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This document comprises a front cover, an inside front cover, pages i to xviii, pages 1/1 to 5/34, pages A/1 to Z/18, pages Al/1 to Ti/12, pages I to XVI, an inside back cover and a back cover.

Foreword

Publishing information

This Published Document is published by BSI Standards Limited, under licence from The British Standards Institution, and came into effect on 31 January 2024. It was prepared by Technical Committee PVE/1, *Pressure vessels*, to provide a specification for the design, manufacture, inspection and testing of pressure vessels manufactured from carbon, ferritic alloy and austenitic steels, aluminium and aluminium alloys, copper and copper alloys, nickel and nickel alloys, titanium and titanium alloys, and duplex and super duplex steels. A list of organizations represented on this committee can be obtained on request to the committee manager.

Supersession

PD 5500:2024 supersedes PD 5500:2021+A3:2023, which is withdrawn.

Information about this document

For this edition, the opportunity has been taken to make some minor editorial changes.

PD 5500 is amended annually and the DPC (draft for public comment) is normally issued in January each year. Users of the specification are encouraged to review and comment on the proposed amendments in the DPC. This can be done by registering on the BSI standards development portal at https://standardsdevelopment.bsigroup.com/ or by obtaining a hard copy from BSI Customer Services.

The form and content of the original issue of PD 5500 was derived, without technical amendment, from the 1997 edition of BS 5500, *Specification for unfired fusion welded pressure vessels*, and all amendments issued thereto, up to and including No. 6 (September 1999). At the time PD 5500 differed from BS 5500 only insofar as it did not retain the latter's status as a national standard. This specification is thus, founded on the experience derived from the application of BS 5500 and the first edition of PD 5500, providing an integrated set of rules which have been shown to provide vessels of suitable integrity for a wide range of duties and risk environments.

BS 5500:1997 was withdrawn because its status as a national standard was incompatible with BSI's obligations to CEN consequent to the development of the European Standard EN 13445, *Unfired pressure vessels*. That European Standard was first published in May 2002. A new edition of EN 13445 was published in 2021.

The process of development of EN 13445 by CEN and its reference in the Official Journal of the European Communities creates, for equipment which conforms to that standard, a presumption of conformity with the essential safety requirements of the EU's pressure equipment directive, 2014/68/EU (see article 12 of that directive). This Published Document does not provide that presumption of conformity. However, this Published Document can be used, for vessels within the scope of directives, subject to:

– adherence of the directive's conformity assessment requirements;

– the manufacturer satisfying himself that this PD covers all the technical requirements of the Directive relevant to the vessel in question.

This use may be to cover the full range of applicable vessel requirements or to cover an issue not, at the time, appropriately supported in EN 13345.

Following the UK's withdrawal from the EU, the PED 2014/68/EU, as implemented by the Pressure Equipment (Safety) Regulations 2016, is no longer applicable for vessels placed on the market in Great Britain (England, Scotland

and Wales). For pressure vessels placed on the market in Northern Ireland, the PED, as implemented by the Pressure Equipment (Safety) Regulations 2016, as modified by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020 is still applicable – see **1.1.3**, **1.4** and Annex Z.

The normative form of wording is used in this Published Document, even though this does not have the status of a national standard, in order to ensure clarity in the definition of its requirements and recommendations.

Reference is made in the text to a number of standards which have been withdrawn. Such standards are identified in the list of references (see page V). Consideration is currently being given as to whether replacement standards are available or are being developed, for example, in the European programme, and to the implications for PD 5500 of such replacement standards. When a decision is made about any replacements standards, these will be identified by the issue of an amendment.

The British Standards Institution will be pleased to receive constructive proposals based on experience or research that may lead to improvements in this Published Document. PVE/1 intends to keep the content and technical status of this specification under review along with the need to publish appropriate supplements covering other types of pressure vessels. If there is sufficient demand from industry, this Published Document will be extended to cover other non-ferrous materials.

The requirements for materials not listed in Section 2. Materials, are given in supplements to the main text, which are to be read in conjunction with the main text so as to provide comprehensive requirements for pressure vessels produced in the relevant material. Annexes to the main text are provided which can be either normative (i.e. requirements) or informative (i.e. recommendations). These annexes can include additional requirements to the main text or informative guidance or recommendations, or can provide worked examples. Enquiry cases are published primarily to give guidance and clarification of possible ambiguities in the main text and will be incorporated into the main text or into an annex at an appropriate stage. Some Enquiry cases are published to provide new information and are identified as “preliminary rules”.

It should be noted that the effective date of amendments to this Published Document will be later than the publication date to allow users time to amend their own working procedures and documentation. See the introduction to the summary of pages table.

The following figures are reproduced by courtesy of the American Welding Research Council.

Figure G.2.5-1 was originally published as Figure 2 on page 21 of WRC Bulletin 90 September 1963.

Figure G.2.5-2 was originally published as Figure 3 on page 21 of WRC Bulletin 90 September 1963.

Figure G.2.5-3 was originally published as Figure 7 on page 24 of WRC Bulletin 90 September 1963.

Figure G.2.5-4 was originally published as Figure 8 on page 24 of WRC Bulletin 90 September 1963.

Figure G.2.5-5 was originally published as Figure 9 on page 25 of WRC Bulletin 90 September 1963.

Figure G.2.5-6 was originally published as Figure 10 on page 25 of WRC Bulletin 90 September 1963.

Figure G.2.5-7 was originally published as Figure 11 on page 26 of WRC Bulletin 90 September 1963.

Figure G.2.5-8 was originally published as Figure 12 on page 26 of WRC Bulletin 90 September 1963.

Figure G.2.6-1 to Figure G.2.6-8 are reproduced by courtesy of the *International Journal of Solids and Structures*, 1967.

This document may be referred to by the UK Health and Safety Executive (HSE) when giving guidance.

Presentational conventions

The provisions of this document are presented in roman (i.e. upright) type. Its requirements are expressed in sentences in which the principal auxiliary verb is "shall".

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

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Section 1. General

1.1 Scope

1.1.1 This Published Document specifies requirements for the design, construction, inspection, testing and verification of compliance of new unfired pressure vessels. The materials of construction are specified in Section 2. The term "pressure vessel" as used in this specification includes branches up to the point of connection to the connecting piping by bolting, screwing or welding, and supports, brackets or other attachments directly welded to the pressure containing shell. The term "unfired" excludes vessels that are subject to direct generated heat or flame impingement from a fired process. It does not exclude vessels subject to electrical heating or heated process streams.

NOTE Whilst this specification is limited to the construction of new vessels, with the agreement of the relevant parties it can be used to guide the maintenance or any modification of existing vessels. Where these existing vessels were designed and constructed using an earlier edition of PD 5500, with the agreement of the relevant parties, that earlier edition can be used to guide the maintenance or any modification.

1.1.2 In addition to the definitive requirements this specification also requires the items detailed in **1.5** to be documented. For compliance with this specification, both the definitive requirements and the documented items have to be satisfied.

1.1.3 This specification applies only to pressure vessels manufactured under the survey of a competent engineering Inspecting Authority or Organization (see **1.3.3** and **1.4.3**). The competent engineering Inspection Authority or Organization shall be one of the following:

a) an organization accredited to BS EN ISO/IEC 17020:2012, to Type A independence criteria, for inspection in the subject matter of this specification, for pressure vessels placed on the market; or

b) competent personnel of a separate engineering inspection department maintained by the purchaser of the vessel and accredited to BS EN ISO/IEC 17020:2012, to Type B independence criteria, for inspection in the subject matter of this specification, for pressure vessels for the purchaser's own use and not for resale. An inspection department maintained by the manufacturer does not satisfy this requirement except:

   1) that specific responsibilities may be delegated at the discretion of the Inspecting Authority or Organization; or

   2) in the case of vessels for the manufacturer's own use and not for resale; or

c) an organization accredited by the local Regulating Authority in countries outside the UK and the EU and in circumstances where the Pressure Equipment (Safety) Regulations 2016 (PER) and the Pressure Equipment Directive 2014/68/EU (PED) do not apply.

NOTE 1 Within the UK, the United Kingdom Accreditation Service (UKAS) acts on behalf of the regulating authority in accrediting inspection bodies.

NOTE 2 In Great Britain (England, Scotland and Wales), approved bodies and user inspectorates are appointed by the UK government for carrying out specified duties under the PER, as amended by Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

NOTE 3 In Northern Ireland, approved bodies and user inspectorates are appointed by the UK government for carrying out specified duties under the PER, as amended by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020.

NOTE 4 In the European Union, notified bodies and user inspectorates are appointed by member states of the EU for carrying out specified duties under the EU PED, or any national implementation thereof.

*NOTE 5 In countries outside the UK and the EU, local regulations can require the appointment of an inspection organization accredited by the local Regulating Authority. The role of these inspection organizations is separate from that of the Inspecting Authority as defined in **1.3.3**, but in practice one organization can perform both functions.*

This specification applies only to vessels made by manufacturers who can satisfy the Inspecting Authority or Organization that they are competent and suitably equipped to fulfil the appropriate requirements of this specification.

The requirements for testing and inspecting serially manufactured pressure vessels are given in Annex V. In all other respects the appropriate requirements in the specification apply.

Glass lined steel vessels require special design considerations subject to the limits imposed by the method of construction which should have the agreement of the Inspecting Authority.

1.1.4 This specification does not cover the following.

a) Storage tanks designed for the storage of liquids at near atmospheric pressures, i.e. where the pressure additional to that due to the hydrostatic head does not exceed 140 mbar[1)] above or 6 mbar below atmospheric pressure.

b) Low pressure, above ground storage tanks which have a single vertical axis of revolution designed for the storage of liquids at a pressure not exceeding 1 bar[1)] .

c) Vessels in which the stresses calculated in accordance with the Equations given in Section 3 are less than 10 % of the design stress permitted by Section 3.

d) Multilayered, autofrettaged, prestressed vessels or other special designs of vessels which may be appropriate for very high pressures.

e) Transport vessels, i.e. vessels used for transport of contents under pressure.

f) Vessels for specific applications which are covered by standards listed in the *BSI Catalogue*.

NOTE 1 PD 5500 may be used for the design and manufacture of liquid and bulk powder road tankers, provided consideration is given to the following:

– *chapter 6.8, clause 6.8.2 of ADR (European Agreement concerning International Carriage of Dangerous Goods by Road) 2019; in particular relating to static and dynamic stresses in motion, protection of the shell and*

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1) 1 mbar = 10^2 N/m^2 = 100 Pa.
1 bar = 10^5 N/m^2 = 0.1 N/mm^2 = 100 kPa.

supports and fittings, minimum thickness and the provision of anti-surge plates;

– *chapter 9.7, clause 9.7.5.1 of ADR relating to stability.*

BS 3441 gives guidance on the design and construction of tanks for the transport of milk and liquid milk products.

Road tankers used to transport non-hazardous substances in the UK and which operate at a pressure above 0.5 bar are subject to the Pressure Systems Safety Regulations, 2000 (SI 128).

NOTE 2 See Note 1 of 3.2.2 regarding the applicability of PD 5500 Section 3 to thick walled vessels.

NOTE 3 The titles of the publications referred to in this specification are listed at the end of the document.

1.1.5 This specification does not address the nature or consequences of a fire in the vicinity of a pressure vessel. Any consideration of the effect of a fire hazard in the design of a pressure vessel would have to be under the direction of the plant owner or his responsible agent such as the plant architect/engineer, with analysis of the consequences of a fire adjacent to a pressure vessel being undertaken in accordance with a comprehensive specification of the fire conditions, impingement parameters, analytical methods and assessment criteria.

1.1.6 This specification addresses materials in various ways.

a) The main text gives requirements for steels.

b) Certain other materials are covered by supplements which identify either where the main text is applicable or where specific requirements of the supplement apply.

1.1.7 When another standard or specification calls for the provisions of PD 5500 to be applied, the responsibility for defining the manner in which the provisions are applied and their appropriateness for the intended duty, is defined in that other document.

1.1.8 Guidance on the application of PD 5500 to pressure vessels that fall within the scope of the UK Pressure Equipment (Safety) Regulations or the EU Pressure Equipment Directive is given in Annex Z.

1.2 Interpretation

If any ambiguity be found or doubt arise as to the meaning or effect of any part of this specification or as to whether anything ought to be done or omitted to be done in order that this specification should be complied with in full, the question shall be referred to the Pressure Vessels Technical Committee (PVE/1) of the British Standards Institution, whose interpretation of the requirements of this specification upon the matter at issue shall be given free of charge and shall be final and conclusive. Parties adopting this specification for the purposes of any contract shall be deemed to adopt this provision unless they expressly exclude it or else import an arbitration provision in terms extending to interpretation of this specification. However, this provision is limited to questions of interpretation and does not confer upon the committee any power, duty or authority to adjudicate upon the contractual rights or duties of any person under a contract except in so far as they may necessarily be affected by the interpretation arrived at by the committee.

Findings or rulings of the committee upon all enquiries, including matters of interpretation, which are of sufficient importance that both enquiries and replies be made public as soon as possible will be published in an enquiry reply form for inclusion in the PD 5500 ring binder as Enquiry Cases. Their availability will be notified in *Update Standards.*

After taking into account any public comment thereon, Enquiry Cases may be incorporated, as appropriate, into this specification as amendments which will form part of the next convenient annual updating.

1.3 Definitions

For the purposes of this specification the following definitions apply.

1.3.1 purchaser

organization or individual who buys the finished pressure vessel for its own use or as an agent for the owner (see **1.4.1**)

1.3.2 manufacturer

organization that designs, constructs and tests the pressure vessel in accordance with the purchaser's order

*NOTE 1 The design function may be carried out by the purchaser or his agent, independently from the organization that constructs and tests the vessel (see **1.4.2**).*

NOTE 2 When this specification is being used to satisfy requirements of the UK PER or the EU PED, or any national implementation thereof, the definition of the manufacturer is different from that given above. For the purposes of this specification the definition given above shall apply.

1.3.3 Inspecting Authority

body or organization that verifies that the vessel has been designed, constructed and tested in accordance with this specification

NOTE When this specification is being used to satisfy requirements of the UK PER or the EU PED, or any national implementation thereof, the approved body, notified body or user inspectorate can also act as the Inspecting Authority, and for permanent joining a Recognized Third Party Organization (RTPO) can act as the Inspecting Authority.

1.3.4 Regulating Authority

authority in the country of installation that is legally charged with the enforcement of the requirements of the law and regulations of that country relating to pressure vessels

1.4 Responsibilities

*NOTE Reference should be made to Annex Z when this specification is being used to satisfy the requirements of the UK PER or the EU PED 2014/68/EU, or any national implementation thereof. Care should be taken to differentiate between the legal responsibilities of the manufacturer under the PER or the PED and the responsibilities of the manufacturer under this specification given in **1.4.2** (see also **Z.2**).*

1.4.1 Responsibilities of the purchaser

The purchaser shall be responsible for furnishing the manufacturer and the Inspecting Authority with the information required by **1.5.1**.

The purchaser shall detail in a purchase specification any requirements additional to those of this specification. Table 1.4-1 summarizes where this specification identifies additional purchase specification options.

Where the Inspecting Authority is nominated by the purchaser, the purchaser shall be responsible for ensuring that any information which the manufacturer is required to supply, as specified in this specification, is made available to the Inspecting Authority.

Where necessary, it shall be the responsibility of the purchaser to ensure that the Inspecting Authority is acceptable to the Regulating Authority.

Where the purchaser elects to perform the design function for the vessel, the purchaser shall be responsible for maintaining a complete design dossier for the vessel (see **1.5.1**) and for ensuring that all the information contained in it, or agreed modifications to it, comply with this specification; the purchaser shall also be responsible for the accuracy of all design calculations for the vessel.

Table 1.4-1 **Purchase specification options**

Clause	Item
1.4.1	Table 1.4-1, identification of purchase specification options
1.4.3b)	Responsibility of IIA regarding purchase specification
1.5.1b)	Information supplied by purchaser regarding purchase specification
2.1.1.2	Any special material limits
2.1.2.1	Use of castings for pressure parts
2.2.4	Requirements for material groups 5 and 6, and sub-group 9.2
3.1.5, Note	Zero negative thickness tolerance
3.2.2	Service life and design margin for fatigue vessels
3.2.4	Design lifetime for time dependent stresses
3.3.1	Identification of the risk of stress corrosion cracking
3.3.2	Required corrosion allowance
3.3.3	Surface finish for coated vessels
3.4.1	Choice of construction category
3.4.2.1	Reduced design stress for operational reasons
3.4.2.1a)	Any special material tests
3.8.1, Note 4	Special requirements for leak tight flanges
3.9.4.1	Tubesheet design for as-new condition
3.9.4.3.4	Tubesheet design for simultaneous shellside and tubeside pressures
3.13.2 Note 2	Additional relief capacity for the event of fire
4.2.1.2	Supplementary NDT in addition to visual inspection
4.2.6	Supplementary structural tolerances
4.3.1c)	Production control test plates
4.5.3.1d)	PWHT
4.6.2	Special finish
Table 5.1-1	Material examination etc.
5.4.1	Requirements for production control test plates for non 9 % Ni
5.6.2	NDT of parent materials
5.6.6.6	Specific method of marking
5.8.2.6	Use of differential pressure in testing of multi-compartment vessels
5.8.8.1	Gas leak test before hydraulic or pneumatic tests
C.1.2.3	Design conditions for cyclic operation in creep range
D.7.2.2	Heat treatment of production test plates with the vessel
D.7.2.2a)	Use of previous results of production test plates
D.7.2.2b)	Requirements for production weld test plates
D.7.2.2	Additional tests on test plates

Table 1.4-1 **Purchase specification options** *(continued)*

Clause	Item
N.1.1	Use of Annex N for non-UK applications
N.1.3c)	Definition of information for purchase specification
N.2.2	Extent of information to be provided by manufacturer
N.2.3	Requirement for manufacturer to recommend periodic examination
Annex S	Optional documentation
T.5.1	Optional tube-end joint tests
V.7	Records for serially produced vessels
Al.5.4.1	Need for production control test plates
Al.5.5.3	Transverse tensile tests for welder approval
Al.5.6.6.6	Specific marking for NDT
Cu.5.4.1	Production control test pieces for C106 alloy
Du.4.2.1.2	Supplementary NDT in addition to visual inspection
Du.4.3.1	Production control test plates
Ti.4.6.2	Method used to remove the alpha case surface layer

1.4.2 Responsibilities of the manufacturer

The manufacturer shall be responsible for the completeness and accuracy of all design calculations and for compliance with all applicable requirements of this specification for the whole vessel. The manufacturer shall, as required at various points in this specification, obtain the agreement of the purchaser and/or Inspecting Authority for aspects of vessel design, material choice, manufacture and testing (summarized in Table 1.4-2). These agreements shall be documented for inclusion in the vessel records [see **1.5.2.2**e)].

If during fabrication unexpected deviations from the requirements of this specification arise, that the manufacturer can justify will not impair the intended functionality or integrity of the vessel, the manufacturer may submit a justification to the purchaser and Inspecting Authority for their approval. Such approved deviations shall be documented for inclusion in the vessel records [see **1.5.2.2**e)] and require the application of suffix XX to Form X (see **1.4.4** and Note 1 to Form X).

Where the Inspecting Authority is not nominated by the purchaser, the manufacturer shall appoint an Inspecting Authority. The manufacturer shall be responsible for ensuring that the Inspecting Authority is provided with any information the manufacturer is required to supply, as specified in this specification.

The organization which discharges the manufacturer's responsibilities for design, construction and testing may elect to have any of these activities carried out by subcontractors. It shall assume overall responsibility for compliance with this specification during all related activities including part manufacture and subsequent fabrication to completion at works and/or site. It shall satisfy the Inspecting Authority, as necessary, under the general provisions of **1.1** that it is competent to ensure by appropriate control or surveillance of such activities, whether carried out by itself or by subcontractors, that all the relevant requirements of this specification are met.

Examinations carried out by the Inspecting Authority do not absolve the manufacturer from his responsibility for compliance with the applicable requirements of this specification.

Where the purchaser elects to perform the design function for the vessel, the manufacturer shall be responsible for ensuring that all the design information he requires to construct and test the vessel is provided by the purchaser and for ensuring that all construction and testing is carried out in compliance with this specification. The manufacturer shall also be responsible for the accuracy of any information he provides to enable the purchaser to fulfil the design function.

Table 1.4-2 **Items for manufacturer, purchaser and/or Inspecting Authority agreement**

Clause	Item
2.1.2.1c)	Use of non BS, BS EN or European approved materials
2.1.2.2.1	Choice of equivalent material group
Table 2.1-3	Use of steels with C content > 0.25 %
2.1.2.2.2.3	Acceptance of properties obtained by other recognized test methods
2.1.2.2.2.4	Acceptance of properties obtained by other recognized test methods
2.1.2.2.2.7, Note 2	Use of rolling process to achieve normalized properties
Table 2.1-4, Note 5	Additional procedure for correction of minor defects in category 3 components
2.2.1b)	Consideration of materials for low temperature testing
2.2.4	Requirements for material in groups 5 and 6, and sub-group 9.2
2.3.1.1a)	Procedure for determination of design strength
2.3.1.1	Extrapolation of material strength data at high temperature
2.3.3.2 Note	Use of $R_{e(T)}$ values equal to those of a material with similar composition and heat treatment
2.3.3.3 Note	Use of $R_{e(T)}$ values equal to those of a material with similar composition and heat treatment
3.2.2b)	Alternative design methods
3.2.2b)4)	Alternative design by comparison with similar vessels
3.2.3	Design vacuum with break valve
3.3.3	Cladding thickness included in calculation in thickness
3.4.2.1b)2)	Alloy design stresses with PWHT not to Table 4.5-1
3.8.1.4	Bolt design stresses higher than Table 3.8-1
3.8.1.5	Use of plate material for hubbed flanges
4.1.1	Commencement of manufacture before drawing approval
4.1.1	Modification of approved drawings
4.1.6	Additional NDT after rectification of departure from tolerance
4.2.1.1	Dressing of cut edges of ferritic steels
4.2.1.2	Major defect notification and rectification
4.2.2.2	NDT method on butt welded plates prior to forming
4.2.2.4.1	Ferritic steel: forming procedure involving plate heating; possible supporting data
4.2.2.4.2	Austenitic steel: forming procedure involving plate heating
4.2.2.5	Sectional flanging for dished ends
4.2.3.1a)	Assembly tolerance of longitudinal joints e > 200 mm
4.2.3.1b)	Assembly tolerance of circumferential joints e > 200 mm
4.2.4.2.3	Circularity tolerance at nozzle
4.3.2.2	Weld metal properties for 9 % Ni

Table 1.4-2 **Items for manufacturer, purchaser and/or Inspecting Authority agreement** *(continued)*

Clause	Item
4.3.2.4	Austenitic stainless steel consumables for 9 % Ni below -101 °C
4.3.6.2	Permanent backing strips for inaccessible second side
4.3.6.2	Backing strips for tubes
4.5.3.1	PWHT ferritic steel thicker than Table 4.5-1
4.5.3.1	Welding of lightly loaded parts after heat treatment
4.5.3.5	PWHT for austenitic stainless steel
4.5.4.4b)	Requirements for local heat treatment of branches
4.5.5.1	PWHT for materials other than sub-groups 1.1, 1.2 or 1.3
5.2.5	Criteria for all weld test
5.3.3	Retake whole or part of welder approval test
5.4.1	Number and details of production control test plates for non-9 % Ni
5.4.2	Number and details of production control test plates for 9 % Ni
5.6.2	Acceptance standards for flaws in parent plates and NDT of repairs
5.6.4.1.2	Method of examination for surface flaws in type A welds
5.6.4.3	Additional testing for visual examination of category 3 components
5.6.5.1	Choice of NDT method
5.6.5.2	Alternative NDT methods to assess depth surface defects
5.6.6.1	Alternative radiographic techniques
5.7.2	Assessment of defects to Annex U
5.7.2.2	Deviation from acceptance criteria of Table 5.7-1, Table 5.7-2 and Table 5.7-3
5.7.3	Repair of welds
5.8.2.2	Pressure test procedure
5.8.2.4	Elevation of test temperature to avoid brittle fracture
5.8.2.5	Pressure test procedure for large vessels
5.8.4.1b)	Consultation for pneumatic testing
5.8.5.5	Testing vacuum vessels with anti-vacuum valves
5.8.6.1	Proof test procedure
5.8.6.2	Strain indicating coating for proof test
5.9.1	Quality specification for castings
5.9.2	Examination and heat treatment of weld repair of cast components
C.3.4.3.1	Raising the weld class for cyclic loading
C.3.4.6.4	Stress factor for machined weld toes
D.1	Alternative to Annex D requirements
D.7.1, Note	Alternative toughness requirements using Annex U
D.7.2.2	Waiving of impact testing of production weld test plates
D.7.2.3a), Note	Position of impact test specimen
H.3.2	PWHT of different material grades with differences in temperature >10 °C
H.5.3	Time at PWHT

Table 1.4-2 **Items for manufacturer, purchaser and/or Inspecting Authority agreement** *(continued)*

Clause	Item
Annex J	Fitting of additional pressure relief valve(s)
K.1.4.1.2	Extrapolation of upper temperature stress values
K.1.4.2, Note 1d)	Design strengths at higher temperatures
K.1.4.2, Note 17	Design strengths for slit rimming tubes
Q.1.3	Selection of test specimen
Q.2.3.3	Maximum strength of all-weld tensiles
Q.2.6	Micro-examination of production control test plates
T.4.3a)	Alternative leak test method for tube-end joints
T.5.3	Tracer gas test of tube-end joints
T.5.4	Radiography for back-face tube to tubesheet welds
T.5.5	Frequency of production control test pieces for tube-end joints
T.5.5	Testing production control test pieces
U.1	Fracture mechanics approach
V.4.4	Reduction in extent of weld examination for serially produced vessels
Al.2.1.2.1b)	Use of non BS EN or European approved materials
Al.2.3.1.3	Extrapolation of material strength data at high temperature
Al.4.2.2.1	Use of local heating and hammering for forming
Al.4.2.2.4.2	Softening for cold worked aluminium
Al.4.3.6.2	Use of backing rings in circumferential butt joints in tubing
Al.4.3.7.7	Alternative to two layers of weld metal at branches etc.
Al.4.5.3.1	Details of any post weld heat treatment
Al.5.4.1	Number and detail of production control test plates
Al.5.4.4	Selection of test specimen
Al.5.5.5	Optional micro examination of production, welder and production control tests
Al.5.6.2	Acceptance standards and techniques for defects in parent materials
Al.5.6.4.1.2	Need for surface flaw examination of full penetration butt welds
Al.5.6.5.2	Alternative method for surface flaw detection
Al.5.7.3.2	Alternative acceptance levels for assessment defects
Al.5.7.4	Repair of welds
Cu.2.1.2.1c)	Use of non BS, BS EN or European approved materials
Cu.2.3.1.6	Extrapolation of material strength data at high temperature
Cu.4.2.2.3	Heat treatment after cold forming
Cu.4.3.2	Non-matching filler rod metals
Cu.4.3.6	Temporary/consumable backing strips for circumferential tube welds
Cu.4.5.1	Preheat requirements for each type of weld
Cu.5.2	Standard for approval testing of welding and brazing procedures
Cu.5.4.1	Details of production control testing of C106 sheet
Cu.5.4.2	Details of production control testing of non-C106 alloys

Table 1.4-2 **Items for manufacturer, purchaser and/or Inspecting Authority agreement** *(continued)*

Clause	Item
Cu.5.5.4	Macro- and micro-examination of test specimen
Cu.5.6.2	Acceptance standards for defects in parent material
Cu.5.6.6.1	Alternative to radiographic techniques
Du.2.1.2.1c)	Use of non BS, BS EN or European approved materials
Du.2.1.2.2	Choice of equivalent grouping for material covered by **Du.2.1.2.1**c)
Du.2.3.1.1	Extrapolation of material strength data at high temperature
Du.4.3.5.3	Dissimilar metal welds
Du.4.5.3	Requirement for post weld heat treatment
Du.4.5.5	Post-weld heat treatment procedure
Du.5.3	Acceptance limits for micro structural analysis of welder and operator approval tests
Du.5.7.3.3	Use of multiple weld repairs
Du.D.2	Material properties for minimum design temperature below -50°C, and/or a nominal thickness of greater than 50 mm
Ni.2.1.2.1c)	Use of non BS or European approved materials
Ni.2.3.1.3	Extrapolation of material strength data at high temperature
Ni.4.5.5	Heat treatment procedure for PWHT
Ti.2.1.2.1b)	Use of non ASTM or European approved materials
Ti.2.3.1.3	Extrapolation of material strength data at high temperature
Ti.4.5.3	Requirement for post-weld heat treatment
Ti.4.5.5	Heat treatment procedure for PWHT
Enquiry cases	Application of any specific enquiry cases

1.4.3 Responsibilities of the Inspecting Authority

The Inspecting Authority shall be responsible for verifying:

a) that all parts of the vessel have been designed in accordance with the requirements of this specification as are applicable for the conditions specified by the purchaser according to **1.5.1**;

b) that the vessel has been constructed and tested in accordance with this specification and any additional requirements in respect of purchaser options covered in the purchase specification (see **1.4.1**).

1.4.4 Certificate of Conformance

On completion of the vessel the manufacturer shall issue Form X to certify that the vessel has been designed, constructed and tested in every respect in accordance with this specification and with any additional requirements in respect of the purchaser's options covered by this specification. Form X shall be countersigned by the Inspecting Authority as required.

Where some of the activities covered by this specification are performed under the surveillance of a second Inspecting Authority, each Inspecting Authority shall attach a statement to Form X, countersigned as required thereon, confirming which part of the total works has been carried out under its surveillance.

The countersigned Certificate and its attachments (if any) shall be furnished to the purchaser with a copy of the Regulating Authority if required.

Where the purchaser or his appointed design consultant/contractor elects to perform the design function for the vessel, the purchaser or his appointed design consultant/contractor shall complete the section of Form X which certifies that the design of the vessel complies with this specification.

If a vessel is subject to deviations from the requirements of this specification the vessel serial number on Form X shall have the suffix XX added (see **1.4.2**).

NOTE Form X may be reproduced as hard copy or by electronic means provided that such reproductions are fair copies of the original. All copies should state "Reproduced from PD 5500" with a reference to the current issue.

Form X — Certificate of Conformance

Vessel description	Type				
	Approximate overall dimensions	. . . .			
	Approved drawing number(s)	. . . .			
	Year of manufacture	Relevant PD 5500 edition including Amd. No(s).			
	Purchaser	Purchaser's serial no.			
Manufacture	Name of manufacturer				
	Manufacturer's serial number . . . .(see NOTE 1)				
	Name of Design Organization (if not above manufacturer).				
	Name of Inspecting Authority				
Design	**Design conditions of principal components** (see NOTES 2 and 3)	**Design pressure** bar	**Design temp.** °C	**Construction category**	**Corrosion allowance** mm
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	. . . .	. . . . . .	. . . . . .	. . . .	. . . .
	Other factors affecting design (e.g. weight, nature of contents, environment) (see NOTES 3 and 4)				
Post -weld heat treatment	**Component**	**Temperature** °C	**Holding time** h		
	. . . .	. . . .	. . . .		
	. . . .	. . . .	. . . .		
	. . . .	. . . .	. . . .		
	. . . .	. . . .	. . . .		
	. . . .	. . . .	. . . .		
	. . . .	. . . .	. . . .		
Pressure test	**Location** (see NOTE 5)	**Test pressure** bar	**Test medium and temperature**	**Date**	
	. . . .	. . . .	. . . .	. . . .	
	. . . .	. . . .	. . . .	. . . .	
	. . . .	. . . .	. . . .	. . . .	
	. . . .	. . . .	. . . .	. . . .	

Form X — Certificate of Conformance *(continued)*

Certificate of Conformance (design)

Approved drawing number(s) . . . .
. . . .
. . . .

We hereby certify that the design of this vessel conforms to PD 5500.

For manufacturer (see NOTE 6): Date:

Position: Name of Company:

We hereby certify that we have checked the design of the above vessel and that this conforms to PD 5500.

For Inspecting Authority: Date:

Position: Name of Company:

Certificate of Conformance (construction and testing)

Manufacturer's serial number:

We hereby certify that this vessel has been constructed and tested in conformance with PD 5500.

For manufacturer : Date:

Position: Name of Company:

We hereby certify that the construction and testing of the above vessel has been carried out under our surveillance and that to the best of our knowledge and belief all aspects of this work conform to PD 5500.

For Inspecting Authority: Date:

Position: Name of Company:

To this certificate shall be attached, where appropriate, the records of any items that caused an XX designation along with the records of the decisions arrived at on any items from Table 1.4-2 that were subject to agreement.

NOTE 1 The suffix 'XX' should be added to the serial number of each vessel for which any deviations or concessions have been authorized (see ***1.4.2*** *and* ***1.4.4****).*

NOTE 2 The design conditions associated with operational duties specified by the purchaser should be given. Where a vessel is subject to more than one loading/temperature condition, each of the conditions should be given.

If a purchaser wishes to change the operational duty of a vessel, revised design conditions, consistent with the vessel scantlings, should be established separately, as appropriate.

NOTE 3 Where the design covers operation below 0 ℃ the various combinations of temperature, pressure and calculated membrane stresses considered in determining the minimum design temperature (see ***3.2.5*** *and Annex D) should be stated.*

NOTE 4 Where appropriate, cross reference to drawings or specifications is acceptable.

NOTE 5 Where a vessel is tested in a different orientation to that in which it will normally operate, this should be stated.

NOTE 6 This part of the Certificate is to be signed by the purchaser in cases where the purchaser elects to perform the design function (see ***1.4.4****).*

1.5 Information and requirements to be agreed and to be documented

1.5.1 Information to be supplied by the purchaser

The following information shall be supplied by the purchaser and shall be fully documented.

a) The normal working conditions of the required vessel, together with details of any transient cyclic and/or adverse conditions in which the vessel is required to operate and any special requirements for in-service inspection.

b) Any requirements relating to the various options covered by the purchase specification (see **1.4.1**).

c) Any special statutory or other regulations with which the finished vessel is required to comply.

d) The name of the Inspecting Authority to be commissioned by the purchaser.

e) The name of the Regulating Authority (if any).

f) The requirement to obtain copies for record purposes of any documents other than those listed in **1.5.2.2**a) to **1.5.2.2**g). (To facilitate the identification of such documents, a check list of optional documents is given in Annex S.)

   Where the purchaser elects to perform the design function for the vessel, the purchaser shall supply any additional design information required by the manufacturer in accordance with **1.4.2**. The design dossier maintained by the purchaser in accordance with **1.4.1** shall cover all the information (whether supplied by the purchaser or by the manufacturer) which the manufacturer would otherwise be required by **4.1.1** to submit before commencing manufacture.

The fulfilment of specified requirements and the availability of documentation/records identified shall both be satisfied before a claim of conformance with the specification can be made and verified.

1.5.2 Information to be supplied by the manufacturer

The information in **1.5.2.1** and **1.5.2.2** shall be supplied by the manufacturer and shall be fully documented.

The fulfilment of specified requirements and the availability of documentation/records identified shall both be satisfied before a claim of conformance with the specification can be made and verified.

1.5.2.1 Before commencement of manufacture

The manufacturer shall submit the information specified in **4.1.1** for approval before commencement of manufacture. In submitting this information, the manufacturer shall identify, in an appropriate manner, any features of the proposed design and/or in the proposed manufacturing, inspection or test procedures which by the terms of this specification require to be approved by the purchaser.

NOTE Table 1.4-2 lists such features. The features should be identified in an appropriate document such as purchase order, approved drawing or an approved working procedure (e.g. weld preparation procedure, heat treatment procedure, welding procedure etc.).

1.5.2.2 On completion of construction

The manufacturer shall supply to the purchaser for record purposes a copy of the following documentation, as finally approved, for each vessel or batch of vessels.

a) A fully dimensioned drawing of the vessel, as built, together with any relevant supporting information as specified in **4.1.1** and which is not covered by items b) to g) as follows.

b) A list of materials (including welding consumables) used in the construction of the vessel with details of any special heat treatments carried out by the material supplier.
For materials specified to a British Standard the date of the standard shall be given. Where other materials are used (see **2.1.2.1**b)) the full specification shall be supplied.

c) Factors affecting the equipment operation or design life – safe operating limits and design basis (which may be on the drawings).

d) Factors affecting the design life i.e.

   1) Creep — design hours of operation at specified temperature.

   2) Fatigue — design number of cycles at specified stress levels.

   3) Corrosion — design corrosion allowance.

   4) Residual hazards not prevented by design or protective measures that might arise from foreseeable misuse.

   5) Information about replaceable parts (liners, wear plates etc.).

e) The welding procedures used during vessel manufacture (see **5.2.2**).

f) The procedures used for radiography, ultrasonic inspection and/or crack detection of welds (see **5.6.6.1**, **5.6.6.2**, **5.6.6.3** and **5.6.6.4**).

g) Records detailing all deviations from this specification and the decision made on all matters subject to agreement (see **1.4.2**).

h) A Certificate of Conformance (Form X) for each vessel (see **1.4.4**).

i) A facsimile of the vessel nameplate (see **5.8.9**).

j) Details of any statutory required marking (e.g. CE marking).

k) Operating instructions for mounting, putting into service its use and maintenance (as far as is relevant).

NOTE 2 The duration for which a manufacturer will retain all records produced during the manufacture of a vessel is influenced by a number of factors which are outside the scope of this specification.

1.6 Thicknesses

Thicknesses are referred to in several ways in this specification in accordance with the following definitions.

a) Minimum thickness; the thickness calculated in Section 3 and some design annexes, to satisfy the relevant design requirement.

b) Nominal thickness; the thickness as specified on the construction drawing which includes all allowances and tolerances.

c) Analysis thickness; the thickness used in design calculations and assessments, which equals the nominal thickness less corrosion allowances and less any other allowances and tolerances.

The nominal thickness, less any negative tolerance permitted by the specification to which the material is ordered, shall not be less than the minimum thickness plus any allowances specified for corrosion, erosion etc.

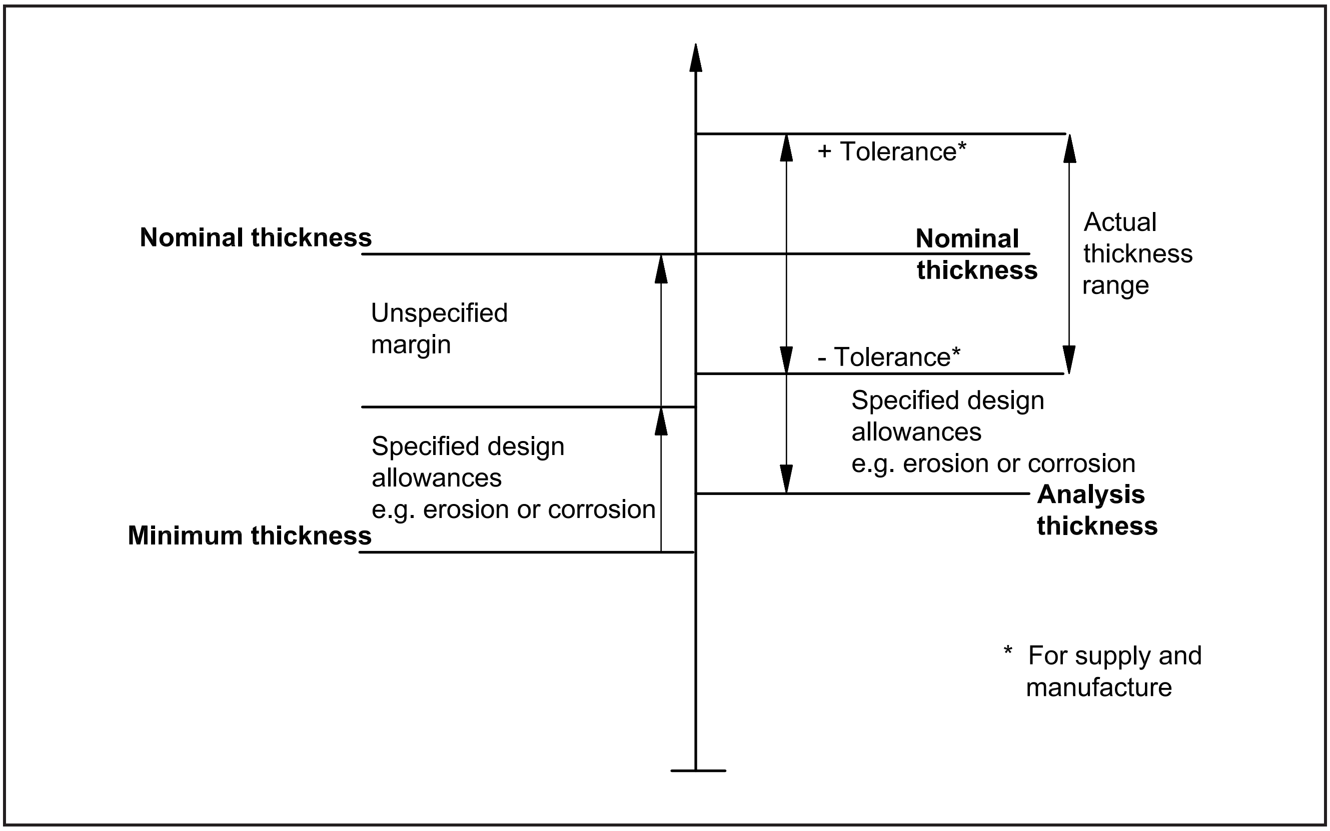
The relationships between the defined thicknesses are shown in Figure 1.6-1. This figure indicates:

a) on the left hand side, that the thickness calculated from the rules is increased by the amount of the specified allowances (for effects such as corrosion) and possibly by an unspecified margin (in consideration, for example, of materials availability);

b) on the right hand side, that the nominal thickness should be reduced by any negative supply (i.e. negative plate thickness) tolerance and any manufacturing (i.e. dishing) allowances as well as by the specified design allowances, to arrive at the analysis thickness.

NOTE Following the replacement of BS 1501 by BS EN 10028, it should be noted that plate conforming to this latter specification is normally supplied in accordance with BS EN 10029 class B which permits a negative tolerance of 0.3 mm for all nominal thicknesses. However, the purchaser may specify a zero negative tolerance.

Figure 1.6-1 **Relationship of thickness definitions**



Section 2. Materials

2.1 Selection of materials

2.1.1 General

2.1.1.1 The selection of the materials of construction for pressure containing parts and their integral attachments shall take into account the suitability of the material with regard to fabrication[1)] and to the conditions under which they will eventually operate.

2.1.1.2 Any special limits for the materials of construction, e.g. composition, heat treatment and non-destructive testing (NDT), which are different to those given in this specification shall be specified in the purchase specification.

2.1.1.3 For the ease of reference throughout this specification, materials, with the exception of clad material, are identified by a group number which has been derived from PD CEN ISO/TR 15608:2017. This grouping is summarized in Table 2.1-1.

2.1.1.4 All material chemical compositions are specified in percentage by mass, in accordance with material standards and industry norms.

2.1.2 Materials for pressure parts

2.1.2.1 General

Pressure parts are defined as components (see **3.4.1**) which constitute the envelope under pressure and which are essential for the integrity of the vessel. Examples of pressure parts are shells, ends, flanges, nozzles, and heat exchanger tubes and tubesheets.

All the materials used in the manufacture of pressure parts shall:

a) conform to BS EN steels given in Table 2.1-2; or

b) conform to BS steels given in Table K.1-2 to Table K.1-12; or

c) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2**b)] and shall conform to **2.1.2.2**; or

d) be the subject of a “European approval for material” in accordance with the EU directive 2014/68/EU (see Note 1); or

e) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE 1 Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Welding material shall conform to Section 4 and Section 5.

The option to be able to use castings for pressure parts shall be specified in the purchase specification. An appropriate “quality specification” for such castings shall be agreed between the manufacturer and material supplier at the time of enquiry, specifying the standards of inspection to be applied and of acceptance for defects. As a minimum, all accessible fillets and changes of section, etc., shall be subject to magnetic particle or penetrant inspection.

[1)] See BS EN 1011 for general guidance on the susceptibility of materials to lamellar tearing during fabrication.

Materials for bolts and nuts shall conform to the specifications listed in Table 3.8-1. Requirements for bolting materials for low temperature application are referred to in **2.2.2**.

NOTE 2 For equipment that is subject to the Pressure Equipment Directive PED 2014/68/EU, requirements for the selection of bolting are given in BS EN 1515-4.

2.1.2.2 Materials covered by 2.1.2.1c)

2.1.2.2.1 General

The material shall be covered by a written specification at least as comprehensive as the nearest equivalent British Standard listed in Table 2.1-2. The written specification shall as a minimum specify the manufacturing process, compositional limits for all constituents, deoxidation practice, heat treatment and appropriate mechanical properties for acceptance and other purposes.

An equivalent grouping for the material (see **2.1.1.3**) shall be chosen and agreed between manufacturer, inspection authority and purchaser. This grouping shall be used to assist in the selection of manufacturing and inspection requirements of Section 4 and Section 5.

Table 2.1-1 **Material grouping**

Sub-group	Type of steel
Group 1	
—	Steels with a specified minimum yield strength $R_{eH} \leq 460$ N/mm^{2}[a] and with analysis in %: C ≤ 0.25 Si ≤ 0.60 Mn ≤ 1.8 Mo ≤ 0.70[b] S ≤ 0.045 P ≤ 0.045 Cu ≤ 0.40[b] Ni ≤ 0.5[b] Cr ≤ 0.3 (0.4 for castings)[b] Nb ≤ 0.06 V ≤ 0.1[b] Ti ≤ 0.05
1.1	Steels with a specified minimum yield strength $R_{eH} \leq 275$ N/mm^2
1.2	Steels with a specified minimum yield strength 275 N/mm^2 < $R_{eH} \leq 360$ N/mm^2
1.3	Normalized fine grain steels with a specified minimum yield strength $R_{eH} > 360$ N/mm^2
1.4	Steels with improved atmospheric corrosion resistance whose analysis may exceed the requirements for the single elements as indicated in group 1
Group 2	
—	Thermomechanically treated fine grain steels and cast steels with a specified minimum yield strength $R_{eH} > 360$ N/mm^2
2.1	Thermomechanically treated fine grain steels and cast steels with a specified minimum yield strength 360 N/mm^2 < $R_{eH} \leq 460$ N/mm^2
2.2	Thermomechanically treated fine grain steels and cast steels with a specified minimum yield strength $R_{eH} > 460$ N/mm^2
Group 3	
—	Quenched and tempered and precipitation hardened fine grain steels, except stainless steels, with a specified minimum yield strength $R_{eH} > 360$ N/mm^2
3.1	Quenched and tempered fine grain steels with a specified minimum yield strength 360 N/mm^2 < $R_{eH} \leq 690$ N/mm^2
3.2	Quenched and tempered fine grain steels with a specified minimum yield strength $R_{eH} > 690$ N/mm^2
3.3	Precipitation hardened fine grain steels except stainless steels
Group 4	
—	Low vanadium alloyed Cr-Mo-(Ni) steels with Mo ≤ 0.7% and V ≤ 0.1%
4.1	Steels with Cr ≤ 0.3% and Ni ≤ 0.7%
4.2	Steels with Cr ≤ 0.7% and Ni ≤ 1.5%

Table 2.1-1 **Material grouping** *(continued)*

Sub-group	Type of steel
Group 5	
—	Cr-Mo steels free of vanadium with C ≤ 0.35%
5.1	Steels with 0.75% ≤ Cr ≤ 1.5% and Mo ≤ 0.7%
5.2	Steels with 1.5% < Cr ≤ 3.5% and 0.7% < Mo ≤ 1.2%
5.3	Steels with 3.5% < Cr ≤ 7.0% and 0.4% < Mo ≤ 0.7%
5.4	Steels with 7.0% < Cr ≤ 10.0% and 0.7% < Mo ≤ 1.2%
Group 6	
—	High vanadium alloyed Cr-Mo-(Ni) steels
6.1	Steels with 0.3% ≤ Cr ≤ 0.75%, Mo ≤ 0.7% and V ≤ 0.35%
6.2	Steels with 0.75% < Cr ≤ 3.5%, 0.7% < Mo ≤ 1.2% and V ≤ 0.35%
6.3	Steels with 3.5% < Cr ≤ 7.0%, Mo ≤ 0.7% and 0.45% ≤ V ≤ 0.55%
6.4	Steels with 7.0% < Cr ≤ 12.5%, 0.7% < Mo ≤ 1.2% and V ≤ 0.35%
Group 7	
—	Ferritic, martensitic or precipitation hardened stainless steels with C ≤ 0.35% and 10.5% ≤ Cr ≤ 30%
7.1	Ferritic stainless steels
7.2	Martensitic stainless steels
7.3	Precipitation hardened stainless steels
Group 8	
—	Austenitic stainless steels, Ni ≤ 35%
8.1	Austenitic stainless steels with Cr ≤ 19%
8.2	Austenitic stainless steels with Cr > 19%
8.3	Manganese austenitic stainless steels with 4.0% < Mn ≤ 12.0%
Group 9	
—	Nickel alloy steels with Ni ≤ 10.0%
9.1	Nickel alloy steels with Ni ≤ 3.0%
9.2	Nickel alloy steels with 3.0% < Ni ≤ 8.0%
9.3	Nickel alloy steels with 8.0% < Ni ≤ 10.0%
Group 10	
—	Austenitic ferritic stainless steels (duplex)
10.1	Austenitic ferritic stainless steels with Cr ≤ 24.0% and Ni > 4.0%
10.2	Austenitic ferritic stainless steels with Cr > 24.0% and Ni > 4.0%
10.3	Austenitic ferritic stainless steels with Ni ≤ 4.0%

Table 2.1-1 **Material grouping** *(continued)*

Sub-group	Type of steel
Group 11	
—	Steels covered by group 1[c] except 0.25%[d] < C ≤ 0.85%
11.1	Steels as indicated under 11 with 0.25%[d] < C ≤ 0.35%
11.2	Steels as indicated under 11 with 0.35% < C ≤ 0.5%
11.3	Steels as indicated under 11 with 0.5% < C ≤ 0.85%

Based on the actual product analysis, group 2 steels may be considered group 1 steels.
If a material has different minimum specified yield strengths depending on the thickness, the highest yield strength shall be used for the determination of the subgroup.

[a] In accordance with the specification of the steel product standards, R_{eH} may be replaced by $R_{p0.2}$ or $R_{t0.5}$.

[b] A higher value is accepted provided that Cr + Mo + Ni + Cu + V ≤ 0.75%.

[c] A higher value is accepted provided that Cr + Mo + Ni + Cu +V ≤ 1%.

