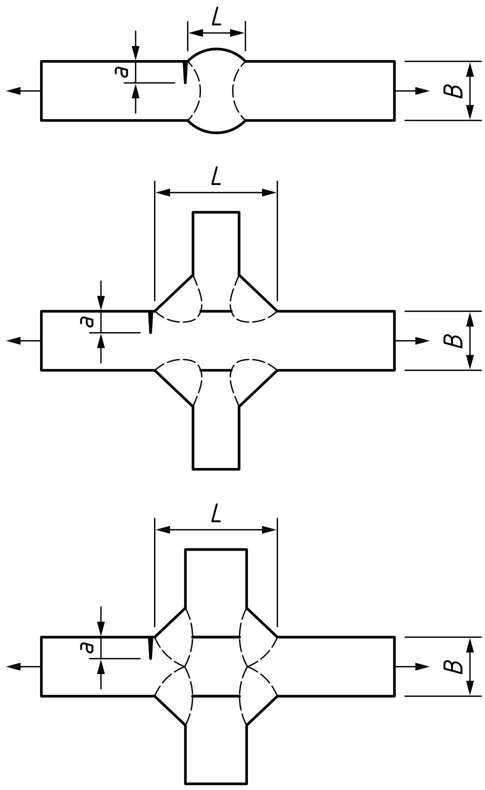
[2)] Length and depth are measured in the direction of propagation.

m	Constant in the crack propagation equation
M_b	Correction factor on stress intensity factor dependent on crack shape and size for bending stress
M_m	Correction factor on stress intensity factor dependent on crack shape and size for membrane stress
M_k	Magnification factor on stress intensity factor to allow for the presence of a stress concentration, such as weld toe (suffices K_m or K_b used to indicate membrane or bending)
N	Number of cycles (in cycles)
N_i	Number of cycles to crack initiation (in cycles)
ΔK	Range of stress intensity factor at the tip of the crack (in N·mm$^{-3/2}$)
ΔK_{th}	Threshold value of ΔK for crack propagation (in N·mm$^{-3/2}$)
φ_o	Complete elliptic integral of the second kind

Figure D.1 **Flaw dimensions**

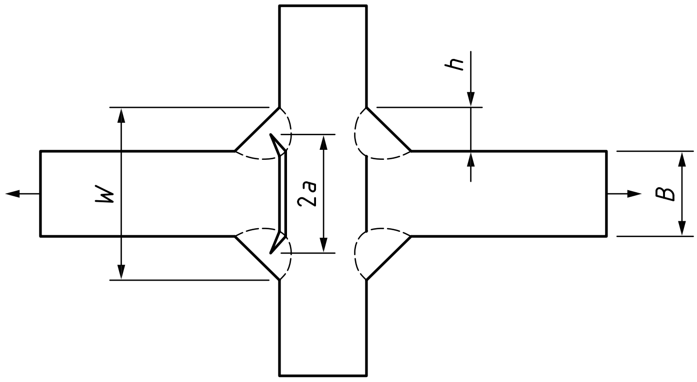


a) Semi-eliptical surface crack



b) Dimensions of joints with cracks at the weld toe

Figure D.2 **Transverse load-carrying cruciform joint**



D.3 General background

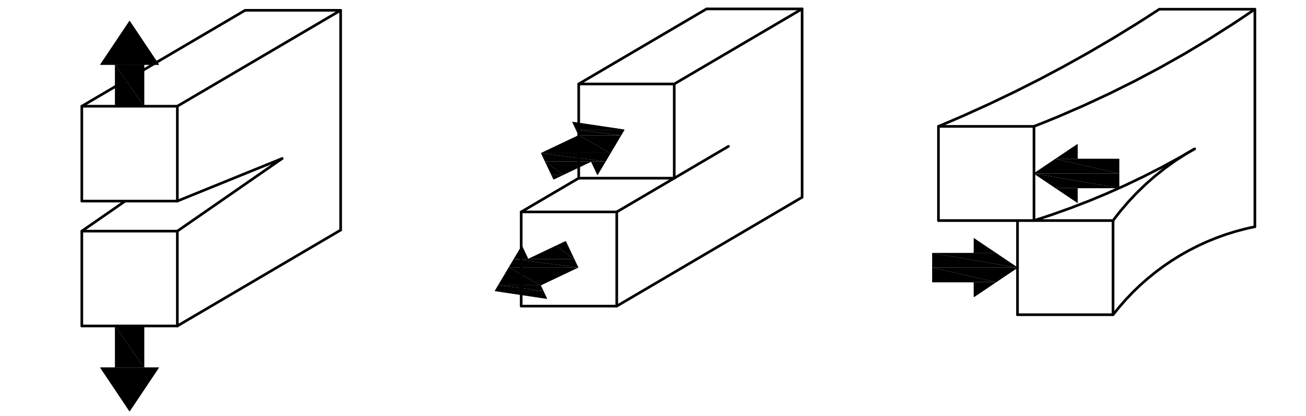
As outlined in **12.1**, fatigue cracks in welded joints can originate either from the toe or root of the weld, depending on the type of joint, or from planar or non-planar flaws in the weld. Cracks originating from the weld toe normally initiate at small flaws while cracks originating from the root often start from areas of deliberate lack of penetration; in both cases the initiating feature can therefore be regarded as a planar discontinuity. Most fatigue cracks in welded joints can therefore be regarded as starting from a pre-existing planar flaw and their behaviour can be described by the use of fracture mechanics analysis.

The objective of such analysis is to calculate the life by integrating the relevant crack growth law. In doing so it is assumed that the real flaws can be idealized as sharp-tipped cracks, which propagate at a rate, da/dN, which is a function of the range of the stress intensity factor, ΔK.

Fracture mechanics is applicable to the behaviour of cracks under loading that can cause them to propagate, including fatigue loading. There are three possible modes of crack opening, depending on the type of loading, as illustrated in Figure D.3. This annex is applicable only to crack opening mode I, generally the most severe, although the principles described would apply for any mode. When more than one mode is relevant, the corresponding stress intensity factor is normally referred to as K_I, but the suffix is not included in this annex for clarity.

The overall relationship between da/dN and ΔK is normally observed to be a sigmoidal curve in a log da/dN versus log ΔK plot. There is a central linear portion. At low values of ΔK the rate of growth falls off rapidly to a threshold stress intensity factor, ΔK_{th}, below which no significant crack growth is likely to occur. At high values of ΔK, when the maximum stress intensity factor in the cycle approaches the critical stress intensity factor for failure under static load, the rate of crack growth accelerates rapidly. However, for practical purposes, it is often sufficiently accurate to ignore the existence both of the threshold and of the failure regions and to assume that the central linear portion applies for all values of ΔK up to failure. An important exception is item f) in **D.1** when ΔK_{th} is used to identify the applied load cycles that can be neglected as being effectively non-damaging.

Figure D.3 **Crack opening modes**



a) Mode I (Opening) b) Mode II (In-plane shear) c) Mode III (Out-of-plane shear)

The relevant equation for the rate of crack propagation da/dN (in mm/cycle) is given by:

$$\frac{da}{dN} = A(\Delta K)^m \qquad (D.1)$$

where

A	is a constant that depends of the material and applied conditions including environment and cyclic frequency;
m	is a constant that depends of the material and applied conditions including environment and cyclic frequency;
ΔK	is the range of stress intensity factor corresponding to the applied stress cycle and instantaneous fatigue crack dimensions.

Integration of Equation D.1 gives the number of cycles, N, required to propagate a crack from an initial size, a_i, to a final size, a_f, as:

$$N = \frac{1}{A}\int_{a_i}^{a_f} \frac{1}{(\Delta K)^m} da \qquad (D.2)$$

If the case being assessed is one in which N_i cycles are required to initiate a fatigue crack, the total fatigue life is $N + N_i$. However, N_i is usually assumed to be zero for fatigue failure from stress concentration features in welded joints.

The application of Equation D.2 for calculating the fatigue life under spectrum loading is equivalent to the use of Miner's rule (see **16.7**) assuming $D = 1$ [3]. However, in contrast to the application of Miner's rule using a constant amplitude *S-N* curve, direct allowance can be made for the damaging effect of stress ranges below the constant amplitude fatigue limit on the basis that only combinations of crack length and stress range that produce ΔK values greater than the threshold value ΔK_{th} are damaging (i.e. cause crack growth). However, again in contrast to the use of Miner's rule, the order of application of the stress spectrum should be taken into account to ensure that stress ranges that actually occur when the crack is long enough for ΔK to exceed ΔK_{th} are not neglected. Therefore, unless the order of application of stress cycles is known, cycles should be applied in the most damaging plausible order, established if necessary by performing the calculations for various orders to determine the worst case.

If there is uncertainty about the validity of Miner's rule for the type of load spectrum being assessed, (see **16.7**), the life N calculated using Equation D.2 should be reduced accordingly (e.g. halved if D = 0.5 is appropriate).

D.4 Values of *A* and *m*

The values of *A* and *m* depend upon the material and the applied conditions, such as stress ratio, environment, test frequency and waveform.

Whenever possible, data relevant to the particular material, product form and service conditions should be used and where any uncertainty exists concerning the influence of the environment such data should be obtained in accordance with BS ISO 12108. Provided that sufficient data are available to enable them to be defined, the chosen values should correspond to the mean plus two standard deviations of log d*a*/d*N*, representing the same 97.7% probability of survival as the design *S-N* curves.