[d] The minimum carbon content in PD CEN ISO/TR 15608:2017 has been increased to 0.30%, but for group 11 materials in PD 5500 the minimum carbon content shall be taken as 0.25%

Table 2.1-2 **List of materials covered by BS EN material standards**

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treat-ment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
1	Plate and strip	BS EN 10028-2	Elevated temperature properties	P235GH	1.0345	N	0	150	1.1	a)
2	Plate and strip	BS EN 10028-2	Elevated temperature properties	P265GH	1.0425	N	0	150	1.1	a)
3	Plate and strip	BS EN 10028-2	Elevated temperature properties	P295GH	1.0481	N	0	150	1.2	a)
4	Plate and strip	BS EN 10028-2	Elevated temperature properties	P355GH	1.0473	N	0	150	1.2	a)
5	Plate and strip	BS EN 10028-2	Elevated temperature properties	16Mo3	1.5415	N	0	150	1.1	a) f)
6	Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5	1.7335	NT	0	60	5.1	a)
7	Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5	1.7335	NT QT	60	100	5.1	a)
8	Plate and strip	BS EN 10028-2	Elevated temperature properties	13CrMo4-5	1.7335	QT	100	150	5.1	a)
9	Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10	1.7380	NT	0	60	5.2	a)
10	Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10	1.7380	NT QT	60	100	5.2	a)
11	Plate and strip	BS EN 10028-2	Elevated temperature properties	10CrMo9-10	1.7380	QT	100	150	5.2	a)
12	Plate and strip	BS EN 10028-2	Elevated temperature properties	11CrMo9-10	1.7383	NT QT	0	60	5.2	a)
13	Plate and strip	BS EN 10028-2	Elevated temperature properties	11CrMo9-10	1.7383	QT	60	100	5.2	a)
14	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P275N	1.0486	N	0	150	1.1	a)
15	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P275NH	1.0487	N	0	150	1.1	a)
16	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P275NL1	1.0488	N	0	150	1.1	a)
17	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P275NL2	1.1104	N	0	150	1.1	a)
18	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P355N	1.0562	N	0	150	1.2	a)
19	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P355NH	1.0565	N	0	150	1.2	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	Product form	EN standard	Material description	Grade	Material number	Heat treat-ment[a]	Thickness mm		Material group to PD CEN ISO/TR 15608:2017	Notes
							min.	max.		
20	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P355NL1	1.0566	N	0	150	1.2	a)
21	Plate and strip	BS EN 10028-3	Fine grain steel normalized	P355NL2	1.1106	N	0	150	1.2	a)
22	Plate and strip	BS EN 10028-4	Low temperature properties	11MnNi5-3	1.6212	N NT	0	50	9.1	a)
23	Plate and strip	BS EN 10028-4	Low temperature properties	13MnNi6-3	1.6217	N NT	0	50	9.1	a)
24	Plate and strip	BS EN 10028-4	Low temperature properties	15NiMn6	1.6228	N NT QT	0	50	9.1	a)
25	Plate and strip	BS EN 10028-4	Low temperature properties	12Ni14	1.5637	N NT QT	0	50	9.2	a)
26	Plate and strip	BS EN 10028-4	Low temperature properties	12Ni19	1.5680	N NT QT	0	50	9.2	a)
27	Plate and strip	BS EN 10028-4	Low temperature properties	X8Ni9	1.5662	N NT QT	0	50	9.3	a)
28	Plate and strip	BS EN 10028-4	Low temperature properties	X7Ni9	1.5663	QT	0	50	9.3	a)
29	Plate and strip	BS EN 10028-6	Fine grain steel, quenched/tempered	P355Q	1.8866	QT	0	150	1.2	a)
30	Plate and strip	BS EN 10028-6	Fine grain steel, quenched/tempered	P355QH	1.8867	QT	0	150	1.2	a)
31	Plate and strip	BS EN 10028-6	Fine grain steel, quenched/tempered	P355QL1	1.8868	QT	0	150	1.2	a)
32	Plate and strip	BS EN 10028-6	Fine grain steel, quenched/tempered	P355QL2	1.8869	QT	0	150	1.2	a)
33	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiN18-7	1.4318	AT	0	75	8.1	a)
34	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNi18-9	1.4307	AT	0	75	8.1	a)
35	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNi19-11	1.4306	AT	0	75	8.1	a)
36	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiN18-10	1.4311	AT	0	75	8.1	a)
37	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X5CrNi18-10	1.4301	AT	0	75	8.1	a)
38	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X5CrNiN19-9	1.4315	AT	0	75	8.1	a)
39	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNi18-10	1.4948	AT	0	75	8.1	a)
40	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNi23-13	1.4950	AT	0	75	8.2	a)
41	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNi25-20	1.4951	AT	0	75	8.2	a)
42	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNiTi18-10	1.4541	AT	0	75	8.1	a)
43	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNiTiB18-10	1.4941	AT	0	75	8.1	a)
44	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiMo17-12-2	1.4404	AT	0	75	8.1	a)
45	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiMoN17-11-2	1.4406	AT	0	75	8.1	a)
46	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X5CrNiMo17-12-2	1.4401	AT	0	75	8.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
47	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X6CrNiMoTi17-12-2	1.4571	AT	0	75	8.1	a)
48	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiMo17-12-3	1.4432	AT	0	75	8.1	a)
49	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiMo18-14-3	1.4435	AT	0	75	8.1	a)
50	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X2CrNiMoN17-13-5	1.4439	AT	0	75	8.1	a)
55	Plate and strip	BS EN 10028-7	Stainless steel, austenitic	X3CrNiMoBN17-13-3	1.4910	AT	0	75	8.1	a)
56	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X1CrNi25-21	1.4335	AT	0	75	8.2	a)
57	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X6CrNiNb18-10	1.4550	AT	0	75	8.1	a)
58	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X8CrNiNb16-13	1.4961	AT	0	75	8.1	a)
59	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X1CrNiMoN25-22-2	1.4466	AT	0	75	8.2	a)
60	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X6CrNiMoNb17-12-2	1.4580	AT	0	75	8.1	a)
61	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X2CrNiMoN17-13-3	1.4429	AT	0	75	8.1	a)
62	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X3CrNiMoN17-13-3	1.4436	AT	0	75	8.1	a)
63	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X2CrNiMoN18-12-4	1.4434	AT	0	75	8.1	a)
64	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X2CrNiMo18-15-4	1.4438	AT	0	75	8.1	a)
66	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X1CrNiMoCuN25-25-5	1.4537	AT	0	75	8.2	a)
67	Plate and strip	BS EN 10028-7	Stainless steel, austenitic, special	X1CrNiMoCuN20-18-7	1.4547	AT	0	75	8.2	a)
69	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNi18-9	1.4307	AT	0	250	8.1	a)
70	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNi19-11	1.4306	AT	0	250	8.1	a)
71	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiN18-10	1.4311	AT	0	250	8.1	a)
72	Bar	BS EN 10272	Stainless steel, austenitic	X5CrNi18-10	1.4301	AT	0	250	8.1	a)
73	Bar	BS EN 10272	Stainless steel, austenitic	X6CrNiTi18-10	1.4541	AT	0	250	8.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	Product form	EN standard	Material description	Grade	Material number	Heat treat-ment[a]	Thickness mm		Material group to PD CEN ISO/TR 15608:2017	Notes
							min.	max.		
74	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMo17-12-2	1.4404	AT	0	250	8.1	a)
75	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMoN17-11-2	1.4406	AT	0	250	8.1	a)
76	Bar	BS EN 10272	Stainless steel, austenitic	X5CrNiMo17-12-2	1.4401	AT	0	250	8.1	a)
77	Bar	BS EN 10272	Stainless steel, austenitic	X6CrNiMoTi17-12-2	1.4571	AT	0	250	8.1	a)
78	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMo17-12-3	1.4432	AT	0	250	8.1	a)
79	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMo18-14-3	1.4435	AT	0	250	8.1	a)
80	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMo17-13-5	1.4439	AT	0	250	8.1	a)
82	Bar	BS EN 10272	Stainless steel, austenitic	X6CrNiNb18-10	1.4550	AT	0	250	8.1	a)
83	Bar	BS EN 10272	Stainless steel, austenitic	X6CrNiMoNb17-12-2	1.4580	AT	0	250	8.1	a)
84	Bar	BS EN 10272	Stainless steel, austenitic	X2CrNiMoN17-13-3	1.4429	AT	0	250	8.1	a)
85	Bar	BS EN 10272	Stainless steel, austenitic	X3CrNiMo17-13-3	1.4436	AT	0	250	8.1	a)
87	Bar	BS EN 10272	Stainless steel, austenitic	X1CrNiMoCuN20-18-7	1.4547	AT	0	250	8.2	a)
89	Bar	BS EN 10273	Elevated temperature properties	P235GH	1.0345	N	0	150	1.1	a)
90	Bar	BS EN 10273	Elevated temperature properties	P250GH	1.0460	N	0	150	1.1	a)
91	Bar	BS EN 10273	Elevated temperature properties	P265GH	1.0425	N	0	150	1.1	a)
92	Bar	BS EN 10273	Elevated temperature properties	P295GH	1.0481	N	0	150	1.2	a)
93	Bar	BS EN 10273	Elevated temperature properties	P355GH	1.0473	N	0	150	1.2	a)
94	Bar	BS EN 10273	Elevated temperature properties	P275NH	1.0487	N	0	150	1.1	a)
95	Bar	BS EN 10273	Elevated temperature properties	P355NH	1.0565	N	0	150	1.2	a)
96	Bar	BS EN 10273	Elevated temperature properties	P355QH	1.8867	QT	0	150	1.2	a)
97	Bar	BS EN 10273	Elevated temperature properties	16Mo3	1.5415	N	0	150	1.1	a) f)
98	Bar	BS EN 10273	Elevated temperature properties	13CrMo4-5	1.7335	NT QT	60	100	5.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
99	Bar	BS EN 10273	Elevated temperature properties	13CrMo4-5	1.7335	QT	100	150	5.1	a)
100	Bar	BS EN 10273	Elevated temperature properties	10CrMo9-10	1.7380	NT	0	60	5.2	a)
101	Bar	BS EN 10273	Elevated temperature properties	10CrMo9-10	1.7380	NT QT	60	150	5.2	a)
102	Bar	BS EN 10273	Elevated temperature properties	11CrMo9-10	1.7383	NT QT	0	60	5.2	a)
103	Bar	BS EN 10273	Elevated temperature properties	11CrMo9-10	1.7383	QT	60	100	5.2	a)
104	Seamless tube	BS EN 10216-1	Room temperature properties	P195TR2	1.0255	N	0	60	1.1	a)
105	Seamless tube	BS EN 10216-1	Room temperature properties	P235TR2	1.0258	N	0	60	1.1	a)
106	Seamless tube	BS EN 10216-1	Room temperature properties	P265TR2	1.0259	N	0	60	1.1	a)
107	Seamless tube	BS EN 10216-2	Elevated temperature properties	P195GH	1.0348	N	0	16	1.1	a)
108	Seamless tube	BS EN 10216-2	Elevated temperature properties	P235GH	1.0345	N	0	60	1.1	a)
109	Seamless tube	BS EN 10216-2	Elevated temperature properties	P265GH	1.0425	N	0	60	1.1	a)
110	Seamless tube	BS EN 10216-2	Elevated temperature properties	20MnNb6	1.0471	N	0	60	1.2	a)
111	Seamless tube	BS EN 10216-2	Elevated temperature properties	16Mo3	1.5415	N	0	60	1.2	a) f)
112	Seamless tube	BS EN 10216-2	Elevated temperature properties	8MoB5-4	1.5450	N	0	16	1.3	a)
113	Seamless tube	BS EN 10216-2	Elevated temperature properties	14MoV6-3	1.7715	NT QT c)	0	60	6.1	a)
114	Seamless tube	BS EN 10216-2	Elevated temperature properties	10CrMo5-5	1.7338	NT QT c)	0	60	5.1	a)
115	Seamless tube	BS EN 10216-2	Elevated temperature properties	13CrMo4-5	1.7335	NT QT c)	0	60	5.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
116	Seamless tube	BS EN 10216-2	Elevated temperature properties	10CrMo9-10	1.7380	NT QT c)	0	60	5.2	a)
117	Seamless tube	BS EN 10216-2	Elevated temperature properties	11CrMo9-10	1.7383	QT	0	60	5.2	a)
118	Seamless tube	BS EN 10216-2	Elevated temperature properties	25CrMo4	1.7218	QT	0	60	5.1	a) b)
119	Seamless tube	BS EN 10216-2	Elevated temperature properties	20CrMoV13-5-5	1.7779	QT	0	60	6.3	a)
120	Seamless tube	BS EN 10216-2	Elevated temperature properties	15NiCuMoNb5-6-4	1.6368	NT QT c)	0	80	4.2	a)
121	Seamless tube	BS EN 10216-2	Elevated temperature properties	X11CrMo5 + I	1.7362 + I	I	0	100	5.3	a)
122	Seamless tube	BS EN 10216-2	Elevated temperature properties	X11CrMo5 + NT1	1.7362 + NT1	NT	0	100	5.3	a)
123	Seamless tube	BS EN 10216-2	Elevated temperature properties	X11CrMo5 + NT2	1.7362 + NT2	NT QT c)	0	100	5.3	a)
124	Seamless tube	BS EN 10216-2	Elevated temperature properties	X11CrMo9-1 + I	1.7386 + I	I	0	60	5.4	a)
125	Seamless tube	BS EN 10216-2	Elevated temperature properties	X11CrMo9-1 + NT	1.7386 + NT	NT QT c)	0	60	5.4	a)
126	Seamless tube	BS EN 10216-2	Elevated temperature properties	X10CrMoVNb9-1	1.4903	NT QT c)	0	100	6.4	a)
127	Seamless tube	BS EN 10216-2	Elevated temperature properties	X20CrMoV11-1	1.4922	NT QT c)	0	100	6.4	a)
128	Seamless tube	BS EN 10216-3	Fine grain steel	P275NL1	1.0488	N	0	100	1.1	a)
129	Seamless tube	BS EN 10216-3	Fine grain steel	P275NL2	1.1104	N	0	100	1.1	a)
130	Seamless tube	BS EN 10216-3	Fine grain steel	P355N	1.0562	N	0	100	1.2	a)
131	Seamless tube	BS EN 10216-3	Fine grain steel	P355NH	1.0565	N	0	100	1.2	a)
132	Seamless tube	BS EN 10216-3	Fine grain steel	P355NL1	1.0566	N	0	100	1.2	a)
133	Seamless tube	BS EN 10216-3	Fine grain steel	P355NL2	1.1106	N	0	100	1.2	a)
134	Seamless tube	BS EN 10216-3	Fine grain steel	P460NH	1.8935	N c)	0	100	1.3	a)
135	Seamless tube	BS EN 10216-3	Fine grain steel	P460NL2	1.8918	N c)	0	100	1.3	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	Product form	EN standard	Material description	Grade	Material number	Heat treat-ment[a]	Thickness mm		Material group to PD CEN ISO/TR 15608:2017	Notes
							min.	max.		
136	Seamless tube	BS EN 10216-4	Low temperature properties	P215NL	1.0451	N	0	10	1.1	a)
137	Seamless tube	BS EN 10216-4	Low temperature properties	P255QL	1.0452	QT	0	40	1.1	a) f)
138	Seamless tube	BS EN 10216-4	Low temperature properties	P265NL	1.0453	N	0	25	1.1	a)
139	Seamless tube	BS EN 10216-4	Low temperature properties	26CrMo4-2	1.7219	QT	0	40	5.1	a) b)
140	Seamless tube	BS EN 10216-4	Low temperature properties	11MnNi5-3	1.6212	N NT c)	0	40	9.1	a)
141	Seamless tube	BS EN 10216-4	Low temperature properties	13MnNi6-3	1.6217	N NT c)	0	40	9.1	a)
142	Seamless tube	BS EN 10216-4	Low temperature properties	12Ni14	1.5637	NT	0	40	9.2	a)
143	Seamless tube	BS EN 10216-4	Low temperature properties	12Ni14 + QT	1.5637	QT	0	40	9.2	a)
144	Seamless tube	BS EN 10216-4	Low temperature properties	X12Ni5	1.5680	N	0	40	9.2	a)
145	Seamless tube	BS EN 10216-4	Low temperature properties	X12Ni5 + QT	1.5680	QT	0	40	9.2	a)
146	Seamless tube	BS EN 10216-4	Low temperature properties	X10Ni9	1.5682	N NT	0	40	9.3	a)
147	Seamless tube	BS EN 10216-4	Low temperature properties	X10Ni9 + QT	1.5682	QT	0	40	9.3	a)
148	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNi18-9	1.4307	AT	0	60	8.1	a)
149	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNi19-11	1.4306	AT	0	60	8.1	a)
150	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNiN18-10	1.4311	AT	0	60	8.1	a)
151	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X5CrNi18-10	1.4301	AT	0	60	8.1	a)
152	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNiTi18-10	1.4541	AT	0	60	8.1	a)
153	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNiNb18-10	1.4550	AT	0	60	8.1	a)
154	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNiMo18-14-3	1.4435	AT	0	60	8.1	a)
155	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNiMo17-12-2	1.4404	AT	0	60	8.1	a)
156	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X5CrNiMo17-12-2	1.4401	AT	0	60	8.1	a)
157	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X1CrNiMoN25-22-2	1.4466	AT	0	60	8.2	a)
158	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNiMoTi17-12-2	1.4571	AT	0	60	8.1	a)
159	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNiMoNb17-12-2	1.4580	AT	0	60	8.1	a)
160	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNiMoN17-13-3	1.4429	AT	0	60	8.1	a)
161	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X3CrNiMo17-13-3	1.4436	AT	0	60	8.1	a)
162	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X1CrNi25-21	1.4335	AT	0	60	8.2	a)
163	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X2CrNiMoN17-13-5	1.4439	AT	0	60	8.1	a)
166	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X1CrNiMoCuN20-18-7	1.4547	AT	0	60	8.2	a)
169	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNi18-10	1.4948	AT	0	60	8.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	Product form	EN standard	Material description	Grade	Material number	Heat treat-ment[a]	Thickness mm		Material group to PD CEN ISO/TR 15608:2017	Notes
							min.	max.		
170	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X7CrNiTi18-10	1.4940	AT	0	60	8.1	a)
171	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X7CrNiNb18-10	1.4912	AT	0	60	8.1	a)
172	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X7CrNiTiB18-10	1.4941	AT	0	60	8.1	a)
173	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X6CrNiMo17-13-2	1.4918	AT	0	60	8.1	a)
176	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X3CrNiMoNB17-13-3	1.4910	AT	0	60	8.1	a)
177	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X8CrNiNb16-13	1.4961	AT	0	60	8.1	a)
178	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X8CrNiMoVNb16-13	1.4988	AT	0	60	8.1	a)
179	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X8CrNiMoNb16-16	1.4981	AT	0	60	8.1	a)
180	Seamless tube	BS EN 10216-5	Stainless steel, austenitic	X10CrNiMoMnNbVB15-10-1	1.4982	AT	0	60	8.1	a)
181	Welded tube	BS EN 10217-1	Room temperature properties	P195TR2	1.0108	N	0	40	1.1	a)
182	Welded tube	BS EN 10217-1	Room temperature properties	P235TR2	1.0255	N	0	40	1.1	a)
183	Welded tube	BS EN 10217-1	Room temperature properties	P265TR2	1.0259	N	0	40	1.1	a)
184	Welded tube	BS EN 10217-2	Elevated temperature properties	P195GH	1.0348	N	0	16	1.1	a)
185	Welded tube	BS EN 10217-2	Elevated temperature properties	P235GH	1.0345	N	0	16	1.1	a)
186	Welded tube	BS EN 10217-2	Elevated temperature properties	P265GH	1.0425	N	0	16	1.1	a)
187	Welded tube	BS EN 10217-2	Elevated temperature properties	16Mo3	1.5415	N	0	16	1.2	a) f)
188	Welded tube	BS EN 10217-3	Fine grain steel	P275NL1	1.0488	N	0	40	1.1	a)
189	Welded tube	BS EN 10217-3	Fine grain steel	P275NL2	1.1104	N	0	40	1.1	a)
190	Welded tube	BS EN 10217-3	Fine grain steel	P355N	1.0562	N	0	40	1.2	a)
191	Welded tube	BS EN 10217-3	Fine grain steel	P355NH	1.0565	N	0	40	1.2	a)
192	Welded tube	BS EN 10217-3	Fine grain steel	P355NL1	1.0566	N	0	40	1.2	a)
193	Welded tube	BS EN 10217-3	Fine grain steel	P355NL2	1.1106	N	0	40	1.2	a)
194	Welded tube	BS EN 10217-4	Low temperature properties	P215NL	1.0451	N	0	16	1.1	a)
195	Welded tube	BS EN 10217-4	Low temperature properties	P265NL	1.0453	N	0	16	1.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment**[a]	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
196	Welded tube	BS EN 10217-5	Sub. Arc, elevated temperature props.	P235GH	1.0345	N	0	40	1.1	a)
197	Welded tube	BS EN 10217-5	Sub. Arc, elevated temperature props.	P265GH	1.0425	N	0	40	1.1	a)
198	Welded tube	BS EN 10217-5	Sub. Arc, elevated temperature props.	16Mo3	1.5415	N	0	40	1.1	a) f)
199	Welded tube	BS EN 10217-6	Sub. Arc, low temperature props.	P215NL	1.0451	N	0	25	1.1	a)
200	Welded tube	BS EN 10217-6	Sub. Arc, low temperature props.	P265NL	1.0453	N	0	25	1.1	a)
201	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNi18-9	1.4307	AT	0	60	8.1	a)
202	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNi19-11	1.4306	AT	0	60	8.1	a)
203	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiN18-10	1.4311	AT	0	60	8.1	a)
204	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X5CrNi18-10	1.4301	AT	0	60	8.1	a)
205	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X6CrNiTi18-10	1.4541	AT	0	60	8.1	a)
206	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X6CrNiNb18-10	1.4550	AT	0	60	8.1	a)
207	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMo17-12-2	1.4404	AT	0	60	8.1	a)
208	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X5CrNiMo17-12-2	1.4401	AT	0	60	8.1	a)
209	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X6CrNiMoTi17-12-2	1.4571	AT	0	60	8.1	a)
210	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMo17-12-3	1.4432	AT	0	60	8.1	a)
211	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMoN17-13-3	1.4429	AT	0	60	8.1	a)
212	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X3CrNiMo17-13-3	1.4436	AT	0	60	8.1	a)
213	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMo18-14-3	1.4435	AT	0	60	8.1	a)
214	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMoN17-13-5	1.4439	AT	0	30	8.1	a)
215	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X2CrNiMo18-15-4	1.4438	AT	0	60	8.1	a)
218	Welded tube	BS EN 10217-7	Stainless steel, austenitic	X1CrNiMoCuN20-18-7	1.4547	AT	0	60	8.2	a)
220	Forging	BS EN 10222-2	Elevated temperature properties	P245GH	1.0352	A	0	35	1.1	a)
221	Forging	BS EN 10222-2	Elevated temperature properties	P245GH	1.0352	N NT QT	35	160	1.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
222	Forging	BS EN 10222-2	Elevated temperature properties	P280GH	1.0426	N	0	35	1.2	a)
223	Forging	BS EN 10222-2	Elevated temperature properties	P280GH	1.0426	NT QT	35	160	1.2	a)
224	Forging	BS EN 10222-2	Elevated temperature properties	P305GH	1.0436	N	0	35	1.2	a)
225	Forging	BS EN 10222-2	Elevated temperature properties	P305GH	1.0436	NT	35	160	1.2	a)
226	Forging	BS EN 10222-2	Elevated temperature properties	P305GH	1.0436	QT	0	70	1.2	a) f)
227	Forging	BS EN 10222-2	Elevated temperature properties	16Mo3	1.5415	N	0	35	1.2	a) f)
228	Forging	BS EN 10222-2	Elevated temperature properties	16Mo3	1.5415	QT	35	500	1.2	a) f)
229	Forging	BS EN 10222-2	Elevated temperature properties	13CrMo4-5	1.7335	NT	0	70	5.1	a)
230	Forging	BS EN 10222-2	Elevated temperature properties	13CrMo4-5	1.7335	NT QT	70	500	5.1	a)
231	Forging	BS EN 10222-2	Elevated temperature properties	15MnMoV4-5	1.5402	NT QT	0	250	1.2	a)
232	Forging	BS EN 10222-2	Elevated temperature properties	18MnMoNi5-5	1.6308	QT	0	200	4.1	a)
233	Forging	BS EN 10222-2	Elevated temperature properties	14MoV6-3	1.7715	NT QT	0	500	6.1	a)
234	Forging	BS EN 10222-2	Elevated temperature properties	15MnCrMoNiV5-3	1.6920	NT QT	0	100	4.1	a)
235	Forging	BS EN 10222-2	Elevated temperature properties	11CrMo9-10	1.7383	NT	0	200	5.2	a)
236	Forging	BS EN 10222-2	Elevated temperature properties	11CrMo9-10	1.7383	NT QT	200	500	5.2	a)
237	Forging	BS EN 10222-2	Elevated temperature properties	X16CrMo5-1	1.7366	A	0	300	5.3	a)
238	Forging	BS EN 10222-2	Elevated temperature properties	X16CrMo5-1	1.7366	NT	0	300	5.3	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	Product form	EN standard	Material description	Grade	Material number	Heat treat-ment[a]	Thickness mm		Material group to PD CEN ISO/TR 15608:2017	Notes
							min.	max.		
239	Forging	BS EN 10222-2	Elevated temperature properties	X10CrMoVNb9-1	1.4903	NT	0	130	6.4	a)
240	Forging	BS EN 10222-2	Elevated temperature properties	X20CrMoV11-1	1.4922	QT	0	330	6.4	a)
241	Forging	BS EN 10222-3	Low temperature properties	13MnNi6-3	1.6217	NT	0	70	9.1	a)
242	Forging	BS EN 10222-3	Low temperature properties	15NiMn6	1.6228	N	0	35	9.1	a)
243	Forging	BS EN 10222-3	Low temperature properties	15NiMn6	1.6228	NT QT	35	50	9.1	a)
244	Forging	BS EN 10222-3	Low temperature properties	12Ni14	1.5637	N	0	35	9.2	a)
245	Forging	BS EN 10222-3	Low temperature properties	12Ni14	1.5637	NT	35	50	9.2	a)
246	Forging	BS EN 10222-3	Low temperature properties	12Ni14	1.5637	QT	50	70	9.2	a)
247	Forging	BS EN 10222-3	Low temperature properties	X12Ni5	1.5680	N	0	35	9.2	a)
248	Forging	BS EN 10222-3	Low temperature properties	X12Ni5	1.5680	NT QT	35	50	9.2	a)
249	Forging	BS EN 10222-3	Low temperature properties	X8Ni9	1.5662	N NT	0	50	9.3	a)
250	Forging	BS EN 10222-3	Low temperature properties	X8Ni9	1.5662	QT	50	70	9.3	a)
251	Forging	BS EN 10222-4	Fine grain steel, high proof strength	P285NH	1.0477	N	0	70	1.2	a)
252	Forging	BS EN 10222-4	Fine grain steel, high proof strength	P285QH	1.0478	QT	70	400	1.2	a) f)
253	Forging	BS EN 10222-4	Fine grain steel, high proof strength	P355NH	1.0565	N	0	70	1.2	a)
254	Forging	BS EN 10222-4	Fine grain steel, high proof strength	P355QH1	1.0571	QT	70	400	1.2	a) f)
255	Forging	BS EN 10222-4	Fine grain steel, high proof strength	P420NH	1.8932	N	0	70	1.3	a)
256	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNi18-9	1.4307	AT	0	250	8.1	a)
257	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiN18-10	1.4311	AT	0	250	8.1	a)
258	Forging	BS EN 10222-5	Stainless steel, austenitic	X5CrNi18-10	1.4301	AT	0	250	8.1	a)
259	Forging	BS EN 10222-5	Stainless steel, austenitic	X6CrNiTi18-10	1.4541	AT	0	450	8.1	a)
260	Forging	BS EN 10222-5	Stainless steel, austenitic	X6CrNiNb18-10	1.4550	AT	0	450	8.1	a)
261	Forging	BS EN 10222-5	Stainless steel, austenitic	X6CrNi18-10	1.4948	AT	0	250	8.1	a)
262	Forging	BS EN 10222-5	Stainless steel, austenitic	X6CrNiTiB18-10	1.4941	AT	0	450	8.1	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
263	Forging	BS EN 10222-5	Stainless steel, austenitic	X7CrNiNb18-10	1.4912	AT	0	450	8.1	a)
264	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiMo17-12-2	1.4404	AT	0	250	8.1	a)
265	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiMoN17-11-2	1.4406	AT	0	160	8.1	a)
266	Forging	BS EN 10222-5	Stainless steel, austenitic	X5CrNiMo17-12-2	1.4401	AT	0	250	8.1	a)
267	Forging	BS EN 10222-5	Stainless steel, austenitic	X6CrNiMoTi17-12-2	1.4571	AT	0	450	8.1	a)
268	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiMo17-12-3	1.4432	AT	0	250	8.1	a)
269	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiMoN17-13-3	1.4429	AT	0	160	8.1	a)
270	Forging	BS EN 10222-5	Stainless steel, austenitic	X3CrNiMo17-13-3	1.4436	AT	0	250	8.1	a)
271	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiMo18-14-3	1.4435	AT	0	75	8.1	a)
272	Forging	BS EN 10222-5	Stainless steel, austenitic	X3CrNiMoN17-13-3	1.4910	AT	0	75	8.1	a)
273	Forging	BS EN 10222-5	Stainless steel, austenitic	X2CrNiCu19-10	1.4650	AT	0	450	8.1	a)
274	Forging	BS EN 10222-5	Stainless steel, austenitic	X3CrNiMo18-12-3	1.4449	AT	0	450	8.1	a)
275	Casting	BS EN 10213	Elevated temperature properties	GP240GR	1.0621	N	0	100	1.1	a)
276	Casting	BS EN 10213	Elevated temperature properties	GP240GH	1.0619	N QT	0	100	1.1	a) f)
277	Casting	BS EN 10213	Elevated temperature properties	GP280GH	1.0625	N QT	0	100	1.2	a) f)
278	Casting	BS EN 10213	Elevated temperature properties	G20Mo5	1.5419	QT	0	100	1.2	a)
279	Casting	BS EN 10213	Elevated temperature properties	G17CrMo5-5	1.7357	QT	0	100	5.1	a)
280	Casting	BS EN 10213	Elevated temperature properties	G17CrMo9-10	1.7379	QT	0	150	5.2	a)
281	Casting	BS EN 10213	Elevated temperature properties	G12MoCrV5-2	1.7720	QT	0	100	6.1	a)
282	Casting	BS EN 10213	Elevated temperature properties	G17CrMoV5-10	1.7706	QT	0	150	6.2	a)
283	Casting	BS EN 10213	Elevated temperature properties	GX15CrMo5	1.7365	QT	0	150	5.3	a)
284	Casting	BS EN 10213	Elevated temperature properties	GX23CrMoV12-1	1.4931	QT	0	150	6.4	a)

Table 2.1-2 **List of materials covered by BS EN material standards** *(continued)*

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min.	**max.**		
285	Casting	BS EN 10213	Low temperature properties	G17Mn5	1.1131	QT	0	50	1.1	a)
286	Casting	BS EN 10213	Low temperature properties	G20Mn5	1.6220	N	0	30	1.2	a)
287	Casting	BS EN 10213	Low temperature properties	G20Mn5	1.6220	QT	0	100	1.2	a) f)
288	Casting	BS EN 10213	Low temperature properties	G18Mo5	1.5422	QT	0	100	1.2	a) f)
289	Casting	BS EN 10213	Low temperature properties	G9Ni10	1.5636	QT	0	35	9.1	a)
290	Casting	BS EN 10213	Low temperature properties	G17NiCrMo13-6	1.6781	QT	0	200	9.2	a)
291	Casting	BS EN 10213	Low temperature properties	G9Ni14	1.5638	QT	0	35	9.2	a)
292	Casting	BS EN 10213	Stainless steel, austenitic	GX2CrNi19-11	1.4309	AT	0	150	8.1	a)
293	Casting	BS EN 10213	Stainless steel, austenitic	GX5CrNi19-10	1.4308	AT	0	150	8.1	a)
294	Casting	BS EN 10213	Stainless steel, austenitic	GX5CrNiNb19-11	1.4552	AT	0	150	8.1	a)
295	Casting	BS EN 10213	Stainless steel, austenitic	GX2CrNiMo19-11-2	1.4409	AT	0	150	8.1	a)
296	Casting	BS EN 10213	Stainless steel, austenitic	GX5CrNiMo19-11-2	1.4408	AT	0	150	8.1	a)
297	Casting	BS EN 10213	Stainless steel, austenitic	GX5CrNiMoNb19-11-2	1.4581	AT	0	150	8.1	a)
298	Casting	BS EN 10213	Stainless steel, austenitic	GX2NiCrMo28-20-2	1.4458	AT	0	150	8.2	a)

a) Materials are considered to meet the requirements of the PER, the PED and BS EN 13445-2.
b) Because of the carbon content special precautions are necessary when the material is welded.
c) See BS EN 10216 for details of heat treatment.
d) *Not used*.
e) *Not used*.
f) Additional requirements for forming and welding shall be considered on a case by case basis.
g) *Not used.*

[a] Heat treatment conditions:
A annealed;
C cold worked;
AT solution annealed;
I isothermally annealed;
N normalized;
NT normalized and tempered;
QT quenched and tempered;
RA recrystalized annealed.

2.1.2.2.2 Steels in material groups 1 to 9 and 11

2.1.2.2.2.1 The chemical composition of steels shall not exceed the values in Table 2.1-3.

Table 2.1-3 **Maximum chemical composition of steels**

Steel group (Table 2.1-1)	Maximum content in ladle analysis		
	% C	% P	% S
1 to 6, 9 and 11	0.25[a)]	0.05	0.05
7.1	0.08	0.04	0.015
7.2	0.06	0.04	0.015
8.1	0.08	0.045	0.015
8.2	0.10	0.035	0.015

a) If the manufacturer proposes to weld steels in material groups 1 to 6, 9 and 11 with a carbon content exceeding 0.25%, this proposal and the welding procedure shall be approved by the purchaser and Inspection Authority.

2.1.2.2.2.2 The deoxidation practice shall be appropriate to the type of steel ordered, particularly where it influences the level of elevated or low temperature properties. It is permitted to use semi-killed steel in accordance with this specification for plates and seamless and welded tubes in carbon and carbon manganese steels in material groups 1, 2 and 11 with an upper limit of the specified tensile strength range of 640 N/mm^2 and with a thickness not exceeding 100 mm. Rimming steel shall only be used for welded tubes in carbon and carbon manganese steel types with an upper limit of the specified tensile strength range of 490 N/mm^2 under service temperature conditions between 0 °C and 380 °C.

2.1.2.2.2.3 Mechanical properties at room temperature shall be specified for acceptance tests in accordance with BS EN ISO 6892-1 covering R_m, R_e (see **2.3**) and minimum elongation at fracture.

Acceptance of properties obtained by other recognized test methods (e.g. other national standards) shall be subject to agreement between the purchaser, manufacturer and Inspection Authority.

The specified minimum percentage elongation at fracture referred to a gauge length of $5.65\sqrt{S_o}$ [2)] shall be appropriate to the type of steel with a lower limit of 16% for plates, 15% for castings and 14% for tubes and forgings, unless the use of the steel is subject to special agreement (see **2.1.1.2**).

The rate of testing and methods of acceptance testing shall generally be consistent with appropriate British Standards for similar product forms.

2.1.2.2.2.4 For materials that will be used above 50 °C, yield point or proof stress properties shall be specified by the manufacturer for acceptance tests in accordance with BS EN ISO 6892-2.

Acceptance of properties obtained by other recognized test methods (e.g. other national standards) shall be subject to agreement between the purchaser, manufacturer and Inspection Authority.

2.1.2.2.2.5 Stress rupture properties shall be specified for materials that will be used in the creep range. These shall be determined in accordance with the procedure laid down in ISO 6303. The manufacturer of the vessel shall be assured that the product supplied is capable of conforming to the specified properties by a

2) S_o is the original cross-sectional area of the gauge length of the tensile test specimen.

statement that the manufacturing processes have remained equivalent to those for the steel for which the test results were obtained.

2.1.2.2.2.6 Charpy V-notch impact test properties at appropriate temperatures shall, where necessary, conform to **2.2.3**, **2.2.4** and **2.2.5**.

2.1.2.2.2.7 Materials shall be supplied in a heat treated condition appropriate to the nearest equivalent British Standard.

NOTE 1 Plates for hot forming may be supplied in any suitable condition as agreed between the manufacturer and the material supplier.

NOTE 2 Any proposal by the manufacturer to use the rolling process to achieve normalized properties of non-BS materials, should be agreed with the purchaser and Inspecting Authority. The properties of the normalized rolled plates should be demonstrated to be similar to the nearest equivalent BS normalized material.

2.1.2.2.2.8 Carbon and carbon manganese steel plates in material groups 1, 2 and 11 for cold forming shall be supplied in the normalized condition except when their thickness is less than 25 mm, when it is permissible to supply plates as-rolled if guaranteed elevated temperature properties are not required.

Low alloy steel plates in material groups 4 to 6 and 9 for cold forming shall normally be supplied in the normalized and tempered condition. Where the plate is metallurgically suitable and where post-weld heat treatment will suffice as the tempering treatment, plates supplied in the normalized condition shall be permitted.

2.1.2.2.2.9 It is permissible to use electric resistance welded or induction welded tubes in the as-welded condition provided the specified upper limit of tensile strength does not exceed 540 N/mm^2 and they are not intended for service below a temperature of 0 °C.

2.1.2.2.2.10 The heat treatment condition to which the specified properties relate shall be clearly stated in the material specifications. These properties can be affected by reheating during fabrication and, where necessary (see **3.4.2** or **4.5**), the manufacturer shall discuss the application and proposed heating or reheating of the steel with the material supplier. However, the test plates shall be supplied and tested in a condition corresponding to the material specification specifically requested by the manufacturer. The material supplier shall, in these circumstances, provide test plates which have been heat treated to a procedure, specified by the manufacturer, which shall detail the heating rate, the holding temperature, the temperature range and time in which any forming takes place.

2.1.2.3 Additional materials for category 3 components

Materials which conform to British Standards listed in Table 2.1-4 shall only be used for the construction of category 3 components. These materials shall satisfy the qualifying requirements of Table 2.1-4.

Table 2.1-4 **Additional materials that may be used for category 3 construction**

Product form	Material standards, BS references		Conditions for use
Plate	BS EN 10025-2	S235JR S235JO S275JR S275JO	a), c) and d)
Plate, sheet or strip	BS 1449-1.1	37/23HR 37/23CR 43/25HR	a), b) and c)
Plate, sheet or strip	BS EN 10088-2: 1.4301 1.4306 1.4307 1.4401 1.4404 1.4432 1.4435 1.4436 1.4541 1.4550 1.4571	 — — 304 S11 316 S31 316 S11 316 S13 — 316 S33 321 S31 347 S31 320 S31	c)

a) The copper content shall not exceed 0.30%.
b) Rimming steel shall not be used.
c) Any negative tolerance on thickness permitted in the material standard shall be taken into account in specifying the ordering thickness (see **3.1.5**).
d) The additional procedure for correction of defects in **7.4.4** of BS EN 10021:2006 only applies with the agreement of the purchaser and Inspecting Authority. Specific inspection should be carried out and a certificate issued.

2.1.3 Materials for non-pressure parts

Materials for supporting lugs, skirts, baffles and similar non-pressure parts welded directly to a pressure component shall be of established identity and shall be compatible with the material to which they are attached. (see **3.10.1.2**, **4.3.5.1** and **D.5.2.6**)

2.2 Materials for low temperature applications

2.2.1 Special consideration shall be given to the selection of materials:

a) for vessels designed to operate below 0 °C;

b) where it is considered by the purchaser, Inspecting Authority or manufacturer that there would otherwise be undue risk of brittle fracture in pressure testing a vessel at the temperature of the available test fluid.

2.2.2 Bolting materials intended for use at low temperature shall be in accordance with Table 2.2-1.

Table 2.2-1 **Bolting materials for low-temperature**

Bolting material	Material specifications and designations	Equivalent BS 4882:1990 grade[a]	Impact requirements[b]	Minimum design temperature °C
Carbon steel	BS EN ISO 898-1:2013 and BS EN ISO 898-2:2022			
	4.8 or **4**		At room temperature	–30
	6.8 or **6**			
Low alloy steel	BS EN ISO 898-1:2013 and BS EN ISO 898-2:2022			
	8.8 or **8**		At room temperature	–50
Low alloy steel	BS EN 10269:2013			
	C45E	2H	Not required[c]	–50
	BS EN 10269:2013			
	42CrMo4	B7	At room temperature	–50
	40CrMoV4-6	B16		
	20CrMoVTiB4-10	B16A		
	BS EN 10269:2013			
	42CrMo4	L7	At –100 °C	–100
9% nickel steel	BS EN 10269:2013			
	X8Ni9	L9	At –196 °C	–196
Austenitic stainless steel	BS EN 10269:2013			
	X5CrNi18-10	B8	Not required[c]	–196
	X5CrNiMo17-12-2	B8M		
	BS EN 10269:2013			
	X5CrNi18-10	L8	At –196 °C	–250
	X5CrNiMo17-12-2	L8M		
Precipitation hardening alloys	BS EN 10269:2013			
	X6NiCrTiMoVB25-15-2	L17B	At –196 °C	–196

[a] BS 4882:1990 is current but partially replaced by BS EN 1515-1:2000 and BS EN 1515-3:2005. The equivalent grades quoted have been taken from Annex B of BS EN 1515-1:2000.

[b] Impact energy values shall be as the material specification at the temperature indicated.

[c] "Not required" indicates that the material may be used, without impact testing, down to the minimum design temperature given in the last column.

2.2.3 The impact requirements for ferritic steels, in material groups 1, 2 and 4, used for vessels designed to operate below 0 °C shall be in accordance with Annex D.

Annex D shall be used when the final pressure test of a vessel is at a temperature higher than that of the available test fluid, as permitted by **5.8.2.4**.

2.2.4 Requirements for the use of ferritic steels in material sub-group 9.2 (3½% Ni) and material groups 5 and 6 shall be specified in the purchase specification. If not specified there, the manufacturer shall propose them for agreement by the purchaser and Inspecting Authority.

2.2.5 The impact requirements for ferritic steels in material sub-group 9.3 (9% Ni) used for vessels designed to operate below 0 °C shall be in accordance with the British Standards listed in Table K.1-1 for material sub-group 9.3.

2.2.6 Austenitic stainless steels in material group 8 (including the high nitrogen and warm worked varieties) are not susceptible to brittle fracture and no special requirements are necessary for their use at temperatures down to –196 °C, however when used at temperatures below –105 °C welds and heat affected zones shall conform to BS EN ISO 21028-1:2016, and impact tests shall be performed as part of the welding procedure qualification and production weld tests.

NOTE For practical reasons, the test temperature of –196 °C is standardized for all austenitic steel testing of any design temperature below –105 °C.

2.2.7 Austenitic-ferritic stainless steels (duplex) in material group 10 shall conform to the requirements in the Duplex supplement.

2.3 Nominal design strength

2.3.1 General

2.3.1.1 The nominal design strength f_N shall be determined:

a) as the lesser of f_E and f_F which are derived by applying the factors of safety from **2.3.3** and **2.3.4** to the property values of the BS EN steels listed in Table 2.1-2 or of the material as agreed between the manufacturer, Inspecting Authority and purchaser [see **2.1.2.1**c)];

b) in the case of BS steels to standards that have been withdrawn, taking values from Table K.1-2 to Table K.1-12.

c) for materials detailed in supplements, annexes or enquiry cases to this specification the nominal design strength shall be taken from the relevant supplement, annex or enquiry case.

The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined by one of the above methods. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

2.3.1.2 It is permissible to use actual material properties instead of specified minimum values to derive time-independent nominal design strengths in order to consider a concessionary acceptance of a specific feature such as a localized manufacturing deviation. This procedure is acceptable provided that appropriate records are retained for material(s) used in the part(s) under consideration, of:

a) the tests made to determine the relevant values of R_m and R_e;

b) the tests or correlation used to derive the relevant $R_{e(T)}$ value, if applicable;

c) the tests made of actual weld properties joining the materials together.

This procedure shall not be used to derive basic design stresses for the original vessel design.

2.3.1.3 Consideration of whether time dependent stresses are applicable shall be given when the design temperature exceeds the values given in Table 2.3-1.

Table 2.3-1 **Temperature above which time dependent properties shall be considered**

Material group or sub-group	Material type	Temperature °C
1, 2 and 11	Carbon and carbon manganese steel	400
1 and 11	Carbon molybdenum steel	
9.1 and 9.2	3½Ni	
9.3	9Ni	
4.1 and 4.2	Low alloy Mn, Cr, Mo, V	470
5.1	1Cr ½Mo	
6.1	½Cr½Mo¼V	490
6.4	12Cr 1Mo V	
5.2	2¼Cr 1 Mo where $R_e \leq 350$ N/mm^2	460
5.3	5Cr ½Mo	
5.4	9Cr 1Mo	
5.2	2¼Cr 1Mo where $R_e > 350$ N/mm^2	350
8.1	Stainless steels	
	Type 321 and Type 347	540
	Type 304 and Type 316	560

NOTE Where the vessel is subject to fatigue, ***C.1.2.3*** *specifies temperature limits for the applicability of the design curves in Annex C.*

2.3.2 Notation

f_E is the nominal design strength corresponding to the short-term tensile strength characteristics;

f_F is the nominal design strength corresponding to the creep characteristics;

f_N is the nominal design strength;

R_e is the minimum value of specified yield strength for the grade of material concerned at room temperature (which shall be tested in accordance with BS EN ISO 6892-1).
Where a standard specifies minimum values of R_{eL} or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels) these values are taken as corresponding to R_e;

R_{eL} is the minimum value of specified yield strength for the grade of material concerned at room temperature (tested in accordance with BS EN ISO 6892-1) or at temperature T (tested in accordance with BS EN ISO 6892-2), as appropriate;

$R_{e(T)}$ corresponds to the minimum value of R_{eL} or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels) specified for the grade of material concerned at a temperature T (which shall be tested in accordance with BS EN ISO 6892-2);

R_m is the minimum tensile strength specified for the grade of material concerned at room temperature (which shall be tested in accordance with BS EN ISO 6892-1);

$R_{p0.2}$ is the minimum value of specified 0.2% proof stress for the grade of material concerned at room temperature (tested in accordance with BS EN ISO 6892-1) or at temperature T (tested in accordance with BS EN ISO 6892-2), as appropriate;

$R_{p1.0}$ is the minimum value of specified 1.0% proof stress for the grade of material concerned at room temperature (tested in accordance with BS EN ISO 6892-1) or at temperature T (tested in accordance with BS EN ISO 6892-2), as appropriate;

S_{Rt} is the mean value of the stress required to produce rupture in time t (at temperature T) for the grade of material in question (which shall be tested in accordance with BS EN ISO 204).

2.3.3 Time-independent design strength

2.3.3.1 General

Time-independent design strength values for materials listed in Table 2.1-2 shall be derived in accordance with the criteria given in **2.3.3.2** or **2.3.3.3**.

The time-independent design strength criteria may be applied to materials not listed in Table 2.1-2, not listed in relevant annexes and not specifically listed in Enquiry Cases, provided they conform to **2.1.2.2.2**. For these materials, the value of $R_{e(T)}$ shall be established by one of the following methods.

a) Derived in accordance with BS EN 10314.

b) Taken from a BS EN standard for steels for pressure purposes at elevated temperatures, when an elevated temperature test is carried out for verification.

c) Taken from a BS EN standard for steels for pressure purposes at elevated temperatures, without any verification of properties at elevated temperature. In this case the value of $R_{e(T)}$ shall be multiplied by 0.90. This reduction factor does not apply in the specific areas recognized by **2.3.1.2**.

d) Verified by tests in accordance with BS EN ISO 6892-2 at the appropriate temperature for each component involved, i.e. each plate as rolled, or forging (or set of forgings as allowed by the appropriate materials specification) and this measured value shall be multiplied by 0.85. This reduction factor does not apply in the specific areas recognized by **2.3.1.2**.

2.3.3.2 Carbon, carbon manganese and low alloy steels

The following strengths shall apply for materials in groups 1 to 7, 9 and 11.

a) Material with specified elevated temperature values:

1) up to and including 50 °C:

$$f_E = \frac{R_e}{1.5} \text{ or } \frac{R_m}{2.35} \tag{2.3-1}$$

whichever gives the lower value;

2) 150 °C and above:

$$f_E = \frac{R_{e(T)}}{1.5} \text{ or } \frac{R_m}{2.35} \tag{2.3-2}$$

whichever gives the lower value;

3) between 50 °C and 150 °C, f_E shall be based on linear interpolation between values obtained from Equations (2.3-1) and (2.3-2).

b) Material without specified elevated temperature values [see Note for values of $R_{e(T)}$]:

1) up to and including 50 °C:

$$f_E = \frac{R_e}{1.5} \text{ or } \frac{R_m}{2.35} \tag{2.3-3}$$

whichever gives the lower value;

2) 150 °C and above:

$$f_E = \frac{R_{e(T)}}{1.6} \text{ or } \frac{R_m}{2.35} \tag{2.3-4}$$

whichever gives the lower value;

3) between 50 °C and 150 °C, f_E shall be based on linear interpolation between values obtained from Equations (2.3-3) and (2.3-4).

NOTE Where agreed by the manufacturer, purchaser and Inspection Authority values of $R_{e(T)}$ can be taken as being equal to those of a material with similar composition and heat treatment which has specified $R_{e(T)}$ values. Where no such values are available the parties can agree that values can be based upon conservative interpretation of other available information.

2.3.3.3 Austenitic stainless steels

The following strengths shall apply for materials in group 8.

a) Material with specified elevated temperature values:

1) up to and including 50 °C:

$$f_E = \frac{R_e}{1.5} \text{ or } \frac{R_m}{2.5} \tag{2.3-5}$$

whichever gives the lower value;

2) 150 °C and above:

$$f_E = \frac{R_{e(T)}}{1.35} \text{ or } \frac{R_m}{2.5} \tag{2.3-6}$$

whichever gives the lower value;

3) between 50 °C and 150 °C, f_E shall be based on linear interpolation between values obtained from Equations (2.3-5) and (2.3-6).

b) Material without specified elevated temperature values [see Note for values of $R_{e(T)}$]:

1) up to and including 50 °C:

$$f_E = \frac{R_e}{1.5} \text{ or } \frac{R_m}{2.5} \tag{2.3-7}$$

whichever gives the lower value;

2) 150 °C and above:

$$f_E = \frac{R_{e(T)}}{1.45} \text{ or } \frac{R_m}{2.5} \tag{2.3-8}$$

whichever gives the lower value;

3) between 50 °C and 150 °C, f_E shall be based on linear interpolation between values obtained from Equations (2.3-7) and (2.3-8).

NOTE Where agreed by the manufacturer, purchaser and Inspection Authority values of $R_{e(T)}$ can be taken as being equal to those of a material with similar composition and heat treatment which has specified $R_{e(T)}$ values. Where no such values are available the parties can agree that values can be based upon conservative interpretation of other available information.

2.3.4 Time-dependent design strength

The time-dependent design strength shall be given by:

$$f_F = \frac{S_{Rt}}{1.3} \tag{2.3-9}$$

Section 3. Design

3.1 General

3.1.1 The minimum thicknesses or dimensions to ensure the integrity of the vessel design against the risk of gross plastic deformation, incremental collapse and collapse through buckling shall be determined using the materials specified in Section 2 and the calculations specified in **3.1.2**, **3.1.3**, **3.1.4**, **3.1.5** and **3.1.6** or **3.2.2**.

3.1.2 Minimum thicknesses or dimensions for particular components of vessels under internal pressure (see **3.5**) shall be calculated in accordance with the subclauses identified in a) to f).

a) cylindrical and spherical vessels (**3.5.1**);

b) dished ends (**3.5.2**);

c) conical ends and truncated cones (**3.5.3**);

d) openings and branch connections (**3.5.4**);

e) flat ends and flat plates (**3.5.5**);

f) spherically domed and bolted ends (see Figure 3.5-39) (**3.5.6**).

There are no provisions in PD 5500 for the design of shells of rectangular cross-section. The required thickness of the shell plates should be determined using an appropriate method, such as that given in BS EN 13445-3:2021, Clause 15. Guidance on the design of flat unstayed ends of non-circular shape and associated flanges is given in Enquiry Case 5500/133.

3.1.3 Minimum thicknesses or dimensions for particular components of vessels under external pressure (see **3.6**) shall be calculated in accordance with the subclauses identified in a) to f).

a) cylindrical shells (**3.6.2**);

b) conical shells (**3.6.3**);

c) spherical shells (**3.6.4**);

d) hemispherical ends (**3.6.5**);

e) torispherical ends (**3.6.6**);

f) ellipsoidal ends (**3.6.7**).

*NOTE The design rules of **3.6** are presented so that for a chosen thickness, an allowable external pressure is calculated for the relevant component. A check can then be made that the allowable external pressure is not less than the design external pressure (see **3.6.1**).*

3.1.4 Minimum thicknesses or dimensions for bolted flange connections, flat heat exchanger tubesheets and jacketted construction shall be calculated in accordance with **3.8**, **3.9** and **3.11**, respectively.

3.1.5 The thicknesses derived from the specified calculations referred to in **3.1.2**, **3.1.3** and **3.1.4** are minimum thicknesses (see **1.6**) and do not include (except where indicated otherwise):

a) corrosion allowance;

b) thinning allowance due to forming;

c) any negative tolerance permitted by the specification to which the material is ordered.

NOTE Following replacement of BS 1501 by BS EN 10028 it should be noted that plate to this specification is normally supplied to BS EN 10029 tolerance class B, which permits a negative tolerance of 0.3 mm for all nominal thicknesses. However the purchaser (in the purchase specification) and/or the manufacturer (in the material enquiry and order) may specify a "zero" negative tolerance by selecting BS EN 10029 tolerance class C instead of B.

3.1.6 Supports, attachments and internal structures (non-pressure parts) shall be designed in accordance with **3.7**.

NOTE Recommended methods of calculating stresses arising from local loads (on spherically or cylindrically shaped vessels) due to nozzles, supports, etc., and thermal gradients are given in Annex G.

3.1.7 Detailed requirements to safeguard against brittle fracture of vessels, ferritic steels in material groups 1, 2 and 4, are given in Annex D (see also **2.2**). Detailed requirements to safeguard the vessel against fatigue failure are given in Annex C.

3.1.8 Several formulae in this section require the derivation of the minimum or maximum of terms within the formulae. This is indicated by the terms which are bracketed, where the terms are separated by semi-colons and where the words min. or max. are given outside the bracket, e.g. x = max. (a ; b).

3.2 Application

3.2.1 Consideration of loads

3.2.1.1 Loads

In the design of a vessel the following loads shall be taken into account, where relevant:

a) internal and/or external design pressure;

b) maximum static head of contained fluid under operating conditions;

c) weight of the vessel;

d) maximum weight of contents under operating conditions;

e) weight of water under hydraulic pressure test conditions;

f) wind, snow and ice loading;

g) earthquake loading;

h) other loads supported by or reacting on the vessel;

i) transportation and handling to final position;

j) test pressure, where a design check is required by **5.8.5.2**.

Consideration shall be given to the effect of the following loads where it is not possible to demonstrate the adequacy of the proposed design, e.g. by comparison with the behaviour of other vessels:

1) local stresses caused by supporting lugs, ring girders, saddles, internal structures or connecting piping or intentional offsets of median lines in adjacent components;

2) shock loads caused by water hammer or surging of the vessel contents;

3) bending moments caused by eccentricity of the centre of pressure relative to the neutral axis of the vessel;

4) stresses caused by temperature differences including transient conditions and by differences in coefficients of thermal expansion;

5) fluctuations of pressure and temperature.

Where portions of a vessel are subjected to high cyclic forces/moments or thermal stresses in service which will not be reproduced during the pressure test specified in **5.8**, the possibility of unacceptable local strain accumulating over the life of the component shall be given appropriate consideration.

3.2.1.2 Notation

For the purposes of **3.2.1.3** the following symbols apply.

B	is the blast load (see **3.2.1.3.7**);
E	is the earthquake load (see **3.2.1.3.5**);
F	are the addition thermal expansion loads from piping (see **3.2.1.3.9**);
f	is the nominal design stress at the design temperature for the vessel section under consideration (see **2.3**);
f_a	is the nominal design stress at ambient temperature for the vessel section under consideration;
G_{corr}	are the corroded dead weight loads (see **3.2.1.3.1**);
G_{lift}	are the lifting dead weight loads (see **3.2.1.3.1**);
G_{max}	are the maximum dead weight loads (see **3.2.1.3.1**);
G_{min}	are the minimum dead weight loads (see **3.2.1.3.1**);
G_{trans}	are the transport dead weight loads (see **3.2.1.3.1**);
H	is the lifting load (see **3.2.1.3.10**);
L	are the live loads for each loading case (see **3.2.1.3.2**);
M	is the wave motion load (see **3.2.1.3.6**);
p_{ex}	is the external design pressure for the vessel section under consideration (see **3.2.3**);
p_i	is the internal design pressure for the vessel section under consideration, including the static head of liquid where applicable (see **3.2.3**);
p_{test}	is the internal test pressure for the vessel section under consideration, including the static head of test liquid (see **5.8**);
R_e	is the minimum value of specified yield strength at ambient temperature for the vessel section under consideration (see **2.3.2**);
$R_{e(T)}$	is the minimum value of specified yield strength at the design temperature for the vessel section under consideration (see **2.3.2**);
T	is the transport load (see **3.2.1.3.8**);
W	is the wind load (see **3.2.1.3.4**);
$\sigma_{z,allow}$	is the allowable longitudinal compressive general membrane stress at the design temperature for the vessel section under consideration (see Annex A);
$\sigma_{z,allow,a}$	is the allowable longitudinal compressive general membrane stress at ambient temperature for the vessel section under consideration;
$\sigma_{z,allow,test}$	is the allowable longitudinal compressive general membrane stress at the test temperature for the vessel section under consideration;
&	is the operator which means superposition of the different load types for the axial and lateral forces, the bending moments and the resulting shear and longitudinal stresses, using beam theory for non-pressure loads and membrane theory for pressure loads.

3.2.1.3 Definitions

3.2.1.3.1 Dead weight loads

The maximum dead weight load (G_{max}) is the weight of the complete vessel in the uncorroded condition with all internals (trays, packing, etc.), attachments, insulation, fire protection, piping, platforms and ladders.

The corroded dead weight load (G_{corr}) is defined as G_{max} but with the weight of the vessel in the corroded condition.

The minimum dead weight load (G_{min}) is the weight of the vessel in the uncorroded condition during the installation phase, excluding the weight of items not already mounted on the vessel before erection (e.g. removable internals, platforms, ladders, attached piping, insulation and fire protection).

NOTE A scaffold is normally self-supported. In this case, the weight of the scaffold is not included in the vessel weight.

The transport dead weight load (G_{trans}) is the weight of the vessel for transportation, including the weight of any items already mounted on the vessel in the workshop (e.g. removable internals and insulation).

The lifting dead weight load (G_{lift}) is the weight of the vessel for the lifting case under consideration, including the weight of any items mounted on the vessel at the time of the lift.

3.2.1.3.2 Live loads

Live loads (L) are the weight loads of the contents (fluids or solids in the bottom of the vessel, on trays and in packing) and traffic loads on platforms and ladders by personnel and machinery.

Where particular values are not specified a uniformly distributed load of 1.5 kN/m^2 may be used for the traffic load on platforms, and a load of 1.0 kN may be used for the traffic load on each ladder. This includes loads from personnel, snow and ice and light machinery. Traffic loads due to heavy machinery on platforms may be based on the weight of the machinery and a uniformly distributed load of 1.0 kN/m^2.

For vessels with more than three platforms it is only necessary to consider the traffic loads on the three largest platforms and associated ladders.

3.2.1.3.3 Snow and ice loads

Snow and ice loads result in additional weight loading on the vessel, platforms and other attachments, and these are classified as live loads. Snow and ice can also cause additional wind load due to the increase in the projected area of the vessel and its attachments.

3.2.1.3.4 Wind loads

Wind loads (W) are horizontal loads caused by wind pressure acting on the projected area of the vessel and its attachments, including the effects of force coefficients (see **3.2.7**, Annex B and Enquiry Case 5500/127).

3.2.1.3.5 Earthquake loads

Earthquake loads (E) are quasi-static horizontal forces on the vessel sections caused by seismic accelerations at the base of vessel (see **3.2.7** and Annex B). Some seismic codes also include a vertical acceleration.

3.2.1.3.6 Wave motion loads

Wave motion loads (*M*) are the forces acting on vessels which are permanently mounted on a ship, caused by horizontal and vertical accelerations due to wave motions. Because wave motion loads are cyclic it might also be necessary to perform a fatigue assessment.

3.2.1.3.7 Blast loads

Blast loads (*B*) are horizontal and vertical loads caused by blast pressure acting on the projected area of the vessel and its attachments. In many cases a static analysis, similar to wind loading, is performed. In some cases a dynamic analysis is required.

3.2.1.3.8 Transport loads

Transport loads (*T*) are the forces acting on the vessel caused by horizontal and vertical accelerations due to motion during transportation. The factors to be applied to G_{trans} shall take into account the method of transportation (e.g. road, rail or ship) and shall be sufficient to ensure that the vessel will not be damaged during transportation. If no special regulations are specified the loads given in the following publications may be used:

a) Road transport – Chapter 6.7, clause 6.7.2.2.12 of ADR (European Agreement Concerning the International Carriage of Dangerous Goods by Road), 2019.

b) Rail Transport – Part 6, clause 6.7.2.2.12 of RID (Regulations Concerning the International Carriage of Dangerous Goods by Rail), 2019.

c) Ship transport – Annex 13 of CSS CODE (Code of Safe Practice for Cargo Stowage and Securing), 2011 Edition. For vessels transported as deck cargo the additional loads due to wind and sea sloshing specified in the CSS Code should be considered.

3.2.1.3.9 Loads from attached external piping

Piping loads are forces from attached external piping resulting from weight (*G*), wind (*W*), earthquake (*E*) and other additional forces (*F*) which influence the global equilibrium of the vessel.

NOTE Forces and moments on nozzles and supports on the vessel caused by attached external piping can act as internal and/or external loads. Internal loads are those that cause local loads only and have no influence on the global equilibrium because they are self-compensating. Furthermore, attached pipes can either load the vessel or restrain it depending on their layout. Consideration of these aspects is given in the following recommendations.

Guidelines for assessing when additional forces are to be considered:

a) Only horizontal and vertical forces need to be taken into account, bending moments should be neglected.

b) Forces act at the elevation where the external horizontal pipe runs arrive at or leave the vessel; therefore they shall be incorporated into the calculation at this elevation. At other elevations the forces are internal forces without influence on the global equilibrium because they result from restraint between nozzles and pipe supports on the vessel.

c) In the piping analysis the local flexibility of the vessel wall may be taken into account. The global flexibility of the whole vessel may be taken into account provided that all essential pipes attached to the vessel are considered in the piping analysis.

d) In cases where multiple pipes are connected to a vessel the resulting horizontal reaction forces and their directions shall be vector combined at each elevation taking into account the direction of each of the single pipe forces. Where actual forces and their directions are not available it is not reasonable to assume that all horizontal forces act in the same direction. The maximum resulting shear force at the base of the vessel shall be vector combined from the horizontal resulting forces and their directions at all elevations. The maximum resulting bending moment at the base of the vessel shall be vector combined from the moments and their directions determined from these horizontal resulting forces with their directions and elevations.

3.2.1.3.10 Lifting loads

Lifting loads (H) are the forces acting on the vessel and its lifting attachments (lugs, trunnions, etc.) caused by the dead weight load of the vessel and by vertical inertia forces during lifting. The dead weight load shall be multiplied by a dynamic magnification factor (shock factor or snatch factor) to take account of the inertia forces. There may be a number of lifting cases to consider, such as lifting for transportation and final site lift. When a vessel is lifted from horizontal to vertical the lifting cases shall include the horizontal lift, the vertical lift and at least one intermediate angle lift.

The dynamic magnification factor for each lifting case shall take account of the lifting conditions, the type of crane or other lifting equipment and its overcapacity. A high factor should be used (typically 2.0 or greater) if the crane capacity is high compared with the weight of the vessel, and for offshore lifting. For lifting of heavy vessels in a well-controlled lifting environment a lower factor may be used (typically 1.2). If neither of these conditions apply the dynamic magnification factor shall be ≥ 1.5.

3.2.1.4 Load combinations

The combinations of pressure and non-pressure loads given in Table 3.2-1 are used in connection with the calculation of stresses in the vessel in accordance with Annex B and the assessment of the stresses in accordance with Annex A. The basic calculations for the pressure envelope subject to internal pressure in accordance with **3.5** and external pressure in accordance with **3.6** (where applicable) shall be performed before the Annex B calculations. The load combinations in Table 3.2-1 are the minimum to be taken into account, if they are relevant. There may also be other loads.

Table 3.2-1 **Load combinations**

Load case	Description	Load combinations including weighting factors	Allowable shell longitudinal tensile stress	Allowable shell longitudinal compressive stress
LC1	Operation with internal pressure	p_i & G_{max} & L	f	$\sigma_{z,allow}$
LC2	Operation with internal pressure, wind and wave motion	p_i & G_{max} & L & F & W & M	1.2f	$\sigma_{z,allow}$
LC3	Operation with external pressure, wind and wave motion	p_{ex} & G_{max} & L & F & W & M	1.2f	$\sigma_{z,allow}$
LC4	Operation with wind and wave motion and no pressure	G_{max} & L & F & W & M	1.2f	$\sigma_{z,allow}$
LC5	Operation with internal pressure and earthquake	p_i & G_{max} & L & F & E	1.2f	$\sigma_{z,allow}$
LC6	Operation with external pressure and earthquake	p_{ex} & G_{max} & L & F & E	1.2f	$\sigma_{z,allow}$

Table 3.2-1 **Load combinations** *(continued)*

Load case	Description	Load combinations including weighting factors	Allowable shell longitudinal tensile stress	Allowable shell longitudinal compressive stress
LC7	Operation with earthquake and no pressure	G_{max} & L & F & E	$1.2f$	$\sigma_{z,allow}$
LC8	Operation with external pressure and blast	p_{ex} & G_{max} & L & F & B	$0.9R_{e(T)}$	$\sigma_{z,allow}$
LC9	Shut down	G_{corr} & W & M	f_a	$\sigma_{z,allow,a}$
LC10	Test, full of test liquid with test pressure and wind	p_{test} & G_{max} & L_{test} & $0.6W$	$0.9R_e$	$\sigma_{z,allow,test}$
LC11	Transport	G_{trans} & T	f_a	$\sigma_{z,allow,a}$
LC12	Lifting	G_{lift} & H	f_a	$\sigma_{z,allow,a}$
LC13	Installation	G_{min} & $0.7W$	f_a	$\sigma_{z,allow,a}$

For LC2 and LC3: If more than one combination of coincident design pressure and design temperature exists then all combinations shall be investigated.
Alternatively a single combination of the maximum pressure and maximum temperature of all the cases may be used. It is not certain that the governing condition of coincident pressure and temperature is also governing for the load combinations.
For LC3, LC6 and LC8: The combined effects of external pressure and applied loads shall be assessed using the interaction formula in Equation (A-3) in **A.3.5.1.2**.
For LC4 and LC7: These load cases are not required when both loading cases LC2 and LC3, or LC5 and LC6 are applicable, i.e. internal and external pressure are applied.
For LC8: This load case may cause possible damage to the vessel but must not cause failure of the pressure envelope.
For LC9: This load case can result in the maximum tensile loads on the supports and anchor bolts.
For LC10: The reduced factor for the wind load is in accordance with BS EN 1991-1-6:2005 for duration times less than 3 days.
For LC11: If multiple modes of transport are used then each mode shall be investigated.
For LC13: The wind load in this case depends on configuration at this time (with or without scaffold, platforms, insulation). The reduced factor for the wind load is in accordance with BS EN 1991-1-6:2005 for duration times less than 12 months.

3.2.2 Design criteria

The adequacy of the design of each component of the vessel shall be demonstrated by one of the following methods:

a) application of the design requirements specified in this section[1)];

b) alternatively, one of the following alternative methods shall be agreed between the purchaser Inspecting Authority and manufacturer:

1) the use of design requirements which meet the criteria given in Annex A and which are given in a document other than this specification;

2) application of one of the sets of criteria in Annex A;

3) derivation of the design pressure of the vessel (or vessel component) using the results of a proof hydraulic test carried out in accordance with **5.8.6**;

4) comparison with other similar vessels or components, as agreed relevant by the manufacturer, purchaser and Inspecting Authority. In making this comparison the following factors shall be amongst those considered:

i) materials of construction;

[1)] The equations in this section may be used with any consistent set of units.

ii) size, geometry and thickness;

iii) design conditions, operating conditions including service life (achieved and intended), operating cycles and corrosion;

iv) method of construction;

v) degree of inspection and quality control.

NOTE It may not be possible to support a comparison solely on the grounds of acceptable experience with a single operated example without some down rating to cover variations in the above factors.

In no case shall the minimum shell thickness for pressure loading be less than required by **3.5.1**, **3.5.2** and **3.5.3** as relevant.

Where, during normal operation, a vessel is subjected to more than one loading[2)] /temperature condition, the thickness shall be determined from that condition which results in the greatest thickness. Each of the loading/temperature conditions shall be specified on Form X (see **1.4.4**).

NOTE 1 The equations in this section are based on mean diameter rules and are not necessarily applicable when the ratio of the outside diameter of the vessel to the inside diameter of the vessel D_o/D_i exceeds 1.3. The design of such vessels should be given special consideration, taking into account that the factor of safety against gross plastic deformation will be greater than that implied by the equations in this section, and that the onset of plasticity (locally at the bore) will occur at relatively low pressure.

*NOTE 2 Where the specified design strength of a material is time dependent, the design procedures covering situations where internal pressure is not a dominant form of loading (e.g. see **3.6**) may not in themselves provide adequate margin against the possibility of creep deformation leading to instability or creep rupture during the agreed design lifetime. In such cases the design procedures specified in this section should, where indicated, be supplemented by appropriate analysis to confirm that this lifetime will be achieved. The design procedures included in this section do not necessarily cover mechanisms (e.g. creep ratchetting) which can significantly increase the deformation rate of components operating in the creep range and subject to frequent temperature cycling. Where such cycling is likely, deformation rates should be confirmed by appropriate supplementary analysis.*

The service life and design margin (see **C.1.2**) for any component requiring fatigue analysis (see **C.2**) shall be given by the purchaser in the purchase specification.

NOTE 3 No such vessel should remain in service once the agreed service life has been completed, without a periodic review based on the inspection/monitoring of the part(s) in question.

3.2.3 Design pressure

The design pressure (i.e. the pressure to be used in the equations for the purposes of calculation) shall be not less than:

a) the pressure which will exist in the vessel when the pressure relieving device starts to relieve, or the set pressure of the pressure relieving device, whichever is the higher (see **3.13**);

b) the maximum pressure which can be attained in service where this pressure is not limited by a relieving device.

The design pressure shall be based on the maximum differential pressure between the inside and outside of the vessel or component, and include the static head where applicable. The design pressure shall be taken as a positive

[2)] In this context, the term "loading" means any combination of loads (including pressure loading) acting simultaneously.

value for use in the equations for the purposes of calculation unless stated otherwise.

Vessels subject to external pressure shall be designed for the maximum differential pressure to which the vessel may be subjected in service. It is recommended that vessels subject to vacuum be designed for a full external pressure of 1 bar[3)] unless a vacuum break valve or similar device is provided, in which case it is permissible for a lower design pressure to be used by agreement between the purchaser, Inspecting Authority and the manufacturer.

3.2.4 Maximum design temperature

The maximum design temperature which is used to determine the appropriate nominal design strength for the selected material shall be not less than the highest actual metal temperature expected in service corresponding to the coincident design pressure. The maximum design temperature shall include an adequate margin to cover uncertainties in temperature prediction. Where different metal temperatures can confidently be predicted for different parts of the vessel, it is permissible to base the design temperature for any point in the vessel on the predicted metal temperature.

The design lifetime for any component whose maximum design temperature is such that the nominal design strength is time dependent (see **2.3.1.3**), shall be given by the purchaser in the purchase specification.

NOTE No vessel designed on this basis should remain in service beyond the agreed design lifetime unless a review is then made of its continued fitness for service based on inspection for creep damage and consideration of its temperature/stress history and the latest materials data. Particular attention should be paid, during inspection, to geometrical discontinuities and details subject to load or temperature cycling. Subject to satisfactory periodic review, it is permissible to extend service lives beyond the original design life.

3.2.5 Minimum design temperature

The minimum design temperature which is used to determine the suitability of the material to resist brittle fracture shall be the lowest metal temperature expected in service. The minimum design temperature shall include a margin to cover any uncertainties in temperature prediction. In the case of components thermally insulated externally, the lowest metal temperature shall be taken to be the minimum temperature of the contents of the vessel at the appropriate loading condition. In the case of components not thermally insulated, the minimum temperature of the components under operating conditions and the method used for assessing the lowest metal temperature shall be subject to agreement.

In cases where the calculated membrane stress can vary with the minimum design temperature, e.g. auto-refrigeration during depressurization, the various combinations of stress and temperature shall be evaluated to determine the one which is most onerous for the purpose of selection of materials (see **D.3.1**).

3.2.6 Thermal loads

Provision shall be made in the design to permit thermal expansion and contraction so as to avoid excessive thermal stresses.

[3)] 1 bar = 10^5 N/m^2 = 0.1 N/mm^2 = 100 kPa.

3.2.7 Wind and earthquake loads

Unless otherwise agreed (see **3.2.2**), wind and earthquake loadings shall be calculated in accordance with Annex B and the higher permissible stresses given in **A.3.6** apply.

3.3 Corrosion, erosion and protection

3.3.1 General

The word "corrosion" as used in this specification shall be taken to mean corrosion, oxidation, scaling, abrasion, erosion and all other forms of wastage.

The purchaser shall give consideration to the likely effect which corrosion (both internal and external) will have upon the useful life of the vessel. The purchaser shall specify in the purchase specification where the contained media (including test media, washing media, etc.), or external media (e.g. seawater) could give rise to stress corrosion cracking and whether associated post-weld heat treatment is required. In these cases the manufacturer shall review the materials used, the material hardness, residual stress and post-weld heat treatment.

Forms of corrosion, including the following, require consideration.

a) Chemical attack where the metal is dissolved by the reagents. It may be general over the whole surface or localized (causing pitting) or a combination of the two.

b) Rusting caused by the combined action of moisture and air.

c) Erosion corrosion where a reagent that is otherwise innocuous flows over the surface at a velocity greater than some critical value.

d) High temperature oxidation (scaling).

When in doubt consideration shall be given to undertaking corrosion tests to be carried out on the actual metal (including welds) or combination of metals under exposure to the actual chemicals used in service.

NOTE 2 It is very dangerous to assume that the major constituent of a mixture of chemicals is the active agent, as in many cases small traces of impurities exert an accelerating or inhibiting effect out of all proportion to the amount of impurity.

Fluid temperatures and velocities should be equivalent to those met in operation. Corrosion tests should be continued for a sufficiently long period to determine the trend of any change in the rate of corrosion with respect to time.

3.3.2 Additional thickness to allow for corrosion

The additional thickness specified over and above that required for design conditions shall be adequate to cover the total amount of corrosion expected on either or both surfaces of the vessel and shall be given by the purchaser in the purchase specification. It shall be at least equal in magnitude to the expected wastage due to corrosion during the specified life of the vessel and shall be a minimum of 1 mm unless a protective lining is employed.

Where corrosion effects are negligible no excess thickness need be specified.

3.3.3 Linings and coatings

It is permissible for vessels to be fully or partially lined (or coated) with corrosion-resistant material. It is permissible for linings to be loose, intermittently attached to the vessel base material or integrally bonded to the vessel base material. This specification does not cover lined vessels where the construction or installation of the lining imposes significant additional membrane stresses on the vessel. Any requirements for the surface finish for coated vessels shall be given by the purchaser in the purchase specification.

Provided contact between the corrosive agent and the vessel base material is excluded, it shall not be necessary to make a corrosion allowance against internal wastage of the base material.

Corrosion-resistant linings shall not be included in the computation of the specified wall thickness except in the case of clad steels, when as agreed between the manufacture, purchaser and Inspecting Authority, the combined thickness of steel and cladding is permitted to be used in calculating the wall thickness.

The design of lining shall take into account the effects of differential thermal expansion; integral linings shall have sufficient ductility to accommodate any strain likely to be imposed on them during service.

3.3.4 Wear plates

Where severe conditions of erosion and abrasion arise, consideration shall be given to fitting local protective or wear plates directly in the path of the impinging material.

3.4 Construction categories and design stresses

3.4.1 Construction categories

For each pressure-containing component of the vessel, the manufacturer shall select a construction category in accordance with Table 3.4-1. The purchaser may require a minimum construction category in which case it shall be specified in the purchase specification. A component is defined as a part of pressure equipment which can be considered as an individual item for the purpose of calculation e.g. flange, end, cylindrical strake.

Table 3.4-1 **Construction categories**

Construction category	Non-destructive testing (NDT)	Permitted material groups and sub-groups	Maximum nominal thickness of component[a] (see 1.6) (mm)	Temperature limits	
				Upper	Lower
1	100% (see **5.6.4.1**)	All	None, except where NDT method limits	See **2.3.1.1**, **K.1.4.1.2** and Note 2 to **3.2.2**	See Annex D limitations for temperatures below 0 °C
2	Limited random (spot) (see **5.6.4.2**)	1[d], 2 and 4	40	See **2.3.1.1**, **K.1.4.1.2** and Note 2 to **3.2.2**	See Annex D limitations for temperatures below 0 °C
		1.2 CMo	30		
		8	40	None	None
3	Visual only (see **5.6.4.3**)	C and CMn steel ($R_m{}^b \le 432$ N/mm^2)	13[c]	300 °C	0 °C
		8	25	300 °C	None

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension of the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.

[b] For definition of R_m see **2.3.2**. The limit of 432 N/mm^2 is not intended to apply to pipe fittings as specified in BS 3799.

[c] This thickness shall only be exceeded if no benefit is taken of it in design.

[d] Excluding 1.2 CMo.

NOTE 1 Any one of the three construction categories in Table 3.4-1 will provide adequate integrity for normal purposes within the material and temperature limitations specified therein. The justification for any special precautions (e.g. additional inspection and/or test requirements, secondary containment) to reduce external risks in the postulated event of an escape of hazardous vessel contents involves consideration of matters by the purchaser (and Regulating Authority) which are beyond the scope of this specification. Any modifications to the requirements of this specification which are required for the purpose should be covered in the purchase specification.

NOTE 2 Construction categories, as defined in Table 3.4-1 are intended to apply to components of a vessel and not necessarily only to complete vessels which may therefore comprise components in two or more categories. Category 3, however, is commonly applied to complete vessels so that design stresses and inspection requirements are consistent throughout the vessel.

NOTE 3 The fatigue assessment of seam welds in accordance with Annex C is influenced by the extent of non-destructive testing performed, and hence by the choice of construction category.

3.4.2 Design stresses

3.4.2.1 Categories 1 and 2

Except as qualified in **3.4.2**, the design strength for materials shall not exceed the appropriate nominal design strength given by **2.3.1.1** for the material of construction at the design temperature.

Where, for operational or other reasons (e.g. to reduce a risk of stress corrosion or to increase the fatigue life), the design stress is required to be less than the relevant nominal design strength for the material, this reduced design stress shall be specified by the purchaser in the purchase specification.

The following points shall also be taken into account.

a) *Carbon and carbon manganese steels in material groups 1 to 3 and 11*

   1) The nominal design strengths given by **2.3.1.1** are intended for general use with the steels listed and acceptance tests on material heat treated with a completed vessel are not required, any reduction in properties of such steels due to post-weld heat treatment being consistent with the overall benefit obtained by stress relief of the structure. A purchaser requiring such tests, or tests on samples subject to non-standard heat treatments, shall specify them in the purchase specification together with appropriate acceptance criteria.

   2) In designs where slight deformation is important or where the proposed post-weld heat treatment times or temperatures will significantly exceed the limits given in **4.4.1**, plate which will meet the properties in the material specification in the normalized plus simulated (3-hour) post-weld heat treated condition is to be specified.

b) *Alloy steels in material groups 4 to 9*

   1) Manufacturers shall discuss the application and proposed heating or reheating of alloy steels with the material supplier before selecting the appropriate nominal design strength.

   2) The nominal design strengths given by **2.3.1.1** shall be used provided the proposed post-weld heat treatment does not exceed the time and temperature limits given in Table 4.5-1. However, for material

sub-groups 5.1 and 5.2 where the temperature limits given in Table 4.5-1 for maximum softening or maximum creep resistance are used, the properties used for design purposes shall be subject to the prior agreement of the purchaser, Inspection Authority and manufacturer. Appropriate time and temperature limits for non-standard heat treatments shall be established at the design stage. If acceptance tests on material heat treated with a completed vessel or in a non-standard manner are required, these shall be specified in the supplementary specification together with acceptance criteria agreed between the manufacturer, purchaser and Inspecting Authority.

3.4.2.2 Category 3

The following design stress limits shall apply irrespective of the orientation of the main welded seams. Main welded seams are defined as type A welds (see Figure 5.6-1).

a) *Carbon and carbon manganese steels in material group 1*

The design stress shall not exceed $R_m/5$.

b) *Austenitic steel in material group 8*

The design stress shall not exceed 120 N/mm^2 or $120\left(\frac{450}{400+t}\right)$, whichever is the smaller, where t is the design temperature (°C).

In cases where the specified minimum yield strength (1.0% proof stress) is less than 230 N/mm^2 the design stress so calculated shall be multiplied by 0.8.

It is permissible to include category 2 components in a category 3 vessel although all type A welds in a category 2 component, including any circumferential welds joining it to any other component, shall meet category 2 requirements.

A welded flat end or a dished end can be treated as a category 2 component even though the circumferential seam joining it to the vessel is category 3, provided that the cylindrical flange meets the thickness requirements for the cylinder.

It is permissible to use category 1 or category 2 stresses in the calculations for the following category 3 components, provided that the components do not include materials listed in Table 2.1-4.

1) Details such as nozzles and attachments remote from type A welds where remote is defined as no closer than $2.5\sqrt{Re}$. R is the internal radius of the cylinder, end or cone and e is the minimum thickness of the cylinder, end or cone calculated to **3.5.1**, **3.5.2** or **3.5.3** using category 3 stresses.
2) Flanges and flanged flat ends.

3.4.2.3 Additional limit for statically cast components

In the case of static castings the design stress shall not exceed 0.7 × the nominal design strength value given by **2.3.1.1**, unless the quality specification (see **2.1.2.1**) makes full provision for the detection and repair of potentially harmful defects in all critical sections (see **5.9**), in which case it is permissible to take this limit as 0.9 × the nominal design strength value given by **2.3.1.1**.

3.5 Vessels under internal pressure

3.5.1 Cylindrical and spherical shells

3.5.1.1 Notation

For the purposes of **3.5.1.2** and **3.5.1.3** the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**).

D_i is the inside diameter of shell;
D_o is the outside diameter of shell;
e is the minimum calculated thickness of shell plate;
f is the nominal design stress;
M is the longitudinal bending moment;
p is the design pressure;
Q is the longitudinal force in cylinder due to M or W, per unit length of inside circumference (positive if tensile); see Equation (3.5.1-7);
R_i is the inside radius of shell;
W (for vessels with a vertical longitudinal axis only);
a) for points above plane of support: is the weight of vessels, fittings, attachments and fluid supported above point considered, the sum to be given a negative sign in Equation (3.5.1-7);
b) for points below plane of support: is the weight of vessels, fittings, attachments and fluid below point considered plus weight of fluid contents not supported above point considered, the sum to be given a positive sign in Equation (3.5.1-7);
σ_z is the net longitudinal compressive stress. See Equation (3.5.1-8).

3.5.1.2 Minimum thickness for pressure loading only

The minimum thickness for pressure loading only shall be calculated from the following equations.

a) *Cylindrical shells*

$$e = \frac{pD_i}{2f - p} \quad (3.5.1\text{-}1)$$

or

$$e = \frac{pD_o}{2f + p} \quad (3.5.1\text{-}2)$$

b) *Spherical shells*

$$e = \frac{pD_i}{4f - 1.2p} \quad (3.5.1\text{-}3)$$

or

$$e = \frac{pD_o}{4f + 0.8p} \quad (3.5.1\text{-}4)$$

3.5.1.3 Minimum thickness for combined loading

3.5.1.3.1 Cylindrical and spherical shells

Where a shell is subjected to loads in addition to internal pressure (see **3.2.2**) it is not possible to give explicit equations for the minimum thickness and solution by trial and error is necessary (see Annex B).

3.5.1.3.2 Approximation for cylinder

Where the effect of such loadings is to produce an axial load W and a bending moment M, a first approximation to the thickness required shall be determined in the following manner. The first approximation is always an overestimate by an amount which is greater for cylinders with larger values of e/R_i.

*NOTE 1 Where Equations (3.5.1-5) and (3.5.1-6) indicate that an increase in the thickness over that given by **3.5.1.2**a) is required, reference should be made to Annex B to establish the minimum thickness.*

The first approximation to the minimum thickness is the largest of the values given by Equations (3.5.1-5) and (3.5.1-6) and by **3.5.1.2**a). Equation (3.5.1-7) is used to calculate Q. When Q has a positive value Equation (3.5.1-5) is used to calculate e, and when it has a negative value Equation (3.5.1-6) is used. Additionally when Q has a negative value, σ_z as calculated by Equation (3.5.1-8) shall not exceed the limit given in **A.3.5**.

$$e = \frac{0.25pD_i + Q}{f - 0.5p} \quad \text{or} \quad e = \frac{0.25pD_o + Q}{f} \tag{3.5.1-5}$$

$$e = \frac{0.25pD_i - Q}{f} \quad \text{or} \quad e = \frac{0.25pD_o - Q}{f + 0.5p} \tag{3.5.1-6}$$

$$Q = \frac{W}{\pi D_i} + \frac{4M}{\pi {D_i}^2} \tag{3.5.1-7}$$

$$\sigma_z = \frac{Q}{e} - \frac{pD_i}{4e} \tag{3.5.1-8}$$

These calculations shall be performed for all combinations of load expected in service.

Conditions during pressure testing shall be the subject of special consideration.

NOTE 2 For dealing with local stresses in the neighbourhood of the points of application of the additional loads see Annex G.

NOTE 3 For dealing with torsional loading, wind or earthquake loading see Annex B.

3.5.2 Dished ends

3.5.2.1 Notation (see Figure 3.5-1)

For the purposes of **3.5.2.2**, **3.5.2.3** and **3.5.2.4** the following symbols apply. All dimensions are in the corroded condition, unless otherwise indicated, and do not include thinning allowance (see **3.1.5**).

D is the outside diameter of end;

D_c is the outside diameter of crown section of torispherical end measured to tangent between crown and knuckle;

e is the minimum calculated thickness after dishing;

e_a is the analysis thickness of the dished end (see **1.6**);

f is the design stress;

h is the outside dished end height, i.e. external height of end measured from plane of junction of end with cylinder skirt;

NOTE $h = (R + e_a) - \{[(R + e_a) - D/2][(R + e_a) + D/2 - 2(r + e_a)]\}^{1/2}$ *(approximately).*

h_e is the smallest of h, $D^2/[4(R + e_a)]$ and $\sqrt{D(r + e_a)/2}$;

p is the design pressure;

R is the inside spherical radius, for torispherical ends;

r is the inside knuckle radius, for torispherical ends.

NOTE The derivation of these rules is given in Part 1 of PD 6550, the Explanatory Supplement to BS 5500.

3.5.2.2 Limitations

The following design limitations shall apply to ellipsoidal and torispherical ends:

a) ellipsoidal:

$e_a \geq 0.002D$

$e \leq 0.12D$

$h_e \geq 0.18D$

b) torispherical ends:

$e_a \geq 0.002D$

$e \leq 0.12D$

$r \geq 0.06D$

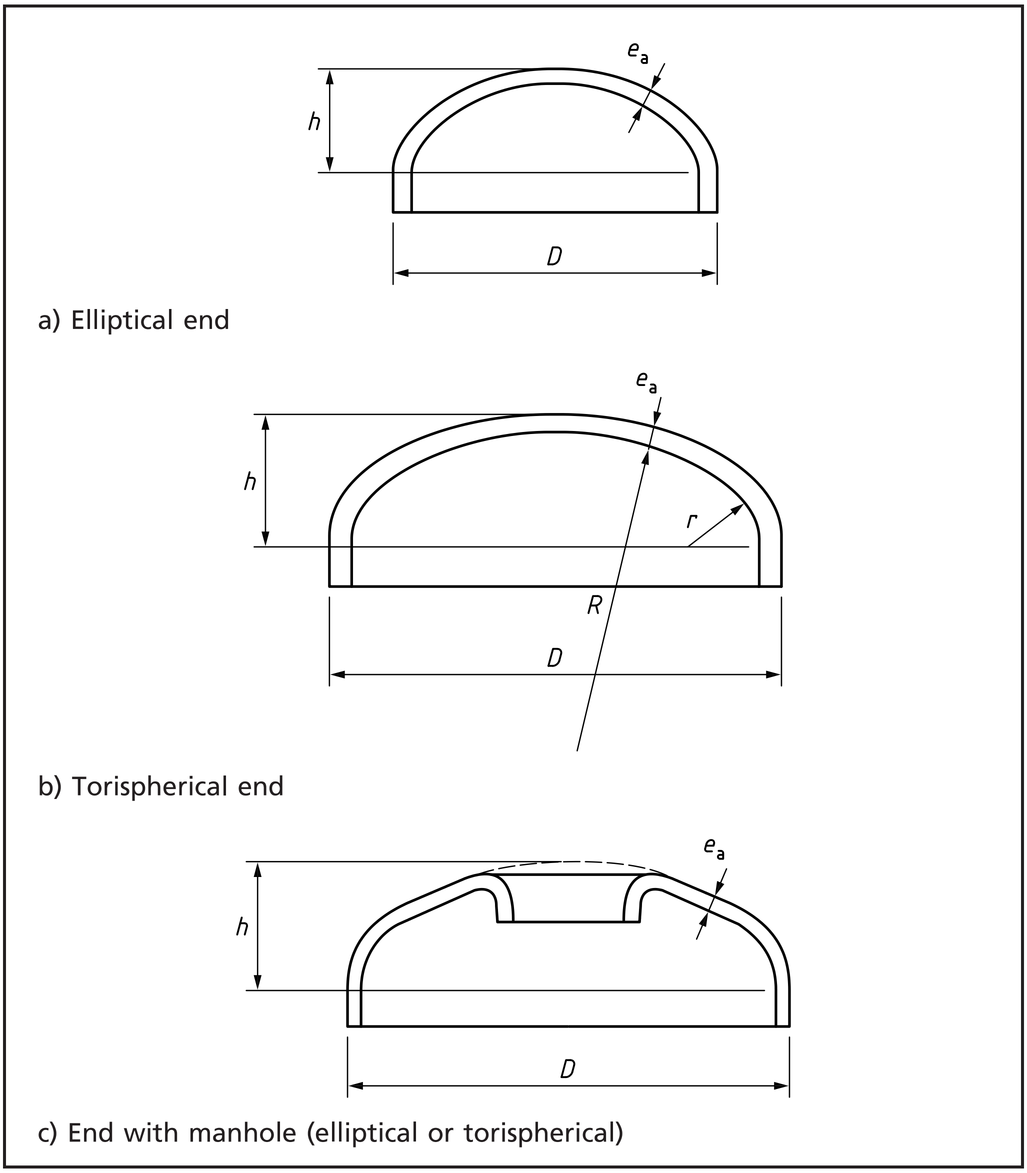
$r \geq 2e$

$R \leq D$

The three relationships in a) and the five relationships in b) shall be fulfilled simultaneously.

The thickness of the cylindrical or straight flange (see Figure 3.10-1 and Figure 3.10-2) of a dished end shall comply with **3.5.1.2**a) unless the length of the flange is less than $0.5\sqrt{De_a}$ in which case it may be the same thickness as the dished end.

Figure 3.5-1 **Dished ends**



a) Elliptical end

b) Torispherical end

c) End with manhole (elliptical or torispherical)

3.5.2.3 Dished ends

3.5.2.3.1 Hemispherical ends

The thickness of hemispherical ends shall be determined using Equations (3.5.1-3) or (3.5.1-4) in **3.5.1.2**.

3.5.2.3.2 Ellipsoidal and torispherical ends (see Figure 3.5-2 and Table 3.5-1)

The thickness of ellipsoidal and torispherical ends shall be determined using the following procedure.

a) Calculate $\frac{p}{f}$ from the design pressure p and the design stress of the chosen material f.

b) Enter Figure 3.5-2 with this value, read up to the appropriate h_e/D line for the proposed end shape and then across to the e/D axis for the corresponding e/D value.

c) Multiply by D to obtain the end thickness.

Interpolation between h_e/D curves is permissible or, alternatively, values may be read from the next highest h_e/D curve.

NOTE 1 The thickness of the spherical portion of a torispherical end may be determined as for a hemispherical end of spherical radius R within the area of diameter $D_c - 2x$, where:

$$x = 0.5\sqrt{R \times \text{torispherical thickness}}$$

NOTE 2 Figure 3.5-2 may be used with values of h_e and D based on internal dimensions, provided $h_e/D < 0.27$; beyond this value external dimensions are to be used.

3.5.2.4 Dished ends with openings

The thickness of hemispherical, torispherical and ellipsoidal ends with openings shall be determined in accordance with **3.5.2.3**. In the procedures of **3.5.4** for openings and nozzle connections the effective mean spherical diameter D_{eff} shall be derived as follows:

D_{eff} = $(D - e)$ for hemispherical ends;

D_{eff} = $(2R + e)$ for torispherical ends;

D_{eff} = $\{[D \times (\text{factor obtained from Table 3.5-1})] - e\}$ for ellipsoidal ends.

NOTE In cases where the design stress is time dependent, these procedures should generally give adequate margins against creep rupture. However, for dished ends made from ferritic steels in material groups 1 to 6, 7.1, 9 and 11 with a large D/e_a(> 100), and also for dished ends made from austenitic steels in material group 8, it is desirable to check that any end of life deformation that may be expected is acceptable.

Figure 3.5-2 **Design curves for unpierced dished ends**

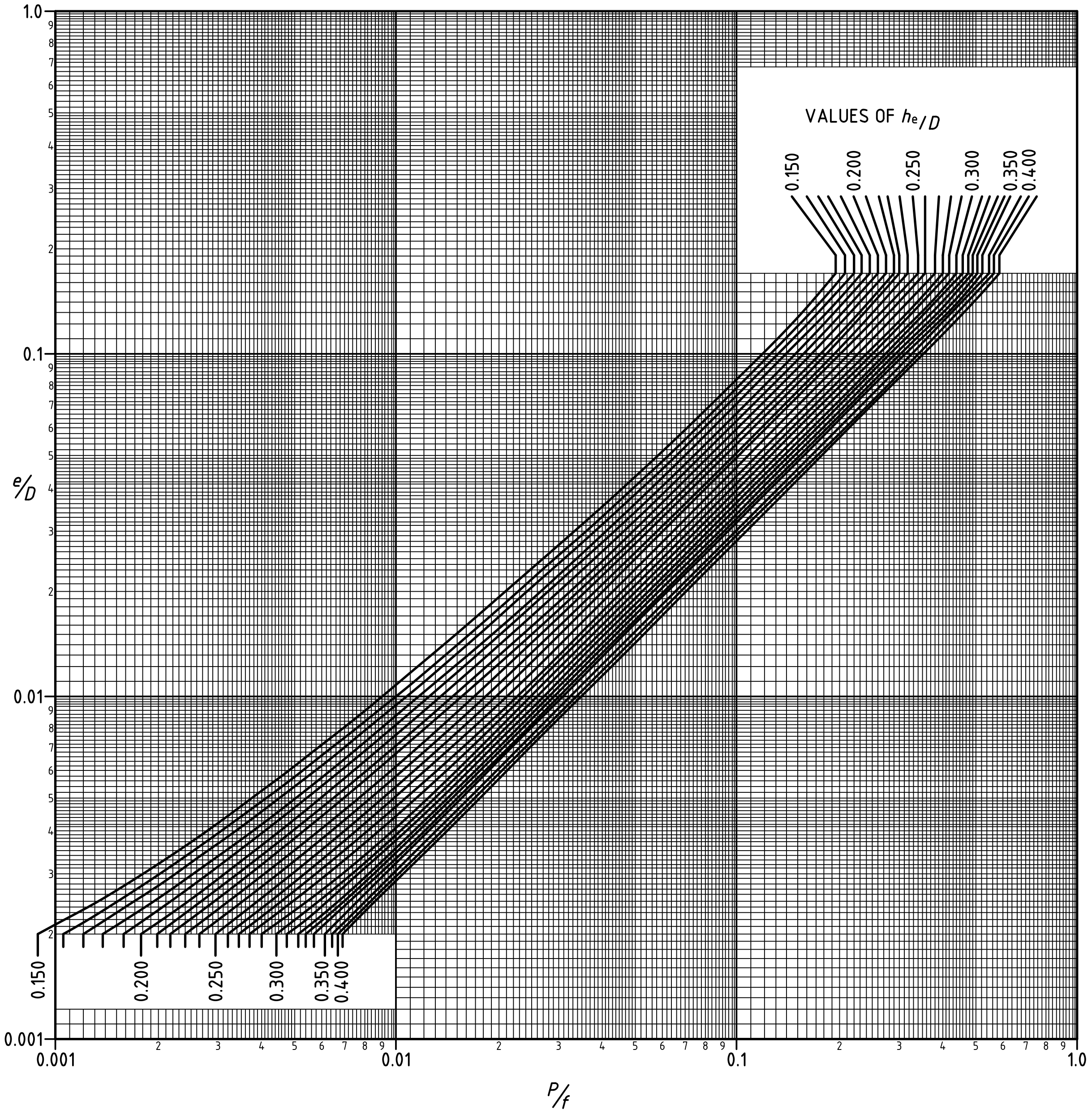


Table 3.5-1 **Values of factor for ellipsoidal ends**

h_e/D	Factor
0.18	2.52
0.192	2.36
0.208	2.17
0.227	1.98
0.25	1.80
0.278	1.63
0.313	1.46
0.357	1.30
0.417	1.14
0.50	1.00

Table 3.5-2 **Values of $e/D \times 10^3$ for unpierced dished ends in terms of h_e/D and p/f**

h_e/D	p/f									
	0.001	**0.0015**	**0.0025**	**0.004**	**0.006**	**0.010**	**0.015**	**0.025**	**0.050**	**greater values**
0.15	2.13	2.70	3.73	5.22	7.20	10.9	15.4	24.0	44.5	$880 \times p/f$
0.16	(1.95)	2.50	3.50	4.90	6.70	10.2	14.3	22.2	41.5	$810 \times p/f$
0.17	(1.80)	2.30	3.24	4.58	6.30	9.6	13.5	21.0	39.2	$770 \times p/f$
0.18	(1.65)	2.11	2.99	4.23	5.80	8.8	12.6	19.7	37.0	$730 \times p/f$
0.19		(1.95)	2.77	3.95	5.43	8.3	11.8	18.5	35.0	$695 \times p/f$
0.20		(1.80)	2.55	3.64	5.00	7.7	11.0	17.3	33.0	$650 \times p/f$
0.21		(1.65)	2.39	3.42	4.75	7.3	10.4	16.2	$620 \times p/f$	
0.22		(1.52)	2.22	3.20	4.45	6.84	9.7	15.4	$585 \times p/f$	
0.23		(1.40)	2.08	2.95	4.12	6.30	9.1	14.5	$555 \times p/f$	
0.24			(1.92)	2.76	3.83	5.90	8.5	13.6	$530 \times p/f$	
0.25			(1.75)	2.58	3.56	5.50	7.8	$500 \times p/f$		
0.26			(1.64)	2.40	3.34	5.15	7.35	$475 \times p/f$		
0.27			(1.52)	2.25	3.12	4.80	6.80	$445 \times p/f$		
0.28			(1.41)	2.12	2.93	4.50	6.45	$425 \times p/f$		
0.29				(2.00)	2.73	4.20	$405 \times p/f$			
0.30				(1.86)	2.54	3.95	$385 \times p/f$			
0.31				(1.71)	2.41	3.80	$370 \times p/f$			
0.32				(1.61)	2.30	3.65	$358 \times p/f$			
0.33				(1.52)	2.20	3.50	$345 \times p/f$			
0.34				(1.45)	2.10	$335 \times p/f$				
0.35					$325 \times p/f$					
0.36					$319 \times p/f$					
0.38					$307 \times p/f$					
0.40					$295 \times p/f$					

NOTE 1 *This table is not valid for values of $e/D \times 10^3 < 2.00$.*

NOTE 2 *Intermediate values may be obtained by logarithmic interpolation.*

NOTE 3 *Values in parentheses are provided for purposes of interpolation.*

3.5.3 Cones and conical ends

3.5.3.1 General

The following gives rules for cones and conical ends subjected to pressure loading. Right circular cones are covered in **3.5.3.3**, offset cones are covered in **3.5.3.7**. Cone to cylinder intersections shall be assessed in accordance with **3.5.3.4**, **3.5.3.5** and **3.5.3.6**. This assessment is not required where a girth flange, flat plate, tubesheet or other large stiffener (see **3.6.1.2.1**) is positioned close to the intersection, see **3.5.3.4**a), **3.5.3.5**a) or **3.5.3.6**a).

Where cone to cylinder junctions are subject only to a proportion of the total axial pressure end load it is not necessary for the junction to meet the reinforcement requirements of **3.5.3.4**, **3.5.3.5** and **3.5.3.6**. This junction arrangement occurs with “vapour belts” fabricated from cones and an external cylindrical shell and where local cut outs in the main cylindrical shell provide vapour distribution but still allow the axial pressure end load to be carried in the main cylindrical shell. This arrangement is shown in Figure 3.5-3.