Comprehensive guidance on *A* and *m* values relevant to welded materials, including allowance for applied stress ratio and various environments, is given in BS 7910. In the case of structural steels under conditions comparable with those experienced by welded joints containing high tensile residual stress, the following conservative fatigue crack growth parameters are recommended:

- in air, $m = 3$, $A = 5.21 \times 10^{-13}$ and $\Delta K_{th} = 63$ N.mm$^{-3/2}$:
- in a marine environment at temperatures up to 20 °C, $m = 3$, $A = 2.3 \times 10^{-12}$ and ΔK_{th} = zero.

D.5 Initial flaw size a_i

The calculated life is usually very sensitive to the assumed value of a_i. Therefore a_i should not be underestimated.

For nominally flaw-free welded joints failing from the weld toe a_i should be assumed to lie within the range 0.1 mm to 0.25 mm [26, 27] unless a larger size is known to be relevant. The correct value to be used is calculated from the calibration calculations (see **D.8**).

In the absence of definite information about the shape of the initial flaws, for joints with welds transverse to the direction of stress it should be assumed that the flaw at the weld toe is long and continuous, i.e. $\frac{a_i}{2c} = 0$ At the ends of longitudinally loaded welds, however, it would be realistic to assume that the initial flaw was a semi-elliptical in shape with $\frac{a_i}{2c} = 0.1$.

D.6 Limit to fatigue crack propagation a_f

In the fatigue assessment, an upper limit should be set to the size a_f to which a crack could grow without failure occurring during operation by any of the following modes, as appropriate:

a) unstable fracture;

b) yielding of the remaining section;

c) leakage (in containment vessels);

d) stress corrosion;

e) instability (buckling); or

f) creep.

D.7 Range of stress intensity factor, ΔK

D.7.1 General

Application of the crack propagation equation (see **D.3**) requires knowledge of the range of stress intensity factor, ΔK, at the crack tip. Hence, for the actual or assumed dimensions and position of the idealized flaw, the value of ΔK corresponding to the range of stress (see **15.2** and **15.3**) should be estimated either from relevant published solutions, notably those in BS 7910, or from specific stress analysis of the structure.

D.7.2 Stress intensity factor correction factor

In general the value of ΔK is of the form:

$$\Delta K = \left(M_{km}M_m\Delta\sigma_m + M_{kb}M_b\Delta\sigma_b\right)\frac{\sqrt{\pi a}}{\Phi_o} \tag{D.3}$$

NOTE φ_o is only relevant in the assessment of elliptical embedded or semi-elliptical surface cracks. It is a function of a/c; for straight-fronted cracks (a/c=0) it is 1.0

BS 7910 provides an extensive range of solutions for the correction factors M_m, M_b and M_k. In cases that apply specifically to welded joints the following apply:

a) Correction factors that do not distinguish between membrane and bending stresses are used in conjunction with the total stress range $\Delta\sigma_m + \Delta\sigma_b$.

b) If the distinction between membrane and bending stresses is not known, the correction factors for membrane stress should be used in conjunction with the total stress range $\Delta\sigma_m + \Delta\sigma_b$.

In the cases shown in Figure D.1 and Figure D.2, the solutions in BS 7910 allow ΔK to be calculated for semi-elliptical or straight-fronted (a/c = 0) cracks at weld toes but only straight-fronted cracks propagating from the weld root in cruciform joints. In a fatigue crack propagation analysis the procedure is to calculate ΔK at the crack tip Q (see Figure D.1) and, in the case of an elliptical flaw, S and use Equation D.1 to determine the increment(s) of crack growth for the relevant ΔK value(s) and the number of applied load cycles. The process is then repeated for the new crack size until $a = a_f$. Depending on the purpose of the analysis this might correspond, for example, to complete failure, leakage of a container, the attainment of a detectable crack or the end of the required life.

D.7.3 Stress range

D.7.3.1 General

BS 7910 provides detailed guidance on the determination of the stress range $\Delta\sigma$ to be used in Equation D.3. However, this annex has some additional guidance to allow a choice to be made between the use of the nominal or hot-spot stress. Table D.1 summarizes the resulting recommendations.

D.7.3.2 Nominal stress range

The guidance in BS 7910 leads to the determination of the same stress range as that described in **15.6** for use in nominal stress-based assessments. Thus, $\Delta\sigma$ in equation D.3 should include allowance for the stress concentration effects of any gross structural discontinuities and misalignment. The additional stress concentration effect of the weld detail, which is not required for nominal stress-based assessments using S-N curves, also needs to be considered. If this is not already included in the K solution it is introduced using the correction factor, M_k, which is a function of crack size, geometry and loading, as follows:

$$M_k = \frac{K \text{ for crack in plate with stress concentration}}{K \text{ for same crack in plate without stress concentration}} \quad (D.4)$$

The solution in BS 7910 for weld toe surface cracks is a function of crack depth a, plate thickness B and the overall length L of the attachment measured from weld toe to weld toe (Figure D.1). M_k normally decreases with increase in crack depth, from a value equal to the stress concentration factor in the absence of a crack down to unity at crack depths of typically 30% of plate thickness. At crack depths greater than that corresponding to M_k = 1.0 it should be assumed that M_k = 1.0.

In the corresponding BS 7910 solution for weld root cracks in cruciform joints (Figure D.2) M_k is a function of crack length a, weld size h and plate thickness B. It is always less than unity and its value might increase or decrease with increase in crack size, depending on the value of h/B.

In all the cases shown in Figure D.1 and Figure D.2 the stress range is that in the loaded plate at the weld toe.

D.7.3.3 Hot-spot stress range

The potential use of the hot-spot stress range is confined to the assessment of fatigue cracking from a weld toe. It is not applicable to the case of fatigue failure from the root of a cruciform joint (Figure D.2). The hot-spot stress includes the stress concentration effects of gross structural discontinuities, misalignment (which in practice is likely to be allowed for by applying k_m to the hot-spot stress determined for an aligned joint, see **B.5.2.1**) and of the weld detail, but excludes the stress concentration effect of the weld toe (see **15.7**). Therefore, the only further correction needed before using it in Equation D.3 is M_k. However, a complication with the M_k solutions in BS 7910 is that they already include the stress concentration effect of the weld detail, as reflected in the L/B value. As that effect is included in the hot-spot stress (except for those cases highlighted in **C.5.4**) the M_k solution for L/B = 0.5 should be used to reduce its influence. However, the actual L/B value should be used if there is any uncertainty that the hot-spot stress has been determined correctly (see **C.5.4**), which is a conservative assumption for any case.

In practice, if the hot-spot stress is determined on the basis of calculation of the through-thickness stress distribution and corresponding membrane and bending stress components (i.e. by the TTI or NF method, see **C.4.3**), these can be used directly in Equation D.3. M_k values would also be required, as described above.

D.8 Calibration

When the problem under consideration is one in which fatigue cracking from a weld toe is involved, the fracture mechanics formulation which is used for the fatigue assessment should be shown to predict, with acceptable accuracy, either:

a) the fatigue strength of a joint class with a detail similar to that under consideration; or

b) test data for joints which are similar to those requiring assessment.

Such calibration checks should be based upon realistic estimates of the mean values of the various parameters.

Table D.1 **Use of stress intensity corrections with nominal or hot-spot stress**

Method	**Stress(es) available**			**Stress type assumed to obtain *M* and M_k**	***L/B* assumed to obtain M_k**	**Additional stress concentrations to be considered**	
	Mem-brane σ_m	**Bending** σ_b	**Total** $\sigma_m + \sigma_b$			**Misalignment**	**Gross structural discontinuities**
Nominal stress	✓	✓	✓	σ_m and σ_b	Actual	✓	✓
	No	No	✓	membrane	Actual	✓	✓
Hot-spot stress	✓	✓	✓	σ_m and σ_b	0.5	✓ [A)]	No
	No	No	✓	membrane	0.5	✓ [A)]	No
Unreliable hot-spot stress (see **C.5**)	✓	✓	✓	σ_m and σ_b	Actual	✓ [A)]	No
	No	No	✓	membrane	Actual	✓ [A)]	No

[A)] Unless effect of misalignment already included in calculation of hot-spot stress

Annex E (normative) Fatigue testing and the use of test data to define design stresses

E.1 Introduction

Fatigue testing might need to be carried out for two different purposes:

a) to establish the relevant *S-N* curve, or joint classification, for the design of some detail which is not adequately covered by the classifications given in Clause **12**; or

b) to establish whether some prototype structure is capable of carrying the fatigue loading expected during the service life of the structure.

In a), the objective is to obtain a design *S-N* curve, under constant amplitude loading, in a similar manner to that used for the standard classes (see **A.1**). Statistical methods that are consistent with BS ISO 12107 are provided in this annex. In fatigue acceptance testing, the objective would normally be to apply to the component or structure loading simulating that to be expected in service. Guidance on the design and production of welded fatigue test specimens is provided in PD ISO/TR 14345.

Any fatigue tests should be performed using equipment and/or testing machines with known calibration (e.g. using equipment that conforms to BS EN ISO 7500-1). The specimens should not be overloaded prior to fatigue testing.

E.2 Fatigue tests to establish joint classification

E.2.1 Fatigue test procedure

Fatigue testing for joint classification purposes involves carrying out constant amplitude tests under tensile cyclic stresses. In the case of welded specimens, ideally these should be full-scale structural components but valid data can be obtained from specimens incorporating the weld detail of particular interest provided plate and weld sizes are full-scale and allowance is made for the likelihood that they would not embody the high tensile residual stress that is likely to exist in the actual structure. Ways of doing this include local spot heating, the introduction of additional weld beads normal to the weld of interest or performing the test at a high positive stress ratio or under conditions of high tensile mean or maximum stress. Tests may be performed at various stresses, with repeat tests at some of them, selected so as to give endurances to failure reasonably evenly distributed over the linear part of the relationship between log (stress range) and log (endurance), (i.e. typically over the range 10^5 to 2×10^6 cycles). Alternatively, all tests may be performed at the same stress level.