Where a conical shell is subjected to loads in addition to internal pressure (see **3.2.2**) it is not possible to give explicit equations for the minimum thickness and solution by trial and error is necessary (see Annex B).

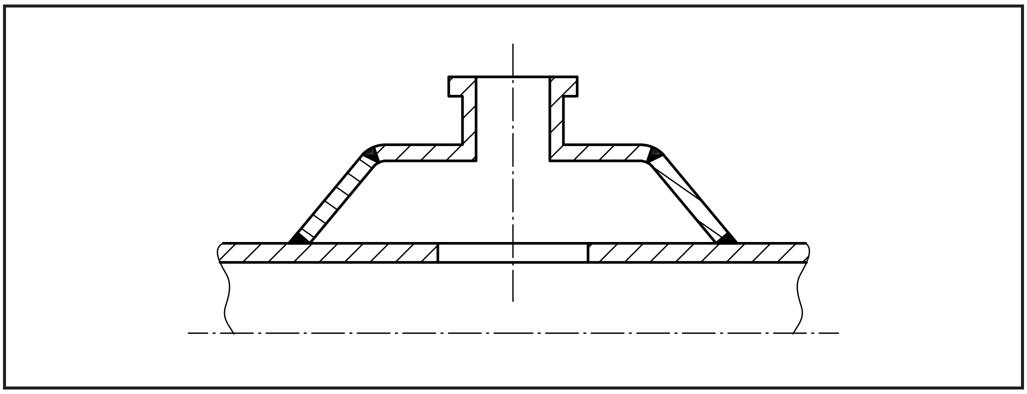
The rules do not apply to cones for which the half angle at the apex of the cone is greater than 60°, or for which

$$\frac{e_a \cos \alpha}{D_c} < 0.001$$

Short cones joining a jacket to a shell are not covered.

If construction category 3 applies, the category 3 design stress shall be used for all calculations in **3.5.3**.

Figure 3.5-3 **Conical shells: Vapour belt arrangement**



3.5.3.2 Notation

For the purposes of **3.5.3** the following symbols apply. All dimensions are in the corroded condition (see **3.1.5**).

- D_i is the inside diameter of the cone;
- D_e is the outside diameter of the cone;
- D_k a diameter used in cone design [see Equation (3.5.3-3)];
- D_m is the mean diameter of the cone;
- D_c is the mean diameter of the cylinder at the junction with the cone;
- e is minimum thickness of a cone as determined in **3.5.3.3**;
- e_a is the analysis thickness of the cone (see **1.6**);
- e_c is the minimum thickness of cylinder as determined in **3.5.1.2**;
- e_j is the minimum thickness at a junction at the large end of a cone [see Equation (3.5.3-4) or (3.5.3-6)];
- e_1 is the minimum thickness of cylinder at junction;
- e_2 is the minimum thickness of cone and knuckle at junction;
- f is the design stress;
- l_1 is the length along cylinder $= \sqrt{D_c e_1}$;
- l_2 is the length along cone at large or small end $= \sqrt{\dfrac{D_c e_2}{\cos \alpha}}$;
- p is the design pressure;
- r is inside radius of curvature of knuckle;
- s is a factor defined in **3.5.3.6**;
- α is the semi angle of cone at apex;
- β is a factor defined in **3.5.3.4**;

β_H is a factor defined in **3.5.3.6**;
γ is a factor defined in **3.5.3.5**;
ρ is a factor defined in **3.5.3.5**;
τ is a factor defined in **3.5.3.6**.

3.5.3.3 Minimum thickness of conical shell

The minimum permissible thickness at any point along the length of a cone is given by one of the following two equations:

$$e = \frac{pD_i}{2f - p} \times \frac{1}{\cos\ \alpha} \tag{3.5.3-1}$$

or

$$e = \frac{pD_e}{2f + p} \times \frac{1}{\cos\ \alpha} \tag{3.5.3-2}$$

At the large end of a cone joined to a cylinder it is permissible to replace D_i in Equation (3.5.3-1) by D_k where

$$D_k = D_c - e_1 - 2r[1 - \cos(\alpha)] - l_2 \sin(\alpha) \tag{3.5.3-3}$$

*NOTE 1 The thickness of the cone may have to be increased at the large and small ends to meet the requirements of **3.5.3.4, 3.5.3.5** and **3.5.3.6**. It may also have to be increased locally or generally to provide reinforcement at branches or openings or to carry non-pressure loads.*

NOTE 2 Since the thickness calculated above is the minimum allowable at that point along the cone, it is permissible to build a cone from plates of different thickness provided that at every point the minimum is achieved.

3.5.3.4 Junction between the large end of a cone and a cylinder without an intermediate knuckle

This subclause applies provided that:

a) the junction is positioned more than $2l_1$ along the cylinder and $2l_2$ along the cone from any other junction or major discontinuity, such as another cone/cylinder junction or a flange;

b) the joint is a butt weld where the inside and outside surfaces merge smoothly with the adjacent cone and cylinder without local reduction in thickness;

c) the weld at the junction shall be subject to 100% non-destructive examination, either by radiography or ultrasonics, unless the design is such that the thickness at the weld exceeds $1.4e_j$, in which case the rules for the relevant construction category shall be applied.

NOTE 1 The junction is defined as the intersection of shell centre-lines, see Figure 3.5-5a).

The minimum thickness e_1 of the cylinder adjacent to the junction is the greater of e_c and e_j where:

$$e_j = \frac{pD_c\beta}{2f} \tag{3.5.3-4}$$

$$\beta = \frac{1}{3}\sqrt{\frac{D_c}{e_j}} \times \left(\frac{\tan\ \alpha}{1 + 1/\sqrt{\cos\ \alpha}}\right) - 0.15 \tag{3.5.3-5}$$

NOTE 2 The above is a trial and error calculation for e_j. The answer is acceptable if the value given by equation (3.5.3-4) is not less than that assumed in Equation (3.5.3-5). Figure 3.5-4 and Table 3.5-3 gives β directly as a function of p/f.

This thickness shall be maintained for a distance of at least $1.4l_1$ from the junction along the cylinder.

The minimum thickness e_2 of the cone adjacent to the junction is the greater of e and e_j. This thickness shall be maintained for a distance of at least $1.4l_2$ from the junction along the cone.

It is permissible to modify a design according to the above rule with the following procedure, provided that the requirements of **3.5.1.2** and **3.5.3.3** continue to be met: the thickness of the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance $1.4l_1$ from the junction is not less than $1.4e_1l_1$. In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance $1.4l_2$ from the junction is not less than $1.4e_2l_2$.

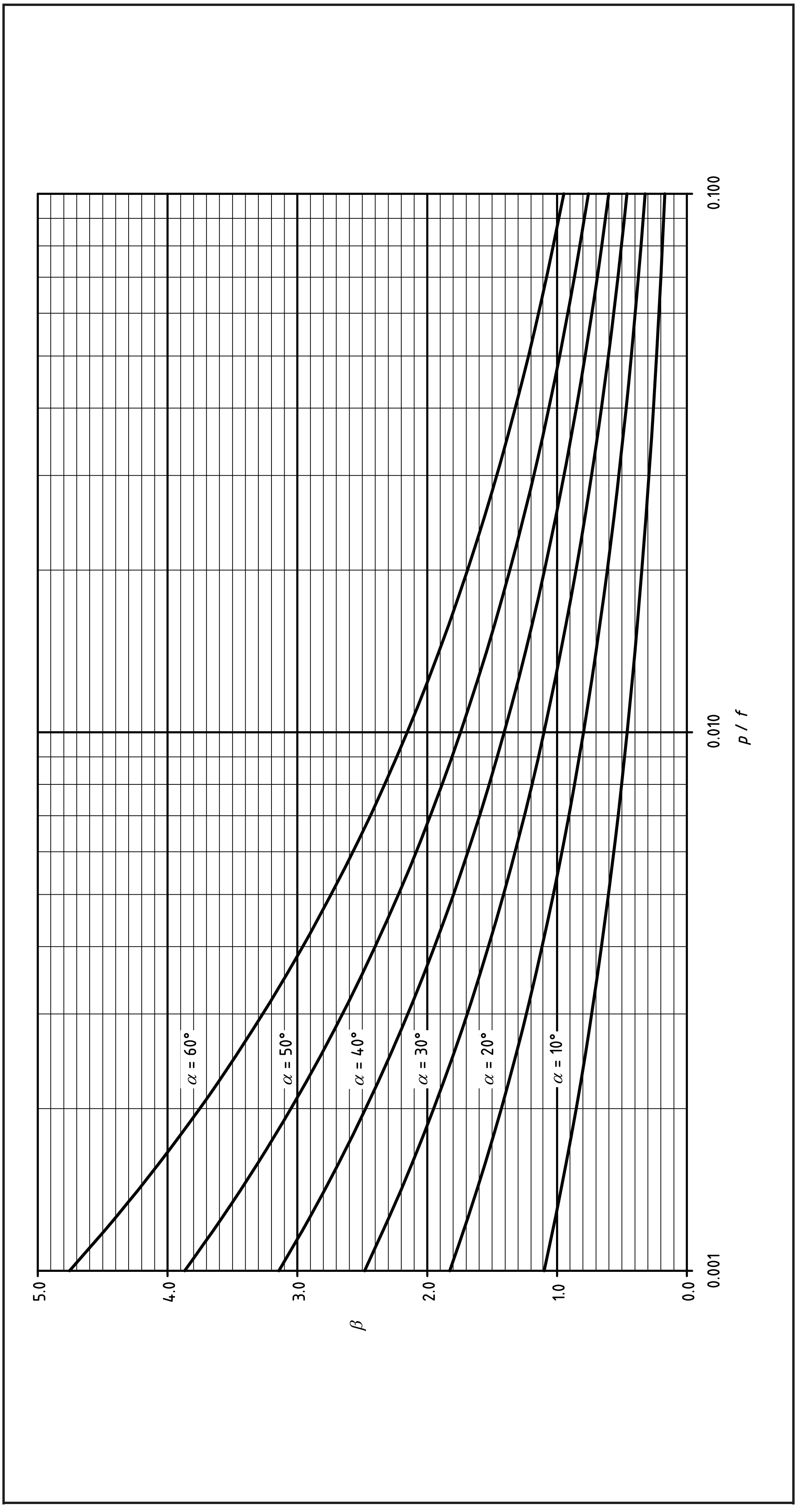


Figure 3.5-4 **Values of coefficient β for cone/cylinder intersection without knuckle**

Table 3.5-3 **Values of coefficient β for cone/cylinder intersection without knuckle (for Figure 3.5-4)**

p/f	α					
	10.0	**20.0**	**30.0**	**40.0**	**50.0**	**60.0**
0.001	1.099	1.826	2.483	3.143	3.869	4.755
0.0015	0.948	1.583	2.157	2.733	3.367	4.141
0.002	0.853	1.429	1.950	2.474	3.051	3.753
0.003	0.733	1.237	1.692	2.149	2.652	3.266
0.005	0.604	1.028	1.412	1.797	2.222	2.740
0.01	0.460	0.796	1.101	1.407	1.743	2.154
0.02	0.347	0.613	0.854	1.097	1.364	1.690
0.05	0.233	0.428	0.605	0.783	0.980	1.220
0.1	0.169	0.321	0.461	0.602	0.758	0.948

3.5.3.5 Junction between the large end of a cone and a cylinder with an intermediate knuckle

This subclause applies provided that:

a) the junction is positioned more than $2l_1$ along the cylinder and $2l_2$ along the cone from any other junction or major discontinuity, such as another cone/cylinder junction or a flange;

b) the knuckle is of toroidal form and merges smoothly with the adjacent cone and cylinder;

c) the inside radius of curvature of the knuckle, $r \leq 0.3D_c$.

NOTE 1 The junction is defined as the meeting point between the mid-thickness lines of cylinder and cone, extended as necessary [see Figure 3.5-5b)].

NOTE 2 This subclause does not prescribe a lower limit to the radius of curvature of the knuckle.

The minimum thickness e_1 of the cylinder adjacent to the junction is the greater of e_c and e_j. This thickness shall be maintained to a distance of at least $1.4l_1$ from the junction along the cylinder and $0.5l_1$ from the cylinder/knuckle tan-line.

The minimum thickness e_2 of the knuckle and the cone adjacent to the junction is the greater of e and e_j (e is to be determined at the diameter of the junction of cone and knuckle). This thickness shall be maintained to a distance of at least $1.4l_2$ from the junction along the cone and $0.7l_2$ from the cone/knuckle tan-line.

The value of e_j is given by:

$$e_j = \frac{pD_c\beta}{2f\gamma} \quad (3.5.3\text{-}6)$$

where

$$\gamma = 1 + \frac{\rho}{1.2\left(1 + \frac{0.2}{\rho}\right)} \quad (3.5.3\text{-}7)$$

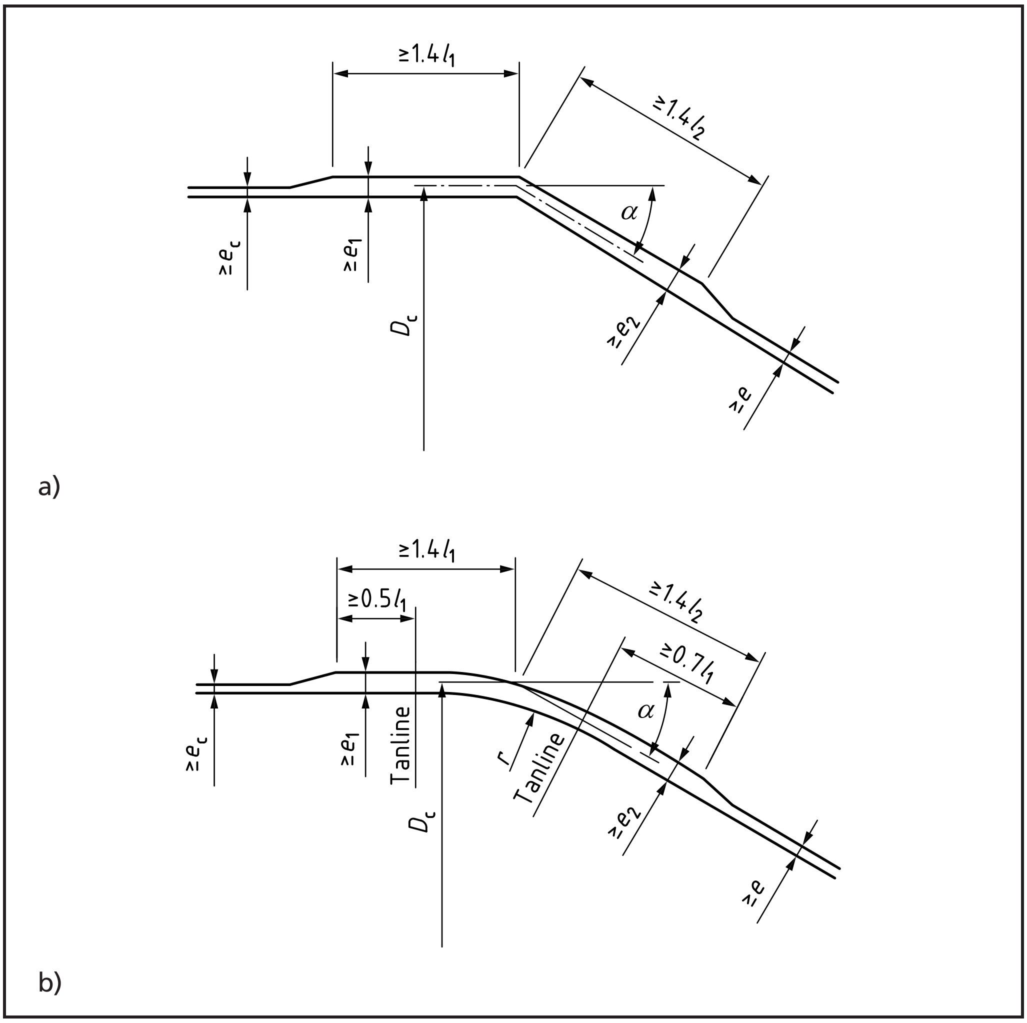
and

$$\rho = \frac{0.028r}{\sqrt{D_c e_j}} \times \frac{\alpha}{1 + 1/\sqrt{\cos\ \alpha}} \quad (3.5.3\text{-}8)$$

where α in the numerator is in degrees.

NOTE 3 The above is a trial and error calculation for e_j*. The answer is acceptable if the value given by Equation (3.5.3-6) is not less than that assumed in Equations (3.5.3-5) and (3.5.3-8).*

Figure 3.5-5 **Geometry of cone/cylinder intersection: large end**



3.5.3.6 Junction between the small end of a cone and a cylinder

This subclause applies provided that:

a) the junction is more than $2l_1$ along the cylinder and $2l_2$ along the cone from any other junction or major discontinuity, such as another cone/cylinder junction or a flange;

b) the minimum thickness of the cylinder e_1 is maintained for a distance l_1 and that of the cone e_2 is maintained for a distance l_2 from the junction (see Figure 3.5-6);

c) the thicknesses meet the requirements of **3.5.1.2** and **3.5.3.3**.

Minimum thicknesses e_1 and e_2 shall be chosen so that:

$$e_1 = \frac{pD_c\beta_H}{2f} \tag{3.5.3-9}$$

where

$$\beta_{\mathrm{H}} = 0.4\sqrt{\frac{D_{\mathrm{c}}}{e_1}}\left(\frac{\tan\ \alpha}{\tau}\right) + 0.5 \qquad (3.5.3\text{-}10)$$

and

$$\tau = s\sqrt{\frac{s}{\cos\ \alpha}} + \sqrt{\frac{1+s^2}{2}} \text{ when } s < 1 \qquad (3.5.3\text{-}11)$$

or

$$\tau = 1 + \sqrt{\frac{1+s^2}{2\cos\alpha}} \times s \text{ when } s \geq 1 \qquad (3.5.3\text{-}12)$$

with, in both cases, $s = \frac{e_2}{e_1}$.

NOTE 1 The above requirements do not provide values for e_1 and e_2 separately. They may be adjusted relative to each other to suit the needs of the design, for example to obtain a favourable value of l_1 or l_2 for use in the procedure that follows.

It is permissible to modify a design according to the above rule with the following procedure, provided that the requirements of **3.5.1.2** and **3.5.3.3** continue to be met: the thickness of the cylinder may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cylinder within a distance l_1 from the junction is not less than l_1e_1. In addition, the thickness of the cone may be increased near the junction and reduced further away provided that the cross-sectional area of metal provided by the cone within a distance l_2 from the junction is not less than l_2e_2.

NOTE 2 When using the modification to check a given geometry, the procedure for finding e_1 is as follows. Guess e_1.

Calculate $l_1 = \sqrt{D_{\mathrm{c}}e_1}$.

Calculate the metal area A_1 within a distance l_1 from the junction along the cylinder. Then a better estimate of e_1 is given by

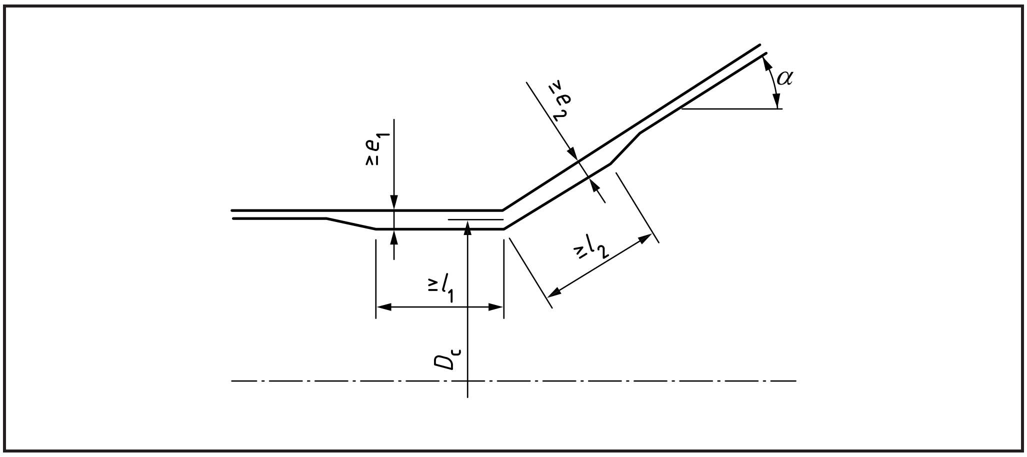
$$e_1 = \frac{A_1}{l_1}.$$

A similar procedure can be used to find e_2.

NOTE 3 Where $e_1 = e_2$ then a torispherical knuckle of the same thickness may be included. l_1 and l_2 continue to be measured from the junction (the point where the centre lines of cone and cylinder meet).

*NOTE 4 The calculations in **3.5.3.6** are for minimum thicknesses. Actual thicknesses may exceed the minima without leading to any increase in l_1 or l_2.*

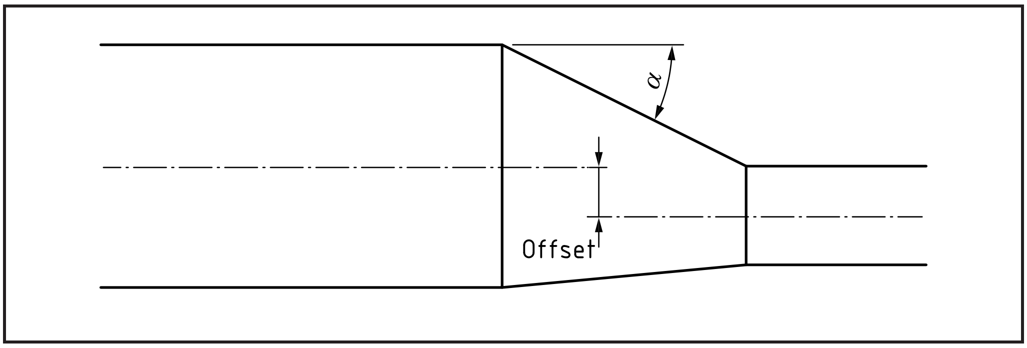
Figure 3.5-6 **Geometry of cone/cylinder intersection: small end**



3.5.3.7 Offset cones

This rule is for offset cones between two cylinders (see Figure 3.5-7). The cylinders shall have parallel centre lines offset from each other by a distance no greater than the difference of their radii. A minimum thickness shall be calculated in accordance with **3.5.3.4** above, for the junction at the large end. A minimum thickness shall be calculated in accordance with **3.5.3.6** above, for the junction at the small end. The greater of these shall apply to the whole cone. The angle (α) is taken as the maximum angle between cone and cylinder.

Figure 3.5-7 **Offset cone**



3.5.4 Openings and nozzle connections

3.5.4.1 General

The amount of compensation to be provided at an opening shall be not less than that specified in **3.5.4.3** and **3.5.4.4** or in accordance with the alternative design method specified in **3.5.4.9**. **3.5.4.3** specifies requirements for the design of isolated openings and nozzle connections in cylinders, spheres and cones, in the form of design procedures. **3.5.4.4** specifies requirements for groups of openings and the procedure allows the checking of chosen geometry. The remainder of **3.5.4** deals with reinforcing pads, external pressure, nozzle pipes and studded, socket welded and screwed connections.

Groups of openings where the openings are arranged in a regular array shall be checked in accordance with **3.5.4.9.3**j).

Figure 3.5-13 to Figure 3.5-23 and Figure 3.5-26, Figure 3.5-27 and Figure 3.5-28 show various nozzle and shell arrangements and indicate key design notation. Figure 3.5-24 and Figure 3.5-25 show the notation associated with groups of nozzles and Figure 3.5-29 and Figure 3.5-30 show the redistribution of compensation close to the opening. The design charts given in Figure 3.5-9, Figure 3.5-10, Figure 3.5-11 and Figure 3.5-12 are based on approximate analyses considering internal pressure loading only, but the effect of other loads shall be taken into account by the selection of an appropriate value of the factor *C* and using the procedure specified in **3.5.4.3**.

The effects of attachments and discontinuities in the proximity of openings shall be calculated in accordance with **3.5.4.10**. In no case shall the thickness of nozzle connections be less than the thickness specified in **3.5.4.7** for branch pipes. Where it is proposed to use material for nozzles or added compensation which is dissimilar to the main shell material, the design shall be in accordance with **3.5.4.3.7**. All nozzle connections, nozzles and openings outside the scope of the application of this clause, as defined in **3.5.4.2**, shall be designed on the basis of special analysis, evidence or tests as permitted by **3.2.2**b).

For the purposes of **3.5.4.2**, **3.5.4.3**, **3.5.4.4**, **3.5.4.5** and **3.5.4.9**, a nozzle is defined as the component(s) up to the coupling point connecting the vessel to other equipment, i.e. up to the face of the first connection flange, or for welded connections up to the first weld beyond the limit of the opening reinforcement.

NOTE 1 The basis on which these requirements are founded is outlined in Part 2 of PD 6550, the Explanatory Supplement to BS 5500.

*NOTE 2 For connection flanges see **3.8.1**.*

NOTE 3 In cases where the design strength is time dependent, these requirements should generally give adequate margins against creep rupture. However, for vessels made from ferritic steels in material groups 1 to 6, 7.1, 9 and 11 with a large D/e_{as} (>100) and also vessels made from austenitic steels in material group 8, it is desirable to check that any end of life deformation that may be expected is acceptable.

For the purposes of **3.5.4.2**, **3.5.4.3**, **3.5.4.4** and **3.5.4.5** the following symbols apply. All dimensions are in the corroded condition (see **3.1.5**) except where otherwise indicated.

A_n, A_s, A_t	are the cross-sectional areas used in calculating compensation for adjacent nozzles (see Figure 3.5-25);
C	is a factor to take account of external loads (see **3.5.4.3.1**);
D	is the mean diameter of spherical or cylindrical section of shell ($2R_c$ for conical section of shell) (see Figure 3.5-13), or in the case of dished ends, the mean diameter of the equivalent sphere derived in **3.5.2.4**;
d	is the mean of the inside and outside diameter of a nozzle, or the bore of an opening not provided with a nozzle; in the case of non-circular openings or nozzles, see **3.5.4.3.6**; in the case of oblique nozzles, see Figure 3.5-15a);
d_A	is the average value of d for any two adjacent openings being considered;
d_x	is half the length of a reinforcing plate in the axial direction (see Figure G.3.1-2);
d_ϕ	is half the length of a reinforcing plate in the circumferential direction (see Figure G.3.1-2);
e_{as}	is the analysis thickness of the reinforced shell (see **1.6**);
e_{ns}	is the nominal uncorroded thickness of the reinforced shell;
e_{ps}	is the required unreinforced shell thickness for pressure loading only;
e_{rs}	is the minimum reinforced thickness of shell as required by **3.5.4**;
e_{ab}	is the analysis thickness of the reinforced nozzle (see **1.6**);
e_{pb}	is the required unreinforced nozzle thickness for pressure loading only;
e_{rb}	is the minimum reinforced thickness of nozzle as required by **3.5.4**;
f_s	is the design stress of shell;
f_b	is the design stress of nozzle or of rim reinforcement;
g	is the arrangement factor (see Figure 3.5-24);

K_1, K_2 are compensation ratios [see Equations (3.5.4-5) and (3.5.4-6)];

L_s is the distance along shell within which shell thickening is assumed to contribute to reinforcement of opening;

L_b is the distance along nozzle within which nozzle thickening is assumed to contribute to reinforcement of opening;

P is the pitch measured between centre lines of two openings along mid-thickness of shell (see Figure 3.5-24);

p is the design pressure;

R_c is the mean radius of conical shell section at opening (see Figure 3.5-13);

s is the shortest distance, measured along the mid-thickness line, between the bores of adjacent openings not provided with nozzles or between the mean diameters of adjacent nozzles (see Figure 3.5-25);

α is the one-half apex angle of cone (see Figure 3.5-13);

ρ is a non-dimensional design parameter as defined in **3.5.4.3** Equation (3.5.4-2).

3.5.4.2 Application

The requirements in **3.5.4.3** and **3.5.4.4** are valid for the design of circular and obround openings and nozzles (including oblique nozzles), arranged singly or in groups, in spherical, cylindrical, and conical shells, and dished ends, positioned to comply with **3.10.1.2**, provided that the following conditions a) to d) are satisfied.

a) Spherical shells

1) Openings and nozzles normal to shell surface

i) The major axis (mean dimension or, where no nozzle is fitted, the bore) of the opening does not exceed one-half of the diameter of the shell.

ii) The ratio of the major to minor axes of the opening shall not exceed 2.

2) Oblique nozzles

The nozzle is of circular cross-section conforming to a)1)i) and the angle between the axis of the nozzle and a line normal to the shell surface shall not exceed 50°.

b) Cylindrical shells

1) Openings and nozzles normal to the shell surface

The ratio of the major to minor axis of the opening shall not exceed 2.

2) Oblique nozzles

The nozzle shall be of circular cross-section and the angle between the axis of the nozzle and a line normal to the shell surface shall not exceed 50°.

3) Protruding nozzles

$(d - e_{pb})$ shall not exceed one-third of the mean diameter of the shell.

4) Flush nozzles

$(d - e_{pb})$ shall not exceed one-third of the mean diameter of the shell if $D/e_{ns} > 200$. If $D/e_{ns} \leq 200$, no limit is placed on the permissible diameter of a flush nozzle.

c) Conical shells

1) Openings and nozzles normal to the shell surface

i) The major axis (mean dimension or, where no nozzle is fitted, the bore) of the opening shall not exceed one-third of the diameter of the shell.

ii) The ratio of the major to minor axes of the opening shall not exceed 2.

2) Oblique nozzles

The nozzle shall be of circular cross-section conforming to c)1)i) and the angle between the axis of the nozzle and a line normal to the shell surface shall not exceed 50°.

d) Dished ends

1) Openings and nozzles normal to the shell surface

i) The major axis of the opening shall not exceed one-half of the diameter of the sphere for a hemispherical end or 0.6 times the outside diameter of the adjoining cylindrical shell for a torispherical or semi-ellipsoidal end.

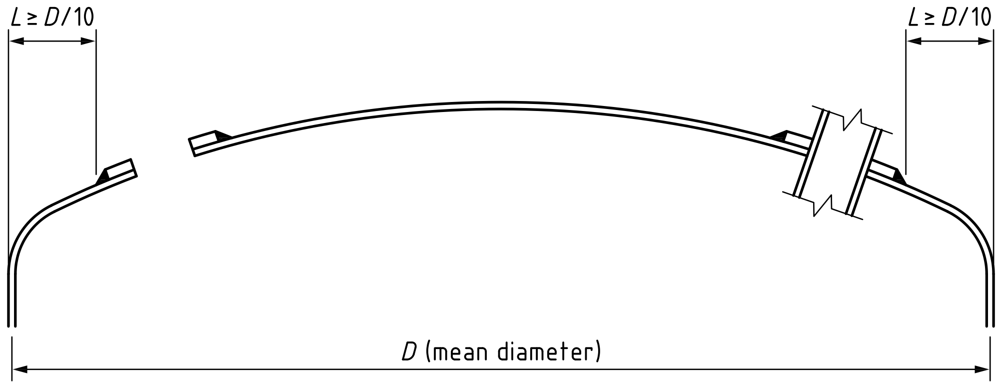
ii) The ratio of the major to minor axes of the opening shall not exceed 2.

iii) Openings and nozzles in torispherical and semi-ellipsoidal ends shall be positioned to conform to Figure 3.5-8. Where reinforcement is provided by means of pads or local thickening of the dished end plate, the distance L shall be measured from the edge of the weld or taper nearest the outside of the vessel. Where a dished end has uniform thickness, the distance L shall be measured from the outside of the nozzle or rim of the opening.

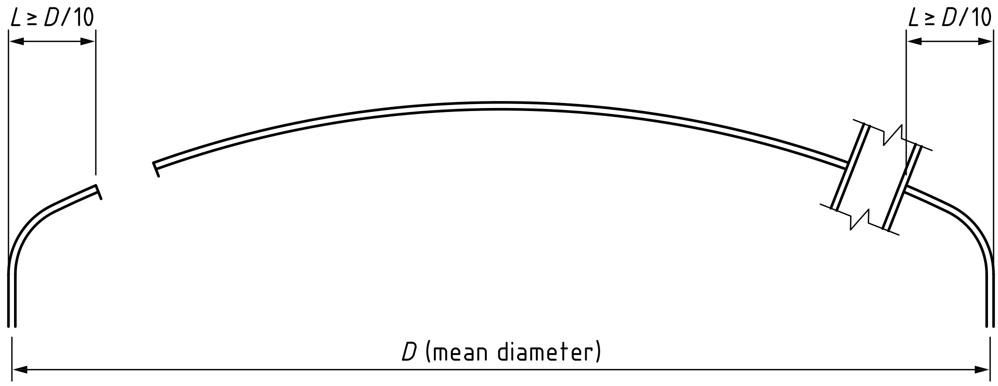
2) Oblique nozzles

The nozzle shall be of circular cross-section conforming to d)1)i) and the angle between the axis of the nozzle and a line normal to the shell surface shall not exceed 50°.

Figure 3.5-8 **Positions of openings or nozzles in dished ends (for weld details see Annex E)**



a) Reinforcement is provided by means of pads (or local thickening of shell)



b) No local reinforcement is provided

3.5.4.3 Design of isolated openings and nozzle connections

3.5.4.3.1 General

Where external loads are negligible, as with a manway, the factor C shall be taken as not more than 1.1.

Where external loadings are not negligible, stresses shall be calculated (for example using Annex G) and assessed in accordance with Annex A. It is permissible to omit this calculation if the value of factor C is taken as 1.0 and if it is possible to demonstrate the adequacy of the design by comparison with similar existing designs.

For vessels operating in the creep range, the factor C shall be taken as not more than 1, and where external loads are not negligible, the stresses shall be calculated (for example using Annex G) and assessed in accordance with Annex A.

3.5.4.3.2 Openings not fitted with nozzle connections

If $\rho = \dfrac{d}{D}\sqrt{\dfrac{D}{2e_{as}}} \leq 0.1$, no further reinforcement is required.

To calculate e_{rs}, the minimum reinforced shell thickness, carry out the following iterative procedure stages a) to g).

a) Calculate e_{ps}, the required unreinforced thickness of the unpierced shell for pressure loading only.

For a cylindrical, spherical or conical shell e_{ps} shall be calculated using the equations in **3.5.1.2** or **3.5.3.3**. For a conical shell the diameter is measured at the location of the opening.

For a hemispherical, torispherical or ellipsoidal end calculate the effective mean spherical diameter D_{eff} given in **3.5.2.4**, and calculate the required thickness e_{ps} using Equation (3.5.4-1):

$$e_{ps} = \frac{pD_{eff}}{4f_s - 0.2p} \quad (3.5.4\text{-}1)$$

b) Select a value of e_{rs}, the minimum reinforced shell thickness, not less than e_{ps}.

c) Calculate the mean shell diameter D using the selected thickness.

d) Determine d, the bore of the opening.

e) Calculate the parameter ρ from Equation (3.5.4-2). If $\rho \leq 0.1$ no further reinforcement is required.

$$\rho = \frac{d}{D}\sqrt{\frac{D}{2e_{rs}}} \quad (3.5.4\text{-}2)$$

f) For larger values of ρ, determine Ce_{rs}/e_{ps} using Figure 3.5-10 or Figure 3.5-11 or Table 3.5-6 with $e_{rb}/e_{rs} = 0$.

g) Taking $C = 1.1$, calculate a new e_{rs} from the value of Ce_{rs}/e_{ps} obtained in f). If this e_{rs} is less than or equal to e_{rs} selected in b) then reinforcement is sufficient. If not, then repeat from step b) selecting an increased value of e_{rs}.

To check an existing geometry carry out steps a) to g) above using e_{as} in place of e_{rs}. The value of e_{rs} calculated in step g) shall not exceed e_{as}.

3.5.4.3.3 Openings fitted with nozzle connections

It is permissible to reinforce nozzle connections by means of an increase in shell thickness or nozzle thickness or by a combination of such increases, subject to the nozzle thickness limits specified in **3.5.4.3.4**c) and external pipework loads. To calculate the minimum reinforced shell and nozzle thicknesses carry out the following iterative procedure stages a) to j).

a) Calculate e_{ps}, the required unreinforced thickness of the unpierced shell for pressure loading using the procedure in **3.5.4.3.2**a).

b) Select a value of e_{rs}, the minimum reinforced shell thickness, not less than e_{ps}.

c) Calculate the mean shell diameter D using the selected thickness.

d) Calculate e_{pb}, the minimum required thickness of the nozzle wall for pressure loading using **3.5.1.2**a).

e) Select a value of e_{rb}, the minimum reinforced nozzle thickness, not less than e_{pb}.

f) Calculate d, the mean nozzle diameter using this selected thickness.

g) Calculate the parameter ρ from Equation (3.5.4-2) and obtain a value of C from **3.5.4.3.1**.

h) Calculate Ce_{rs}/e_{ps} and d/D, then use the relevant Figure 3.5-9, Figure 3.5-10 or Figure 3.5-11 to obtain a value for e_{rb}/e_{rs} and thus e_{rb} using the value e_{rs} selected in step b). In the case of flush nozzles in cylindrical vessels, if $d/D > 0.3$ use Figure 3.5-12 to obtain a value of e_{rb} directly, or where $0.2 < d/D < 0.3$, a value of e_{rb1} shall be derived using Figure 3.5-11 and a value of e_{rb2} shall be derived from Figure 3.5-12, then e_{rb} shall be obtained from Equation (3.5.4-3) as follows:

$$e_{rb} = e_{rb1} + 10(d/D - 0.2)(e_{rb2} - e_{rb1}). \tag{3.5.4-3}$$

i) If e_{rb} differs from the value selected in step e), select a new value and repeat from step e) until the iteration has converged. Check that e_{rb} is not greater than $(2 - d/D)e_{rs}$. If it is, select a larger value of e_{rs} and repeat from step b). See **3.5.4.3.7** if nozzle design stress f_b, is less than the shell design stress f_s.

j) From the calculated e_{rb} select the nominal nozzle thickness and thus the analysis thickness e_{ab}. Check that e_{ab} is not less than the thickness given in Table 3.5-4. If it is, select a larger value of e_{rs} and repeat from step b).

To check an existing geometry carry out steps a) to h) above using e_{ab} and e_{as} in place of e_{rb} and e_{rs} respectively. The value of e_{rb} obtained in step h) shall not exceed e_{ab}.

Extrapolation of Figure 3.5-9, Figure 3.5-10 and Figure 3.5-11 is not permitted. If the calculated value of ρ is greater than the limit for the relevant figure then e_{rs} shall be increased in order to reduce ρ to an acceptable value.

NOTE 1 If the calculated value of Ce_{rs}/e_{ps} is greater than the value from the appropriate figure at the calculated value of ρ and with e_{rb}/e_{rs} = 0, then no reinforcement is required in the nozzle and the thickness can be chosen from Table 3.5-4.

NOTE 2 Figure 3.5-9, Figure 3.5-10 and Figure 3.5-11 are provided for ease of application in manual calculations. Definitive values from these figures are obtained by linear interpolation from the data given in Table 3.5-5. To assist in the interpolation for small values of e_{rb}/e_{rs} tabulated values of Ce_{rs}/e_{ps} against ρ are given in Table 3.5-6. Figure 3.5-12 gives both curve and data for manual calculation.

Figure 3.5-9 **Design curves for protruding nozzles in spherical vessels *(d/D* < 0.5*)* and for protruding nozzles in cylindrical and conical vessels *(d/D* < 1/3)**

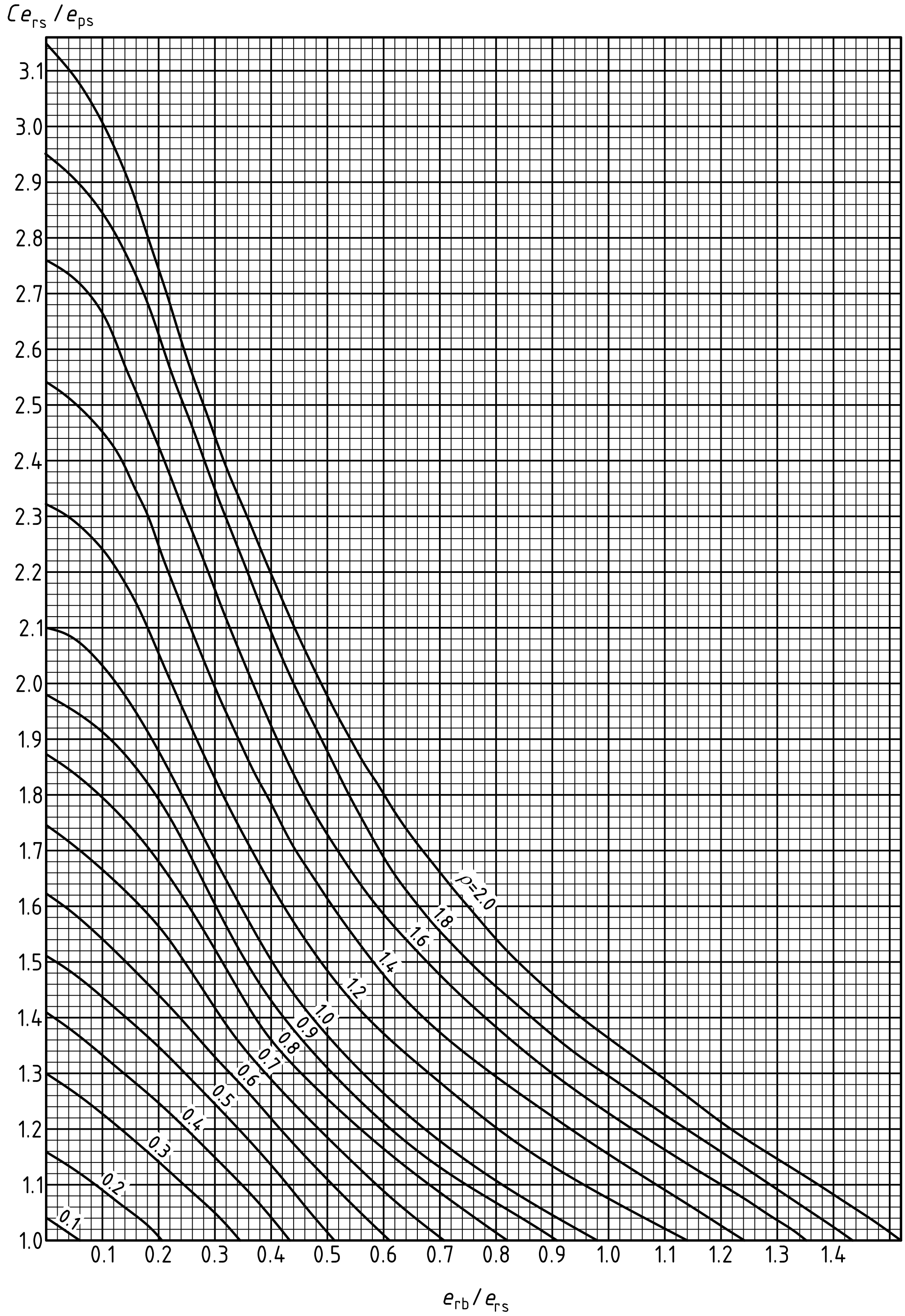


Figure 3.5-10 **Design curves for flush nozzles in spherical shells (*d*/*D* < 0.5) and for flush nozzles in conical shells (*d*/*D* < 1/3)**

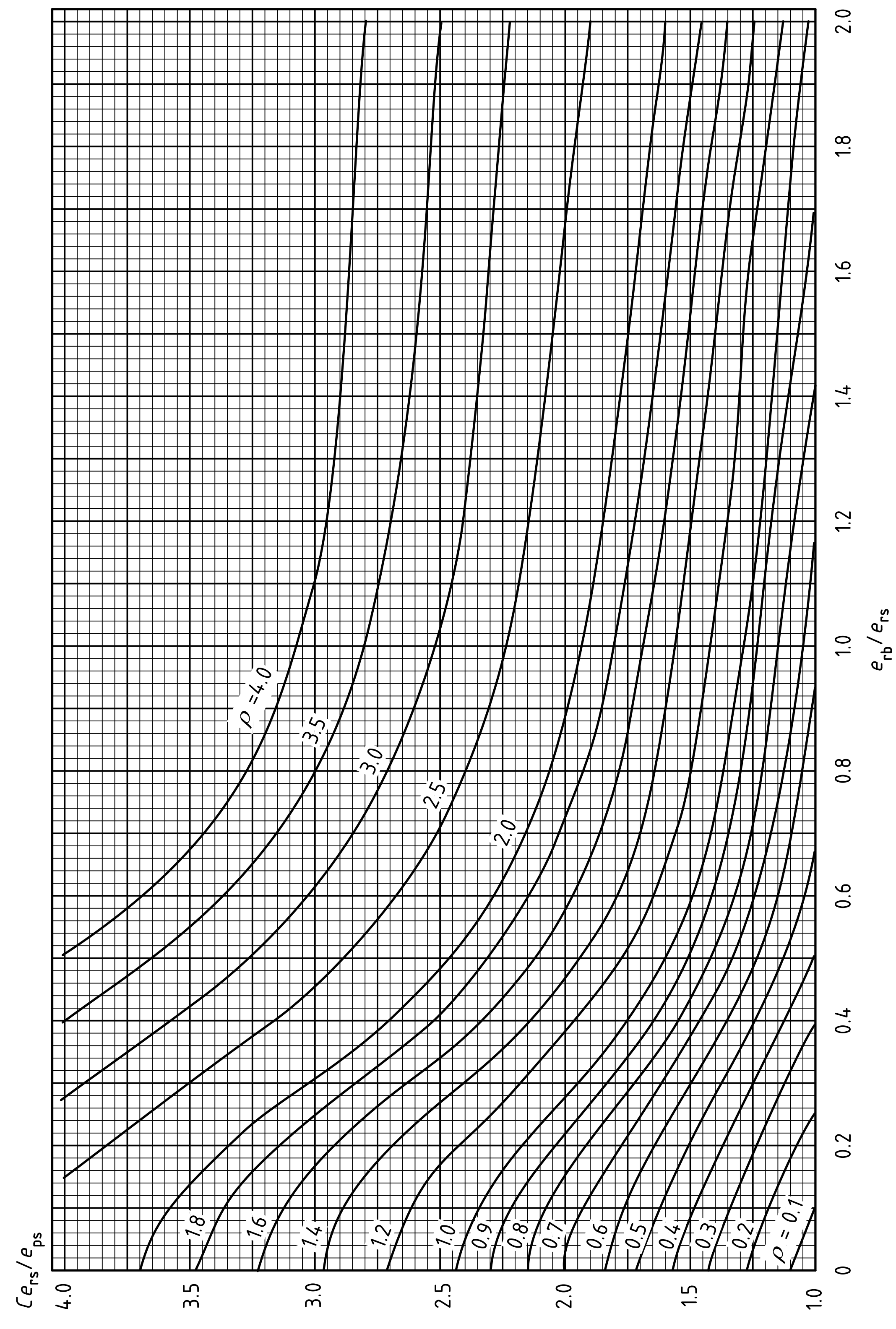
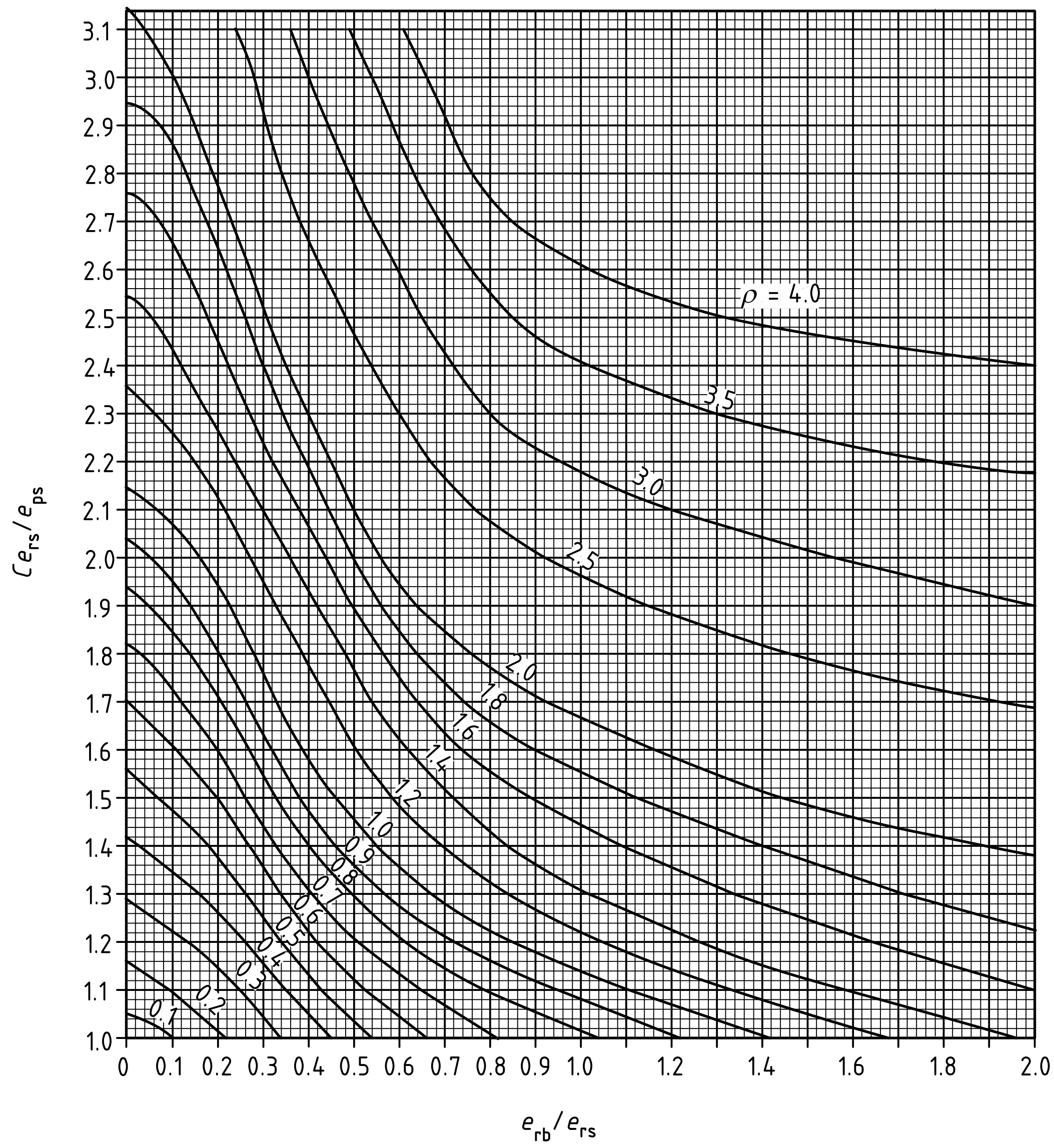
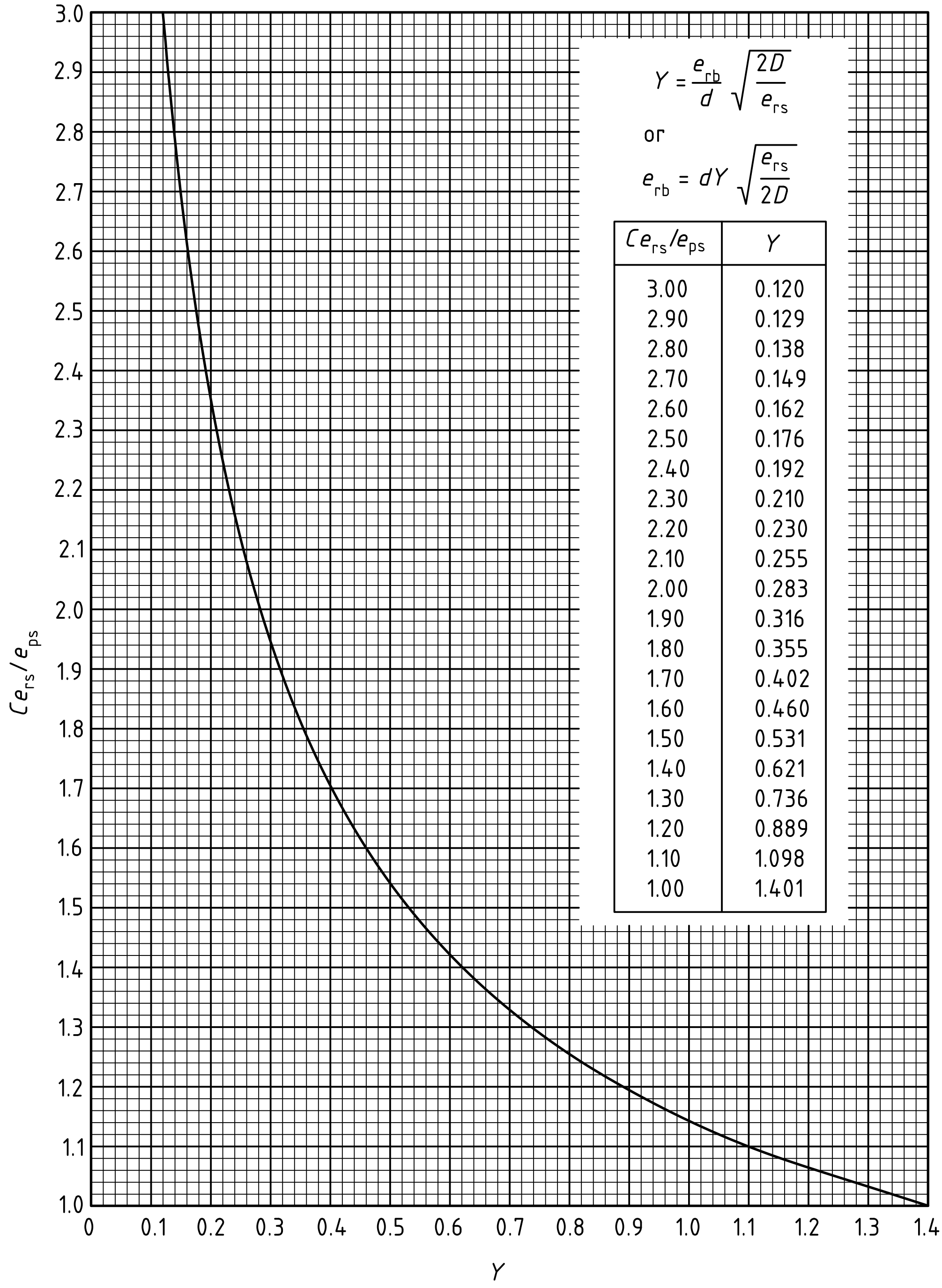


Figure 3.5-11 **Design curves for flush nozzles in cylindrical shells (0 < d/D < 0.3)**



*NOTE For values of d/D: 0.2 < d/D < 0.3 see **3.5.4.3.3**.*

Figure 3.5-12 **Design curves for flush nozzles in cylindrical shells (0.2 < d/D ≤ 1.0)**



$$Y = \frac{e_{rb}}{d}\sqrt{\frac{2D}{e_{rs}}}$$

or

$$e_{rb} = dY\sqrt{\frac{e_{rs}}{2D}}$$

Ce_{rs}/e_{ps}	Y
3.00	0.120
2.90	0.129
2.80	0.138
2.70	0.149
2.60	0.162
2.50	0.176
2.40	0.192
2.30	0.210
2.20	0.230
2.10	0.255
2.00	0.283
1.90	0.316
1.80	0.355
1.70	0.402
1.60	0.460
1.50	0.531
1.40	0.621
1.30	0.736
1.20	0.889
1.10	1.098
1.00	1.401

NOTE 1 This figure may be represented by the following expression which may also be used for Ce_{rs}/e_{ps} values greater than 3:

$$Y = \frac{64}{\left[4Ce_{rs}/e_{ps} + 0.8 + \left\{16\left(Ce_{rs}/e_{ps}\right)^2 - 12.8Ce_{rs}/e_{ps} + 0.64\right\}^{0.5}\right]^2}$$

NOTE 2 For 0.2 < *d/D* < 0.3 *see* ***3.5.4.3.3***.