It is usually appropriate to test not less than eight nominally identical specimens representative of the detail under consideration.

If the detail is subsequently to be used in an environment other than air at normal ambient temperatures, then the service environment (e.g. corrosion conditions, temperature) should be simulated in the fatigue tests. In those circumstances it is also important that the loading frequency should be similar to that expected in service.

E.2.2 Tests performed to validate a design *S-N* curve

E.2.2.1 Approach

Use is made of the mean *S-N* curve for the class selected, and its standard deviation of log *N*, SD_d. Initially the assumption is made that the new test results form part of the same population as that used to determine the design *S-N* curve (this is called the null hypothesis). Then, hypothesis testing is used to show that, under this assumption, it is very unlikely (at a specified significance level) that the new results would be as high as they are. This is the basis for regarding the null hypothesis as implausible, and for accepting the alternative hypothesis that the new results actually belong to a population having longer fatigue lives than the main database. Thus, use of the selected class would be valid.

Assuming a 5% significance level, the condition for accepting the alternative hypothesis is:

$$\overline{\log N_{test}} \geq \log N_d + \frac{1.645.SD_d}{\sqrt{n}} \qquad (E.1)$$

where

$\overline{\log N_{test}}$ is the mean logarithm of the fatigue life from the tests at a particular stress;

$\log N_d$ is the logarithm of the corresponding fatigue life from the mean *S-N* curve for the design class;

n is the number of fatigue test results.

NOTE 1 The value 1.645 is obtained from standard normal probability tables for a probability of 0.95.

NOTE 2 The 5% level of significance is commonly considered to give a sufficiently low probability of concluding that the populations are different in the case where they are actually the same. Alternatives include 1.285 at the 10% level of significance, 1.960 at 2.5% and 2.33 at 1%.

Equation E.1 can also be expressed:

$$\overline{N_{test}} \geq N_d \times 10\left(\frac{1.645.SD_d}{\sqrt{n}}\right) \qquad (E.2)$$

where :

$$S_r^m . N = C_o \times 10\left(\frac{1.645.SD_d}{\sqrt{n}}\right)$$

is the geometric mean of the test fatigue lives.

Basic assumptions are that:

a) the slope of the mean *S-N* for the test results is the same as the slope *m* of the design curve, and

b) the standard deviation of log *N* about that mean *S-N* curve (assuming that its slope is *m*) is the same as that for the database that produced the design curve. (i.e. SD_d).

Tests are available to check these assumptions if there is any uncertainty (see 28).

E.2.2.2 Tests performed at the same stress level

Each specimen is tested to failure. Equation E.2 is then satisfied, where $\overline{N_{test}}$ is the geometric mean fatigue life obtained at the stress level used and n is the total number of tests. Unless the new results lie on an *S-N* curve with the same slope as the design curve, this approach only validates the selected class at the stress level used for the tests.

E.2.2.3 Repeat tests at a number of stress levels

Each specimen is tested to failure. Equation E.2 is then applied in turn for each stress level. This approach validates the design curve over the range of stress levels used, even if the *S-N* curve for the new test results does not have the same slope m as the design curve for the selected class.

E.2.2.4 Tests performed to produce an *S-N* curve

The specimens are tested to failure at various stress levels and an *S-N* curve is fitted by regression analysis (see **A.1**). This should be of the same form as the selected design curve and have the same slope m (see **E.2.2.1**) such that Equation E.2 can be modified to compare this curve and the mean *S-N* curve for the design class being validated to give the condition:

$$C_{test} \geq C_O \times 10^{\left(\frac{1.645 . SD_d}{\sqrt{n}}\right)} \tag{E.3}$$

Consequently, the mean curve fitted to the test results is expected to be on or above the following *S-N* curve:

$$S_r^m . N = C_o \times 10^{\left(\frac{1.645 . SD_d}{\sqrt{n}}\right)} \tag{E.4}$$

Rather than testing every specimen to failure, Equation E.4 could be used as a target curve to be achieved in every test. In this case the specimens fatigue tested do not need to fail, simply to achieve lives on or above the target curve.

E.3 Fatigue acceptance test

The objective of the fatigue acceptance test is to establish if some prototype component or structure is capable of carrying the fatigue loading expected during its service life.

Where the service loads vary in a random manner between limits, they should be represented by an equivalent series of variable amplitude load cycles. Occasional high service loads in the test spectrum should not be unrepresentatively large or too numerous as, if they are, the fatigue lives which are obtained might not be representative (due to the fact that occasional high stresses can retard fatigue crack growth).

Alternatively, the test could be performed under constant amplitude loading at the maximum imposed service load, with the required number of repetitions being estimated to produce fatigue damage equivalent to that produced by the actual service loading spectrum, allowing for fatigue crack growth retardation, if relevant.

As in the case of tests to establish the classification of a joint (see **E.2**), if the structure is to be used in an abnormal environment then the environment and loading frequency should be simulated in the acceptance tests.

The geometric mean life obtained from the effective number of specimens should be at least equal to the design life multiplied by the factor, F, from Table E.1.

Owing to the great increase in scatter for tests carried out near the fatigue limit, stresses should be selected to give specimen lives not exceeding 2×10^6 cycles.

Table E.1 **Fatigue test factor, *F***

Number of test result, n	Fatigue test factor, *F*
1	5.4
2	4.3
3	3.9
4	3.7
10	3.2

NOTE Test factor obtained from Equation E.2 modified to compare $\overline{N_{test}}$, assuming SD = 0.2.

Annex F (normative) Weld toe improvement techniques

F.1 Background

The weld toe is a primary source of fatigue cracking because of the severity of the stress concentration it produces. Apart from a relatively sharp transition from the plate surface to the weld, dependent on the weld profile, the stress concentration effect is enhanced by the presence of minute crack-like flaws, extending to depths (below any undercut) of a few tenths of a mm [26, 27]. These flaws are an inherent feature of fusion welds and are not regarded as defects. Fatigue cracks readily initiate at these flaws.

The weld toe improvement methods described in this annex rely on two main principles:

a) *Reduction of the severity of the weld toe stress concentration*. Two methods are given, burr grinding and re-melting by TIG or plasma dressing. The primary aim is to remove or reduce the size of the weld toe flaws and thus extend the crack initiation part of the fatigue life. A secondary aim is to reduce the local stress concentration due to the weld profile by achieving a smooth blend at the transition between the plate and the weld face.

b) *Introduction of beneficial compressive residual stress*. An alternative, or additional, approach is to introduce beneficial compressive residual stresses in the weld toe region. These have the effect of clamping the weld toe in compression, with the result that an applied tensile stress would first overcome the residual stress before it became damaging. The applied stress range is therefore less damaging. This annex covers techniques that all achieve this aim by plastic deformation of the weld toe region, namely hammer, needle, high-frequency and shot peening. Compressive residual stresses are then produced as a result of the constraint imposed by the surrounding elastic material.

An important practical limitation on the use of improvement techniques that rely on the presence of compressive residual stresses is that the fatigue performance of the treated weld is strongly dependent on the applied mean stress of the subsequent fatigue loading. In particular, their beneficial effect decreases as the maximum applied stress approaches tensile yield, disappearing altogether at maximum stresses above yield. Thus, in general the techniques are not suitable for structures operating at applied stress ratios (R) of more than 0.4 or maximum applied stresses above around 80% yield. The occasional application of high stresses, in tension or compression, can also be detrimental in terms of relaxing the compressive residual stress.

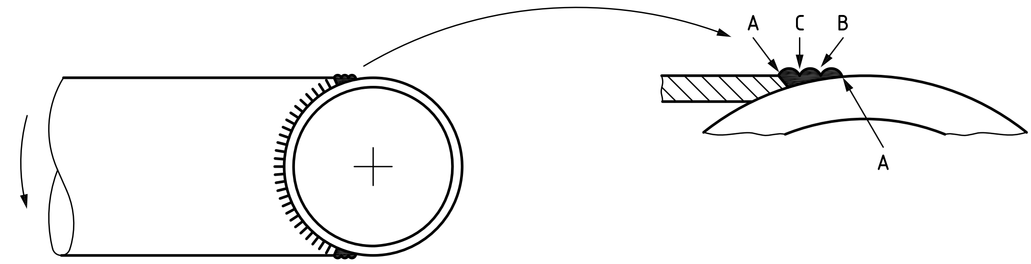
The techniques in this annex are particularly suitable for treating transverse fillet weld toes. Although in principle they could equally be used to treat butt weld toes, in practice it is usually more convenient simply to grind the butt weld overfill flush to improve its fatigue performance. The information provided in this annex refers specifically to transverse fillet weld toes, including the toes of external fillets in full or partial penetration joints and the toes of fillet welds that pass around the ends of longitudinal attachments, and the specified extents of improvement in fatigue performance resulting from the use of the improvement techniques apply only to such cases.

This annex provides only general guidance on the application of the improvement techniques. For more detailed guidance reference can be made to IIW Recommendations [29]. The extent of the improvement in fatigue performance resulting from the use of the techniques is also defined in this annex.

The improvement techniques covered by this annex are intended to increase the fatigue performance of the welded joint treated with respect to potential fatigue failure from the weld toe. In practice, the possibility of failure initiating at some other location with little improvement in fatigue life should be taken into account. This is especially relevant to joints with load-carrying fillet or partial penetration welds where fatigue cracking might still propagate from the weld root. Even nominally non-load-carrying fillet welds can fail from the root when the toe has been improved. Consequently, when weld improvement is planned, full penetration welds or fillet welds with extra-large throats should be used where possible, particularly for welds at the ends of cover plates and longitudinal stiffeners.

In the case of a multi-run weld, more than one weld toe might need to be treated. This is particularly the case in situations where the stress distribution over the weld surface is fairly uniform, such as a nodal joint between tubes of nearly equal diameter under out-of-plane bending (Figure F.1); treatment of the weld toe on the member surface would merely transfer failure to an adjacent toe between surface beads of the weld. Consequently, some or all other weld toes would need to be treated, in which case the most practical approach would be to grind the whole weld surface rather than to treat individual weld toes (see Figure F.1).

Figure F.1 **Multi-run weld in tubular nodal joint requiring improvement of every weld toe**



If the stress over the weld surface is fairly uniform grinding the toes A merely transfers failure to the inter-run toes B and C. In such situations the whole weld needs to be ground.

This annex applies to the treatment of weld toes on members at least 6 mm thick in any arc welded steel structure that is subjected to fatigue loading. Due to lack of experimental data for extra high strength steels, the specified improvements in fatigue performance apply only to structural steel and stainless steel grades up to maximum specified yield strength of 960 N/mm^2. However, it is reasonable to expect that, in principle, the methods would also improve the fatigue performance of welded higher strength steels, and welds on members less than 6 mm in thickness. In the absence of relevant published data, such benefit should be quantified by special testing (see Annex E).

A1 With regard to service loading conditions, the rules in **16.7** for variable amplitude loading assessments are applicable to joints with improved weld toes, including the treatment of stresses below the CAFL detailed in **16.4**. The tensile stress limitations in **16.1** still apply. A1 The specified benefits of burr grinding and TIG dressing extend to low-cycle fatigue strain cycling conditions in regions of geometric stress concentration. However, special restrictions are imposed regarding applied peak stresses and stress ratios in the case of the peening techniques.

F.2 Preparation

The ease of application and benefit from any of the improvement techniques covered by this annex is improved if the weld to be treated has a favourable profile with a well-defined weld toe. A favourable profile has a low weld toe contact angle, 45° or less, a generous weld toe radius, 1 mm or more, and freedom from weld toe undercut or cold laps. Therefore, if it is planned to use an improvement technique, it is advisable to try to produce fillet welds with such features. If the weld to be treated has a poor profile, it is advisable to grind a shallow groove along the toe to help guide the improvement technique tool (TIG or plasma torch, peening hammer, etc.). Such preparation might not be required before shot peening, but light grinding makes it easier to check that the treatment has been applied correctly and, reduce the stress concentration. In the case of the treatment of welds that pass around the ends of longitudinal attachments, including corner gussets, it is also advisable to use full-penetration welds, at least for the first 30 mm, to reduce the risk of fatigue failure from the weld root. Further benefit comes from tapering the attachment end to provide a smoother transition to the main plate.

In all cases the weld to be treated should be de-slagged and cleaned by wire-brushing to remove scale, rust or paint. It is also advisable to remove traces of oil or other surface contaminants before TIG or plasma dressing.

F.3 Weld toe dressing techniques

F.3.1 Burr grinding

The primary aim of grinding is to remove or reduce the size of the weld toe flaws from which fatigue cracks propagate and to reduce the local stress concentration effect of the weld profile by smoothly blending the transition between the plate and the weld face.

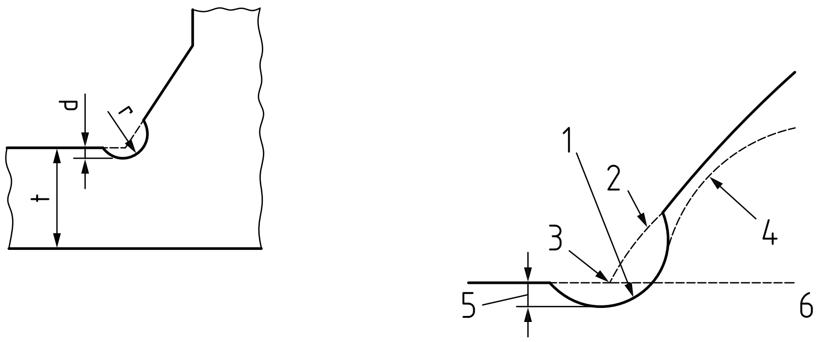
Toe grinding is normally carried out with a high speed pneumatic, hydraulic or electric rotary burr grinder with rotational speed from 15 000 to 40 000 rpm. The tool bit is normally a tungsten carbide burr (or rotating file) with a hemispherical end. To avoid a notch effect from too small a groove radius (r), it should be scaled to the plate thickness (t) at the weld toe being ground and the grinding depth (d) such that the root radius of the groove is no less than 0.25t or 4d (see Figure F.2). If a suitable burr for achieving such a radius directly is not available, as is likely to be the case with thick sections, the final groove might need to be produced by blending the sides of the sharp groove by additional grinding.

Figure F.2 **Recommendations for weld toe grinding**

d = min. 0.5 mm
below undercut

r/t > 0.25

r/d > 4



NOT TO SCALE

Key

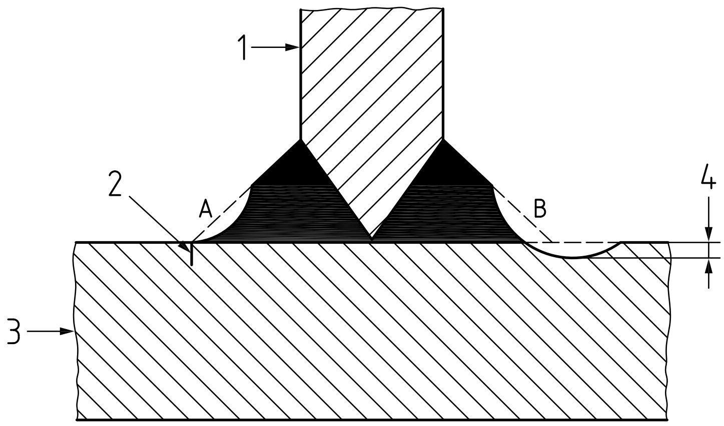
1	Ground profile	4	Minimum throat to be maintained
2	Original profile	5	Grinding depth
3	Original toe	6	Weld root

The tool is centred over the weld toe and pushed or pulled along the toe to produce a smooth concave profile at the weld toe. The depth of the depression penetrating into the plate surface should be at least 0.5 mm below the bottom of any undercut or flaw at the weld toe, which can be located by magnetic particle or liquid penetrant inspection (Figure F.3). The maximum depth of local machining or grinding should not exceed 2 mm. If the reduction in plate thickness exceeds 5% this should be taken into account in the stress calculation. The ends of fillet welds, such as those attaching longitudinal members, can only be treated effectively if the weld can be carried around the end of the attachment member to provide a distinct weld toe.

Where toe grinding is used to improve the fatigue life of fillet welded connections, care should be taken to ensure that the required throat thickness is maintained.

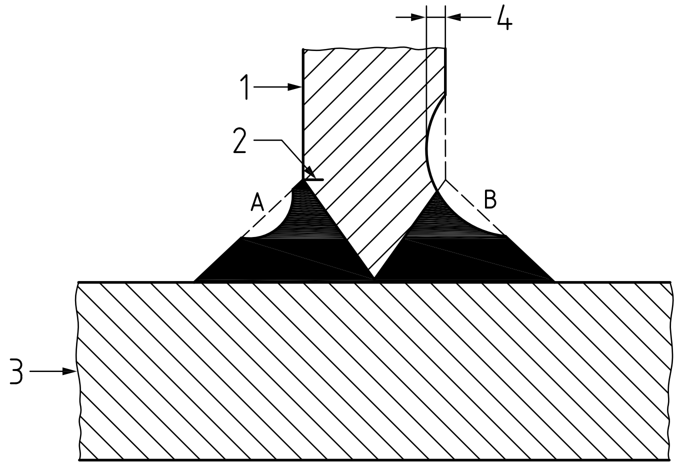
The final ground surface should be inspected by suitable NDT to confirm that the weld toe has been removed completely and to ensure freedom from surface-breaking flaws that might have been exposed as a result of the grinding.

Figure F.3 **Toe grinding to improve fatigue strength**



a) Stress in member X

Figure F.3 **Toe grinding to improve fatigue strength**



NOTE Grinding a weld toe tangentially top the plate surface as at A produces little improvement in strength. Grinding should extend below the surface as at B, in order to remove toe flaws.

b) Stress in member Y

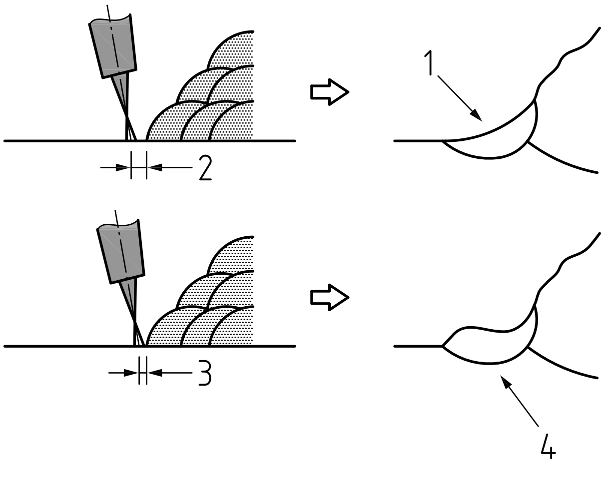
Key

1	Member Y	3	Member X
2	Flaw	4	Depth of grinding should be at least 0.5 mm but ≤ 2 mm below bottom of any flaw at the weld toe

F.3.2 TIG or plasma dressing

The objective of TIG or plasma dressing is to remove the weld toe flaws by re-melting the material at the weld toe and to reduce the local stress concentration effect of the local weld toe profile by providing a smooth transition between the plate and the weld face.