Table 3.5-4 **Thickness of nozzles**

Nozzle nominal size mm	Minimum analysis thickness mm
15	2.4
20	2.4
25	2.7
32	3.1
40	3.1
50	3.6
65	3.9
80	4.7
100	5.4
125	5.4
150	6.2
200	6.9
250	8.0
300	8.0
350	8.8
400	8.8
450	8.8
500	10.0
600	10.0

*NOTE These tabular values incorporate a margin of strength, suggested by experience, to cover additional loading by connected pipework of the order normally to be expected with a properly designed and supported piping arrangement. The use of Table 3.5-4 is described in **3.5.4.7**b).*

Table 3.5-5 **Design values of e_{rb}/e_{rs}**

a) Associated with Figure 3.5-9

Ce_{rs}/e_{ps}	ρ														
	2.0	**1.8**	**1.6**	**1.4**	**1.2**	**1.0**	**0.9**	**0.8**	**0.7**	**0.6**	**0.5**	**0.4**	**0.3**	**0.2**	**0.1**
3.2															
3.1	.04														
3.0	.11														
2.9	.15	.06													
2.8	.19	.13													
2.7	.22	.17	.07												
2.6	.25	.21	.13												
2.5	.29	.24	.17	.06											
2.4	.33	.28	.21	.13											
2.3	.36	.31	.25	.18	.04										
2.2	.40	.35	.29	.22	.13										
2.1	.45	.39	.33	.26	.18	.00									
2.0	.49	.44	.37	.30	.23	.12									
1.9	.55	.49	.42	.35	.28	.19	.11								
1.8	.61	.54	.46	.39	.32	.25	.20	.10							
1.7	.68	.61	.52	.45	.37	.30	.26	.19	.07						
1.6	.75	.68	.59	.51	.42	.35	.31	.26	.17	.04					
1.5	.85	.77	.68	.58	.49	.40	.36	.31	.25	.15	.02				
1.4	.95	.88	.79	.68	.58	.47	.43	.37	.32	.24	.15	.01			
1.3	1.08	1.00	.91	.80	.68	.56	.51	.45	.39	.33	.25	.14			
1.2	1.22	1.14	1.04	.93	.81	.68	.61	.55	.48	.41	.34	.25	.14		
1.1	1.37	1.29	1.19	1.08	.96	.82	.75	.67	.58	.51	.43	.35	.25	.09	
1.0	1.53	1.44	1.35	1.24	1.13	.99	.91	.81	.71	.62	.52	.44	.34	.21	.06

Table 3.5-5 **Design values of e_{rb}/e_{rs}**

b) Associated with Figure 3.5-10																			
Ce_{rs}/e_{ps}	**ρ**																		
	4.0	**3.5**	**3.0**	**2.5**	**2.0**	**1.8**	**1.6**	**1.4**	**1.2**	**1.0**	**0.9**	**0.8**	**0.7**	**0.6**	**0.5**	**0.4**	**0.3**	**0.2**	**0.1**
4.0	.5	.38	.27	.14															
3.9	.53	.41	.30	.17															
3.8	.56	.44	.33	.21															
3.7	.59	.47	.36	.24	.00														
3.6	.63	.51	.38	.27	.09														
3.5	.67	.54	.42	.29	.14														
3.4	.72	.58	.45	.32	.17	.06													
3.3	.78	.62	.48	.35	.21	.12													
3.2	.86	.67	.52	.38	.24	.17	.02												
3.1	.96	.73	.56	.41	.27	.21	.10												
3.0	1.12	.80	.61	.45	.30	.24	.16												
2.9	1.42	.89	.66	.49	.33	.27	.20	.07											
2.8	2.00	1.01	.72	.53	.36	.30	.24	.14											
2.7		1.20	.80	.58	.40	.33	.27	.19	.00										
2.6		1.52	.90	.64	.43	.36	.30	.23	.10										
2.5		2.00	1.04	.71	.47	.40	.33	.26	.16										
2.4			1.26	.80	.53	.44	.37	.30	.21	.04									
2.3			1.68	.91	.60	.49	.41	.33	.25	.12									
2.2			2.08	1.09	.66	.55	.45	.37	.29	.18	.10								
2.1				1.34	.76	.62	.50	.42	.33	.23	.17	.06							
2.0				1.68	.90	.72	.56	.46	.37	.27	.22	.15							
1.9				2.00	1.10	.84	.65	.52	.42	.32	.27	.20	.10						
1.8					1.36	1.02	.77	.60	.47	.37	.31	.26	.18	.04					
1.7					1.68	1.26	.96	.71	.54	.42	.36	.31	.24	.14	.00				
1.6					2.00	1.58	1.22	.90	.65	.49	.42	.36	.30	.22	.11				
1.5						1.90	1.54	1.16	.82	.59	.50	.42	.36	.29	.20	.07			
1.4						2.33	1.90	1.50	1.09	.73	.61	.51	.43	.36	.27	.17	.02		
1.3							2.15	1.80	1.46	.94	.78	.65	.53	.43	.34	.25	.12		
1.2								2.13	1.80	1.30	1.06	.85	.66	.52	.42	.33	.22	.08	
1.1									2.09	1.72	1.38	1.16	.86	.69	.52	.42	.31	.17	.0
1.0										2.12	1.69	1.44	1.16	.93	.67	.50	.39	.25	.10

Table 3.5-5 **Design values of e_{rb}/e_{rs}**

c) Associated with Figure 3.5-11																			
Ce_{rs}/e_{ps}	ρ																		
	4.0	**3.5**	**3.0**	**2.5**	**2.0**	**1.8**	**1.6**	**1.4**	**1.2**	**1.0**	**0.9**	**0.8**	**0.7**	**0.6**	**0.5**	**0.4**	**0.3**	**0.2**	**0.1**
3.1	.61	.49	.36	.24	.04														
3.0	.66	.54	.40	.28	.10														
2.9	.71	.58	.44	.31	.15	.07													
2.8	.76	.63	.49	.34	.19	.13													
2.7	.85	.69	.54	.38	.23	.17	.07												
2.6	1.02	.76	.59	.43	.27	.22	.13												
2.5	1.32	.85	.65	.48	.31	.26	.18	.05											
2.4	2.00	1.02	.72	.54	.35	.30	.22	.12											
2.3		1.30	.80	.60	.40	.34	.27	.18	.06										
2.2		1.78	.95	.68	.45	.39	.32	.24	.15										
2.1			1.20	.77	.50	.44	.38	.30	.21	.06									
2.0			1.56	.92	.56	.49	.44	.36	.27	.16	.05								
1.9			2.00	1.15	.64	.56	.49	.42	.33	.23	.14	.05							
1.8				1.46	.75	.63	.56	.48	.39	.28	.21	.14	.03						
1.7					.90	.74	.64	.54	.44	.33	.26	.21	.12						
1.6					1.16	.90	.74	.62	.50	.39	.32	.27	.20	.11					
1.5					1.44	1.12	.89	.72	.59	.46	.38	.33	.26	.20	.07				
1.4					1.90	1.40	1.09	.84	.69	.55	.46	.40	.33	.27	.17	.03			
1.3						1.71	1.34	1.02	.84	.67	.57	.50	.41	.34	.26	.15			
1.2							1.64	1.26	1.05	.85	.72	.61	.51	.42	.34	.26	.13		
1.1							2.00	1.59	1.33	1.11	.95	.79	.65	.52	.43	.35	.25	.09	
1.0								1.96	1.68	1.42	1.22	1.04	.82	.66	.54	.45	.34	.22	.10

Table 3.5-6 **Values of Ce_{rs}/e_{ps} for Figure 3.5-9, Figure 3.5-10 and Figure 3.5-11 when $e_{rb}/e_{rs} = 0$**

ρ	Ce_{rs}/e_{ps}		
	Figure 3.5-9	**Figure 3.5-10**	**Figure 3.5-11**
0.1	1.04	1.10	1.05
0.2	1.16	1.27	1.16
0.3	1.29	1.42	1.29
0.4	1.41	1.56	1.42
0.5	1.51	1.70	1.56
0.6	1.63	1.83	1.70
0.7	1.74	1.98	1.82
0.8	1.87	2.14	1.94
0.9	1.99	2.28	2.04
1.0	2.10	2.44	2.15
1.2	2.32	2.70	2.36
1.4	2.54	2.95	2.54
1.6	2.76	3.22	2.76
1.8	2.94	3.46	2.94
2.0	3.14	3.70	3.14

3.5.4.3.4 Limits on reinforcement

The reinforced shell and nozzle thicknesses shall satisfy the following.

a) For reinforcements provided as in Figure 3.5-15, Figure 3.5-16, Figure 3.5-17, Figure 3.5-19, Figure 3.5-20 or Figure 3.5-21, e_{ab} shall not be reduced to a thickness less than e_{rb}, within a distance L_b measured from the relevant surface of the shell of thickness e_{rs}, where:

$L_b = \sqrt{de_{rb}}$

b) For reinforcements provided as in Figure 3.5-14, Figure 3.5-17, Figure 3.5-18, Figure 3.5-21 or Figure 3.5-26, e_{as} shall not be reduced to a thickness less than e_{rs}, within a distance L_s measured from the outer surface of the nozzle of thickness e_{rb} where $L_s = \sqrt{De_{rs}}$ or $d/2$ whichever is the smaller.

c) For reinforcements provided as in Figure 3.5-16 or Figure 3.5-20, if the actual length of the internal protrusion in the corroded condition is less than L_b then the reinforcement shall satisfy the requirements of d) ii); otherwise the nozzle shall be treated as a flush nozzle.

d) It is permissible to modify the distribution of reinforcement so as to concentrate the material close to the opening, as shown in Figure 3.5-29 and Figure 3.5-30, provided the following requirements are met.

   i) For flush nozzles, the cross-section area of the nozzle on each side of the nozzle centre line falling within distance L_b shall not be less than $L_b e_{rb}$.

   ii) For protruding nozzles, the cross-section area of the nozzle on each side of the nozzle centre line falling within distance L_b shall not be less than $2L_b e_{rb}$ and the reinforcement shall be approximately equally disposed about the shell mid-thickness. The cross-section area of the internal protrusion shall take account of the corrosion allowance on the inside and outside diameter and on the protrusion length.

   iii) For shell thickening, half the total cross-section area taken between the outermost extremity of the dimension L_s on one side of the nozzle and a similar point on the opposite side of the nozzle, but excluding any area included in i) or ii) shall not be less than $e_{rs}(L_s + e_{rb})$.

   For these calculations L_b and L_s shall be established as in a) and b) before local thickening of the nozzle and shell respectively.

   *NOTE For forged nozzle inserts, the procedure in **3.5.4.3.5** includes the necessary thickening correction factor.*

e) The nozzle thickness e_{rb} (except for studded pads, see **3.5.4.8**) shall not exceed $(2 - d/D)e_{rs}$. For local thickening, as permitted in d), e_{rb} and e_{rs} shall be taken as the modified thickness at the opening.

f) The transitions between sections of shell or between sections of nozzle or nozzle connections of different thicknesses shall be achieved by means of a smooth taper. The requirements of **3.10.2** shall apply in the case of shell sections.

3.5.4.3.5 Rim reinforcements and set-in nozzle forgings

Reinforcement for smooth profiled reinforcement shall be derived from the following procedures.

a) Protruding rim (see Figure 3.5-22).

   i) Using Figure 3.5-9 and the procedure given in **3.5.4.3.3**, determine e_{rb} and e_{rs}.

   ii) Calculate L_b and L_s from **3.5.4.3.4**a) and **3.5.4.3.4**b).

iii) One half of the total cross-section area falling within the outermost extremities of L_b and L_s shall not be less than:

$[2L_b e_{rb} + e_{rs}(L_s + e_{rb})](f_s/f_b)$.

iv) The area of the internal protrusion shall take account of the corrosion allowance on the inside and outside diameter and on the protrusion length.

b) Flush rim (see Figure 3.5-23).

i) Using Figure 3.5-10, Figure 3.5-11 or Figure 3.5-12 and the procedure given in **3.5.4.3.3**, determine e_{rb} and e_{rs}.

ii) Calculate L_b and L_s from **3.5.4.3.4**a) and **3.5.4.3.4**b).

iii) One half of the total cross-section area falling within the outermost extremities of L_b and L_s shall not be less than:

$[L_b e_{rb} + e_{rs}(L_s + e_{rb})](f_s/f_b)$.

NOTE The cross-sectional area of the rim required in these derivations will vary depending on the particular combination of e_{rs} and e_{rb}. A trial procedure using different combinations of e_{rs} and e_{rb} may be employed to establish the minimum area required. For the protruding rim, the cross-sectional area should be equally disposed about the shell mid-thickness.

3.5.4.3.6 Obround, elliptical openings and oblique nozzles

Non-circular openings and oblique nozzles shall meet the same requirements as circular openings and nozzles normal to the shell except for the following.

a) For cylindrical and conical shells, dimension d shall be measured along the opening axis which is parallel to the centre line axis of the shell.

b) For shells other than cylinders or cones, dimension d shall be measured along the major axis of the opening.

c) In determining dimensions L_b and L_s, the value of d shall be as given in a) or b).

d) In the case of multiple openings the value of d_A used in **3.5.4.4.2** shall be determined using d as given in a) or b).

3.5.4.3.7 Dissimilar materials

The design procedure normally assumes the use of similar materials in the nozzle and shell but dissimilar materials may be used provided that the design strength of the nozzle, f_b, is within the range $0.6f_s$ to $1.5f_s$. The following shall apply.

a) If $0.6f_s < f_b < f_s$ then the mean nozzle diameter calculated in **3.5.4.3.3** stage f) shall be based on the outside diameter of the nozzle and the final value of e_{rb} calculated in **3.5.4.3.3** shall be increased in the ratio of (f_s/f_b). Having thus determined e_{rb} it is not necessary to carry out any further reiteration in the procedures of **3.5.4.3.3** or **3.5.4.3.4**. The selected nozzle analysis thickness e_{ab} shall not be less than the increased thickness e_{rb}.

b) If $f_b > f_s$, there shall be no reduction in e_{rb}.

Figure 3.5-13 **Nozzle in a conical shell**

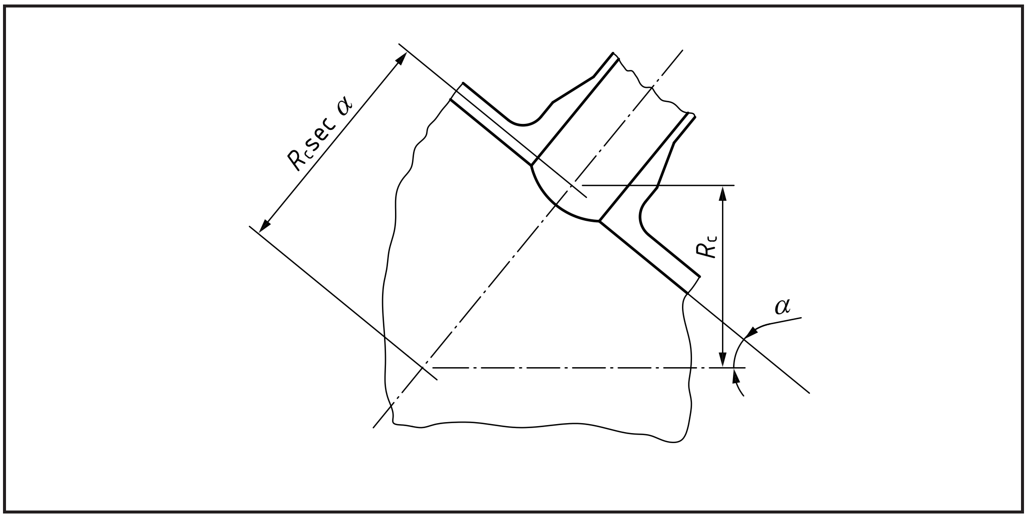


Figure 3.5-14 **Notation applicable to spheres**

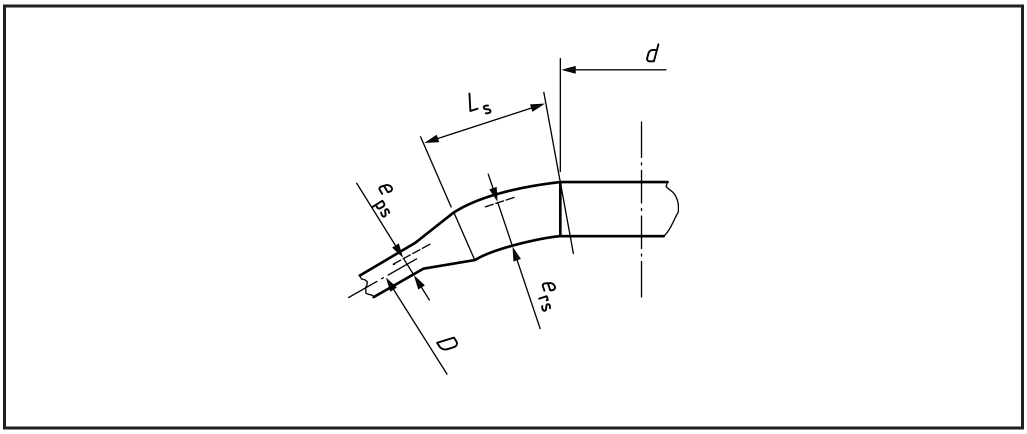


Figure 3.5-15 **Notation applicable to spheres**

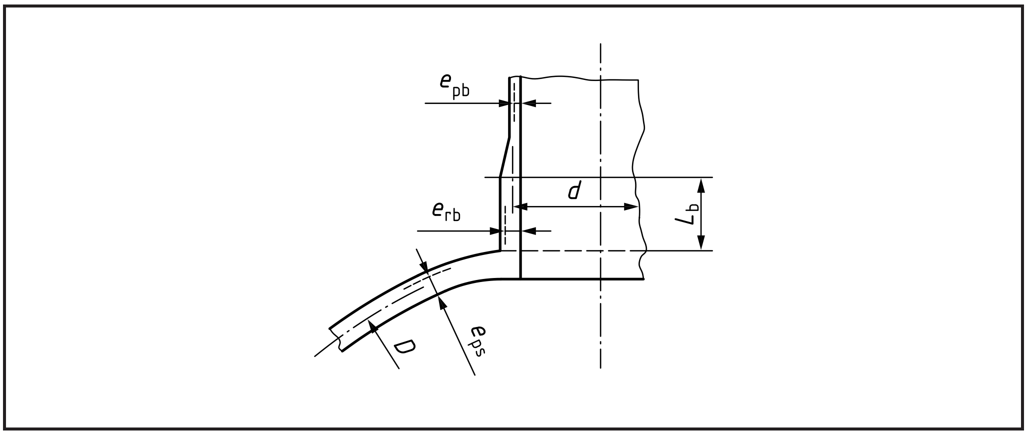


Figure 3.5-15a **Notation applicable to oblique nozzles in spheres**

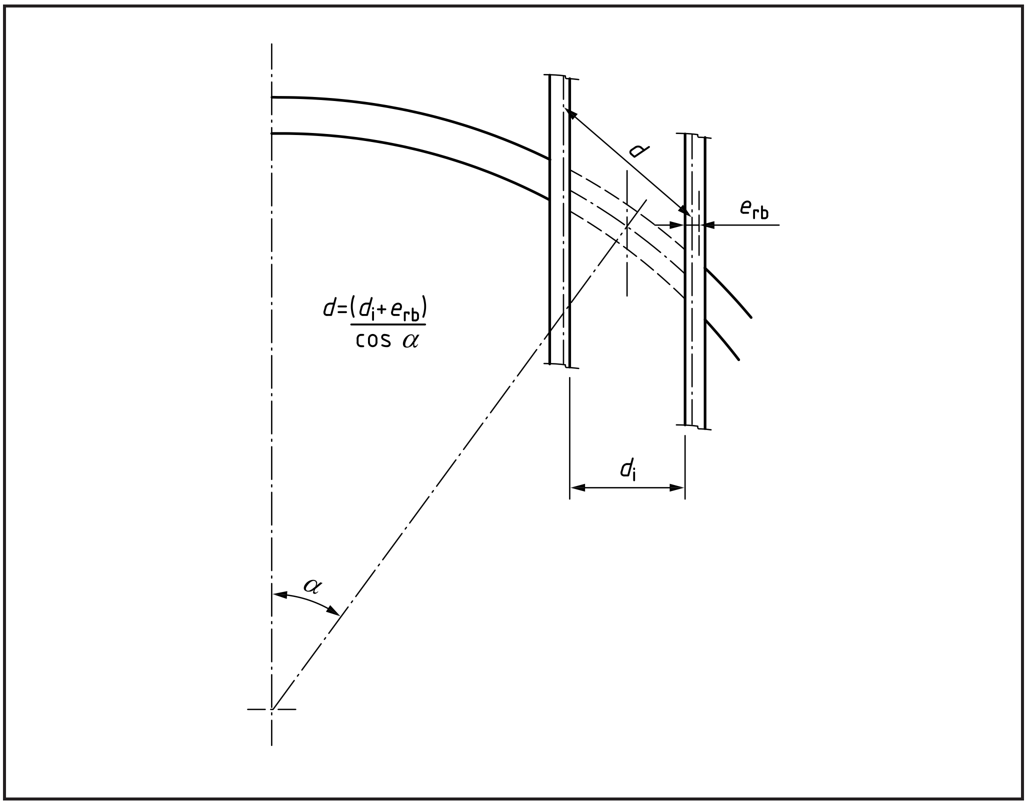


Figure 3.5-16 **Notation applicable to spheres**

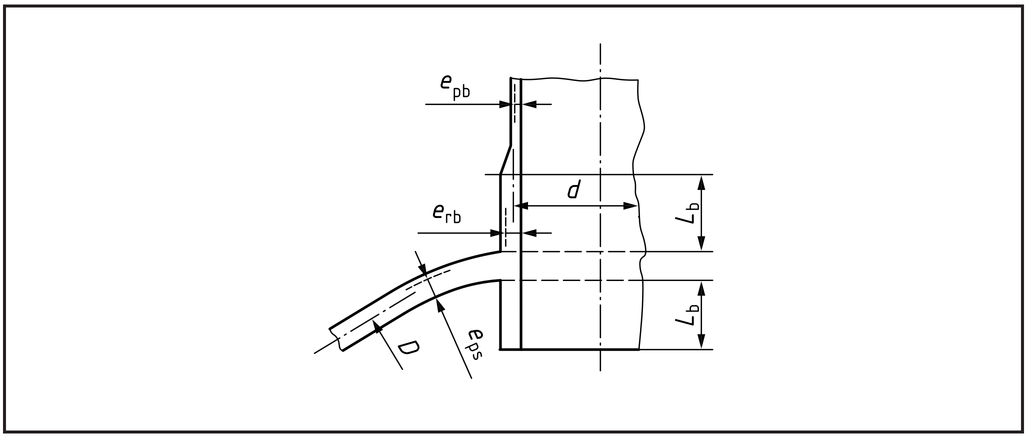


Figure 3.5-17 **Notation applicable to spheres**

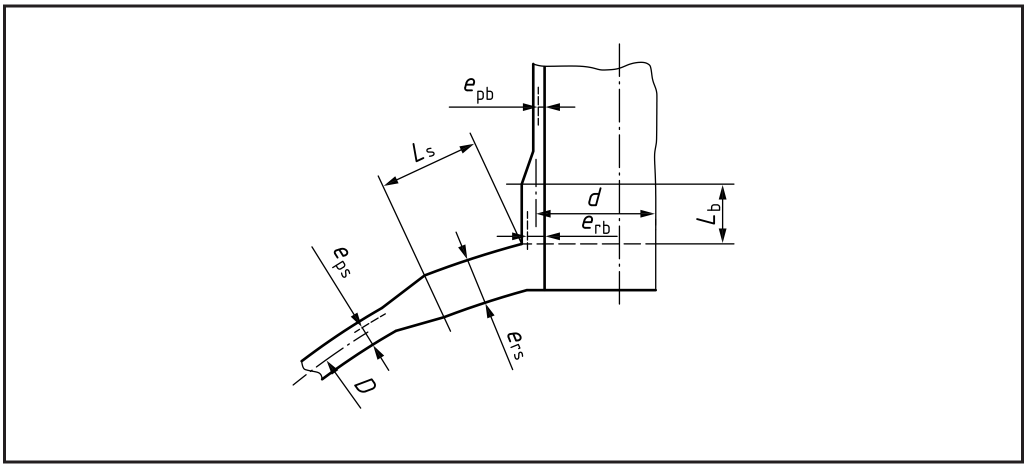


Figure 3.5-18 **Notation applicable to cylinders**

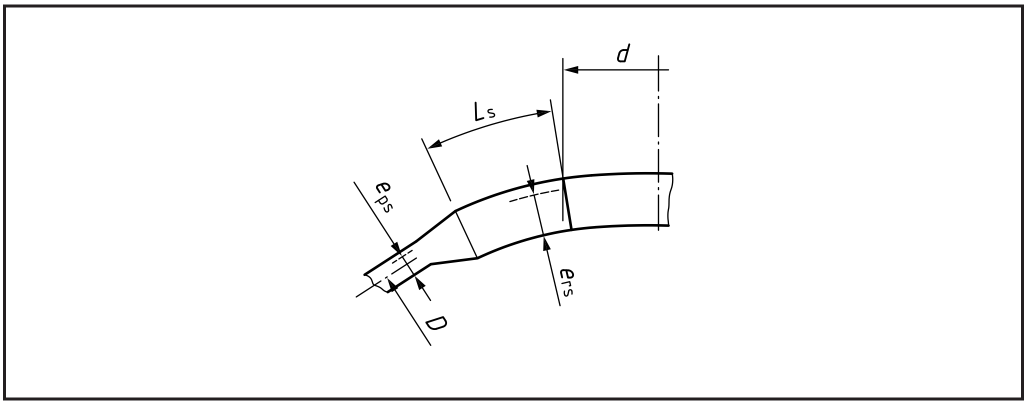


Figure 3.5-19 **Notation applicable to cylinders**

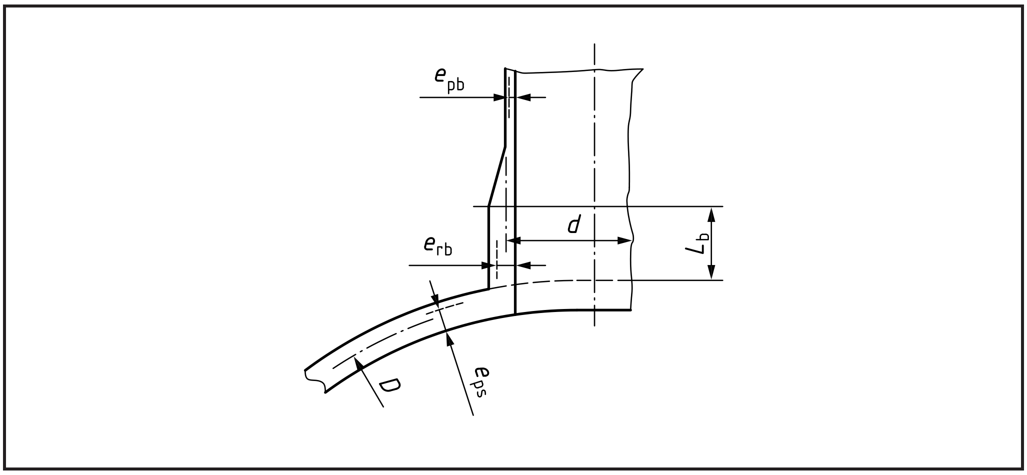


Figure 3.5-20 **Notation applicable to cylinders**

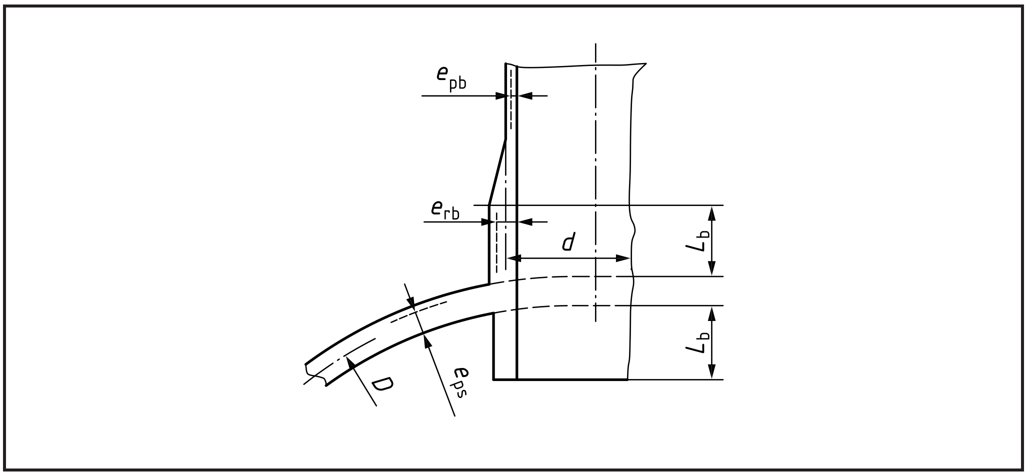


Figure 3.5-21 **Notation applicable to cylinders**

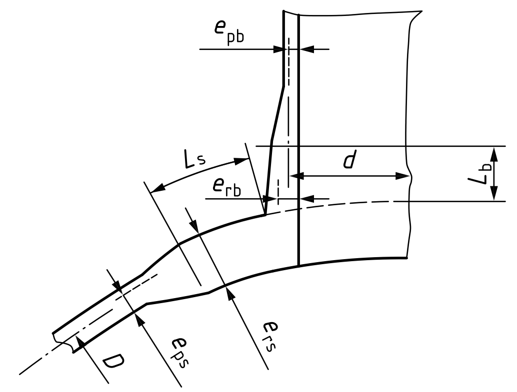
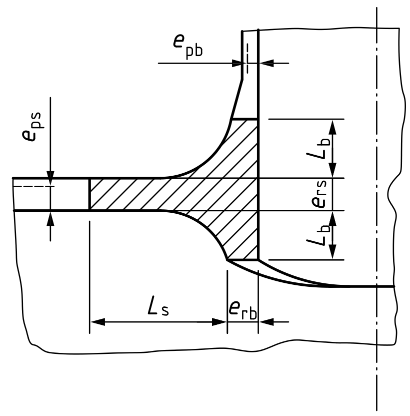
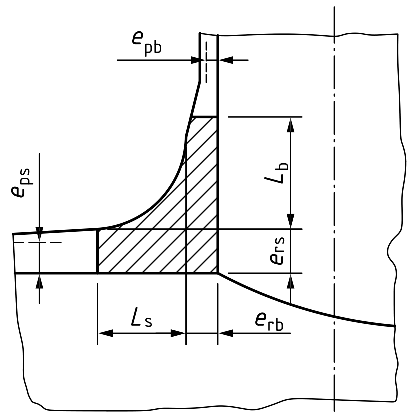


Figure 3.5-22 **Protruding rim**



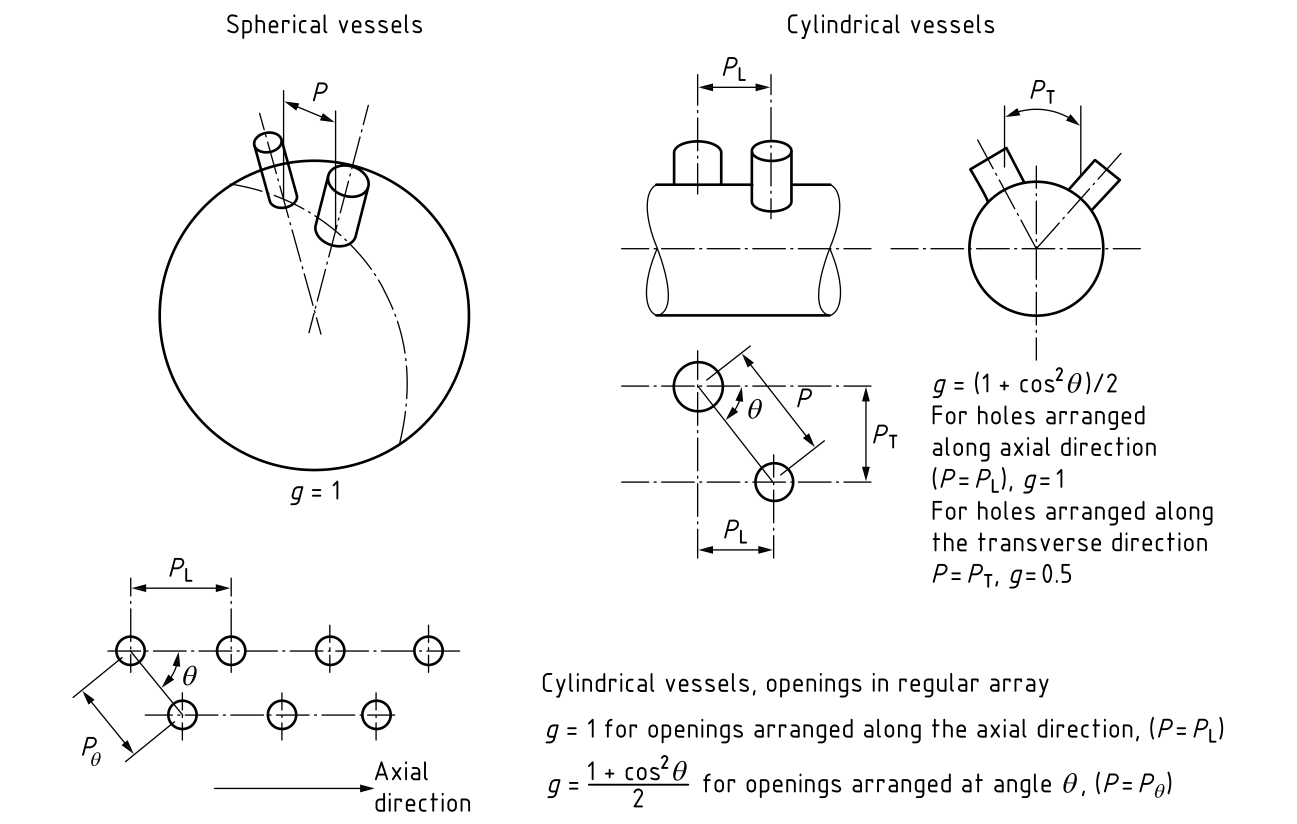
The dimensions e_{rs}, L_s, e_{rb} and L_b refer to the design of a nozzle connection using components of constant thickness from which the design of the profiled rim is derived on an area basis.

Figure 3.5-23 **Flush rim**



The dimensions e_{rs}, L_s, e_{rb} and L_b refer to the design of a nozzle connection using components of constant thickness from *w* the design of the profiled rim is derived on an area basis.

Figure 3.5-24 **Arrangement factor *g***



NOTE For elliptical and oblique nozzles the value of d is the dimension of the opening in the direction of the relevant pitch.

Figure 3.5-25 **Nozzle compensation**

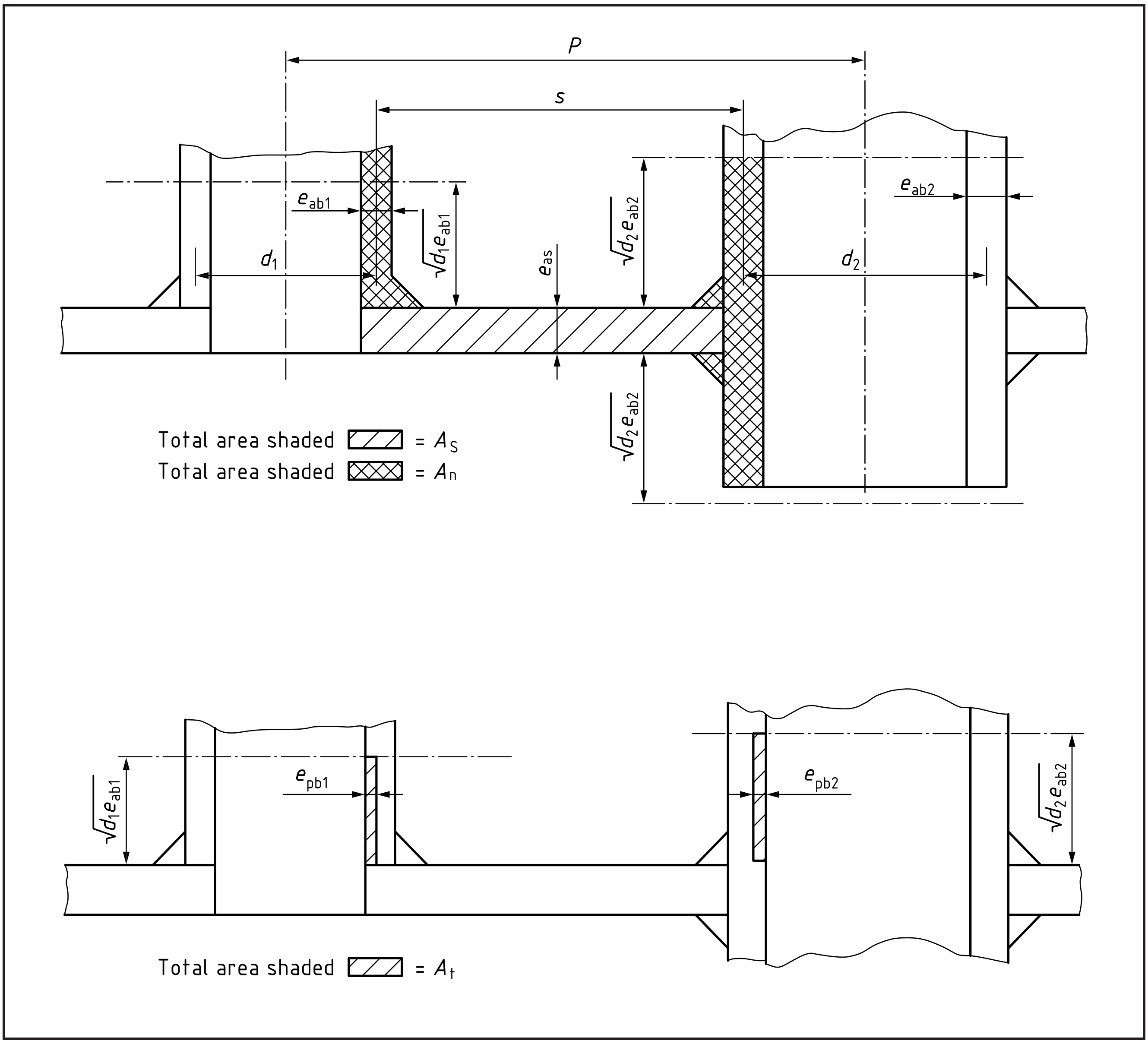


Figure 3.5-26 **Notation applicable to spheres and cylinders**

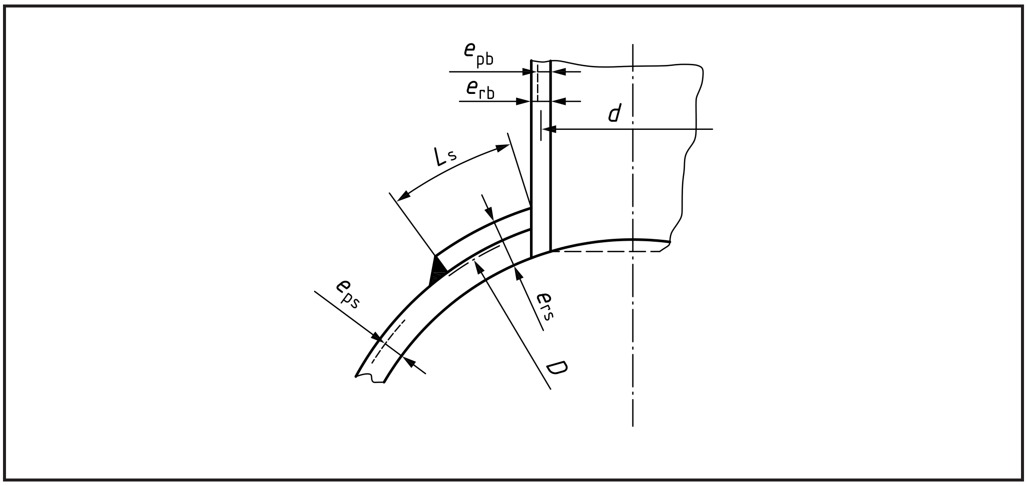


Figure 3.5-27 **Notation applicable to spheres and cylinders**

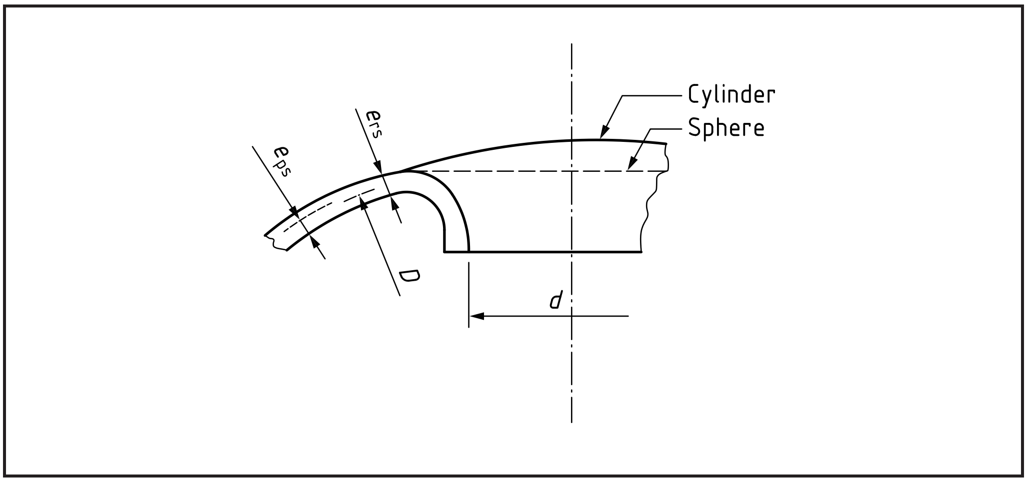


Figure 3.5-28 **Notation applicable to spheres and cylinders**

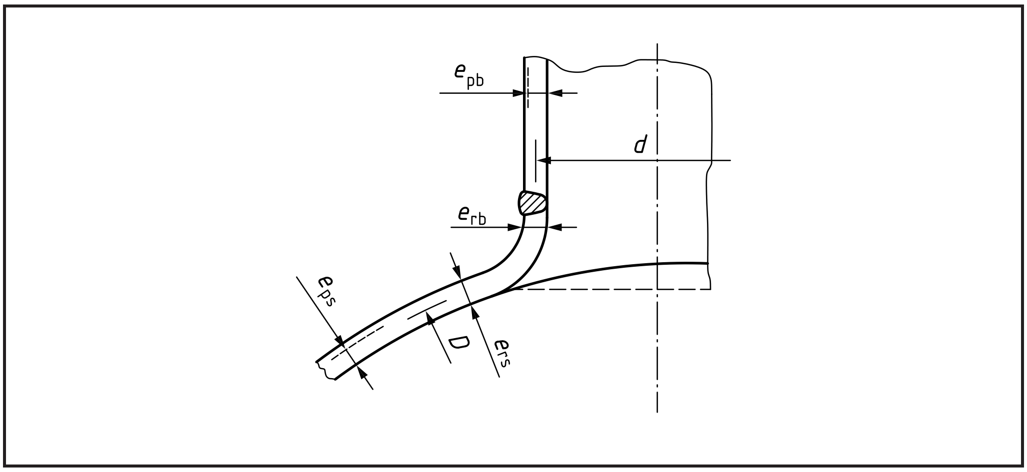
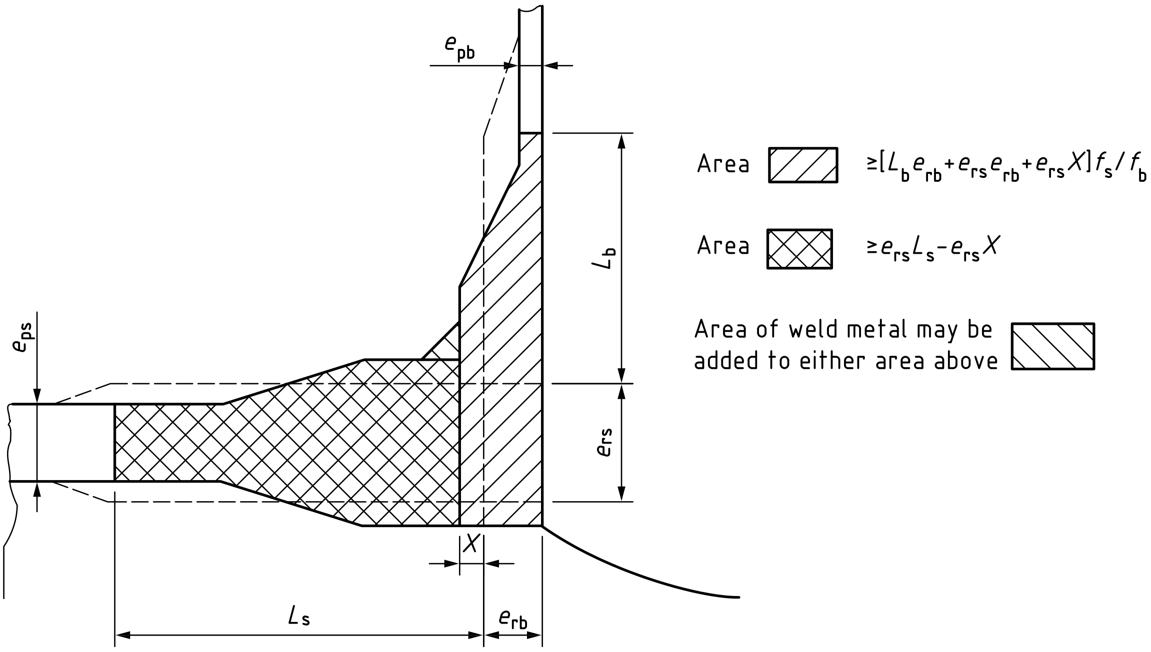
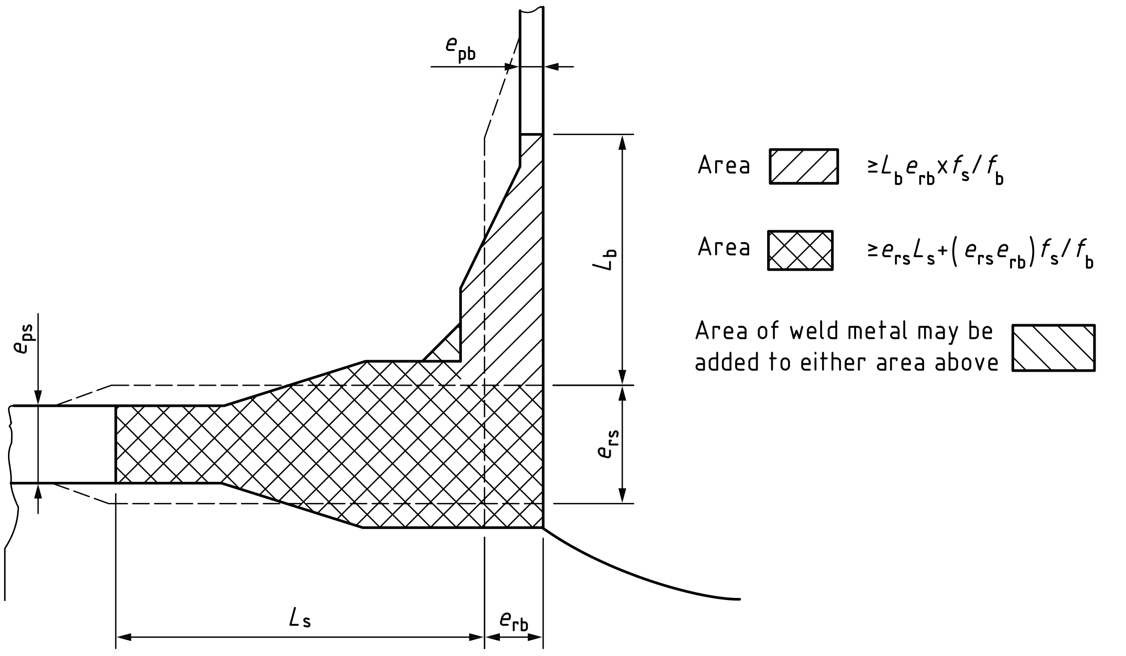


Figure 3.5-29 **Modified flush nozzle compensation**

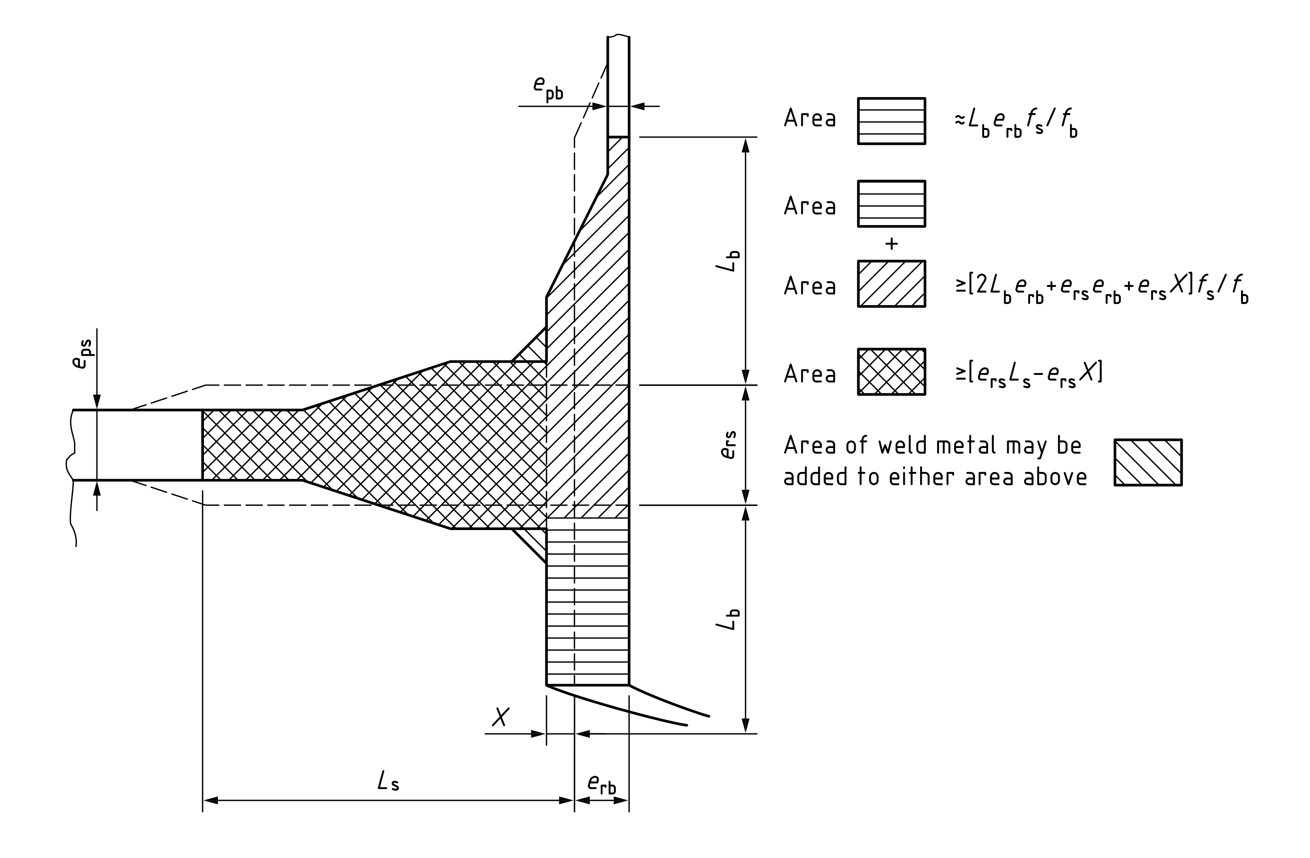


a) Set-in



b) Set-on

Figure 3.5-30 **Modified protruding nozzle compensation**



3.5.4.4 Design of groups of openings and nozzle connections

3.5.4.4.1 General

When s is less than $3\sqrt{De_{ps}}$ the compensation available between the bores of adjacent openings shall be calculated and adjusted in accordance with **3.5.4.4.2**.