Figure F.4 **Effect of TIG or plasma torch position on resulting weld profile**



Key

1	Optimum shape	3	Toe ±0.5 mm
2	0.5 mm to 1.5 mm from weld toe	4	Unacceptable shape

Standard TIG or plasma welding equipment is used, without the addition of any filler material. Argon is normally used as shielding gas. The addition of helium in TIG dressing is beneficial as this gives a larger pool of melted metal due to a higher heat input. However, one advantage of plasma dressing over TIG is that a wider re-melted material pool is produced.

If pre-heating would be required for welding the steel concerned then pre-heating to the lower end of the range of recommended temperatures should be used before TIG or plasma dressing. In both cases the position of the welding torch arc with respect to the original weld toe should be controlled. Trials should be carried out before applying TIG or plasma dressing to establish the correct conditions. The aim is to produce a profile that blends smoothly from the weld face to the plate surface, as shown in. This should be possible with the arc centred close to the weld toe on the plate side, ideally 0.5 mm to 1.5 mm from the toe. An unacceptable result, Figure F.4, may be produced if the arc is too close to the weld toe, within 0.5 mm, and the dressing conditions should be modified to prevent this.

Further details of the techniques can be found in [27, 29 and 30].

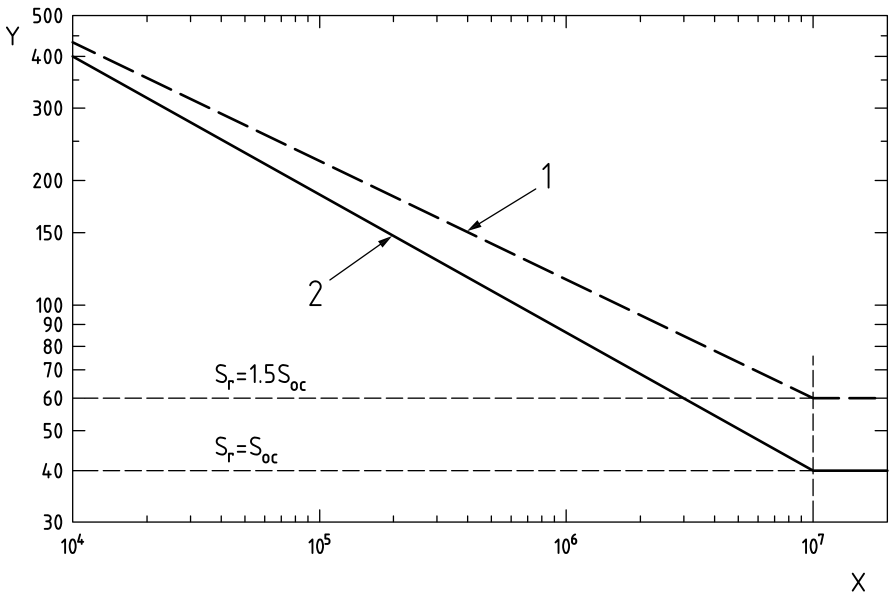
F.3.3 Benefit

The classification of welds might, where indicated in Table 4 to Table 10, be raised when dressing of the weld toes is carried out in accordance with this annex (see also **13.2**).

For assessments based on either nominal or hot-spot stresses, the *S-N* curve for the untreated weld can be assumed to be increased in fatigue strength at 10^7 cycles by a factor of 1.5 and the S_r-*N* curve rotated to a slope of $m = 3.5$, as illustrated in Figure F.5.

Weld toe grinding should not be assumed to be effective in the presence of a corrosive environment as this can cause pitting of the dressed surface. The full potential of weld toe dressing might not be achieved if fatigue cracking could initiate at a location other than the treated weld toe. The design process should still address all potential sites for fatigue crack initiation and classify each one accordingly.

Figure F.5 **Modification to design *S-N* curve for untreated weld resulting from weld toe dressing**



Key

X Endurance *N*, cycles

Y Stress range S_r N/mm²

1 [A1] Toe dressed [/A1] $S_r^{3.5}.N = (1.5S_{OC})^{3.5}.10^7$

2 Untreated weld [A1] $S_r^3.N = C_2$ [/A1]

Finally, the fatigue strength of toe dressed welds is limited by that of the parent material. Thus, for applied stress ranges greater than that where the *S-N* curves for the treated weld and parent material (class B) intersect, the lower of the two *S-N* curves should be used.

F.4 Weld toe peening techniques

F.4.1 General

Four weld toe peening techniques are covered by this annex:

a) hammer;

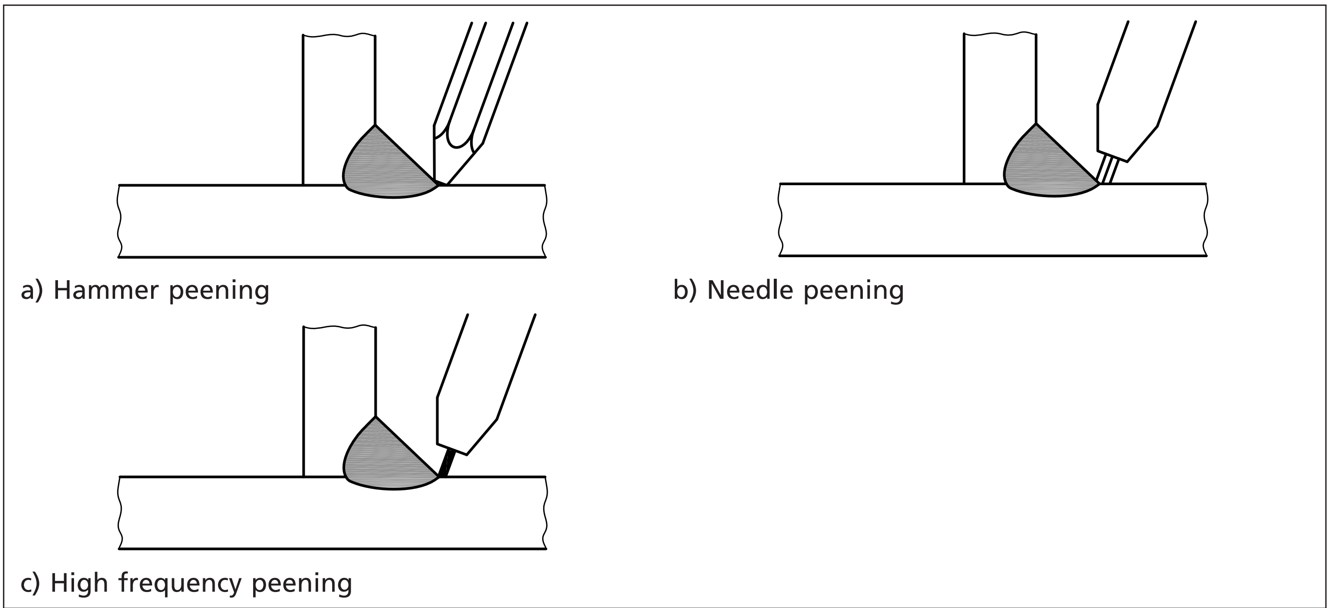
b) needle;

c) high-frequency; and

d) shot peening.

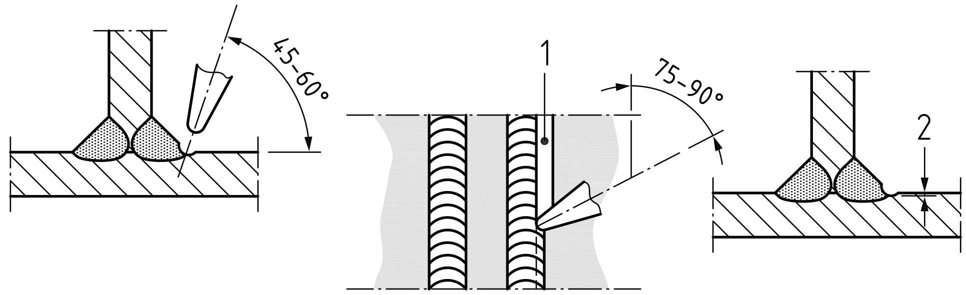
Methods a) – d) aim to induce compressive residual stresses by plastic deformation produced either by repeatedly hammering the weld toe region with solid blunt-nosed tools or a bundle of small-diameter rods, both with smooth, rounded tips (Figure F.6), or by directing high-velocity shot at the weld toe. In use, the tips of the solid tools wear, causing flattening, and the surface becomes rougher. Therefore, the end shape should be checked after every use and, if necessary, the tool tip re-ground and polished.

Figure F.6 **Weld toe peening methods**



In the case of the hammering techniques, the tool is centred over the weld toe and moved along it so that the repeated hammering produces an indentation at the weld toe. Effective treatment requires reasonably accurate positioning of the tip of the tool over the weld toe, which is facilitated by prior grinding, so that metal on each side (both weld metal and parent plate) is deformed. This is normally achieved by supporting the hammer firmly with the tool between 45° and 60° to the plate surface and approximately perpendicular (75° to 90°) to the direction of travel (Figure F.7) and keeping the tool tip in close contact with the weld toe as it is moved along. The peening operation is repeated until there is a clear indication of uniform plastic deformation in the form of a groove with a uniform surface appearance. In the case of needle peening this is a surface bright in appearance with a uniform distribution of small indentations.

Figure F.7 **Weld toe peening**



Key

1 Bright surface

2 Peening depth 0.1 mm to 0.5 mm depending on peening method

In order to ensure full coverage of the weld toe region, the following procedure should be followed.

a) Apply needle peening until the relevant area is free of untreated spots, check for 100% coverage using a low power magnifying glass and record the peening time taken to achieve this.

b) Repeat the peening for the same length of time to achieve what is termed 200% coverage.

c) A satisfactory result can normally be achieved with a total of four passes.

In the case of hammer or high-frequency peening the requirement is for a surface that is smooth and free from obvious individual indentations. Again, four passes are usually sufficient. Surface flaws should also be avoided. In this respect, both peening methods, especially when applied to peaky or severely convex weld profiles, can cause the plastically deformed metal to fold over the original weld toe and leave a crack-like lap feature. The resulting fatigue performance of the welded joint might actually be less than that of the original. Therefore, it is advisable to grind the weld lightly before peening to improve its shape and create a groove which facilitates a steady movement of the peening tool. Any signs of folded laps or heavy flaking of the surface should be removed by local grinding, followed by re-peening.

Table F.1 summarizes the main characteristics of the equipment and objectives of the three hammer or needle peening techniques.

An important feature of shot peening is the need to ensure full coverage of the surface to be treated. The usual technique for assessing adequate shot peening coverage is to apply a fluorescent dye before peening and then inspect it with the aid of a black light. Checks can be made during shot peening so that particular attention can be paid to regions where access by the shot is restricted.

As all the peening techniques rely on the retention of compressive residual stress they cannot be relied upon to provide any benefit unless they are applied after any operations that could introduce tensile residual stress (e.g. the assembly of components when fit-up is imperfect).

F.4.2 Practical comments

F.4.2.1 Hammer peening

The diameter of the tool tip influences the resulting appearance of a hammer peened surface. In general, the smaller the diameter, the greater the likelihood that the actual weld toe itself would be peened and eventual-ly disappear. Peening with a large diameter tool (greater than 12 mm) does not usually reach the weld toe but instead deforms material either side of it. Although in general the desired effect is achieved with fewer passes using a large diameter tool, the presence of the original weld toe is a disadvantage from the viewpoint of inspection. In particular, it is not obvious that the toe has been correctly treated (i.e. left in a state of compressive residual stress) and remnant traces of weld toe confuse in-service inspection as it is difficult to distinguish between them and fatigue cracks. The use of a small diameter tool, or a combination of small and large diameter tools, with the aim of deforming the actual weld toe offers the best compromise. Inspection would then ensure that all traces of the original weld toe had disappeared.

Hammer peening, even using modern silenced hammers, is a noisy operation; the operator and others working in the vicinity should use ear protection. Vibration from peening equipment can cause physical discomfort or harm, and the operator should wear vibration-damping gloves and not perform the operation for extended periods of time.

Table F.1 **Summary of weld toe peening methods**

Details	Method		
	Hammer peening	**Needle peening**	**High frequency peening**
Tool	3–9mm diameter round-tipped chisel	Bundle of round-tipped steel rods (e.g. around thirty 2mm diameter needles)	One or more single- or double-radius ended tools, typically 3–5mm in diameter
Machine	Pneumatic, hydraulic or electrical impacting hammer	Standard weld de-slagging needle gun	Purpose-built impacting hammer activated by vibrations produced by ultrasonic transducer or pneumatically
Typical hammering frequency	25–100 blows per second	40 blows per second	100–400 blows per second
Typical travel speed, mm/minute	500	800	500
Number of passes	Typically 4	Typically 4	As required to produce required surface finish
Typical depth of plastic deformation, mm	0.15–0.5 (0.15mm minimum required)	0.1–0.2	0.15–0.4
Required surface finish	Uniform groove with smooth surface free from obvious individual indentations	Bright surface with uniform distribution of small indentations over complete surface	Uniform groove with smooth surface free from obvious individual indentations

F.4.2.2 Needle peening

Needle peening is noisy and ear protection should be used. However, it is less noisy than hammer peening and the equipment is lighter and generally easier to use than hammer peening equipment.

F.4.2.3 High frequency peening

The relatively low frequencies of hammer and needle peening mean that their effectiveness depends on the pressure on the tool against the treated surface (typically at least 20 kgf). The result is that the vibrations of the tool are transmitted directly to the hands of the operator and some effort might be required to maintain alignment of the peening tool along the weld. In contrast, the high-frequency peening methods are based on the generation and utilization of impacts from very high frequency vibrations, with the result that effective treatment is virtually independent of the pressure on the tool (~3 kgf). Noise and vibration are also considerably lower. In view of these factors, the high frequency peening techniques offer advantages over hammer and needle peening in terms of ease of use, health and safety and quality control of the peening operation.

F.4.2.4 Shot peening

Shot peening entails impacting a surface with shot (round metallic, glass or ceramic particles) varying in diameter from 50 μm to 6 mm with a force sufficient to create plastic deformation. It is a non-contact process so that particles are fired at the surface remotely and can be applied manually but generally in an automatic or mechanized manner for consistency of the residual stress profile. Checking for that consistency can be achieved using fluorescent dyes sprayed or brushed onto the surface before shot peening and subsequently checked using a black light during and after processing. Any residue of fluorescence, which might be localized if it reflects the presence of a sharp discontinuity like a steep weld profile or weld toe undercut, provides an indication of inadequate coverage and the need for further peening. The selection of shot size and type depends on the material and geometry of the part and the strain sensitivity of that material.

F.4.3 Benefits

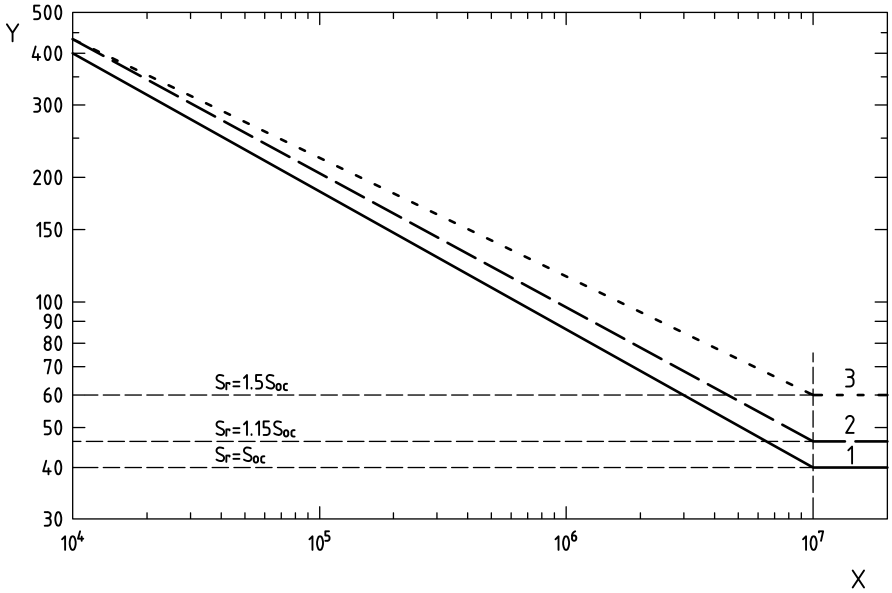
The classification of welds might, where indicated in Table 4 to Table 10, be raised by peening the weld toes (see also **13.2**). However, in contrast to the behaviour of toe dressed welds the improvement in fatigue strength depends on the applied stress ratio, R, and the maximum applied tensile stress, such that the benefit decreases with increase in either. If practical, some compensation for this problem arises if the peening operation is carried out while the welded joint is subjected to a tensile stress. For example, such an approach might be possible if part of the tensile stress experienced by the joint in service is due to the dead load on the structure. However, in general the peening techniques are not suitable for structures operating at applied $R > 0.4$ or maximum applied tensile stresses above around 80% yield. Similarly, their value is questionable under loading conditions that include the occasional application of high stresses, in tension or compression, as they can be detrimental in terms of relaxing the compressive residual stress.

With the above limitations, on the basis of the application of weld toe peening in accordance with this annex, for assessments based on either nominal or hot-spot stresses the fatigue strength of the untreated weld at 10^7 cycles may be increased by a factor of up to 1.5 and the slope of the S_r-N curve changed to $m = 3.5$, depending on R and the maximum applied tensile stress (S_{max}), as indicated in Table F.2 and illustrated in Figure F.8. As in the case of weld toe dressing, the fatigue strength of peened welds is limited by that of the parent material. For applied stress ranges greater than that where the S-N curves for the treated weld and class B intersect, the lower of the two S-N curves should therefore be used. Similarly, if curve 2 in Figure F.8 crosses curve 3 then curve 3 should be used for higher applied stress ranges.