3.5.4.4.2 Calculation method

a) From Figure 3.5-24 calculate g, the arrangement factor, for the adjacent openings.

b) If $s \leq 2\sqrt{De_{ps}}$ calculate the reinforcement as required in d) below, using $K_2 = 1.0$.

c) If $2\sqrt{De_{ps}} < s < 3\sqrt{De_{ps}}$ calculate the reinforcement of adjacent nozzles using **3.5.4.3**, assuming that they are located such that $s = 3\sqrt{De_{ps}}$.

Using Figure 3.5-25 calculate area A'_s using the minimum reinforced shell thickness e_{rs}; calculate area A'_n using the minimum reinforced thicknesses e_{rb1} and e_{rb2} and calculate area A'_t using the minimum unreinforced nozzle thicknesses for pressure loading only, e_{pb1} and e_{pb2}. Calculate K_1 and K_2 from Equations (3.5.4-5) and (3.5.4-6) then calculate the reinforcement as required in d) below.

$$K_1 = \left[\frac{A'_s + A'_n - A'_t}{ge_{ps}(d_A + 3\sqrt{De_{ps}})}\right] \quad (3.5.4\text{-}5)$$

$$K_2 = 1 - (1 - K_1)\frac{(s - 2\sqrt{De_{ps}})}{\sqrt{De_{ps}}} \quad (3.5.4\text{-}6)$$

d) Using Figure 3.5-25 calculate area A_s using the analysis thickness of the reinforced shell e_{as}, calculate area A_n using the analysis thicknesses of the reinforced nozzles e_{ab1} and e_{ab2}. The area of the internal protrusions, if any, shall take account of the corrosion allowance on the inside and outside diameter and on the protrusion length. Calculate area A_t using the minimum unreinforced nozzle thicknesses for pressure loading only, e_{pb1} and e_{pb2}. Any additional material from reinforcing pads should be similarly introduced. When the nozzle or pad material is weaker than the shell material any relevant area shall be reduced in the ratio of the design stresses. The reinforcement shall be increased, if required, such that for each ligament:

$$A_s + A_n - A_t \geq K_2 gPe_{ps} \quad (3.5.4\text{-}7)$$

e) The thickness of the nozzle, e_{ab}, shall be less than $(2 - d/D)e_{as}$.

f) The transitions between sections of shell or between sections of nozzle or nozzle connections of different thickness shall be achieved by means of a smooth taper. The requirements of **3.10.2** shall apply in the case of shell sections.

3.5.4.5 Reinforcing pads

3.5.4.5.1 Pressure considerations

It is permissible for the requirements in **3.5.4.3** and **3.5.4.4** for the design of integral reinforcement to be used for reinforcement of penetrations or openings incorporating pads, doubling plates, or studded, socket welded and screwed connections (see **3.5.4.8**) but all of the following conditions shall be observed.

a) The *d/D* ratio shall not be greater than:

one-third for double-sided pads;

one-quarter for single-sided pads.

b) The width of the pad shall not be less than $L_s/2$.

c) The thickness of a pad shall not exceed 40 mm or the as-built shell thickness, whichever is the lesser.

d) The thickness of the pad shall not be less than $e_{ps}/4$.

The amount of compensation to be provided shall be equal to the amount which would have been had the compensation been integral.

The design of reinforcing pads for nozzles where one or more of the criteria in a) to d) are not satisfied shall be checked by an alternative design method as permitted in **3.2.2**b).

3.5.4.5.2 Non-pressure considerations

Conditions a) to d) of **3.5.4.5.1** do not apply to reinforcing pads which are used to limit the local stresses due to mechanical loads on nozzles, supports or mounting. However, the maximum thickness of a pad which can be counted as effective reinforcement of a nozzle for pressure loading shall be limited to the value given in **3.5.4.5.1**c).

If the thickness of the reinforcing pad is greater than the vessel shell thickness, its size ($d_\phi \times d_x$ in Figure G.3.1-2) shall be such that the design leg length of the attachment welds to the vessel shell does not exceed the vessel thickness (see **G.2**).

3.5.4.5.3 General

Reinforcing pads are permitted to have one ventilation hole which shall remain open during welding and/or post-weld heat treatment.

Reinforcing pads shall not be used under conditions where severe corrosion/oxidation is possible or where there is the possibility of severe temperature gradients occurring, in service, across the thickness of the shell.

3.5.4.6 Vessels subject to external pressure

3.5.4.6.1 Compensation of openings in single-walled vessels subject to external pressure shall be designed in accordance with the requirements for vessels subject to internal pressure specified in **3.5.4.1**, **3.5.4.2**, **3.5.4.3**, **3.5.4.4** and **3.5.4.5**, or in accordance with the alternative design method specified in **3.5.4.9**, using an internal design pressure equal to the external design pressure.

3.5.4.6.2 Compensation of openings in each shell of a double-walled vessel shall conform to **3.5.4.6.1** for the shell, subject to external pressure, and with the requirement for vessels subject to internal pressure, specified in **3.5.4.1** and **3.5.4.2**, irrespective of whether there is a common nozzle connection rigidly attached to both shells or not.

3.5.4.7 Nozzle and nozzle pipe minimum thicknesses

Nozzle pipes shall be designed to satisfy the following criteria.

a) They shall be able to withstand design pressure. For this purpose the minimum thickness of a nozzle pipe shall be calculated in accordance with **3.5.1** for cylindrical shells.

b) They shall be able to withstand superimposed loading by connected pipework or fittings, whether operational or non-operational. Notwithstanding the required minimum thickness specified in a) or the requirements specified in **3.5.4.3**, the nozzle, and its connection to the vessel, shall either be analysed to demonstrate the ability to withstand all relevant loads or the nominal thickness of a nozzle intended for connection to external piping shall be not less than the smaller of:

   1) the value given in Table 3.5-4 increased by the amount of any required corrosion and any manufacturing tolerances; or

   2) the nominal thickness of the main portion of the vessel shell. The thickness of the nozzle need not be increased if the nominal thickness of the shell is increased for plate availability reasons.

c) They shall be suitable for the recommended forms of nozzle to shell attachment welds (see Annex E).

*NOTE In the case of stainless steel pressure vessels for the food industry where the thickness to meet **3.5.4.7**a) is less than 2.5 mm, and where nozzle connections are formed by belling out as typified in Figure E.22, the minimum nozzle thickness may be less than the thickness of the vessel, provided that the compensation requirements of **3.5.4** are satisfied, and that the minimum thickness is not less than the minimum thickness specified in BS 4825 for stainless steel pipes and fittings for the food industry.*

3.5.4.8 Studded, socket welded and screwed connections

Construction of studded, socket welded and screwed connections shall be in accordance with Figure E.29 and Figure E.30. Where required by **3.5.4** appropriate reinforcement shall be incorporated.

The thread forms on which joints are to be made shall conform to BS EN 10266-1 or BS 21 unless otherwise specified (see **3.2.2**). Such joints shall not exceed the 1½ thread size designation and shall not have a design temperature exceeding 260 °C unless taper/taper thread joints are used. Taper/taper thread joints are permitted for the sizes and design pressures given in Table 3.5-7 as long as the design temperature does not exceed 495 °C and provided the pressure/temperature ratings in the appropriate component standards, e.g. BS 3799:1974, are not exceeded.

If parallel threads are used, a collar and a facing around the hole shall be arranged to provide a joint face. If welded sockets are used they shall conform to BS 3799. The maximum diameter of holes tapped in plates shall not exceed the thickness of the plate before addition of the corrosion allowance.

Stud holes shall straddle the centre line of the vessel where practicable and shall be tapped to a depth of not less than the diameter of the stud plus 3 mm. There shall be a minimum of 6 mm of metal between the bottom of the stud hole and the pressure retaining surface of the vessel before the addition of the corrosion allowance.

The corroded thickness of a studded connection shall be not less than the largest of the following:

a) e_{rs}, the minimum thickness required for compensation (see **3.5.4.3.2**);

b) t, the minimum thickness required for a flange (see **3.8.3**); and

c) the minimum thickness as given above for tapped holes.

Table 3.5-7 **Allowable taper/taper thread sizes**

Nominal pipe size mm	**Maximum design pressure N/mm^2**	**Minimum pipe thickness mm**
8	10.35	3.02
10	10.35	3.20
15	10.35	3.75
20	10.35	3.91
25	8.3	4.55
32	4.15	4.85
40	4.15	5.08
50	4.15	5.54
65	2.75	7.01
80	2.75	7.62

3.5.4.9 Alternative "pressure area" design method

3.5.4.9.1 General

The use of this design method is limited to the compensation of openings that conform to the geometric limitations specified in **3.5.4.9.2**.

NOTE This method has extensive satisfactory use in European codes of practice and has been adopted in BS EN 13445-3.

For the purposes of **3.5.4.9** the following symbols, as shown in Figure 3.5-32 and Figure 3.5-33, are additional to those defined in **3.5.4.1**. All dimensions are in the corroded condition, unless otherwise indicated (see **3.1.5**).

A_{fb} cross-sectional area of branch within the compensation limits;

A_{fs} cross-sectional area of shell within the compensation limits;

A_{fp} cross-sectional area of pad or compensation plate within the compensation limits;

A_{fw} cross-sectional area of fillet weld between nozzle, pad or compensation plate and shell, within the compensation limits [see **3.5.4.9.3**f) and Figures 3.5-32 and 3.5-33];

A_p pressure loaded area (as shown in Figure 3.5-32 and Figure 3.5-33, calculated using internal dimensions);

d_i inside diameter of opening or branch;

D_i inside diameter of cylindrical or spherical shell or straight cylindrical flange of dished end;

d_o outside diameter of branch;

e_{ap} analysis thickness of pad or compensation plate (see **1.6**); [see Equations (3.5.4-16), (3.5.4-18) and (3.5.4-21)];

f_b the lower of the design stress of the branch and f_s ;

f_p the lower of the design stress of the pad or compensation plate and f_s ;

f_w the lower of the design stresses of the components being joined by the fillet weld;

h_i inside height of an ellipsoidal dished end;

L_{bi} length of internal branch considered as effective compensation, measured from the inside surface of the main body (ignoring an additional compensation plate) [see Equation (3.5.4-20)];

L_e is the distance from inside surface of shell or dished end to outside surface of pad [see Figure 3.5-32e) and Figure 3.5-32f)];

L_p maximum length of pad or compensation plate considered to be effective as compensation, measured along the material centre line from the edge of the opening or outside of the branch [see Equations (3.5.4-15) and (3.5.4-17)];

r_{ih} inside radius of spherical shell, hemispherical end or spherical portion of torispherical end;

r_{is} inside radius of shell, as specified in **3.5.4.9.2**b);

ψ angle (≤ 50°) between the branch axis and a line normal to the main body wall in an oblique branch connection.

3.5.4.9.2 Application

The design method only applies to cylindrical shells, spherical shells and dished ends having circular or elliptical openings, where the following assumptions and conditions are satisfied.

a) The geometry of the openings or branches and main shell shall fall within the following limits:

1) cylindrical shells,

$$\frac{d_i}{D_i} \leq 1 \qquad (3.5.4\text{-}8)$$

2) spherical shells and dished ends,

$$\frac{d_i}{D_i} \leq 0.6 \qquad (3.5.4\text{-}9)$$

3) the ratio of branch thickness to main body thickness e_{ab}/e_{as} shall conform to the limits of Figure 3.5-31.

b) The distance between openings, branches or pads, measured from the edge of the opening or outside of the branches and pads shall be not less than $2L_s$, where:

$$L_s = \sqrt{(2r_{is} + e_{as})e_{as}} \qquad (3.5.4\text{-}10)$$

The values to be used for r_{is} are:

1) for cylindrical shells,

$$r_{is} = D_i/2 \qquad (3.5.4\text{-}11)$$

2) for spherical shells and hemispherical or torispherical ends,

$$r_{is} = r_{ih} \qquad (3.5.4\text{-}12)$$

3) for semi-elliptical ends,

$$r_{is} = D_i\left(\frac{0.22D_i}{h_i} + 0.02\right) \qquad (3.5.4\text{-}13)$$

Where the distance between openings is less than $2L_s$, the requirements of **3.5.4.9.3**j) shall apply.

c) Openings and branches in dished ends shall be located in accordance with the limits illustrated in Figure 3.5-8.

d) Cylindrical shells, spherical shells and ends with openings shall be reinforced where necessary. The reinforcement of the main body can be obtained by the following measures:

1) by an increased wall thickness of the main body compared with that of the shell without openings [see Figure 3.5-32a) and Figure 3.5-32b)];

2) by set-on welded compensation plates [see Figure 3.5-32c) and Figure 3.5-32d)];

3) by set-in welded pads [see Figure 3.5-32e) and Figure 3.5-32f)];

4) by set-on or set-in welded branches [see Figure 3.5-32g) and Figure 3.5-32 h)];

5) by combinations of the above mentioned measures [see Figure 3.5-32i) and Figure 3.5-32j)].

e) The reinforcement area of the main body with openings cannot be calculated directly but shall be assumed in the first instance. That assumption shall be verified by means of the method specified in **3.5.4.9.3**. The applied method is based on basic pressure thicknesses derived from Equation (3.5.1-1) for cylindrical shells and from Equation (3.5.1-3) for spherical shells and spherical sections of dished ends respectively and leads to relationships between a pressure loaded area A_p and a stress loaded cross-sectional area which is the sum of A_{fs}, A_{fp} , A_{fb} and A_{fw} (see Figure 3.5-32). The calculation may need to be repeated using a corrected assumption of the reinforcement area.

f) Where necessary, sufficient reinforcement shall be provided in all planes through the axis of the opening or branch.

g) In the case of elliptical openings, the ratio between the major and the minor axis shall not exceed 1.5. For design purposes, the diameter of elliptical openings in cylindrical shells shall be taken as the opening axis parallel to the longitudinal axis of the cylinder. For elliptical openings in spherical shells and dished ends the diameter shall be taken as the major axis of the elliptical opening.

h) Expanded branches shall not be considered as reinforcement and shall be calculated in accordance with **3.5.4.9.3**a). Set-on or set-in branches may be considered as reinforcement provided that the attachment weld dimensions conform to Annex E.

i) Reinforcement of openings by compensation plates is permitted, subject to the limitations specified in **3.5.4.9.3**c). However, the effective width of such plates shall be calculated taking only the main shell thickness, not the combined thickness.

3.5.4.9.3 Design requirements

a) Fundamental criteria

All openings shall satisfy the following general relationship:

$$p\left[A_P + 0.5\left(A_{fs} + A_{fb} + A_{fp} + A_{fw}\right)\right] \leq f_s A_{fs} + f_p A_{fp} + f_b A_{fb} + f_w A_{fw} \quad (3.5.4\text{-}14)$$

NOTE Simple formulae for calculation of A_p, A_{fs}, A_{fp} and A_{fb} for various geometries are given below the diagrams in Figure 3.5-32 and Figure 3.5-33. These formulae are considered to give acceptable results within the accuracy of the method. However, if so desired, the designer may calculate more precise values based on the true geometry.

b) Reinforcement by increased wall thickness

Where reinforcement is attained by an increased wall thickness of the main body, compared with that of the shell without openings, this wall thickness shall exist for no less than a distance L_s [see Equation (3.5.4-10)] measured from the edge of the opening, as shown in Figure 3.5-32a) and Figure 3.5-32b).

c) Reinforcement by compensation plates

Compensation plates shall only be considered as contributing to the reinforcement if the inside diameter of the opening or branch d_i does not exceed the inside radius of the shell r_{is}, as defined in **3.5.4.9.2**b).

Such plates shall:

i) have close contact with the main body;

ii) be of similar material to the main body they are welded on to. No credit shall be taken for stronger material in calculating compensation areas; and

iii) have a nominal thickness no greater than 1.5 times the nominal thickness of the main body.

The width of compensation plates L_p, considered as contributing to the reinforcement, shall not exceed L_s, as shown in Figure 3.5-32c) and Figure 3.5-32d).

$$L_p \leq L_s \quad (3.5.4\text{-}15)$$

The value of e_{ap} used in the determination of A_{fp} in Equation (3.5.4-14) shall not exceed e_{as}.

$$e_{ap} \leq e_{as} \quad (3.5.4\text{-}16)$$

d) Reinforcement by pads

Only pads of set-in welded type in accordance with Figure 3.5-32e) and Figure 3.5-32f) shall be used. The width of the pads L_p, considered as contributing to the reinforcement, shall not exceed L_s.

$$L_p \leq L_s \quad (3.5.4\text{-}17)$$

The value of e_{ap} used in the determination of A_{fp} in Equation (3.5.4-14) shall not exceed e_{as}.

$$e_{ap} \leq 2e_{as} \quad (3.5.4\text{-}18)$$

e) Reinforcement by branches

Branch pipes shall meet the requirements of **3.5.4.7** and **3.5.4.9.3**g), **3.5.4.9.3**h) or **3.5.4.9.3**i), as applicable.

f) Fillet weld areas

A_{fw} is the area of any welds connecting together the different components (shell to nozzle, compensation plate to nozzle, shell to reinforcing pad or shell to compensation plate) which is located within the length L_s on the shell [see Equation (3.5.4-10)] and lengths L_b and L_{bi} on the nozzle [see Equations (3.5.4-19) and (3.5.4-20)]. Areas of welds already included in other areas, e.g. A_{fs}, A_{fp} or A_{fb}, shall be omitted from A_{fw}.

g) Branch connections normal to the vessel wall

For branch connections normal to the vessel wall, the areas A_p, A_{fs}, A_{fb}, A_{fp} and A_{fw} shall be determined in accordance with Figure 3.5-32g) to Figure 3.5-32l), where the lengths contributing to the reinforcement shall be not more than L_s, for the shell [see Equation (3.5.4-10)], and L_b for the branch.

$$L_b = \sqrt{(d_o - e_{ab})e_{ab}} \tag{3.5.4-19}$$

The maximum value to be used in the calculation of the part extending inside, if any, in the case of set-through branches [see Figure 3.5-32h), Figure 3.5-32i) and Figure 3.5-32j)] and intruded branches [see Figure 3.5-32l)] shall be

$$L_{bi} = 0.5L_b \tag{3.5.4-20}$$

The area of the part extending inside shall take account of the corrosion allowance on the inside and outside diameter, on the protrusion length and on the fillet weld throat thickness.

The dimensions of any compensation plate to be used in the calculation shall be

$$e_{ap} \leq 2e_{as} \text{ and } L_p \leq L_s \tag{3.5.4-21}$$

h) Oblique branch connection in cylindrical shells

For branches on cylindrical shells lying in a plane perpendicular to the longitudinal axis of the shell and having an angle ψ, not exceeding 50° to the normal, the higher stress may occur in the lateral section (Figure 3.5-33a partial view I) or in the longitudinal section [Figure 3.5-33a) partial view II]. Equation (3.5.4-14) shall apply to both cases with the areas A_p, A_{fs}, A_{fb}, A_{fp} and A_{fw} as shown in Figure 3.5-33a) partial views I and II, to be used in the calculation. In both cases, L_b shall be based on the diameter of the branch, (not the chord of the opening) using Equation (3.5.4-19).

Where branches on cylindrical shells lie in a radial plane and have an angle ψ, in the longitudinal direction, not exceeding 50° to the normal, as shown in Figure 3.5-33b), the reinforcement of the opening shall be calculated as for a normal connection in accordance with **3.5.4.9.3**g), except that the value of A_p shall be based on the major axis of the resultant opening whereas L_b shall be based on the diameter of the branch (not the chord of the opening), using Equation (3.5.4-19).

i) Oblique branch connection in spherical shells or dished ends

For branches in spherical shells, or dished ends, lying in a plane that contains the axis of the branch and the centre of the spherical shell, or dished end, having an angle ψ not exceeding 50°, as shown in Figure 3.5-33c), the reinforcement of the opening shall be calculated as for a normal connection in accordance with **3.5.4.9.3**g), except that the value of A_p shall be based on the major axis of the resultant opening whereas L_b shall be based on the diameter of the branch (not the chord of the opening), using Equation (3.5.4-19).

j) Openings and branches less than $2L_s$ apart

For groups of openings or branches in cylindrical or in spherical shells, not in a regular array, each pair of openings or branches shall satisfy Equation (3.5.4-22). Where the openings or branches are arranged in a regular array, pairs of openings or branches on each array axis shall satisfy Equation (3.5.4-22).

$$p[(gA_{p0} + A_{p1} + A_{p2}) + 0.5(A_{fs0} + A_{fb1} + A_{fb2} + A_{fp1} + A_{fp2} + A_{fw1} + A_{fw2})] \leq f_s A_{fs0} + f_b A_{fb1} + f_b A_{fb2} + f_p A_{fp1} + f_p A_{fp2} + f_w A_{fw1} + f_w A_{fw2} \tag{3.5.4-22}$$

where g, the arrangement factor for the adjacent openings, is obtained from Figure 3.5-24. The relevant pressure areas and material areas for cylindrical and spherical shells are obtained from Figure 3.5-32m) and Figure 3.5-32n).

The reinforcement of each opening or branch shall in addition be checked individually in accordance with the relevant requirement of **3.5.4.9.3**.

Figure 3.5-31 **Maximum branch to body thickness ratio**

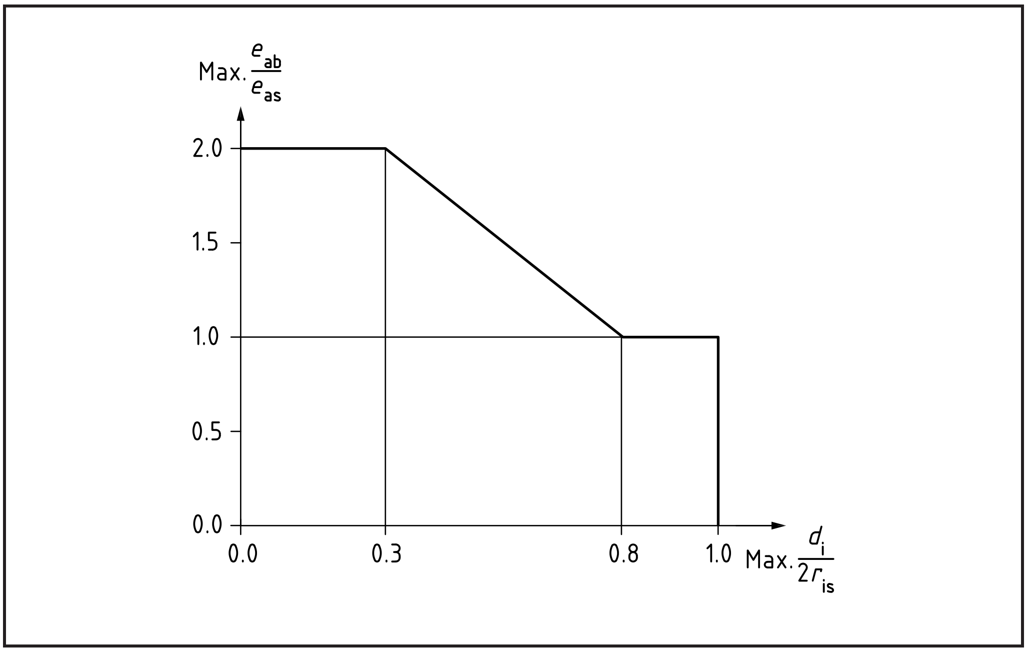
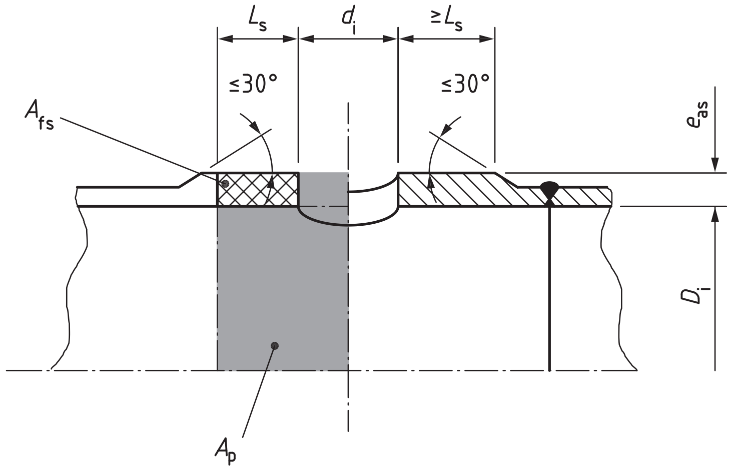


Figure 3.5-32 **Reinforcement of openings and branches**



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_i}{2}\right) + \frac{d_i e_{as}}{2}$$

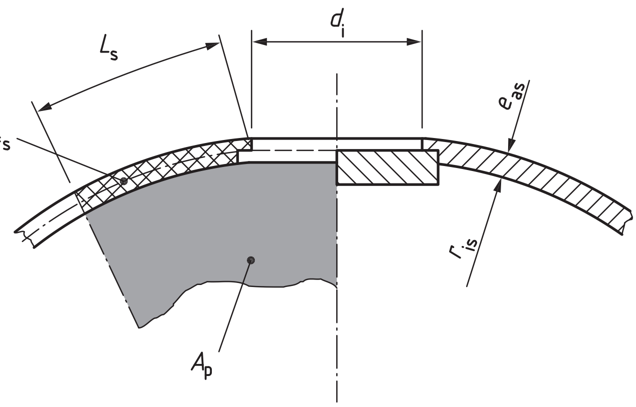
If the closure of the opening is located inside the shell, as shown in Figure 3.5-32 b), then the second term in this equation is omitted.

$$A_{fs} = e_{as} L_s$$

a) Cylindrical shells with isolated openings

Reinforcement by increased wall thickness.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_i}{2}\right) + \frac{d_i e_{as}}{2}$$

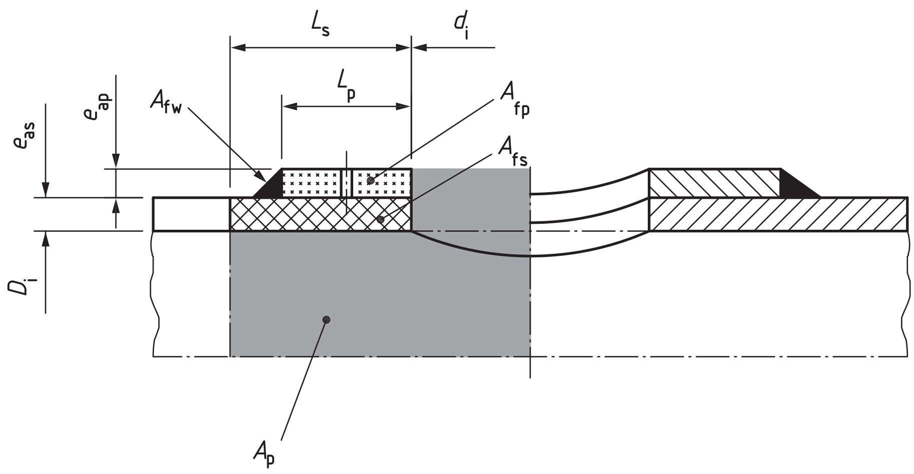
If the closure of the opening is located inside the shell or dished end, as shown above, then the second term in this equation is omitted.

$$A_{fs} = e_{as} L_s$$

b) Spherical shells and dished ends with isolated openings

Reinforcement by increased wall thickness.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_i}{2}\right) + \frac{d_i}{2}(e_{as} + e_{ap})$$

If the closure of the opening is located inside the shell, as shown in Figure 3.5-32 b), then the second term in this equation is omitted.

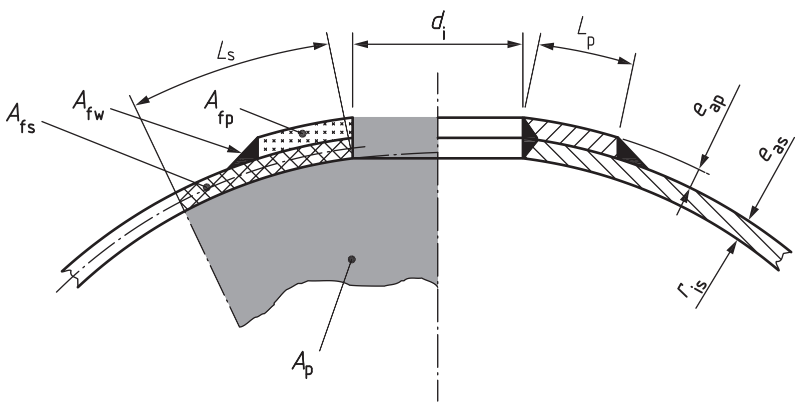
$$A_{fs} = e_{as}L_s$$

$$A_{fp} = e_{ap}L_p$$

c) Cylindrical shells with isolated openings

Reinforcement by compensating plates.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_i}{2}\right) + \frac{d_i}{2}(e_{as} + e_{ap})$$

If the closure of the opening is located inside the shell or dished end, as shown in Figure 3.5-32 b), then the second term in this equation is omitted.

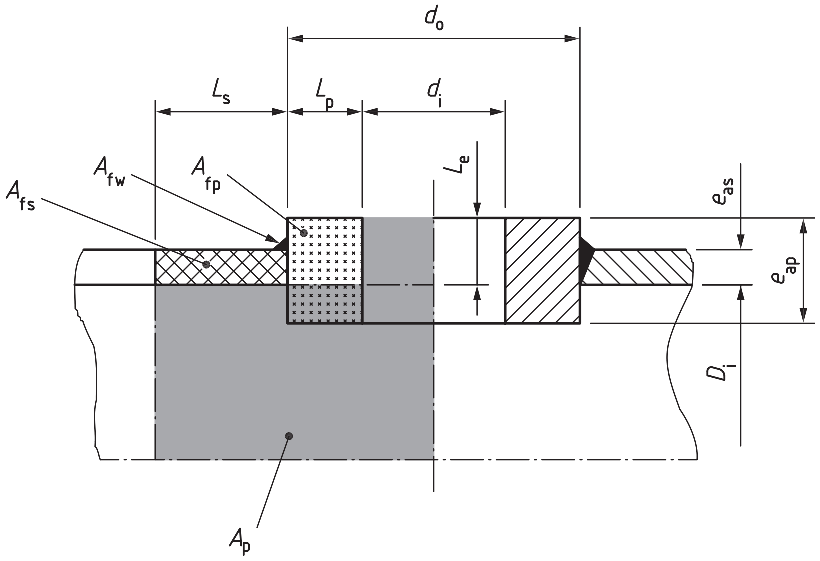
$$A_{fs} = e_{as}L_s$$

$$A_{fp} = e_{ap}L_p$$

d) Spherical shells and dished ends with isolated openings

Reinforcement by compensating plates.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i L_e}{2}$$

If the pad is fitted with an internal blind flange then the second term in this equation is omitted.

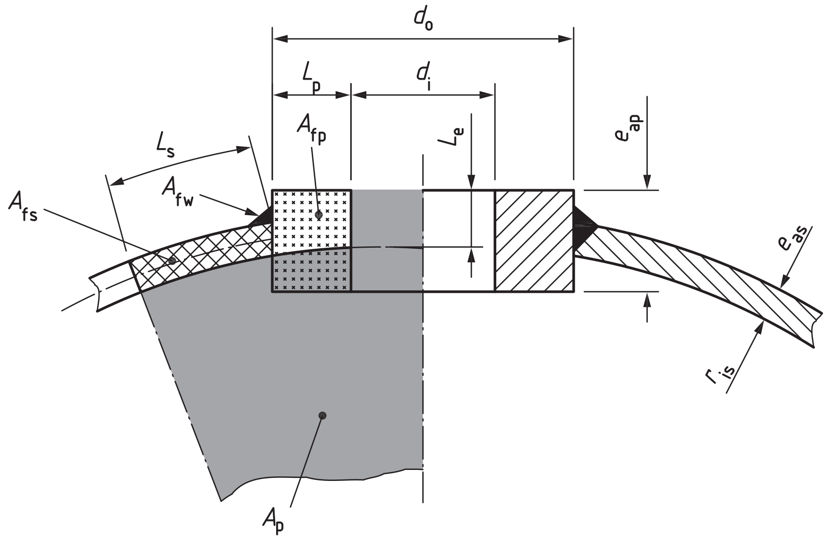
$$A_{fs} = e_{as} L_s$$

$$A_{fp} = e_{ap} L_p$$

e) Cylindrical shells with isolated openings

Reinforcement by pads.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i L_e}{2}$$

If the pad is fitted with an internal blind flange then the second term in this equation is omitted.

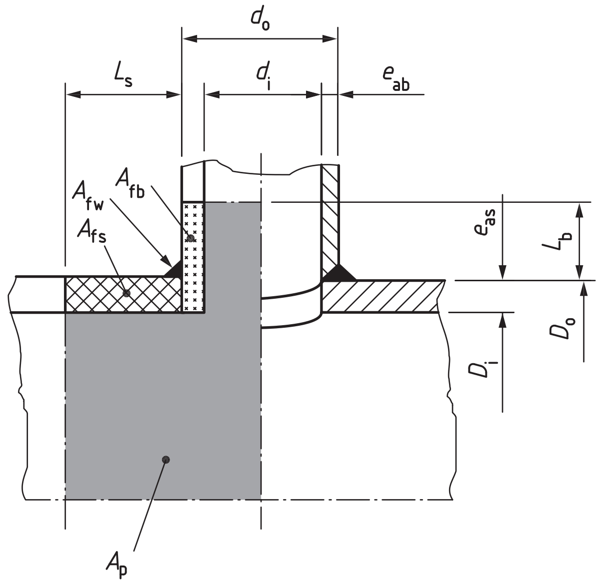
$$A_{fs} = e_{as} L_s$$

$$A_{fp} = e_{ap} L_p$$

f) Spherical shells and dished ends with isolated openings

Reinforcement by pads.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

$A_{fs} = e_{as}L_s$ (set-in)

$A_{fs} = e_{as}(L_s + e_{ab})$ (set-on)

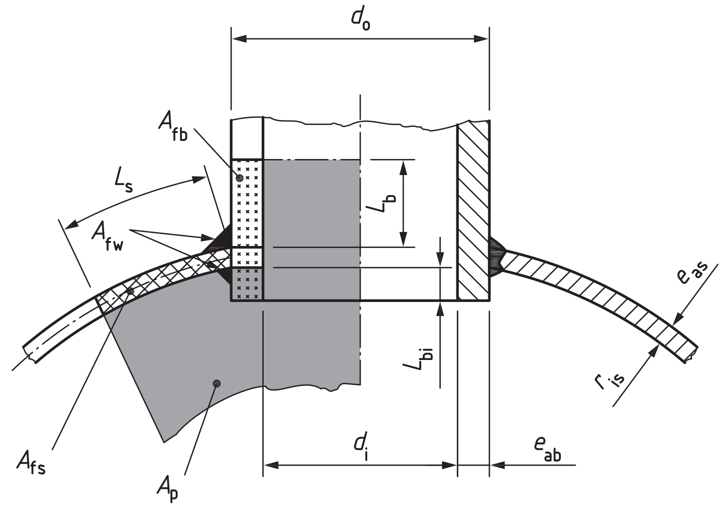
$A_{fb} = e_{ab}(L_b + e_{as})$ (set-in)

$A_{fb} = e_{ab}L_b$ (set-on)

g) Cylindrical shells with isolated openings

Reinforcement by branches.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

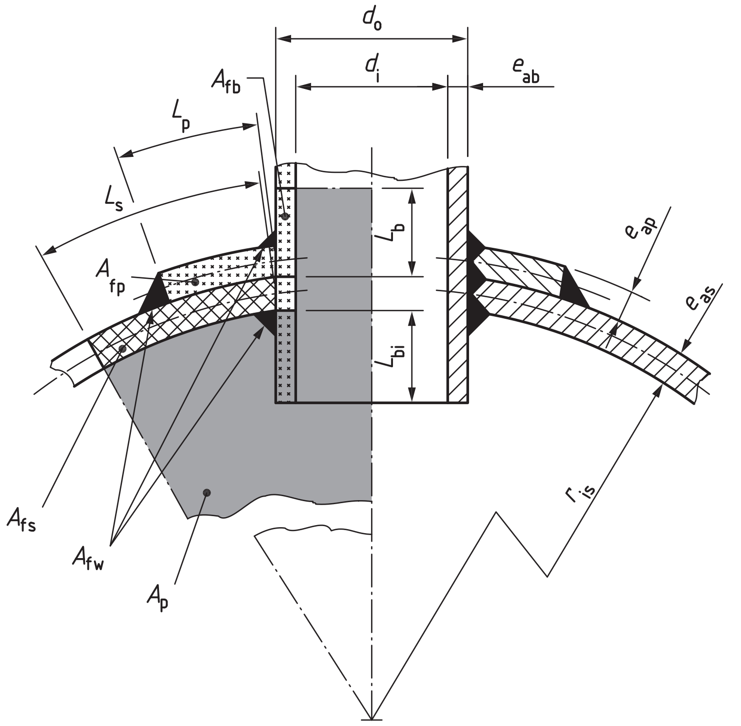
$$A_{fs} = e_{as}L_s$$

$$A_{fb} = e_{ab}(L_b + e_{as} + L_{bi})$$

h) Spherical shells and dished ends with isolated openings

Reinforcement by branches.

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

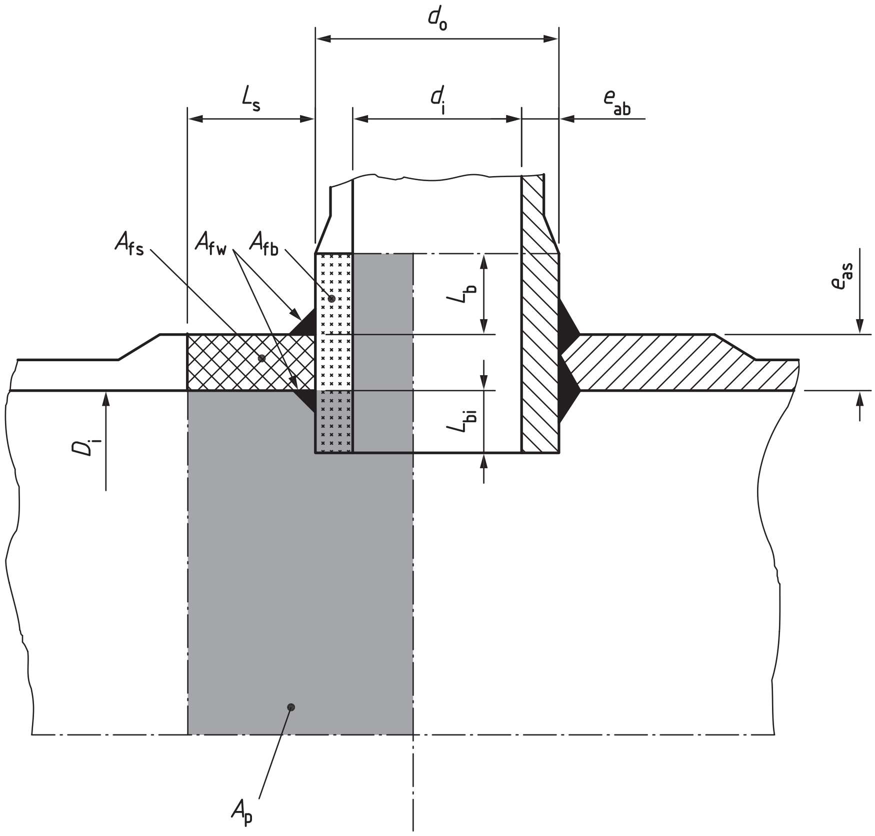
$$A_{fs} = e_{as}L_s$$

$$A_{fp} = e_{ap}L_p$$

$$A_{fb} = e_{ab}(L_b + e_{as} + L_{bi})$$

i) Combined reinforcement

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



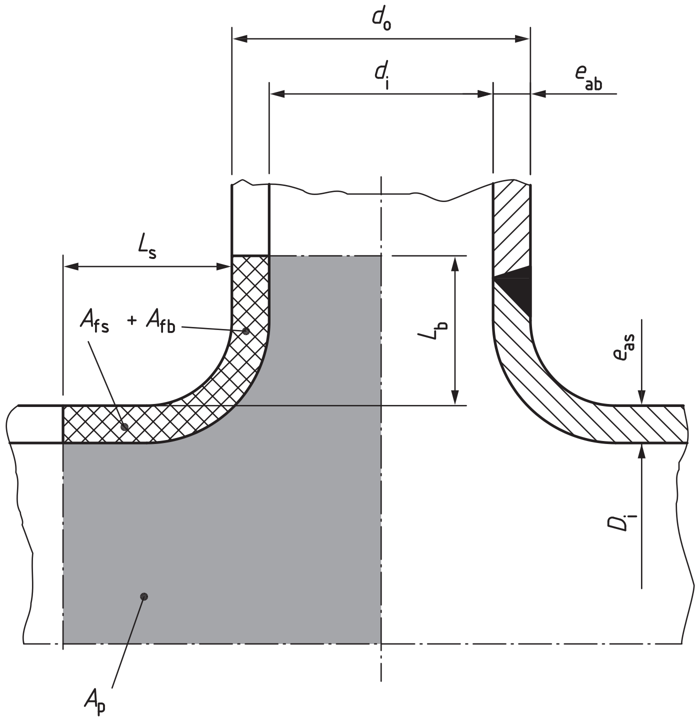
$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

$$A_{fs} = e_{as}L_s$$

$$A_{fb} = e_{ab}(L_b + e_{as} + L_{bi})$$

j) Combined reinforcement

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

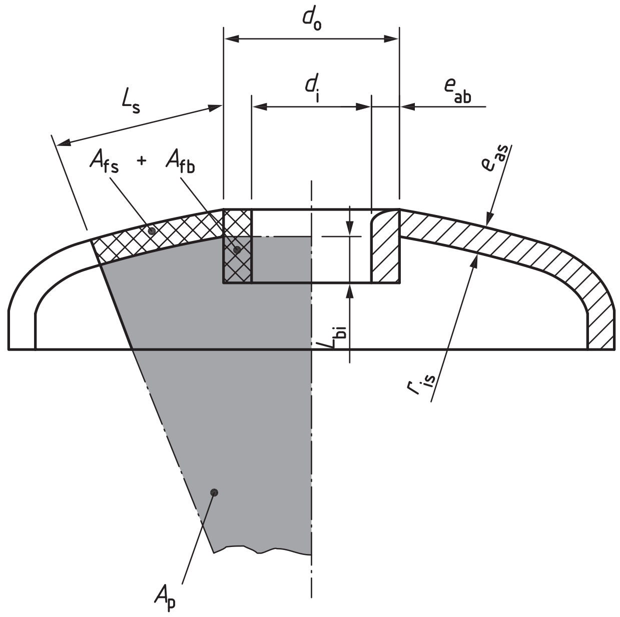
$$A_{fs} = e_{as}L_s$$

$$*\ A_{fb} = e_{ab}(L_b + e_{as})$$

*This formula shall be adjusted if weld joins branch of weaker material to the shell.

k) Extruded branch in a cylindrical shell

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



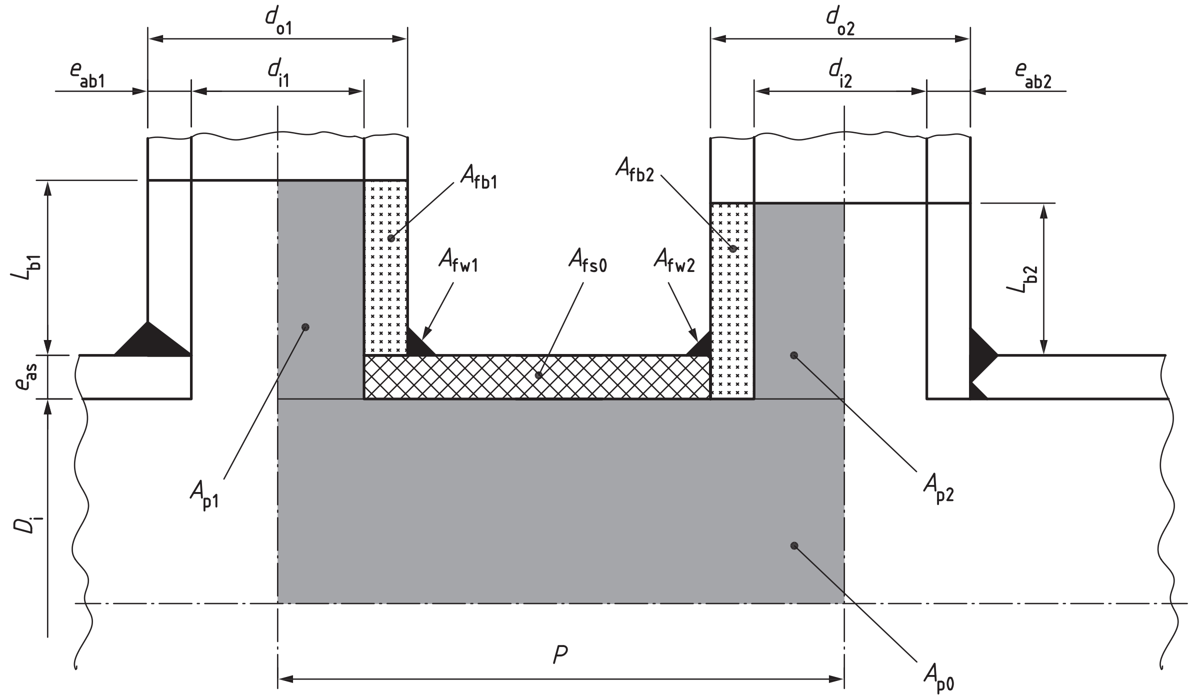
$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_o}{2}\right)$$

$$A_{fs} = e_{as}L_s$$

$$A_{fb} = e_{ab}(e_{as} + L_{bi})$$

l) Intruded branch in a dished end

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



The pitch of the opening P is measured at the shell mid-surface.