Table F.2 **Improvement in fatigue strength due to weld toe peening**

Conditions	Improvement
$R < 0$ A1, including fully compressive loading A1	Increase fatigue strength at 10^7 cycles by factor of 1.5, change slope of S_r-N curve to $m = 3.5$ and treat detail as stress-relieved in accordance with **16.3.6**.
$0 \le R \le 0.28$, $S_{max} \le 80\%$ yield	Increase fatigue strength at 10^7 cycles by factor of 1.5, change slope of S_r-N curve to $m = 3.5$.
$0.28 < R \le 0.4$, $S_{max} \le 80\%$ yield	Increase in fatigue strength by factor of 1.15 but no change in slope of S_r-N curve.
$R > 0.4$ or $S_{max} > 80\%$ yield	No benefit unless proved by fatigue testing (see Annex E)

Figure F.8 **Modification to design *S-N* curve for untreated weld resulting from weld toe peening**



Curve 1: Basic design curve for untreated weld if $R > 0.4$ or $S_{max} > 80\%$ yield

Curve 2: $S_r^3 N = (1.15S_{oc})^3.10^7$ if $0.28 < R \leq 0.4$ [A1] and $S_{max} \leq 80\%$ yield [A1]

Curve 3: $S_r^{3.5}\ N = (1.5S_{oc})^{3.5}.10^7$

a) $R < 0$, $S_{r\ effective} = S_{max} + 0.6S_{min}$

b) $0 \leq R \leq 0.28$ and $S_{max} \leq 80\%$ yield, $S_{r\ effective} = S_r$

Key

X Endurance N, cycles

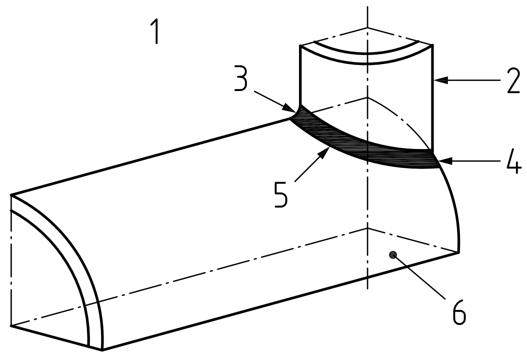
Y Stress range S_r, N/mm²

Annex G (normative) Assessment of tubular node joints

G.1 Fatigue of tubular joints

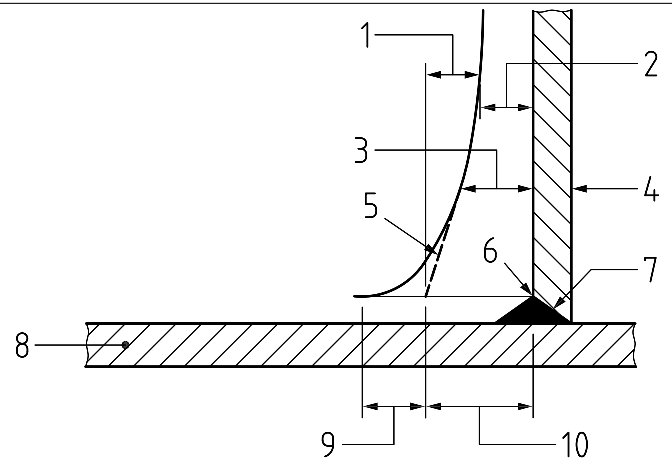
The stress range that should be used in the fatigue analysis of a tubular node joint is the hot-spot stress range at the weld toe. This should be evaluated at sufficient locations to characterize fully the fatigue performance of each joint. For example, in the case of a tubular set-on connection at least four equally spaced points around the joint periphery should be taken into account. For any particular type of loading, this hot-spot stress range can be defined as the product of the nominal stress range in the brace and the appropriate stress concentration factor (SCF), as indicated in Figure G.1, which also details the nomenclature used to describe tubular joints.

Figure G.1 **Example of hot-spot stresses in a tubular node joint**



Key
1 Load applied in the brace
2 Brace
3 Crown point
4 Saddle point
5 Weld
6 Chord

a) Nodal joint nomenclature

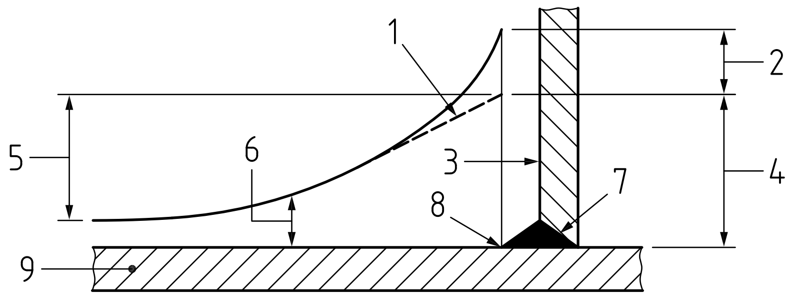


Key

1	Increase in stress due to overall joint geometry
2	Nominal stress
3	Stress in brace
4	Brace wall
5	Extrapolation of geometric stress distribution to weld toe
6	Weld toe
7	Weld
8	Chord wall
9	Stress increase due to weld geometry
10	Brace hot-spot stress = nominal stress × SCF_{Brace}

b) Stress distribution in brace

Figure G.1 **Example of hot-spot stresses in a tubular node joint**



c) Stress distribution in chord

Key

1 Extrapolation of geometric stress distribution to weld toe
2 Stress increase due to weld geometry
3 Brace well
4 Chord hot-spot stress = nominal stress × SCF_{Chord}
5 Increase in stress due to overall joint geometry
6 Stress in chord
7 Weld
8 Weld toe

In the fatigue assessment of tubular joints using the class TJ design curve presented in Clause **16**, the greatest values of the hot-spot stress ranges on both the brace and chord sides should be taken into account.

G.2 Stresses arising in tubular node joints

The stresses in tubular nodal joints arise from the following three main causes:

a) the basic structural response of the joint to the applied loads (nominal stresses);

b) the need to maintain compatibility between the tubes (geometric stresses); and

c) the highly localized deformations in part of the tube wall near to the brace-chord intersection (local stresses).

Nominal stresses arise due to the tubes behaving as beam-columns, and can be calculated by frame analysis of the structure.

Geometric stresses result from the differences in deformation between the brace and chord under load. For example, in a T-joint under axial tensile brace load, the brace extends only very slightly, whereas the circular cross section of the chord becomes significantly elongated to a pear-shape section. The differences in deformation require the tube walls to bend so that the brace and chord remain in contact at the weld. They can also cause a maldistribution of the nominal membrane stresses around the brace circumference.

Local stresses arise because of the geometric discontinuity of the tube walls at the weld toes where an abrupt change of section occurs which increases until the weld root is reached. Local stresses are not propagated far through the wall thickness, however, and therefore a local region of three-dimensional stress occurs. The sharper the corner at the weld toe, and the greater the angle of the overall weld profile to the tube wall, the higher the local stress.

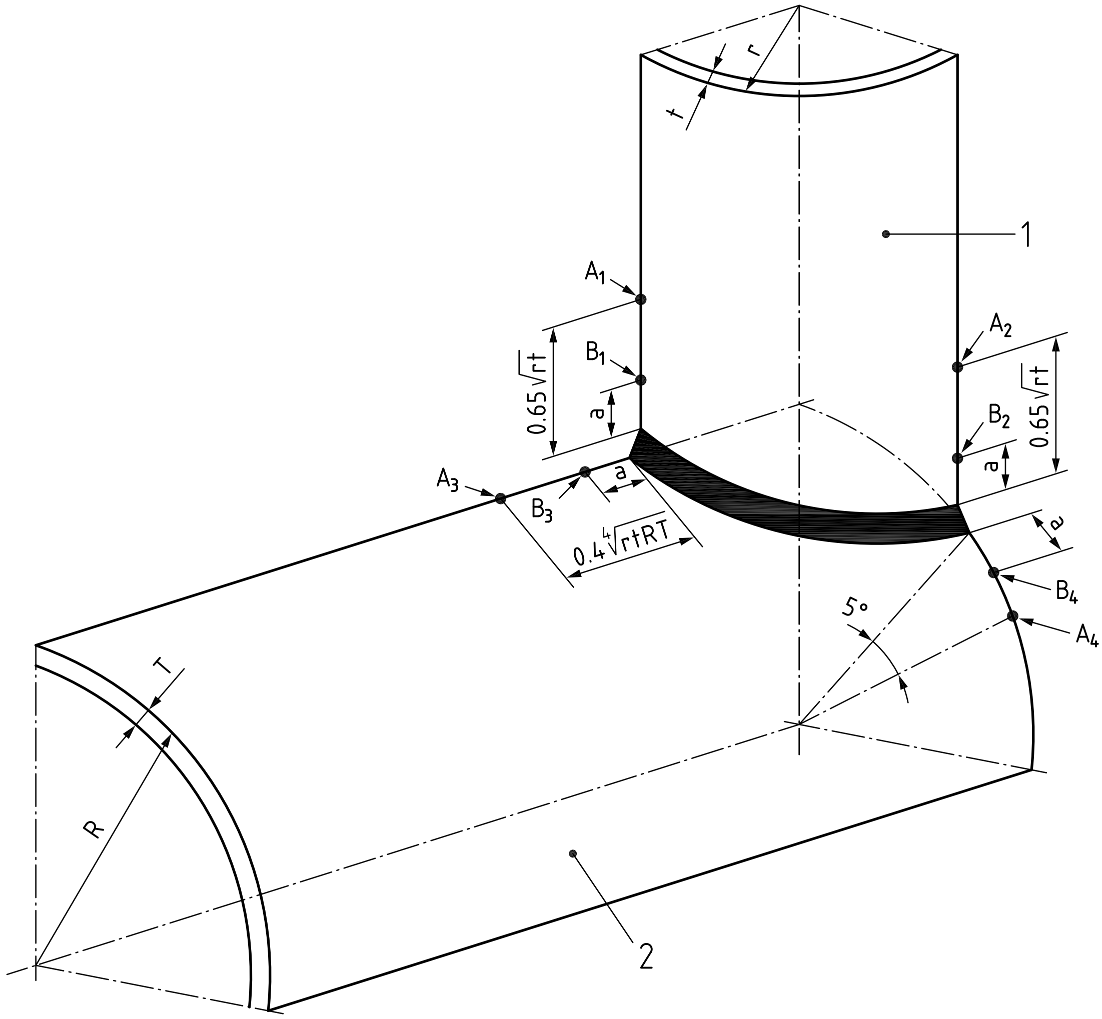
G.3 Hot-spot stresses in tubular joints

Any of the methods described in Annex C may be used to establish the hot-spot stress but the most widely used, including in the establishment of parametric formulae for calculating the hot-spot stress concentration factor, is linear surface stress extrapolation (SSE). However, the locations of the stresses used to perform the linear extrapolation for tubular joints are different from those defined in Annex C for analysing plate joints, as shown in Figure G.2. These are based on the assumption that the maximum extent of the local region where the stress is influenced by the "notch" effect of the weld itself is the greater of 4 mm or $0.2\sqrt{rt}$, where r and is the brace outer radius and t is the wall thickness.