$A_{p0} = 0.5PD_i^2/\ (D_i + e_{as}\sin\theta)$

$A_{p1} = 0.5d_{i1}(L_{b1} + e_{as})$

$A_{p2} = 0.5d_{i2}(L_{b2} + e_{as})$

$A_{fs0} = e_{as}(P - 0.5d_{o1} - 0.5d_{o2})$ (for set-in) or $= e_{as}(P - 0.5d_{i1} - 0.5d_{i2})$ (for set-on)

$A_{fb1} = e_{ab1}(L_{b1} + e_{as})$ (for set-in) or $= e_{ab1}L_{b1}$ (for set-on)

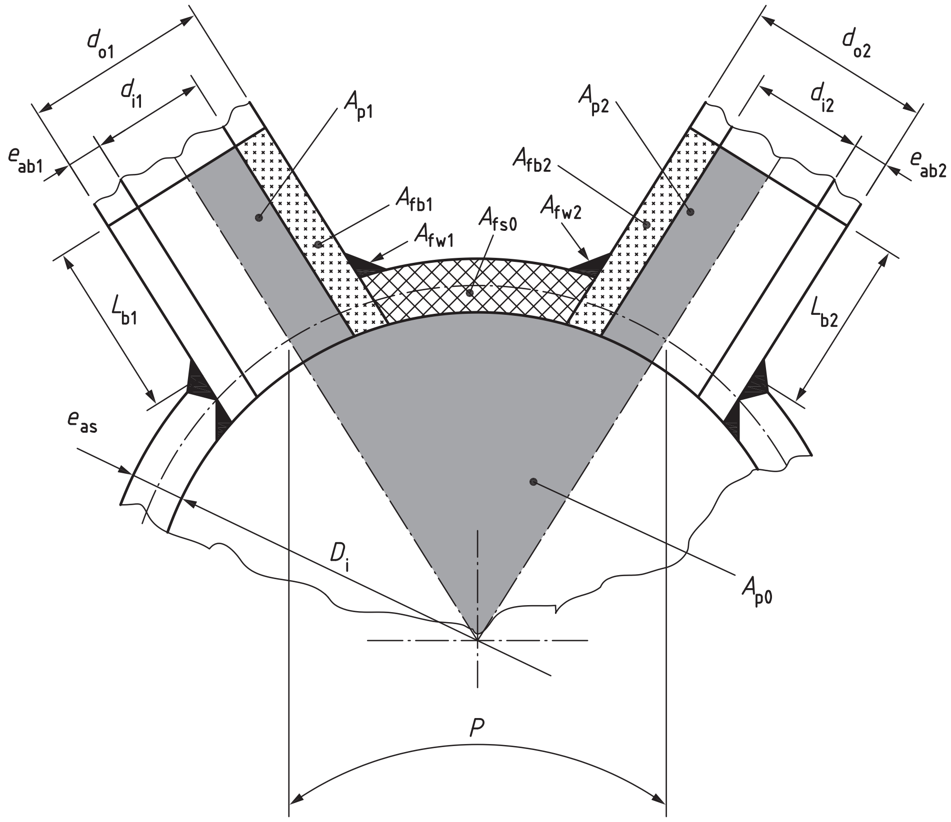
$A_{fb2} = e_{ab2}(L_{b2} + e_{as})$ (for set-in) or $= e_{ab2}L_{b2}$ (for set-on)

A_{fp1} and A_{fb2} for reinforcing plates, if applicable, shall be derived from Figure 3.5-32i).

NOTE For elliptical and oblique nozzles the value of d is the dimension of the opening in the direction of the relevant pitch.

m) Openings and branches less than $2L_s$ apart, in cylindrical shells

Figure 3.5-32 **Reinforcement of openings and branches *(continued)***



The pitch of the opening P is measured at the shell mid-surface.

$A_{p0} = 0.25PD_i^2/(D_i + e_{as})$

$A_{p1} = 0.5d_{i1}(L_{b1} + e_{as})$

$A_{p2} = 0.5d_{i2}(L_{b2} + e_{as})$

$A_{fs0} = e_{as}(P - 0.5d_{o1} - 0.5d_{o2})$ (for set-in) or $= e_{as}(P - 0.5d_{i1} - 0.5d_{i2})$ (for set-on)

$A_{fb1} = e_{ab1}(L_{b1} + e_{as})$ (for set-in) or $= e_{ab1}L_{b1}$ (for set-on)

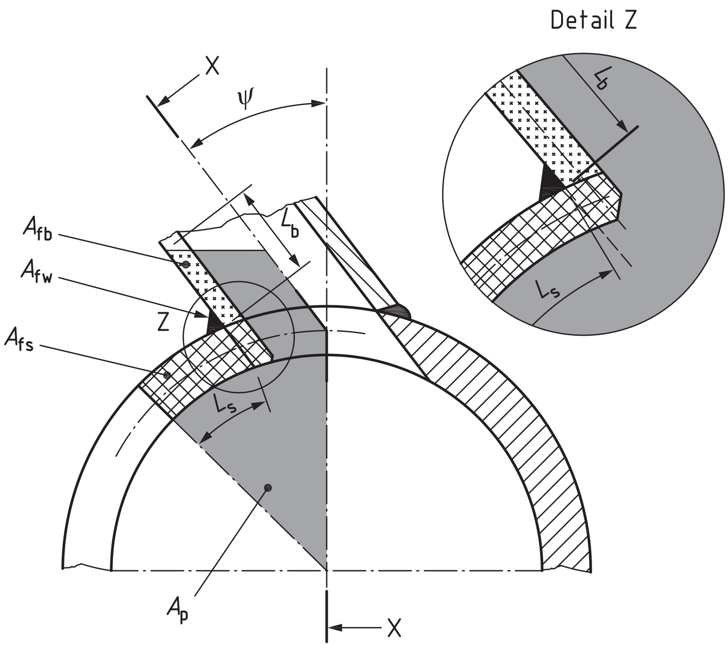
$A_{fb2} = e_{ab2}(L_{b2} + e_{as})$ (for set-in) or $= e_{ab2}L_{b2}$ (for set-on)

A_{fp1} and A_{fb2} for reinforcing plates, if applicable, shall be derived from Figure 3.5-32i).

NOTE For elliptical and oblique nozzles the value of d is the dimension of the opening in the direction of the relevant pitch.

n) Openings and branches less than $2L_s$ apart, in spherical shells

Figure 3.5-33 **Reinforcement of non-radial branches**



$$A_p = \frac{D_i}{4}\left(L_s + \frac{d_o}{2\cos\psi}\right) + \frac{d_i}{2}\left(L_b + \frac{e_{as}}{\cos\psi}\right)$$

$A_{fs} = e_{as}L_s$ (set-in)

$A_{fs} = e_{as}\left(L_s + \dfrac{e_{ab}}{\cos\psi}\right)$ (set-on)

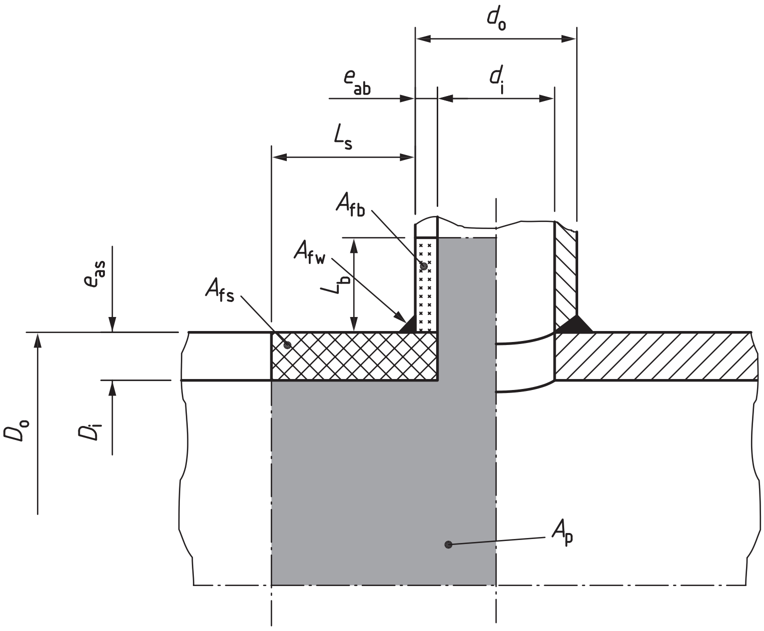
$A_{fb} = e_{ab}\left(L_b + \dfrac{e_{as}}{\cos\psi}\right)$ (set-in)

$A_{fb} = e_{ab}L_b$ (set-on)

Partial view I

a) Cylindrical shell with a branch not radially arranged (off centre)

Figure 3.5-33 **Reinforcement of non-radial branches *(continued)***



$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2}\right) + \frac{d_i}{2}\ (L_b + e_{as})$$

$A_{fs} = e_{as} L_s$ (set-in)

$A_{fs} = e_{as}(L_s + e_{ab})$ (set-on)

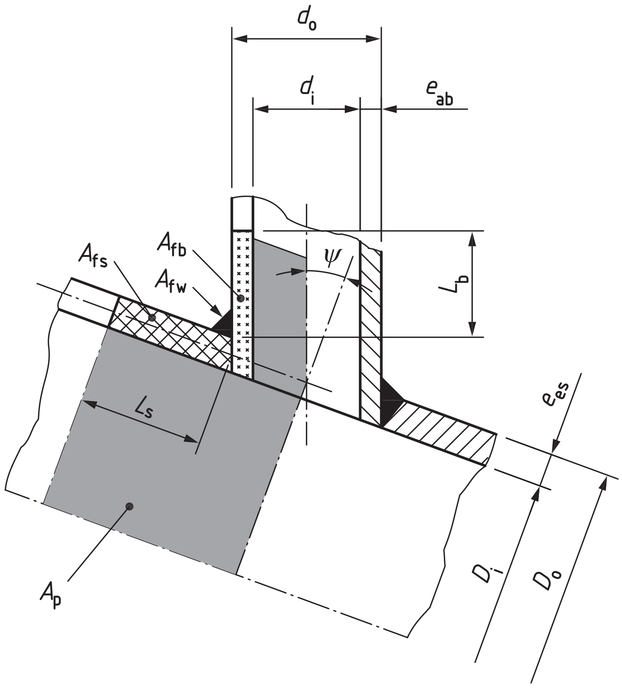
$A_{fb} = e_{ab}(L_b + e_{as})$ (set-in)

$A_{fb} = e_{ab} L_b$ (set-on)

Partial view II

a) Cylindrical shell with a branch not radially arranged (off centre)

Figure 3.5-33 **Reinforcement of non-radial branches *(continued)***



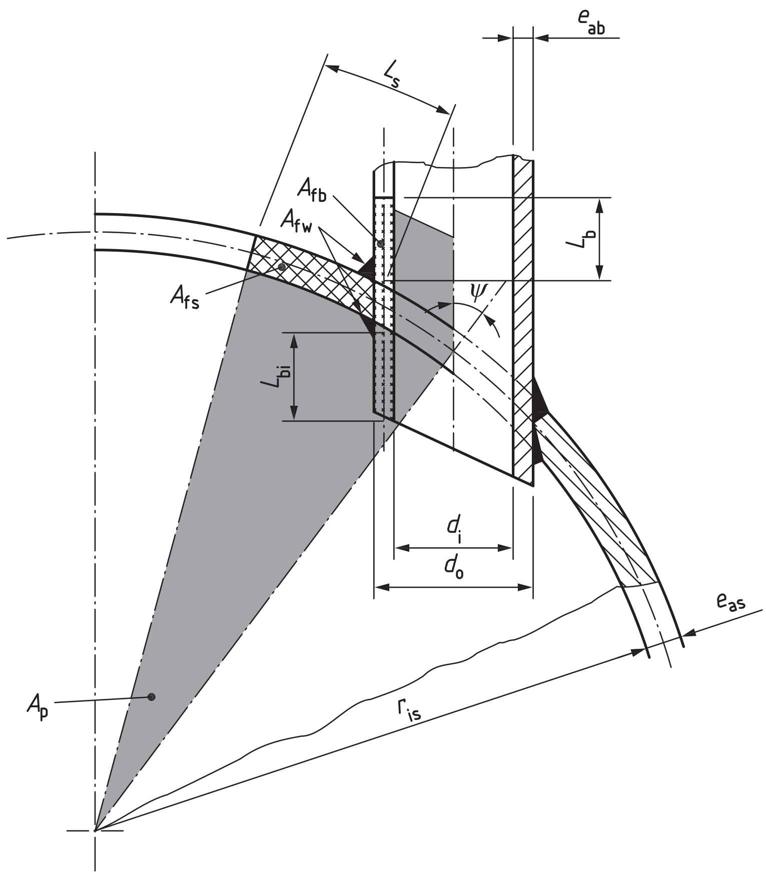
$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_o}{2\cos\psi}\right) + \frac{d_i}{2}\left(L_b + \frac{e_{as}}{\cos\psi}\right)$$

$$A_{fs} = e_{as}L_s$$

$$A_{fb} = e_{ab}\left(L_b + \frac{e_{as}}{\cos\psi}\right)$$

b) Cylindrical shell with a branch not radially arranged (oblique)

Figure 3.5-33 **Reinforcement of non-radial branches *(continued)***



$$A_p = \frac{r_{is}}{2}\left(L_s + \frac{d_o}{2\cos\psi}\right) + \frac{d_i}{2}\left(L_b + \frac{e_{as}}{\cos\psi}\right)$$

$$A_{fs} = e_{as}L_s$$

$$A_{fb} = e_{ab}\left(L_b + \frac{e_{as}}{\cos\psi} + L_{bi}\right)$$

c) Spherical shell with a branch not radially arranged

3.5.4.10 Openings and nozzle connections close to discontinuities

Where an opening or nozzle connection in a shell is close to a discontinuity junction in the following list a) to h), the reinforcement rules of **3.5.4** shall be met for the opening or nozzle connection assuming that the shell continues beyond the discontinuity with minimum thickness in accordance with **3.5.1**, but that reinforcement is only available up to the discontinuity junction.

a) Flange/shell.

b) Flat end/shell.

c) Tubesheet/shell.

d) Dished end/cylindrical shell.

e) Hemispherical end/cylindrical shell.

f) Large end of cone/shell.

g) Small end of cone/shell.

h) Nozzle with opening in nozzle wall/shell.

In the case of a taper hub flange the discontinuity junction is at the small end of the hub.

NOTE 1 This approach is based upon the concept that the opening and the discontinuity should both satisfy their appropriate requirements without any overlap of the relevant reinforcements.

*NOTE 2 **3.5.4.3.4**c)iii) allows shell thickening so as to modify the distribution of reinforcement.*

*NOTE 3 In the above list item d) is not intended to apply to the junction between the central part of a dished end and the knuckle. The limitations on openings and nozzles in dished ends are specified in **3.5.4.2**d)1) and Figure 3.5-8.*

The reinforcement rules for discontinuities f), g) and h) above shall additionally be satisfied without overlapping the reinforcement required for the shell opening or nozzle connection. A recommended procedure for checking these discontinuities is as follows.

– Choose reinforced thicknesses for the opening and junction.

– Choose a dividing line between the opening and junction reinforcement regions.

– Check the opening and junction reinforcements available up to the dividing line, assuming the shell continues beyond the dividing line with minimum thickness.

– If one check fails but the other succeeds it might be possible to move the dividing line to satisfy both checks. If not, or if both checks fail, increase the thicknesses or move the opening further from the junction and repeat the procedure from the second stage.

In the case of the large end of a cone/cylinder intersection without a knuckle, the requirements specified in **3.5.3.4** that the cross-sectional area of metal provided by the cylinder, within a distance $1.4l_1$ from the junction, shall not be less than $1.4e_1l_1$, may be modified to an area of e_1l_1 within a distance l_1 from the junction.

The allowable proximity, or design, of relevant welds shall be as specified in Section 3, in particular in **3.10**.

3.5.5 Flat ends and flat plates

3.5.5.1 Introduction

The following rules provide requirements for the determination of the thickness required against pressure loading of:

– circular flat unstayed ends, with and without openings; and

– stayed plates.

For the design of heat exchanger tubesheets refer to **3.9**.

*NOTE Calculations for circular flat ends with radial reinforcement ribs are given in BS EN 13445-3:2021, Clause **21**.*

Where the external load on the end (or the loading due to reverse pressure) can exceed 10% of the load due to design pressure, or where in the case of welded ends the temperature difference between the end and the vessel wall or between the end and a nozzle wall exceeds 50 °C (30 °C for austenitic steel), alternative methods shall be used (see **3.2.2**).

In cases where the design stress is time dependent, components designed by the procedure specified in this section shall be reviewed to ensure that creep deformation (local or general) will be acceptable throughout the agreed design lifetime.

3.5.5.2 Notation

For the purposes of **3.5.5.3** the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**).

C	is a factor (for welded circular ends see Figure 3.5-34, Figure 3.5-36 or Figure 3.5-37, for which Table 3.5-8 gives data points, for bolted circular ends, see Figure 3.5-35);
d	is the mean of the inside and outside diameter of a nozzle, or the bore of an opening not provided with a nozzle;
d_e	is the effective diameter of an opening;
D	is the diameter measured as in Figure 3.5-34 and Figure 3.5-35;
D_i	is the diameter measured as in Figure 3.5-34;
e	is the minimum thickness of end or plate;
e_1	is the minimum thickness at and beyond a gasket [see Figure 3.5-35c)];
e_a	is the analysis thickness of end or plate;
e_{cyl}	is the analysis thickness of cylindrical shell (see **1.6**);
e_{cylo}	is the minimum thickness of cylindrical shell = $pD/(2f)$;
F	is the total bolt load for a cover with self-sealing joint (see Figure 3.5-35);
F_{amb}	is the force in a single bolt in the ambient condition;
F_{op}	is the force in a single bolt in the operating condition;
f	is the design stress;
f_{amb}	is the design stress at ambient condition;
H	is the total hydrostatic end load for a cover with self-sealing joint [see **3.5.5.3.1**a)];
h	is the minimum distance between the centre of opening and the shell inside diameter;
h_G	is the mean distance between the gasket reaction and the bolt circle diameter [see Figure 3.5-35c)];
j	is an opening factor;
k	is the distance between the centres of a pair of openings;
p	is the design pressure (for flat ends subject to external pressure this shall be taken as a positive value for use in the equations – see **3.2.3**);
r	is the corner radius (see Figure 3.5-34);
t_B	is the mean bolt pitch for a bolted flat end;
u	is the distance between flat end and the end of thickness reduction [see Figure 3.5-34a)];
w	is the minimum length of shell of thickness e_{cyl} in Figure 3.5-34b) and Figure 3.5-34c): $w \geq \sqrt{(D + e_{cyl})e_{cyl}}$;
Y_1, Y_2	are parameters for ends with openings [see Equation (3.5.5-6) and Equation (3.5.5-7)].

3.5.5.3 Flat unstayed ends[4)]

3.5.5.3.1 Circular flat ends

a) Flat ends without openings.

1) The minimum thickness e of a welded or bolted circular flat end as shown in Figure 3.5-34 or Figure 3.5-35a), Figure 3.5-35b), or Figure 3.5-35d), is given by the following:

$$e = CD\sqrt{p/f} \qquad (3.5.5\text{-}1)$$

where the factor C for the various configurations given in Figure 3.5-34 or Figure 3.5-35 is:

Figure 3.5-34a),	$C = 0.35$ if $u \geq r + \left[1.1 - 0.8\left(\frac{e_{cyl}}{e}\right)^2\right]\sqrt{D_i e}$ otherwise $C = 0.41$;
Figure 3.5-34b) and Figure 3.5-34c),	C is given by Figure 3.5-36, Figure 3.5-37 or Table 3.5-8;
Figure 3.5-35a) and Figure 3.5-35b),	$C = 0.41$;
Figure 3.5-35d),	$C = \sqrt{0.17 + 0.75F/H}$ where F is the total bolt load and $H = \pi p D^2/4$.

For welded ends, the thickness of the cylinder, e_{cyl} (see Figure 3.5-34) need only be maintained over a distance w from the end. The thickness of the cylinder may be increased above e_{cyl} (but not so as to exceed e) local to the end and be reduced to a value not less than e_{cylo} at more distant points provided that the total cross-sectional area of the shell walls falling within the distance w from the end is not less than we_{cyl}.

In the case of welded ends, if the design stresses of the cylinder and the end are different, the lower value shall be used.

NOTE 1 This ensures that at the junction with the cylinder, the cylindrical end of the flat end is not thinner than the adjacent cylinder (see Figure 3.5-34).

For a bolted flat end with full face gasket, bolt spacing shall not exceed the dimension given by Equation (3.5.5-2). If necessary the flat end thickness shall be increased to enable this requirement to be met.

$$2 \times \text{bolt o.d.} + (E/200\,000)^{0.25} \times 6e/(m + 0.5) \qquad (3.5.5\text{-}2)$$

where

E is the modulus of elasticity of flat end material, at the design temperature, given in Table 3.6-3 (in N/mm^2);

m is the gasket factor given in Table 3.8-4.

2) The minimum thickness e of a bolted circular flat end as shown in Figure 3.5-35c) is given by Equation (3.5.5-3) and in addition shall take into account e_1, the minimum thickness at and beyond the gasket, as given by Equation (3.5.5-4), and any grooves and steps at and beyond the gasket:

[4)] For supporting information, see Annex R.

$$e = \max\left(\sqrt{\frac{1.909Wh_G}{DS_{FA}}}; \sqrt{\frac{0.3D^2p}{S_{FO}} + \frac{1.909W_{m1}h_G}{DS_{FO}}}\right) \tag{3.5.5-3}$$

and

$$e_1 = \max\left(\sqrt{\frac{1.909Wh_G}{DS_{FA}}}; \sqrt{\frac{1.909W_{m1}h_G}{DS_{FO}}}\right) \tag{3.5.5-4}$$

where W, W_{m1}, h_G, S_{FA} and S_{FO} are as defined in **3.8.2**; and D is the gasket diameter as defined as G in **3.8.2**, for blind flanges with gasket entirely within the bolt circle [see Figure 3.5-35c)].

For flat ends subject to external pressure W_{m1} = 0 (see **3.8.3.5**).

b) Flat ends with openings, where $d/D \leq 0.5$.

The following rules are applicable to single and multiple openings. Where there are multiple openings, each opening shall be checked as a single opening, and then each pair of openings checked. Holes may extend up to the inside of the shell if the end is welded or the inside of the gasket if the end is bolted.

1) The minimum thickness e of a welded flat end with an opening, as shown in Figure 3.5-34, is given by:

$$e = \max(CY_1; 0.41Y_2)D\sqrt{p/f} \tag{3.5.5-5}$$

2) The minimum thickness of a bolted flat end, as shown in Figure 3.5-35 but with an opening, is the thickness as calculated from the following, for the relevant equations:

i) (3.5.5-1) multiplied by Y_2;

ii) (3.5.5-2);

iii) (3.5.5-3) or (3.5.5-4), whichever is the greater, multiplied by Y_2.

For bolted flat ends to Figure 3.5-35c), even if edge bolting controls the thickness, reinforcement of openings to these rules is required.

Y_1 and Y_2 are obtained from Equations (3.5.5-6) and (3.5.5-7).

$$Y_1 = \min\left(2; \sqrt[3]{\frac{j}{j - d_e}}\right) \tag{3.5.5-6}$$

$$Y_2 = \sqrt{\frac{j}{j - d_e}} \tag{3.5.5-7}$$

where d_e, the effective diameter of an opening ($d_e \geq 0$ for calculation purposes):

= d for a single plain opening;

= $d - e_{ab} - 2A/e_a$ for a single set-on nozzle, with $A = L_b(e_{ab} - e_{pb}) + A_w$;

= $d + e_{ab} - 2A/e_a$ for a single set-in or set-through nozzle,

with $A = e_a e_{ab} + L_b(e_{ab} - e_{pb}) + A_w$ for set-in nozzles and
$A = (L_{bi} + e_a)e_{abi} + A_{wi} + L_b(e_{ab} - e_{pb}) + A_w$ for set-through nozzles;

= mean of the relevant effective diameters given above, for a pair of openings

where

i. $L_b = \sqrt{de_{ab}}$ or the actual length of the nozzle neck measured from the outside surface of the flat end, whichever is the smaller; and $L_{bi} = \sqrt{de_{ab}}$ or the actual length of the internal protrusion measured from the inside surface of the flat end, whichever is the smaller;

ii. the thicknesses e_{ab} and e_{pb} are defined in **3.5.4**;

iii. the thickness *e*abi is the analysis thickness of the nozzle internal protrusion, taking account of the corrosion allowance on the inside and outside diameter;

iv. *A*w is the cross-sectional area of the external fillet weld between the nozzle and the flat end located within the length *L*b, and for set-through nozzles *A*wi is the cross-sectional area of the internal fillet weld between the nozzle and the flat end located within the length *L*bi, taking account of the corrosion allowance;

v. *A* shall be multiplied by *f*b/*f*, if the nozzle design stress *f*b is lower than the flat end design stress *f*.

Factor *j*:

= 2*h* in Equation (3.5.5-6), for a single opening;

= *D* in Equation (3.5.5-7), for a single opening;

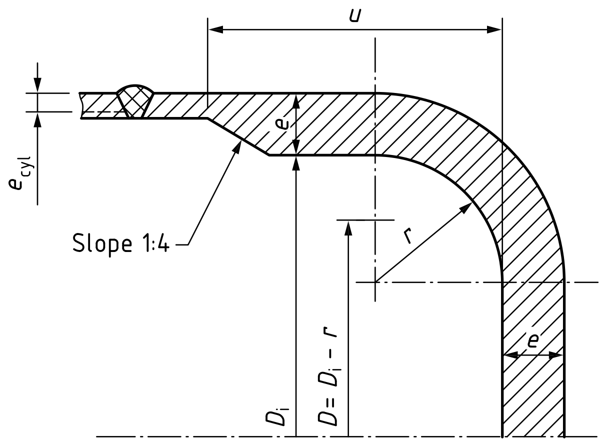
= *k* in Equations (3.5.5-6) and (3.5.5-7), for a pair of openings.

c) Flat ends with an opening, where $d/D > 0.5$.

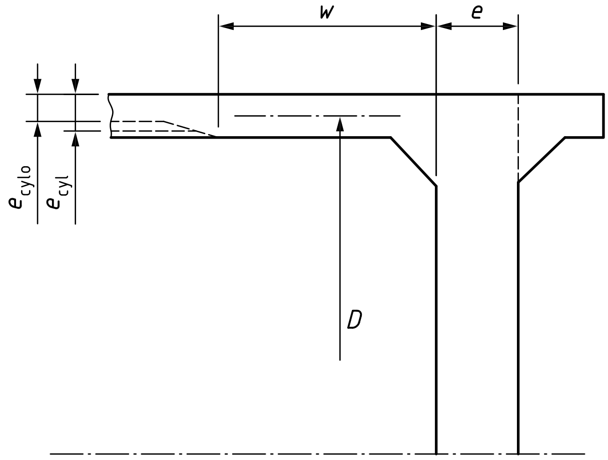
It is permissible to use the method given in b) above where an opening has a diameter greater than *D*/2.

*NOTE 2 This method will produce a more conservative design as the d/D ratio increases. A less conservative design for central holes may be based upon the basic principles used in **3.8** for flange design.*

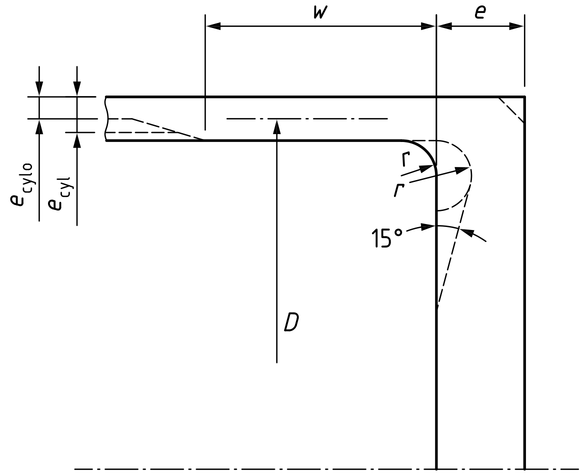
Figure 3.5-34 **Typical welded flat ends and covers (for typical weld joint details, see Figure E.38**



a) $r \geq$ Twice the nominal thickness of end or plate;



b) Dotted lines indicate alternative profiles.

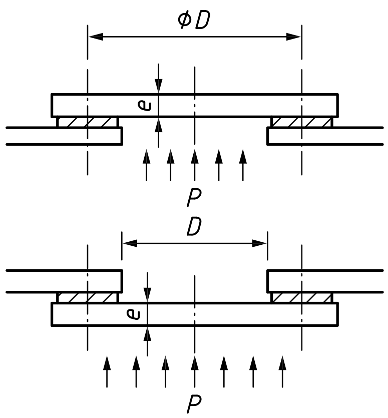


c) $r \geq 5$ mm or $e_{cyl}/4$. The end may be undercut as shown to make provision for the radius r or to improve access for non-destructive testing of the weld (see Figure E.38) provided that $r < e_{cyl}$ and the thickness of the end is nowhere less than e_{cyl}.

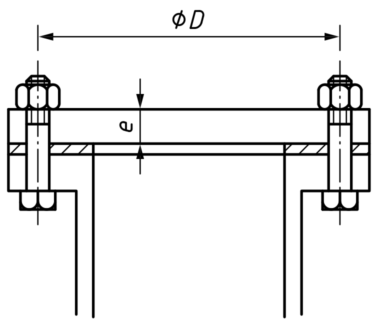
NOTE 1 The analysis thickness of the cylinder, e_{cyl} need only be maintained over a distance w from the end. The analysis thickness of the cylinder may be increased above e_{cyl} (but not so as to exceed e) local to the end and be reduced to a value not less than e_{cylo} at more distant points provided that the total cross-sectional area of the shell walls falling within the distance w from the end is not less than we_{cyl}. C is determined from Figure 3.5-35, Figure 3.5-37 or Table 3.5-8.

NOTE 2 In Figure 3.5-34b) and Figure 3.5-34c), D = mean diameter of cylinder.

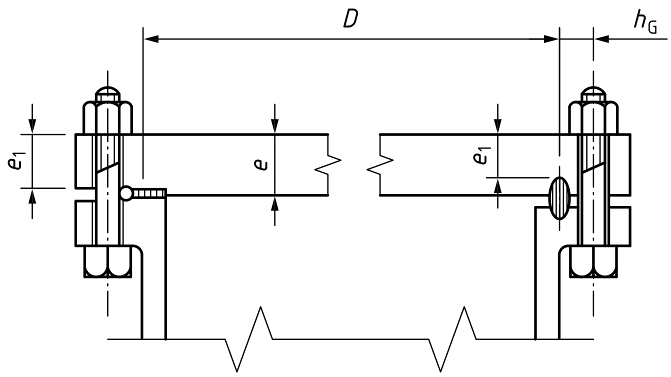
Figure 3.5-35 **Typical non-welded flat ends and covers**



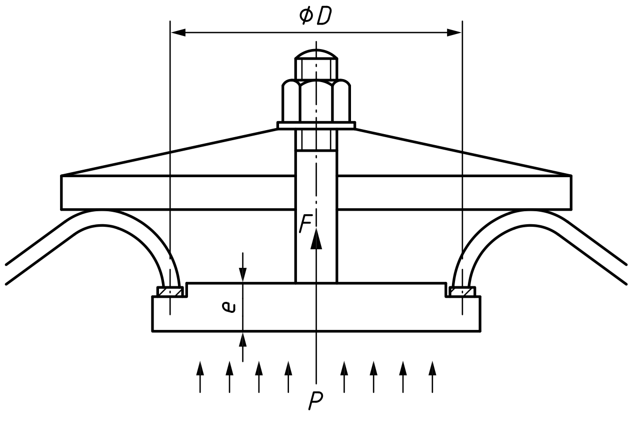
a) Flat cover with a full face gasket C = 0.41.



b) Blind flange with a full face gasket C = 0.41.

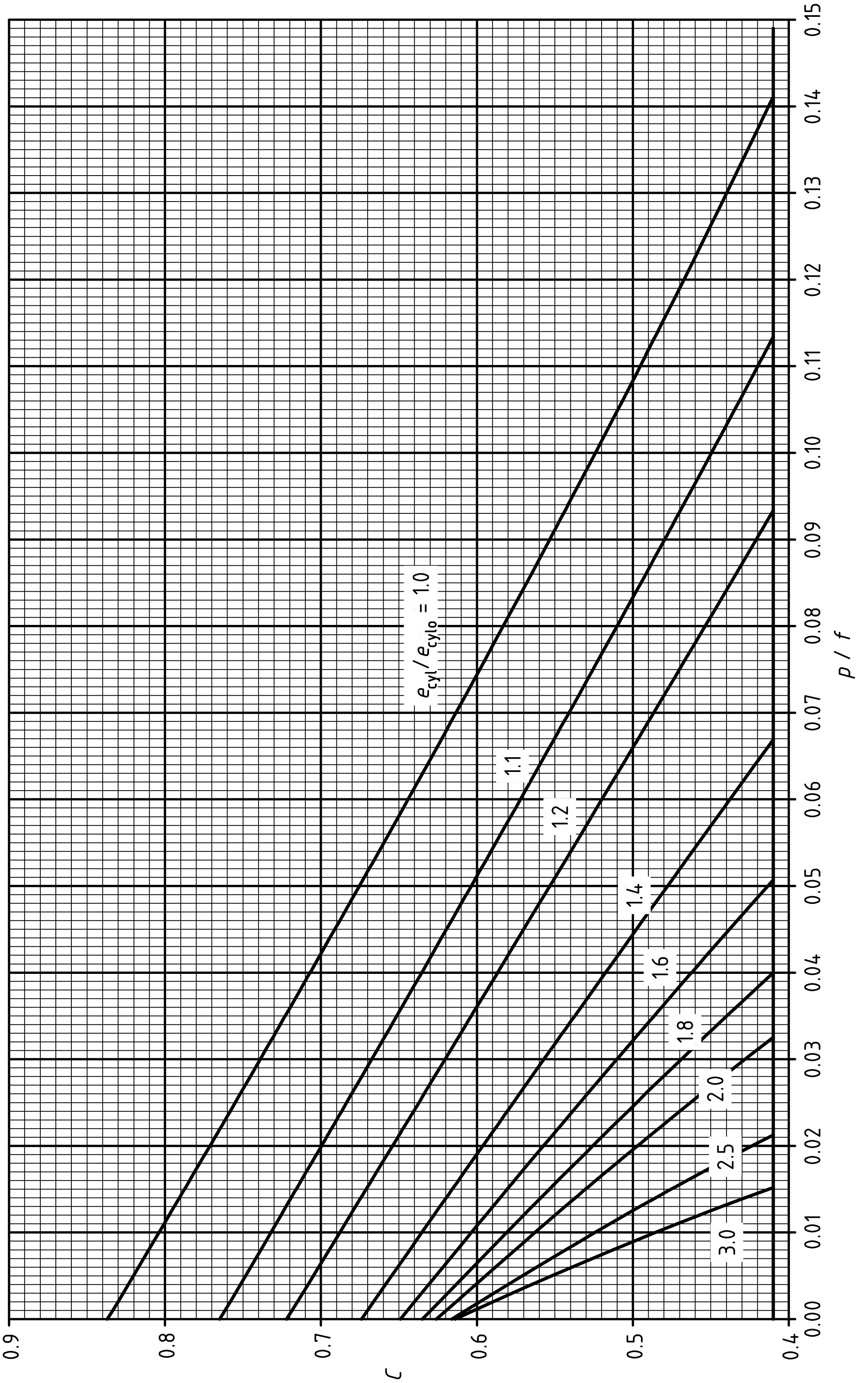


c) Blind flange with gasket entirely within the bolt circle.



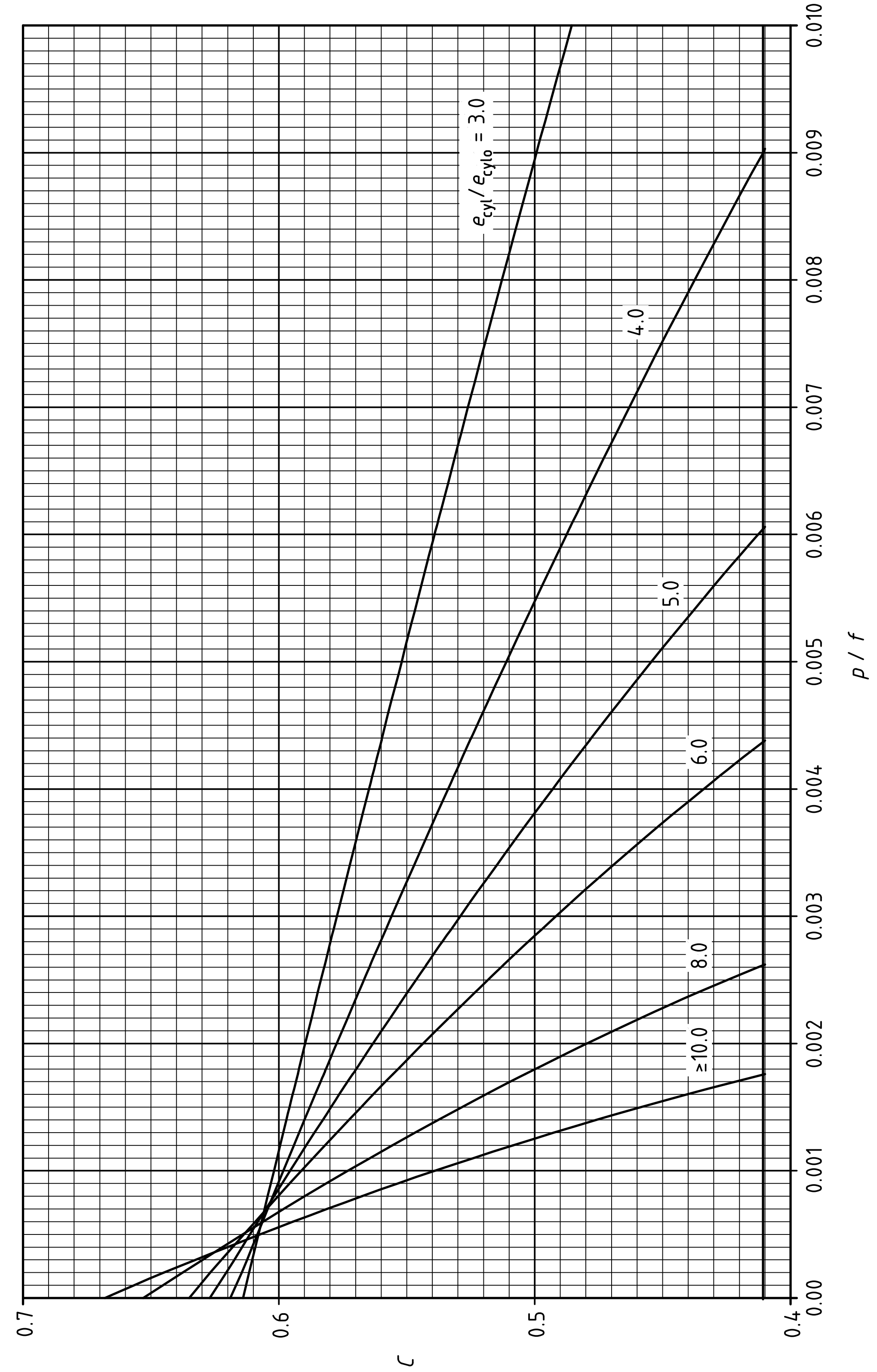
d) Cover with self-sealing joint.

Figure 3.5-36 **Factor C for welded flat ends [to Figure 3.5-34b) and Figure 3.5-34c)] for e_{cyl}/e_{cylo} = 1 to 3**



NOTE C = 0.41 is always valid when $e/e_{cyl} \le 1.5$.

Figure 3.5-37 **Factor C for welded flat ends [to Figure 3.5-34b) and Figure 3.5-34c)] for e_{cyl}/e_{cylo} = 3 to > 10**



NOTE C = 0.41 is always valid when $e/e_{cyl} \leq 1.5$.

Table 3.5-8 **Values of factor *C* for welded flat ends to Figure 3.5-34b) and Figure 3.5-34c) (for Figure 3.5-36 and Figure 3.5-37)**

p/f	e_{cyl}/e_{cylo}													
	1.0	**1.1**	**1.2**	**1.4**	**1.6**	**1.8**	**2.0**	**2.5**	**3.0**	**4.0**	**5.0**	**6.0**	**8.0**	**10.0**
0.00	0.837	0.765	0.722	0.674	0.649	0.635	0.626	0.616	0.614	0.619	0.627	0.635	0.653	0.668
0.01	0.804	0.732	0.688	0.635	0.604	0.581	0.563	0.524	0.486	—	—	—	—	—
0.02	0.771	0.700	0.654	0.596	0.557	0.526	0.497	0.424	—	—	—	—	—	—
0.03	0.739	0.667	0.620	0.557	0.510	0.469	0.428	—	—	—	—	—	—	—
0.04	0.707	0.635	0.587	0.518	0.462	—	—	—	—	—	—	—	—	—
0.05	0.675	0.604	0.553	0.478	0.413	—	—	—	—	—	—	—	—	—
0.06	0.644	0.572	0.520	0.438	—	—	—	—	—	—	—	—	—	—
0.07	0.613	0.541	0.487	—	—	—	—	—	—	—	—	—	—	—
0.08	0.583	0.510	0.454	—	—	—	—	—	—	—	—	—	—	—
0.09	0.553	0.480	0.421	—	—	—	—	—	—	—	—	—	—	—
0.10	0.524	0.450	—	—	—	—	—	—	—	—	—	—	—	—
0.11	0.495	0.420	—	—	—	—	—	—	—	—	—	—	—	—
0.12	0.467	—	—	—	—	—	—	—	—	—	—	—	—	—
0.13	0.440	—	—	—	—	—	—	—	—	—	—	—	—	—
0.14	0.413	—	—	—	—	—	—	—	—	—	—	—	—	—
Factor *C* = 0.41 for *p/f* greater than or equal to the values tabulated below														
	0.141	0.113	0.0934	0.0669	0.0507	0.0400	0.0325	0.0213	0.0152	0.00903	0.00606	0.00438	0.00262	0.00176

3.5.5.4 Flat stayed plates without openings

3.5.5.4.1 Plate thickness

The thickness of stayed and braced plates shall be calculated in accordance with the following:

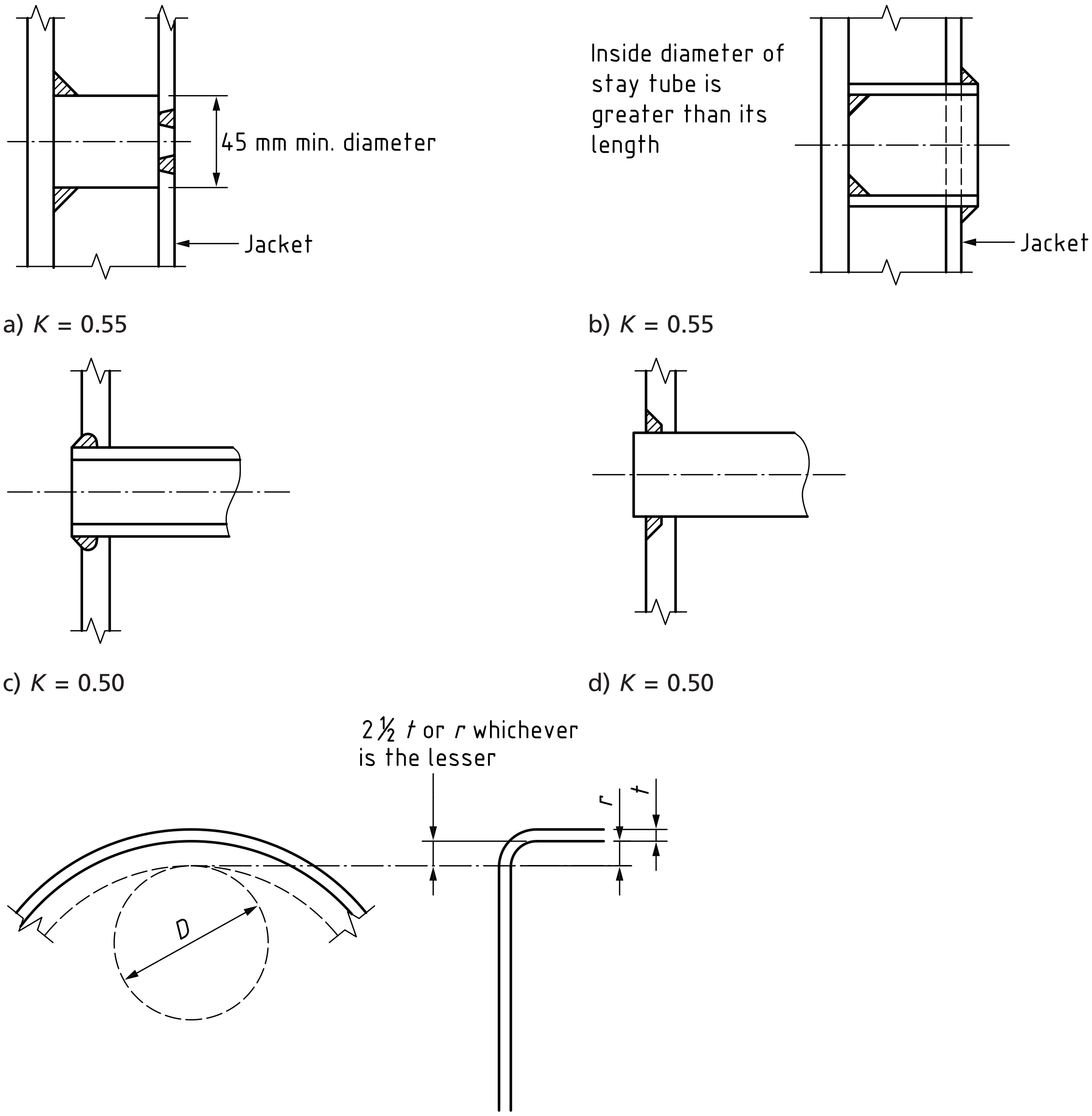
$$e = KD\sqrt{\frac{p}{f}} \tag{3.5.5-8}$$

where

- D is the diameter of a circle drawn through at least three points of support pitched at reasonably regular intervals circumferentially;
- e is the minimum calculated thickness;
- f is the design stress;
- K is a constant depending on the method of attachment of stay to plate (see Figure 3.5-38). K shall be a mean value when more than one type of support is involved;
- p is the design pressure.

Designs in which plate deflection and/or differential expansion are significant require alternative methods (see **3.2.2**).

Figure 3.5-38 **Typical stays: areas supported by stays**



a) K = 0.55

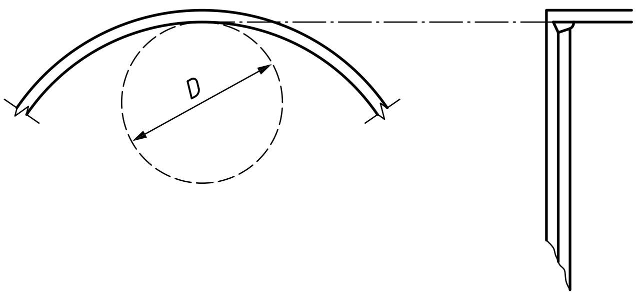
b) K = 0.55

c) K = 0.50

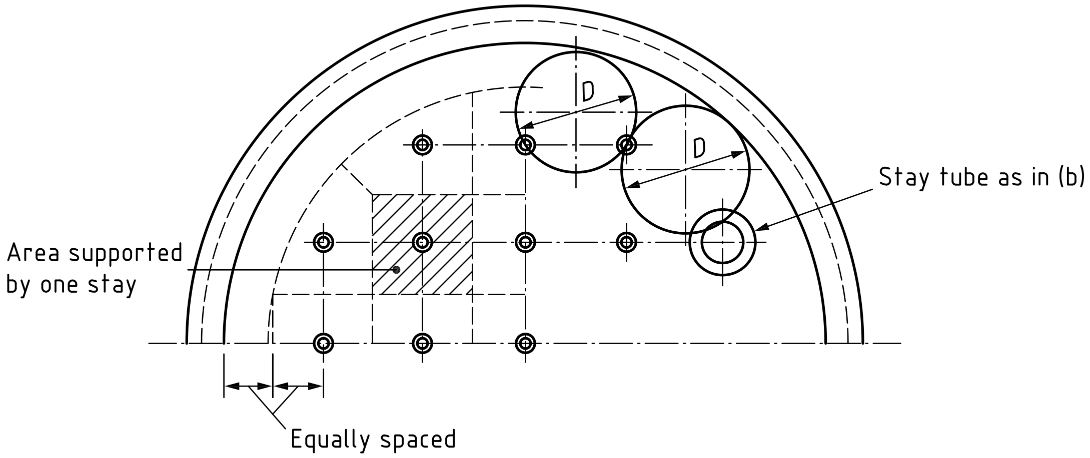
d) K = 0.50

e) K = 0.45

Figure 3.5-38 **Typical stays: areas supported by stays *(continued)***



f) $K = 0.45$



g)

NOTE For weld details see Annex E.

3.5.5.4.2 Methods of support

The method of support shall be chosen from the typical methods shown in Figure 3.5-38a) to Figure 3.5-38f).

NOTE When it is undesirable to drill a plate for the attachment of stays, e.g. when the plate is to be lined, the use of stays of the type shown in Figure 3.5-38a) or Figure 3.5-38b) is recommended.

3.5.5.4.3 Stays

The design stress of stays, calculated on the least cross-sectional area, shall not be greater than the following:

for solid staybars:	$0.75f$;
for staytubes:	$0.70f$;
for staybolts:	$0.65f$;

where f is the design stress derived from **2.3.1.1** or **3.8.1.4**, as appropriate.

For the purposes of calculation, the gross area supported by each stay shall be as shown in Figure 3.5-38g). In the case of stays of the type shown in Figure 3.5-38b) it is permissible to use the net area supported, in the equations.