NOTE The alternative value of 4 mm is only used in practice for very small tubular joints with disproportionately large welds. The local region in such cases is determined by the geometry of the weld toe bead (typically a few millimetres in size) as the stresses rise into the weld which, being disproportionately large, forms a ring of very stiff and inflexible material joining the brace to the chord.

This expression was obtained empirically, though the dependence on the parameter $\sqrt{rt}$ was originally drawn from a similar dependence on the wavelength of bending stress in tubes (see references [31] and [32]). The hot-spot stress range is the maximum principal stress range at the weld toe, based on the extrapolated values of the stress components at the weld toe. Therefore, the maximum and minimum values of each stress component are determined at each extrapolation point during a load cycle, the corresponding ranges of each stress component at the weld toe are then determined by linear extrapolation and finally these are used to calculate the principal stress range.

Figure G.2 **Locations A and B of stresses used for linear extrapolation to weld toes to determine hot-spot stresses in tubular joints**



Key

1 Brace

2 Chord

G.4 Calculation of the hot-spot stress

The calculation of hot-spot stress can be undertaken in a variety of ways, e.g. by physical model studies, finite element analysis or from hot-spot SCFs obtained by use of semi-empirical parametric formulae. The first two allow the hot-spot stress to be determined at any location whereas parametric equations tend to concentrate on the positions of maximum hot-spot stress, usually at the crown and saddle positions. When physical models are used, the geometric stress extrapolated to the weld toe should be obtained as described in this sub-clause. When finite element calculations do not allow for any effect of weld geometry, the hot-spot stress at the weld toe can be estimated from the value obtained at the brace/chord intersection. Parametric formulae should be used with caution in view of their inherent limitations; in particular they should only be used within the bounds of applicability relevant to the formula under consideration.

Extensive guidance on available parametric formulae for both co-planar and multi-planar tubular connections and recommendations on the choice of solution is given in BS EN ISO 19902. Guidance is also given on the effects of ring stiffening. More recent solutions are included in DNV-RP-C203. The hot-spot SCFs are defined as:

$$SCF = \frac{\text{Hot-spot stress at weld toe considered}}{\text{Nominal stress in loaded brace}} \qquad (G.1)$$

with the hot-spot stress in terms of the maximum principal stress at the weld toe obtained by extrapolation as described above.

The parametric formulae were developed by fitting analytical or model stress concentration factor data to equations which are functions of non-dimensional tubular joint geometric parameters, as follows:

$$\alpha = \frac{\text{length of chord cylinder}}{\text{mean radius of chord cylinder}} \qquad (G.2)$$

$$\beta = \frac{\text{mean radius of brace cylinder}}{\text{mean radius of chord cylinder}} \qquad (G.3)$$

$$\gamma = \frac{\text{mean radius of chord cylinder}}{\text{wall thickness of chord cylinder}} \qquad (G.4)$$

$$\mathrm{T} = \frac{\text{wall thickness of brace cylinder}}{\text{wall thickness of chord cylinder}} \qquad (G.5)$$

The form of these equations assumes that some dependence of the parameters can be deduced logically (by, for example, using simple beam theory to calculate a component of stress at the chord crown); otherwise a polynomial or power law dependence is assumed.

Each set of parametric formulae is limited in application in the following three ways:

a) types of joint geometry;

b) parametric validity range; and

c) loading cases.

There are also restrictions on the range of the other four variables: α, γ, β, τ. These are due to limitations in the range of original data obtained, or the methods used to obtain those data, but also because marked changes in the nature of the stress distribution occur at extreme values of these parameters. For example, high β ($\beta > 0.9$) connections show a radically different peak stress distribution from similar lower β connections, with the hot-spot moving from the normal saddle position to the crown, and for low γ connections thin shell idealization used for finite element calculation is not valid. Non-compliance with parametric validity range can result in a considerable increase in the variation of performance, both conservative and non-conservative, of these equations.

G.5 Rectangular hollow sections

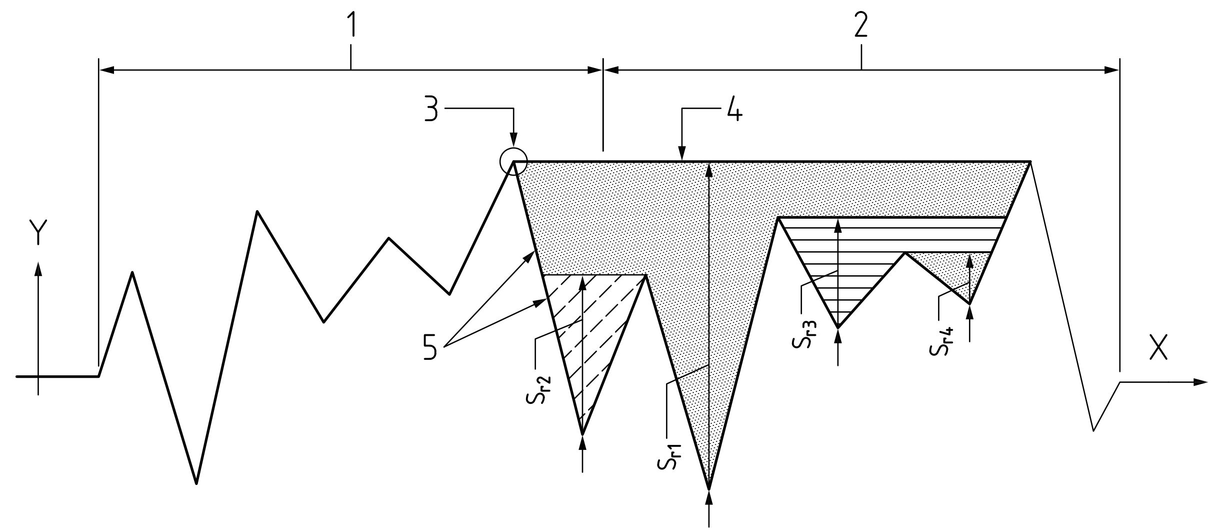
The class TJ design curve is based on fatigue test data obtained from joints between relatively large circular section tubes, with wall thickness generally above 16 mm. Alternative guidance directed more at thinner (down to 4 mm), circular and rectangular section tubes is available in BS ISO 14347.

Annex H (normative) Cycle counting by the reservoir method

H.1 General

The purpose of cycle counting is to reduce an irregular series of stress fluctuations to a simple list of constant amplitude stress ranges that are likely to produce the same fatigue damage. Various methods exist but the most widely used is the rainflow method, as described in ASTM E1049. However, in practice it is often necessary to deal with relatively short stress histories, such as those produced by individual loading events. In such cases, the reservoir method given in **H.2** and shown in Figure H.1 is more convenient. It consists of imagining a plot of the graph of each individual stress history as a cross section of a reservoir, which is successively drained from each low point, counting one cycle for each draining operation. The result, after many repetitions of the loading event, is the same as that obtainable by the rainflow method.

Figure H.1 **Example of cycle counting by reservoir method**



Key

X	Time	Y	Stress
1	First occurance	4	Imaginary reservoir
2	Second occurance	5	Shaded areas are those parts of the reservoir which successively became empty
3	Highest peak		

H.2 Method

H.2.1 Derive the peak and trough values of the stress history, due to one loading event. Sketch the history due to two successive occurrences of this loading event. The calculated values of peak and trough stresses may be joined with straight lines if desirable. Mark the highest peak of stress in each occurrence. If there are two or more equal highest peaks in one history, mark only the first peak.

H.2.2 Join the two marked points and asses only the part of the plot that falls below this line, like the section of a full reservoir.

H.2.3 Drain the reservoir from the lowest point leaving the water that cannot escape. If there are two or more equal lowest points the drainage can be from any one of them. List one cycle having a stress range S_{r1} equal to the vertical height of water drained.

*NOTE See **15.9** for low stress ranges that may be ignored.*

H.2.4 Repeat **H.2.3** successively with each remaining body of water until the whole reservoir is emptied, listing one cycle at each draining operation.

H.2.5 Compile the final list which contains all the individual stress ranges in descending order of magnitude S_{r1}, S_{r2}, etc. Where two or more cycles of equal stress range are recorded, list them separately.

H.2.6 For non-welded details only, a horizontal line representing zero stress should be plotted and those parts of the stress ranges in the compression zone modified as in **15.4**.

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