The design stress in attachment welds shall not exceed $0.5f$ in fillet welds and $0.6f$ in penetration welds.

Stays shall be of welding quality wrought materials complying with Section 2 and shall be compatible with the material of the plates which they support.

 •

Stays shall not be welded, except at the point of attachment. Where necessary, long stays shall have additional support to prevent sagging.

3.5.5.4.4 Tube to tubeplate connections

The central line of tubes that are to be expanded shall not be closer together than $1.125d + 12.7$ mm, measured at the tubeplate:

where

d is the outside diameter of the tube in millimetres.

*NOTE This subclause does not apply to tubeplates covered in **3.9**.*

3.5.6 Spherically domed and bolted ends of the form shown in Figure 3.5-39

3.5.6.1 General

Except as specified as follows for bolted ends of the form shown in Figure 3.5-39, conical and domed and bolted ends shall be designed by treating the domed end and the bolted flange as two separate components in compliance with the relevant clauses of this specification. The design rules for full faced flange rings given in **3.5.6** are only applicable where the domed and bolted ends are attached to tubesheets.

NOTE 1 The method of determining the thickness of the flange ring involves assessing the final thickness in order to arrive at the location of the centroid and hence the value of h_r and is thus a "trial and error" calculation.

These equations are approximate in that they do not take account of the structural continuity that exists at the junction of the domed end and flange. A more exact (and often less conservative) analysis is given by Soehrens[5]. The stresses calculated using this approach should be assessed in accordance with Annex A.

*NOTE 2 Suggested working forms with sketches, covering spherical domed and bolted ends are provided at the end of **3.5.6**. Working form 3.5.6-1 covers narrow faced gasketted ends subject to internal or external pressure and Working form 3.5.6-2 covers full faced gasketted ends subject to internal or external pressure.*

For the purposes of **3.5.6.2** and **3.5.6.3** the following symbols apply. All dimensions are in the corroded condition, unless otherwise indicated (see **3.1.5**).

A is the outside diameter of flange or, where slotted holes extend to outside of flange, the diameter to bottom of slots;

B is the inside diameter of flange;

C_F is the bolt pitch correction factor (see **3.8.2**);

f is the design stress for material of spherical crown section;

H_D is the hydrostatic end force on area inside of flange (i.e. force applied via connection to flange);

H_G is the gasket load;

H_R is a balancing reaction force for full face gasketted flanges [see Equations (3.5.6-8) and (3.5.6-11)];

H_r is the radial component of membrane force developed in spherical crown section, acting at edge;

H_T is the hydrostatic end force due to pressure on flange face;

h_D is the radial distance from bolt circle to circle on which H_D acts (see Figure 3.5-39);

h_G is the radial distance from gasket load reaction to bolt circle (see Figure 3.5-39);

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[5] J. E. SOEHRENS. The design of floatingheads for heat exchangers. *Pressure Vessel and Piping Design. Collected Papers 1927 to 1959*, ASME.

h_r is the axial distance from mid-surface of crown section at edge to centroid of flange ring cross-section (see Figure 3.5-39); h_r has a positive value when the horizontal line of intersection between crown and flange is between the centre of gravity of flange and the gasket face.

NOTE Figure 3.5-39 and the working form show a negative h_r.

h_T is the radial distance from bolt circle to circle on which H_T acts (see Figure 3.5-39);

M_{atm} is the total moment acting upon flange for gasket sealing conditions as defined in **3.8.3.3**;

M_{op} is the total moment acting on flange for operating conditions [see Equations (3.5.6-5) or (3.5.6-10)];

p is the design pressure (this shall be taken as a positive value for use in the equations – see **3.2.3**);

R_1 is the inside radius of curvature of spherical crown section (see Figure 3.5-39);

S_{FA} is the design stress for flange material at atmospheric temperature;

S_{FO} is the stress for flange material at design temperature (operating conditions);

T is the minimum flange ring thickness;

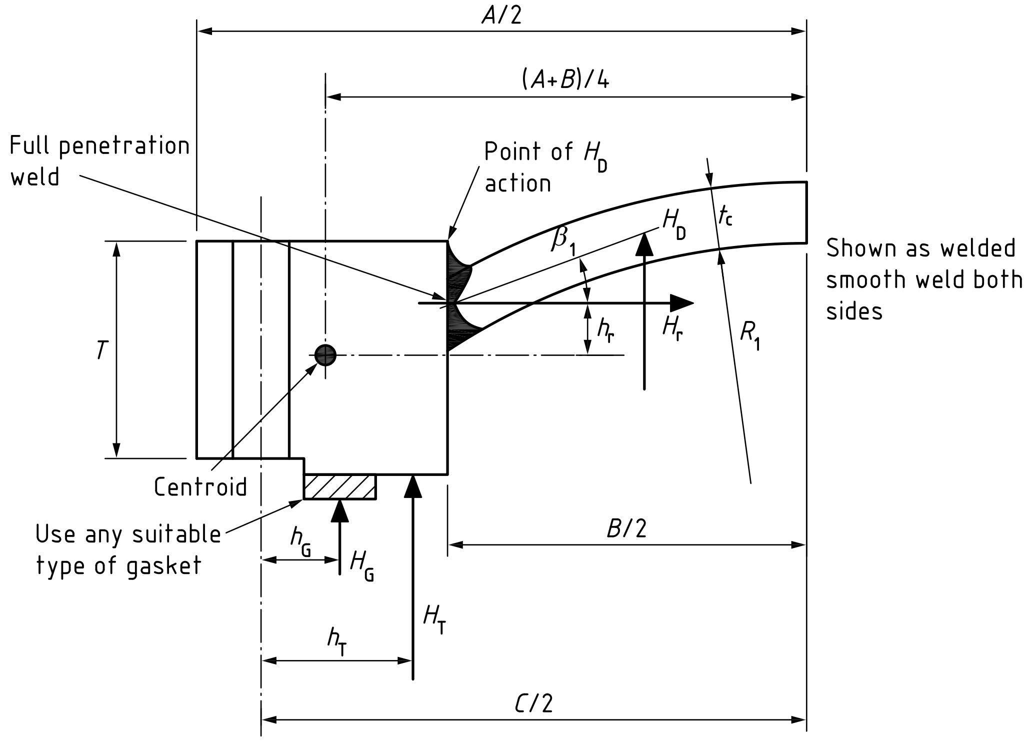
T_{fo} is the flange minimum thickness required for operating condition [see Equation (3.5.6-2)];

T_{fA} is the flange minimum thickness required for bolting-up condition [see Equation (3.5.6-7)];

t_c is the minimum thickness of spherical crown section [see Equation (3.5.6-1)];

β_1 is the angle between tangent to domed crown section at its edge and a plane parallel to flange face (see Figure 3.5-39).

Figure 3.5-39 **Spherically domed and bolted end (narrow faced gasket)**



NOTE 1 Directions of forces are shown for internal pressure.

*NOTE 2 h_r is shown having a negative value (see **3.5.6.1**).*

3.5.6.2 Subject to internal pressure (concave to pressure)

3.5.6.2.1 Crown section

The minimum thickness of the spherical crown section shall be:

$$t_c = \frac{5pR_1}{6f} \qquad (3.5.6\text{-}1)$$

3.5.6.2.2 Flange ring

The bolting area required, the bolt loads and the gasket width check shall be calculated in accordance with **3.8**.

3.5.6.2.3 Flange ring thickness

The minimum thickness, T, of the flange ring shall be the greater of T_{fo} or T_{fA} determined as follows, but shall be not less than twice the crown thickness, i.e. $T \geq 2t_c$.

3.5.6.2.3.1 Narrow faced gasketted flange

a) Operating condition:

$$T_{fo} = F + \sqrt{F^2 + J_o} \qquad (3.5.6\text{-}2)$$

where

$$F = \frac{pB\sqrt{4R_1^2 - B^2}}{8S_{FO}(A - B)} \qquad (3.5.6\text{-}3)$$

$$J_o = \frac{|M_{op}|C_F(A + B)}{BS_{FO}(A - B)} \qquad (3.5.6\text{-}4)$$

$$M_{op} = H_D h_D + H_G h_G + H_T h_T + H_r h_r \qquad (3.5.6\text{-}5)$$

$$H_r = H_D cot\beta_1 \qquad (3.5.6\text{-}6)$$

where H_D, h_D, H_G, h_G, H_T and h_T are defined in **3.8.2** and h_r is defined in **3.5.6.1**.

b) Bolting up condition:

$$T_{fA} = \sqrt{\frac{C_F(A + B)M_{atm}}{(A - B)BS_{FA}}} \qquad (3.5.6\text{-}7)$$

3.5.6.2.3.2 Full faced flanges

$$H_R = \frac{H_D h_D + H_T h_T + H_G h_G + H_r h_r}{h_R} \qquad (3.5.6\text{-}8)$$

where H_D, h_D, H_T, h_T, H_G, h_G, and h_R are defined in **3.8.4.1** and h_r is defined in **3.5.6.1**. H_r is defined in Equation (3.5.6-6).

Bolt loads and areas shall be in accordance with **3.8.4.2**. Flange design shall be in accordance with **3.8.4.3**. Additionally the flange thickness shall be checked so that:

$$T > 2F\left(\frac{A - B}{A - B - 2d}\right) \tag{3.5.6-9}$$

where

A and *B* are defined in **3.5.6.1**;

d is the diameter of the bolt holes;

F is determined from **3.5.6.2.3.1**.

3.5.6.3 Subject to external pressure (convex to pressure)

The crown section and flange ring shall comply with the following.

a) *Crown section*

The minimum thickness of the spherical crown section shall be the greater of:

1) thickness determined in accordance with **3.5.6.2.1**; or

2) thickness of a spherical shell of radius R_1 under external pressure determined in accordance with **3.6.4**.

b) *Flange ring*

The thickness of a narrow faced gasketted flange ring shall be determined in accordance with **3.5.6.2.3.1** except that

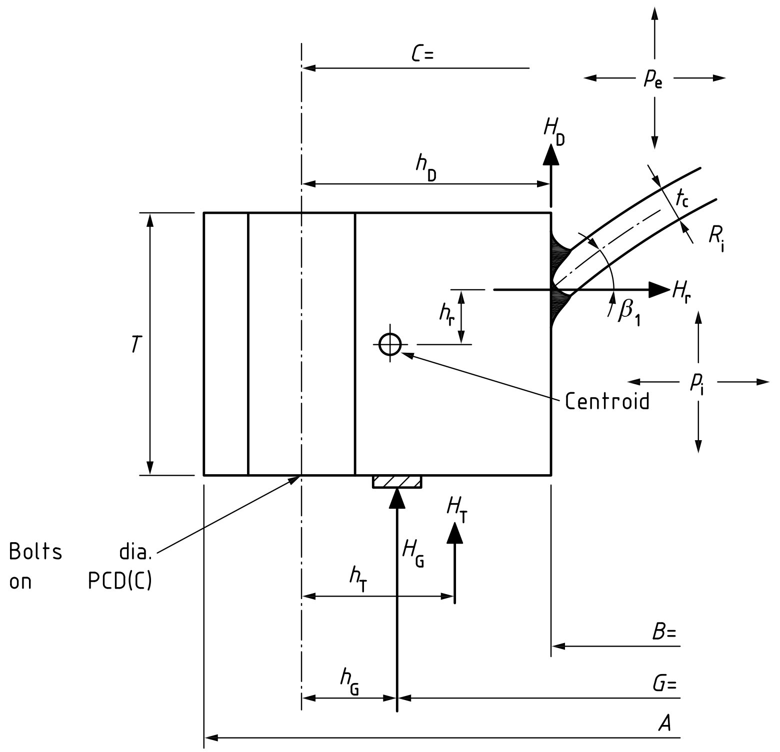
$$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G) + h_r H_r \tag{3.5.6-10}$$

NOTE The gasket should be checked against excessive deformation under the action of the bolt load and the external pressure thrust.

The thickness of a full faced flange ring shall be determined in accordance with **3.5.6.2.3.2** except that

$$H_R = \frac{H_D(h_D - h_G) + H_T(h_T - h_G) + H_r h_r}{h_R} \tag{3.5.6-11}$$

Suggested working form 3.5.6-1		
Spherically domed and bolted end: narrow faced gasket		
Design pressure: Internal pressure p_i =	External pressure p_e =	
Design temperature =		
Corrosion allowance =		
Flange material:		
Gasket type:		
Bolting material:		

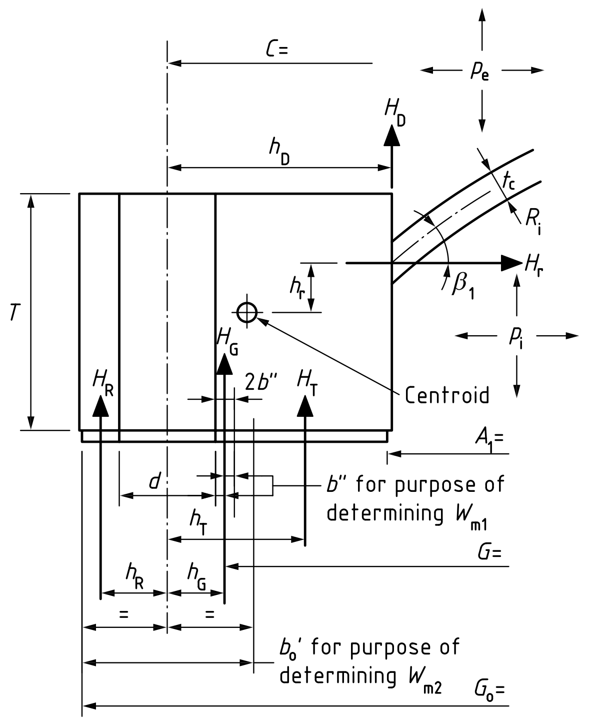


NOTE 1 Directions of forces are shown for the internal pressure case.
NOTE 2 Negative values of h_r indicated (see 3.5.6.1).

Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Bolts	S_b =	S_a =

Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Bolting		
$H = (\pi/4)G^2 p_i$ =	$H_G = 2b \times \pi G m p_i$ =	
$W_{m1} = H_G + H$ =	$W_{m2} = \pi b G y$ =	
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =	
$A_m = \max(A_{m1}\ ;\ A_{m2})$ =		
A_b = number of bolts × root area =		
$W = (A_b + A_m)S_a/2$ =		
Moment arms		
$h_D = (C - B)/2$ =	$h_G = (C - G)/2$ =	
$h_T = (2C - B - G)/4$ =		
Shape constants		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	$(C_F \geq 1)$	
F from equation (3.5.6-3) =		
Bolting-up condition		
$M_{atm} = W h_G$ =		
T_{fA} (from equation 3.5.6-7) =		
Operating condition	**Internal pressure**	**External pressure**
$H = (\pi/4)G^2 p_i$ (or p_e) =		
$H_D = (\pi/4)B^2 p_i$ (or p_e) =		
$H_G = 2b \times \pi G m p_i$ (or p_e) =		
$H_T = H - H_D$ =		
$H_r = H_D \cot\beta_1$ =		
Assume T_{fo} ($\geq T_{fA} \geq 2t_c$) =		
Thus h_r =		
Calculate M_{op}	from equation (3.5.6-5)	from equation (3.5.6-10)
M_{op} =		
J_o from equation (3.5.6-4) =		
T_{fo} from equation (3.5.6-2) =		
Thus min thickness T =		

Suggested working form 3.5.6-2		
Spherically domed and bolted end: full faced gasket		
Design pressure: Internal pressure p_i =	External pressure p_e =	
Design temperature =		
Corrosion allowance =		
Flange material:		
Gasket type:		
Bolting material:		



NOTE 1 Directions of forces are shown for the internal pressure case.
NOTE 2 Negative values of h_r indicated (see 3.5.6.1).

Design stresses	Operating	Ambient
Flange material	S_{FO} =	
Bolts	S_b =	S_a =
Elastic modulus for flange	E =	

Gasket details (dimensions in millimetres)		
b'_o = lesser of $(G_o - C)$ and $(C - A_1)$ =	$2b''$ = 5 mm	
$b' = 4\sqrt{b'_o}$ =	m =	
y =		
Moment arms		
$h_D = (C - B)/2$ =		
Assume bolt hole diameter d =	$h_G = (d + 2b'')/2$ =	
$G = C - d - 2b''$ =		
$h_T = (C + d + 2b'' - B)/4$ =	$h_R = (G_o - C + d)/4$ =	
Operating condition	**Internal pressure**	**External pressure**
Assume T ($\geq 2t_c$) =		
Thus h_r =		
Loads and moments		
$H = (\pi/4)(C - d)^2 p_i$ (or p_e) =		
$H_D = (\pi/4)B^2 p_i$ (or p_e) =		
$H_G = 2b'' \times \pi G m p_i$ (or p_e) =		
$H_T = H - H_D$ =		
$H_r = H_D \cot\beta_1$ =		
Calculate H_R	from equation (3.5.6-8)	from equation (3.5.6-11)
H_R =		
$M = H_R h_R$ =		
Bolting		
$W_{m1} = H_G + H + H_R$ =	$W_{m2} = \pi C b' y$ =	
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =	
$A_m = \max(A_{m1}\,;A_{m2})$ =	A_b = number of bolts × root area =	
Bolt pitch/bolt o.d. (d_b) =		
Check assumed bolt hole diameter (d)		
Thickness	**Internal pressure**	**External pressure**
Moment criteria: min T $T = \sqrt{\dfrac{6M}{S_{FO}(\pi C - nd)}} =$		
Bolt spacing criteria: min T $T = \dfrac{(\text{bolt sp} - 2d_b)(m + 0.5)}{6(E/200\,000)^{0.25}} =$		
F from equation (3.5.6-3) =		
T from equation (3.5.6-9) =		
Thus min thickness T =		

3.6 Vessels under external pressure

3.6.1 General

Shells subject to external pressure shall conform to the requirements of **3.6**. These requirements apply to stiffened and unstiffened cylinders and cones, spherical components and dished ends. Where other loadings listed in **3.2.1** are present, support shall be provided, if necessary, by increasing the shell thickness or by other means.

The thickness of a component subject to external pressure shall not be less than the thickness calculated to the relevant equation of **3.5.1**, **3.5.2** or **3.5.3** using a design internal pressure equal to the design external pressure.

Using a chosen thickness the allowable external pressure for the component shall be calculated in accordance with the requirements of **3.6** and shall not be less than the design external pressure. For external pressure the component dimensions shall meet the following tolerances.

a) The procedures given in **3.6.1** and **3.6.2** for cylinders, cones and their stiffeners only apply if they are circular to within ±0.5% on radius as measured from the true centre; a procedure to measure and calculate this departure from a true circle is given in **3.6.8**. This radial tolerance of 0.5% may be increased at the design stage if the design external pressure is increased in the ratio of (increased % radial tolerance/0.5). For stiffeners only, an alternative to increasing the design external pressure is to replace the figure of 0.005 (i.e. 0.5% tolerance/100) in Equation (3.6.2-15) or Equation (3.6.3-5) by the figure (increased % tolerance/100).

   When the tolerances of the cylindrical components are outside these limits an allowable external pressure shall be calculated using the procedure given in Annex M.

   NOTE 1 The method given in Annex M is a more refined method but requires measurements of the as-built shape. Calculations to Annex M often show that the 0.5% tolerance on radius can be exceeded without penalty, but this should not be assumed without calculation since the result depends upon the deformed shape as well as the achieved tolerance.

b) The specified tolerances and procedure for increased tolerances for spherical shells are given in **3.6.4**.

   Any allowable deviation from shape or increased tolerance assumed in the design shall be specified on the relevant drawing and associated documentation.

The compensation of openings in components subject to external pressure, shall be in accordance with **3.5.4.6**.

NOTE 2 In cases where the design strength is time dependent, components designed by the procedure specified in this clause should be reviewed to ensure that creep deformation (local or general) will be acceptable throughout the agreed design lifetime.

NOTE 3 For more information on the background to this section see proceedings of I.Mech.E. Conference, 7 December 1972, on Vessels under Buckling Conditions, and in particular the following papers.

a) C187/72, *Buckling under external pressure of cylinders with either torispherical or hemispherical end closure*, by G.D. Galletly and R.W. Aylward.

b) C190/72, *Collapse of stiffened cylinders under external pressure*, by S.B. Kendrick.

c) C191/72, *Collapse of domes under external pressure*, by C.N. Newland.

NOTE 4 The derivation of these rules is given in PD 6550-3, the explanatory supplement to BS 5500. The rules generally give a safety factor of 1.5 on the lower bound of experimental collapse results.

NOTE 5 Some worked examples for the design method are given in Annex W.

3.6.1.1 Notation

For the purposes of **3.6.2** to **3.6.7** the following major symbols apply. All dimensions are in the corroded condition (see **3.1.5**).

A	is the modified area of stiffener [see Equation (3.6.2-4)];
A_c	is the cross-sectional area of stiffener plus effective length of shell = (A_s + eL_e);
A_s	is the cross-sectional area of stiffener;
B	is a parameter in the interstiffener collapse equation [see Equation (3.6.2-5)];
b	is the width of stiffener in contact with shell;
C	is a parameter dependent on stiffener proportions (see Figure 3.6-6);
$\overline{d}$	is the distance between the centroid and the extremity of a stiffener [see Equation (3.6.2-16)];
d	is the radial height of stiffener between flanges, if any (see Figure 3.6-6);
E	is the modulus of elasticity of material of part under consideration at design temperature from Table 3.6-3;
e	is the analysis thickness of shell plate (see **1.6**);
e_f	is the analysis thickness of flange of stiffener section (see Figure 3.6-6) (see **1.6**);
e_w	is the analysis thickness of web of stiffener section (see Figure 3.6-6) (see **1.6**);
f, f_s	are the design stresses for shell and stiffener respectively;
G	is a parameter in the interstiffener collapse equation [see Table 3.6-1 or Equation (3.6.2-3)];
h	is the internal depth of a dished end;
I_c	is the second moment of area of the composite stiffener cross-section and effective length (L_e) of shell acting with it about axis parallel to cylinder axis passing through the centroid of the combined section [see Equation (3.6.2-12)];
I_s	is the second moment of area of stiffener cross-section about axis through centroid parallel to cylinder axis;
k	is a stiffener fabrication factor: = 1.8 for fabricated or hot formed stiffeners (i.e. low residual stress); = 2.0 for cold formed stiffeners (i.e. high residual stress);
L	is the effective unsupported length of shell (see Figure 3.6-1 or Table 3.6-2);
L_c	is the distance between heavy stiffeners (see Figure 3.6-1);
L_{con}	is the total length of the cone;
L_{cyl}	is the cylinder length between tangent lines (see Figure 3.6-1);
L_e	is the effective length of shell acting with stiffener (see Table 3.6-6);
L_e', L_e'', etc.	are components of L_e;
L_s	is the distance between light stiffeners (see Figure 3.6-1);
N	is a parameter of the interstiffener collapse equation [see Table 3.6-1 or Equation (3.6.2-2)];
n	is an integer used in stiffener design calculations;

n_{cyl} is the node number for the minimum buckling pressure (see Figure 3.6-3);

p is the design external pressure;

p_a is the calculated allowable external pressure;

p_e is the elastic instability pressure for collapse of spherical shell [see Equation (3.6.4-2)];

p_m is the elastic instability pressure for collapse of cylindrical shell [see Equation (3.6.2-8)];

p_{mc} is the elastic instability pressure for collapse of conical section between stiffeners [see Equation (3.6.3-2)];

p_n is the elastic instability pressure of stiffened shell [see Equation (3.6.2-11) for cylinder and Equations (3.6.3-3) or (3.6.3-6) for cone];

p_y is the pressure at which mean circumferential stress in cylindrical shell midway between stiffeners reaches yield point of material [see Equation (3.6.2-7)];

p_{yc} is the pressure at which mean circumferential stress in conical section between stiffeners reaches yield point of material [see Equation (3.6.3-1)];

p_{ys} is the pressure causing circumferential yield of stiffener [see Equation (3.6.2-14) for cylinders and Equation (3.6.3-4) for cones];

p_{yss} is the pressure at which membrane stress in spherical shell reaches the yield point of material [see Equation (3.6.4-1)];

R is the mean radius of cylindrical, conical or spherical shells or sections, or crown radius of torispherical ends;

R_f is the radius of standing flange of stiffener (see Figure 3.6-6);

R_s is the radius of centroid of ring stiffener cross-section (see Figure 3.6-6);

R_{max} is the maximum conical radius for a check on interstiffener collapse (see Figure 3.6-8);

$\overline{R}_{max}$ is the maximum conical radius for a check on overall collapse (see Figure 3.6-8);

R_{mean} is the mean conical radius for a check on interstiffener collapse (see Figure 3.6-8);

$\overline{R}_{mean}$ is the mean conical radius for a check on overall collapse (see Figure 3.6-8);

r is the mean knuckle radius of torispherical ends;

r_i is the inside radius of stiffener (see Figure 3.6-6);

s is the factor relating f to effective yield point of material; for the purposes of **3.6**, s shall be taken to be 1.4 for carbon, carbon manganese and ferritic alloy steels in material groups 1 to 6, 9 and 11, and 1.1 for austenitic steels in material group 8 and ferritic, martensitic or precipitation hardened stainless steels in material group 7[A)] ;

w_f is the outstanding width of flange of stiffener (see Figure 3.6-6);

w_i', w_i'' are part widths of stiffener in contact with the shell (see Figure 3.6-6);

X_c is a parameter in the calculation of elastic instability [see Equation (3.6.2-13)];

Z is a parameter for strain calculation [see Equation (3.6.2-10)];

Z' is a parameter in the approximate calculation of L_e (see Table 3.6-7);

α is a dimensional parameter [see Equation (3.6.2-1)];

β is a parameter for stiffened cylinders (see Figure 3.6-7);

Δ is a parameter used in calculating the required external design pressure (see Figure 3.6-4);

ε is the mean elastic circumferential strain at collapse [see Figure **3.6-2** or Equation (3.6.2-9)];

θ is the semi-angle to the axis of a conical shell;

λ	is a parameter = +1 for internal stiffeners, –1 for external stiffeners;
σ_s	is the maximum stress in a light stiffener [see Equation (3.6.2-15) for cylinders, or Equation (3.6.3-5) for cones];
ν	is Poisson's ratio (to be taken as 0.3);
γ	is a stiffener parameter [see Equation (3.6.2-6)].

[A] It is permitted for carbon, carbon manganese and ferritic alloy steels in material groups 1 to 6, 9 and 11 to take *sf* as $1.4R_e/1.5$ or $1.4R_{e(T)}/1.5$ (whichever is the lower) for applications and temperatures where time dependent properties (see **2.3.1.3**) not govern the values of *f*.

3.6.1.2 Definitions

For the purposes of **3.6.2** to **3.6.7** the following definitions apply.

3.6.1.2.1 heavy stiffener
dished end, girth flange, diaphragm or other similar substantial support which resists external collapse

3.6.1.2.2 light stiffener
ring, tee, angle or I-section which stiffens a shell to resist external collapse

3.6.1.2.3 interstiffener collapse
collapse of a section of shell between two stiffening rings, or between a stiffening ring and a shell end

3.6.1.2.4 overall collapse
collapse of a section of shell which includes a light or heavy stiffener

3.6.1.2.5 stiffener tripping
sideways twisting of a stiffener about its point of connection to the shell

3.6.2 Cylindrical shells

*NOTE Annex W illustrates, in **W.1.2.1** and **W.1.2.2**, calculations for cylinders and light stiffeners subject to external pressure.*

3.6.2.1 Cylinder thickness

The thickness of unstiffened cylinders, or cylindrical lengths between points of support, shall not be less than that determined by the following procedure. This procedure only applies to cylinders that are circular to within 0.5% on the radius measured from the true centre. A procedure to calculate the departure from the mean circle is given in **3.6.8**.

For cylinders outside this tolerance the allowable external pressure shall be calculated using the procedure given in Annex M.

a) Estimate a value for e, not less than that required for internal pressure (see **3.5**) and calculate p_y as follows.

$$\alpha = \frac{1.28}{\sqrt{Re}} \tag{3.6.2-1}$$

$$N = \frac{(\cosh \alpha L - \cos \alpha L)}{(\sinh \alpha L + \sin \alpha L)} \tag{3.6.2-2}$$

(see also Table 3.6-1)

L is determined from Figure 3.6-1 or Table 3.6-2

$$G = \frac{2\left(\sinh\frac{\alpha L}{2}\cos\frac{\alpha L}{2} + \cosh\frac{\alpha L}{2}\sin\frac{\alpha L}{2}\right)}{\sinh \alpha L + \sin \ \alpha L} \quad (3.6.2\text{-}3)$$

$$A = \frac{R^2 A_s}{{R_s}^2} \quad (3.6.2\text{-}4)$$

$$B = \frac{2eN}{\alpha(A + be)} \quad (3.6.2\text{-}5)$$

$$\gamma = \frac{A(1 - \nu/2)}{(A + be)(1 + B)}; \quad (3.6.2\text{-}6)$$

$= 0$ when no stiffeners are fitted

$$p_y = \frac{sfe}{R(1 - \gamma G)} \quad (3.6.2\text{-}7)$$

NOTE It is permissible to use the approximation γ = 0 to simplify the calculation of Equation (3.6.2-7) but this may lead to an underestimation of the allowable external pressure p_a.

b) Calculate p_m as follows using the same value for e assumed in calculating p_y.

$$p_m = \frac{Ee\varepsilon}{R} \quad (3.6.2\text{-}8)$$

The value of E is obtained from Table 3.6-3.

The value of ε is obtained directly from Figure 3.6-2 or calculated from:

$$\varepsilon = \frac{1}{{n_{cyl}}^2 - 1 + \frac{Z^2}{2}} \times \left\{ \frac{1}{\left(\frac{{n_{cyl}}^2}{Z^2} + 1\right)^2} + \frac{e^2}{12R^2(1 - \nu^2)}\left({n_{cyl}}^2 - 1 + Z^2\right)^2 \right\} \quad (3.6.2\text{-}9)$$

where

n_{cyl} is an integer ≥ 2 obtained from Figure 3.6-3 so as to minimize the value of p_m;

and

$$Z = \frac{\pi R}{L} \quad (3.6.2\text{-}10)$$

c) Calculate p_m/p_y and determine p_a/p_y from curve a) of Figure 3.6-4.

d) Calculate the allowable external pressure p_a. If this value is less than the design external pressure p, the assumed value of e shall be increased or the spacing of the stiffeners, if any, shall be adjusted until p_a is greater than or equal to p.

3.6.2.2 Stiffening rings for cylindrical shells, general requirements

Any stiffening rings assumed to act in the derivation of p_a shall conform to **3.6.2.2** and **3.6.2.3**.

NOTE The size of the light stiffeners[6)] (acting at L_s) necessary to comply with these requirements will depend significantly upon the use that is made of occasional heavy stiffeners or diaphragms (acting at L_c) to control the effective length and overall collapse of the stiffened cylinder.

Stiffening rings and other features used as stiffeners shall, where practicable, extend and be completely attached around the circumference; any joints shall be so designed as to develop the full stiffness of the ring (see also **3.10**). Stiffening rings arranged with local spaces between the shell and the ring, as shown in Figure 3.6-5, shall be subject to special consideration, but in no case shall the length of the unsupported shell plate exceed the value: (circumference)/($4n_{cyl}$), where n_{cyl} is derived from Figure 3.6-3.

Structural members used as stiffeners, which may be attached internally, externally or partially internally and externally, are detailed in Figure 3.6-6. Rings for supporting trays, etc. in fractioning columns and similar constructions may be used as stiffeners provided it can be demonstrated that they are adequate for the duty and that they conform to **3.6.2.3**.

Welds attaching stiffening rings to the shell shall be designed in accordance with **3.10**. Intermittent welds shall not be used where crevice corrosion may occur.

To ensure lateral stability and resistance against stiffener tripping, stiffeners shall conform to the following.

a) The proportions of stiffeners (other than flat bar stiffeners) shall be such that:

   1) $C > \dfrac{sf_s p}{Ep_{ys}}$

   2) for stiffeners flanged at the edge remote from the vessel shell:

   either $\dfrac{d}{e_w} \leq 1.1\sqrt{\dfrac{E}{sf_s}}$;

   or $\dfrac{d}{e_w} \leq 0.67\sqrt{\dfrac{Ep_{ys}}{sf_s p}}$;

   also either $\dfrac{w_f}{e_f} \leq 0.5\sqrt{\dfrac{E}{sf_s}}$;

   or $\dfrac{w_f}{e_f} \leq 0.32\sqrt{\dfrac{Ep_{ys}}{sf_s p}}$.

b) For flat bar stiffeners:

   $p < \dfrac{\sigma_e p_{ys}}{4sf_s}$, where σ_e is obtained from Table 3.6-4 or Table 3.6-5, depending

   on whether the stiffener is internal or external, using the value of n_{cyl} from Figure 3.6-3. To ensure that there is no local instability of the flat bar stiffener, the maximum permitted value of $(\sigma_e/E)(d/e_w)^2$ from Table 3.6-4 or Table 3.6-5 is 1.14 and the values of this expression in these tables shall not be extrapolated beyond that value.

3.6.2.3 Stiffener design

To resist overall collapse, the design of each stiffener shall be checked using the relevant procedures **3.6.2.3.1.1** to **3.6.2.3.2.2**.

A conservative method of assessment of stiffeners for horizontal vessels subject to external pressure and dead load is given in **G.3.3.4**.

[6] An approximate first estimate of the size of stiffener likely to be required can be obtained by designing each stiffener with a cross-section of 10% of the cross-section of the shell wall between stiffeners. A full calculation shall be subsequently carried out to check design adequacy.

3.6.2.3.1 Light stiffeners

*NOTE It is permissible to use the simplifications β = 0 and n = 2 in **3.6.2.3.1** but this will lead to a conservative design.*

3.6.2.3.1.1 Design against elastic instability

a) Calculate p_n for values of n = 2, 3, 4, 5 and 6 using:

$$p_n = \frac{Ee\beta}{R} + \frac{(n^2 - 1)}{R^3 L_s} EI_c \qquad (3.6.2\text{-}11)$$

where β is obtained from Figure 3.6-7; and

$$I_c = \frac{e^3 L_e}{3} + I_s + A_s[e/2 + \lambda(R - R_s)]^2 - A_c X_c{}^2 \qquad (3.6.2\text{-}12)$$

and

$$X_c = \left\{\frac{e^2}{2} L_e + A_s[e/2 + \lambda(R - R_s)]\right\} / A_c \qquad (3.6.2\text{-}13)$$

L_e is obtained as follows:

1) if, $\frac{L_s}{2\pi R} \leq 0.1$, from Table 3.6-6; or

2) if, $\frac{L_s}{2\pi R} > 0.1$, (and approximately for $0.07 \leq \frac{L_s}{2\pi R} \leq 0.1$) from

$$\frac{L_e}{L_s} = \frac{x}{100a^2} + \frac{(a - 0.1)Z'}{6.28a^2}$$

where

a $= \frac{L_s}{2\pi R}$;

x is the value of L_e/L_s from Table 3.6-6 at $\frac{L_s}{2\pi R}$ = 0.1;

Z' is the value from Table 3.6-7;

NOTE It is always safe to use the approximation L_e = Z′R to determine L_e.

L_s is obtained from Table 3.6-2

3) If $12 \times 10^{-7} \leq e^2/R^2 \leq 12 \times 10^{-4}$, it is equally valid to use the following formula. Whilst the formula is not exact, the degree of inaccuracy is of a similar order to that obtained when using the interpolation procedure required with Table 3.6-6. When $e^2/R^2 > 12 \times 10^{-4}$ it is permissible to obtain a value for L_e using the formula with the actual value of L_s/R and $e^2/R^2 = 12 \times 10^{-4}$.

$$L_e = \frac{A\sqrt{Re}}{\sqrt{\sqrt{1 + Bx^2} + Cx}}$$

where

<table>
<tr><th>For u =</th><th>A =</th><th>B =</th><th>C =</th></tr>
<tr><td>$u \leq 1$</td><td rowspan="3">$u/(1/1.098 + 0.03u^3)$</td><td>0</td><td rowspan="3">$0.6(1 - 0.27u)u^2$</td></tr>
<tr><td>$1 < u < 2.2$</td><td>$u - 1$</td></tr>
<tr><td>$2.2 \leq u \leq 2.9$</td><td rowspan="4">1.2</td></tr>
<tr><td>$2.9 < u < 4.1$</td><td>$1.2 + 1.642/u$</td><td rowspan="2">$0.75 + 1/u$</td></tr>
<tr><td>$4.1 \leq u < 5$</td><td rowspan="2">$1.556 + 0.183/u$</td></tr>
<tr><td>$u \geq 5$</td><td>$0.65 + 1.5/u$</td></tr>
</table>

when $x = n^2\left(\frac{e}{R}\right)$ and $u = \frac{L_s/R}{\sqrt{e/R}}$.

b) Check that for n = 2, 3, 4, 5 and 6, $p_n \geq kp$. If not, increase cylinder thickness or stiffening.

3.6.2.3.1.2 Stiffener maximum stress

a) Calculate p_{ys} using:

$$p_{ys} = \frac{sf_s e R_f}{R^2(1 - \nu/2)}\left(1 + \frac{A}{be + \frac{2Ne}{\alpha}}\right) \tag{3.6.2-14}$$

NOTE It is permissible to use the simplification A = 0 in Equation (3.6.2-14) but this will result in a larger stiffener section.

b) Calculate σ_s for n = 2 to n = 6 with the relevant value of p_n for each n, using:

$$\sigma_s = \frac{kpsf_s}{p_{ys}} + \frac{E\overline{d}}{R}\left\{\frac{(n^2 - 1)0.005kp}{p_n - kp}\right\} \tag{3.6.2-15}$$

where

$$\overline{d} = \max\{[\lambda(R - R_f) - X_c + e/2]; X_c\} \tag{3.6.2-16}$$

c) Check that for n = 2, 3, 4, 5 and 6, $sf_s \geq \sigma_s > 0$. If not increase cylinder thickness or stiffening.

3.6.2.3.2 Heavy stiffeners

3.6.2.3.2.1 Design against elastic instability

a) Calculate p_n using Equation (3.6.2-11), taking the first term as zero and n = 2. In the use of Table 3.6-6 to determine the value of L_e for heavy stiffeners, L_c shall be used instead of L_s. Where stiffeners are spaced unequal distances apart, L_e shall be taken as the average of two values of L_e, calculated using the lengths of bays on either side of the stiffener under consideration.

b) Check that $p_n \geq kp$. If not, increase cylinder thickness or stiffening.

3.6.2.3.2.2 Stiffener maximum stress

a) Calculate p_{ys} using Equation (3.6.2-14).

b) Calculate σ_s for n = 2 using Equation (3.6.2-15).

c) Check that $sf_s \geq \sigma_s > 0$. If not, increase cylinder thickness or stiffening.

3.6.3 Conical shells

The procedures specified in **3.6.3.1, 3.6.3.2** and **3.6.3.3** shall be used to determine the thickness of conical shells where the semi-angle at the apex $\theta \leq 75°$ and to check the dimensions of any associated stiffeners.

3.6.3.1 Conical thickness

The thickness of unstiffened cones, or conical lengths between points of stiffener support [see Figure 3.6-8a)], shall not be less than that determined by the following procedure.

a) Assume a value for e and calculate p_{yc} as follows:

$$p_{yc} = \frac{sfe\cos\theta}{R_{max}} \tag{3.6.3-1}$$

NOTE 1 This equation is the same as Equation (3.6.2-7), substituting ($e\cos\theta$) for e, R_{max} for R and taking $\gamma = 0$.

b) Calculate p_{mc} as follows:

$$p_{mc} = \frac{Ee\varepsilon \ \cos^3 \theta}{R_{mean}} \tag{3.6.3-2}$$

ε is determined from Figure 3.6-2 using [$L/(2R_{mean} \cos \theta)$] in place of [$L/(2R)$] and ($2R_{mean} \cos \theta/e$) in place of ($2R/e$).

NOTE 2 This equation is the same as Equation (3.6.2-8), substituting ($e\cos\theta$) for e, ($R_{mean}\cos^2\theta$) for R, ($\varepsilon \cos^4 \theta$) for ε and ($L\cos\theta$) for L.

c) Calculate p_{mc}/p_{yc} and determine p_a/p_{yc} from curve a) of Figure 3.6-4.

d) Calculate the allowable external pressure p_a. If this value is less than the design external pressure p, the assumed value of e shall be increased or the spacing of the stiffeners, if any, shall be adjusted until p_a is greater than or equal to p.

3.6.3.2 Stiffener design for conical shells

The requirements for stiffening ring proportions to resist stiffener tripping, given for cylinders in **3.6.2.2** shall apply without modification.

NOTE Internal stiffeners on conical shells are not covered.

3.6.3.2.1 Constant conical shell thickness, stiffener size and spacing [see Figure 3.6-8b)]

The alternative methods of assessment for stiffened cylinders in **3.6.2.3.1** or **3.6.2.3.2** shall apply except that p_n, p_{ys} and σ_s shall be determined from the following Equations (3.6.3-3), (3.6.3-4) and (3.6.3-5) respectively.

$$p_n = \frac{Ee\beta \ \cos^3 \theta}{\overline{R}_{mean}} + \frac{(n^2 - 1)EI_c' \ \cos \theta}{(\overline{R}_{max})^3 L_s} \tag{3.6.3-3}$$

where

β is determined from Figure 3.6-7 taking $\left[L_c \ / (2\overline{R}_{mean} \cos \ \theta)\right]$ in place of [$L_c/(2R)$];

I'_c is the second moment of area of composite cross-section including stiffener and effective length (L_e) of shell acting with it [see Figure 3.6-6e)].

For the purposes of evaluating I'_c, the effective length of each bay on either side of the stiffener under consideration shall be taken as one-half of L_e as derived from Table 3.6-6 taking:

$\dfrac{(e\ \cos\ \theta)^2}{12R_i^{\ 2}}$ in place of $\dfrac{e^2}{12R^2}$ and $\dfrac{L_s}{2\pi R_i\ \cos\ \theta}$ in place of $\dfrac{L_s}{2\pi R}$ where R_i is the cone mean radius measured in the plane of stiffener under consideration. The value of L_e shall then be obtained by taking the appropriate value of L_e/L_s from Table 3.6-6 and multiplying it by $L_s/\cos\theta$.

$$p_{ys} = \frac{sf_s e\cos\theta R_f}{\overline{R}_{max}{}^2(1-\nu/2)}\left[1+\frac{A\cos\theta}{be+2Ne/\alpha'}\right] \quad (3.6.3\text{-}4)$$

where $\alpha' = 1.28\sqrt{\dfrac{\cos\theta}{Re}}$

NOTE It is permissible to use the simplification A = 0 in Equation (3.6.3-4) but this will result in a larger stiffener section.

$$\sigma_s = \frac{kpsf_s}{p_{ys}} + \frac{E\overline{d}'}{\overline{R}_{max}}\left[\frac{(n^2-1)0.005kp}{p_n - kp}\right] \quad (3.6.3\text{-}5)$$

where $\overline{d}' = X_f + e_f/2$ [see Figure 3.6-6e)].

3.6.3.2.2 Varying conical shell thickness, stiffener size or spacing [see Figure 3.6-8c)]

The minimum shell thickness for any length between planes of substantial support shall be determined using **3.6.3.1**. It is permissible to use the alternative methods of assessment for stiffened cylinders in **3.6.2.3.1** or **3.6.2.3.2** with Equation (3.6.3-3), Equation (3.6.3-4) and Equation (3.6.3-5) with any of the following.

a) Where the stiffener pitch and size is constant, use the minimum thickness anywhere along the length of the section under consideration [i.e. e_1 in Figure 3.6-8c)] in calculating p_n and p_{ys}, take I'_c as defined in c).

b) Consider each stiffener separately using the appropriate minimum shell thickness and R_{max} for the two half bays on either side of the stiffener and $\beta = 0$ [i.e. ignoring the first term in Equation (3.6.3-3) in the calculation of p_n].

c) Consider each stiffener separately using the appropriate minimum shell thickness and R_{max} for the two half bays on either side of the stiffener.

Where $n > 2$, calculate p_n as in b) i.e. with $\beta = 0$, and where $n = 2$, calculate p_n from the following equation:

$$p_n = \frac{E\overline{e}\beta\ \cos^3\theta}{R_{mean}} + \frac{2E\ \cos\ \theta(n^2-1)}{L_c} \times \sum_{i=0}^{i=N-1} \frac{I'_{ci}\ \sin^2\theta\left[\dfrac{\pi X_i}{L_c}\right]}{R_i^3} \quad (3.6.3\text{-}6)$$

where

$\overline{e}$ is the minimum thickness in total cone length;

β is determined from Figure 3.6-7 taking $[L_c/(2R_{mean}\cos\theta)]$ in place of $[L_c/(2R)]$;

R_i is the cone mean radius in plane of stiffener under consideration at axial distance X_i from small end of the cone;

N is the number of bays between light stiffeners in length L_c, including the bays between the ends of the cone and the first and last stiffeners;

I'_{ci} is the combined second moment of area of stiffener and shell at axial distance X_i from the small end of the cone using L_e as determined in **3.6.2.3.2** and taking values for e separately for each bay.

3.6.3.3 Cone/cylinder intersections

The junction of a cylinder to cone intersection subject to external pressure shall be checked to the requirements of **3.6.3.3.1**, **3.6.3.3.2** and **3.6.3.3.3**. If a torispherical knuckle is used at this junction, the same requirements shall be checked taking the lengths of the cone and cylinder as measured from their projected intersection (see Note 1 to **3.5.3.4**).

3.6.3.3.1 Planes of substantial support

The intersection between a cone and a cylinder (at both large and small ends) shall be considered a plane of substantial support if:

a) $\theta \geq 30°$; and

b) n_{cyl} [the node number for the minimum buckling pressure obtained from Figure 3.6-3, or when light stiffeners are present by applying Equation (3.6.2-11)] does not equal 2 for either cone or cylinder.

If both of the above conditions are met, the distance L between planes of substantial support, shall be considered to extend up to the intersection of the cone and cylinder and it follows that the cone and cylinder are treated separately.

If either of the above conditions are met, the distance L between planes of substantial support, shall be the sum of the effective unsupported length(s) of the cylinder(s) plus the axial length of the cone [see Figure 3.6-1b)]. The thickness of the cone and the small cylinder shall be not less than the small end cylinder thickness required by **3.6.2.1**. If there are light stiffeners they shall be applied to the cone and small cylinder as well as to the large cylinder, at the pitch and size determined in **3.6.3.2**.

3.6.3.3.2 Reinforcement of large end intersection

It is not necessary to provide additional thickening or local stiffening at this intersection, for external pressure considerations.

3.6.3.3.3 Reinforcement of small end intersection

Reinforcement in the form of additional thickening and/or local stiffening shall be provided, if necessary, to keep the maximum local hoop stress at the small end of the cone within acceptable limits, using the following procedure.

a) Calculate p_a, the allowable external pressure on the small end cylinder, from **3.6.2.1**. In using **3.6.2.1**, R and e are for the small end cylinder and the value of L depends on whether the junction is a plane of substantial support. If the conditions for substantial support at the small end, as given in **3.6.3.3.1**, are met then L is the length of the small end cylinder. If the conditions are not met, L is the length of this cylinder plus the length of the cone; in addition if the conditions for substantial support at the large end are not met then the length of the large end cylinder up to the first plane of substantial support shall be added to L.

b) Calculate p_y, the pressure at which the mean hoop stress in the cylinder reaches yield point, using Equation (3.6.2-7). Values for R, e and L are taken as in a) above.

c) Using a suitable stress analysis technique, calculate σ_j the maximum mean hoop stress at the junction due to pressure p.

d) Calculate p_{yj}, the pressure at which the mean hoop stress at the junction reaches yield point, from Equation (3.6.3-7)

$$p_{yj} = \frac{psf}{\sigma_j} \quad (3.6.3\text{-}7)$$

e) If $p_{yj} \geq p_y$ then no additional reinforcement is required.

f) If $p_{yj} < p_y$ increase the local stiffening and/or the cone/cylinder thicknesses and repeat the calculation from step a) until e) is satisfied. If the cylinder thickness has been increased it might be less conservative to recalculate p_a using p_{yj} in place of p_y in step c) of **3.6.2.1**.

3.6.4 Spherical shells

The thickness of a spherical shell shall be not less than that given by the following procedure.

a) Assume a value for e and calculate p_{yss} as follows:

$$p_{yss} = \frac{2sfe}{R} \quad (3.6.4\text{-}1)$$

b) Calculate p_e as follows (using the same value for e assumed in calculating p_{yss}).

$$p_e = \frac{1.21Ee^2}{R^2} \quad (3.6.4\text{-}2)$$

c) Calculate p_e/p_{yss} and determine p_a/p_{yss} from curve b) of Figure 3.6-4.

d) Calculate the allowable external pressure p_a. If this value is less than the design external pressure p, the assumed value of e shall be increased until p_a is greater than or equal to p.

The design curve in Figure 3.6-4 applies only to spheres that are spherical to within 1% on the radius and in which the radius of curvature, measured over an arc length of $2.4\sqrt{R_{max}e}$, does not exceed the nominal value by more than 30%.

NOTE Enquiry Case 5500/33 gives guidance on verification of shape of vessels subject to external pressure.

For applications in which this criterion for applicability cannot be met, possibly due to difficulties of manufacture and measurement, it is permissible to divide the pressure obtained from the above procedure by the factor $(R_{max}/[1.3R])^2$ where R_{max} is the maximum local radius of curvature either measured or estimated conservatively.

The thickness of a cylindrical or straight flange (see Figure 3.10-1 and Figure 3.10-2) of a spherical, or dished end, shall conform to **3.6.2** unless the length of the flange is less than $0.5\sqrt{De}$ in which case it may be the same thickness as the dished end. D is defined as the outside diameter of the end and e as the analysis thickness of the end.

3.6.5 Hemispherical ends

Hemispherical ends shall be designed as for spherical shells.

3.6.6 Torispherical ends

Torispherical ends shall be designed as spherical shells of mean radius R equal to the external dishing or crown radius. The shape limitations in **3.5.2** shall apply.

3.6.7 Ellipsoidal ends

Ends to true semi-ellipsoidal form shall be designed as spherical shells of mean radius R equal to the maximum radius of the crown, i.e. $D^2/(4h)$. The shape limitations in **3.5.2** shall apply.

3.6.8 Procedure by which the departure from the mean circle may be obtained (see 3.6.2)

Where difficulty is experienced in determining the departure from the mean circle by more direct methods, use of the following procedure is permitted. Radii are measured at 24 equally spaced intervals around the circumference. This can be done either by swinging an arm internally or by external measurements with the cylinder mounted in a lathe. It is necessary to rotate the internal arm or cylinder about an axis near to the true centre of circularity of the section under consideration[7].

NOTE Enquiry Case 5500/33 gives guidance on the verification of the shape of vessels subject to external pressures.

The radial measurements need to be corrected for the mean and for the error in positioning the centre. This is done by finding the coefficients b_0, a_1 and b_1 in the Fourier series expansion:

$$R_\mathrm{r} = b_0 + a_1 \sin r\varphi + b_1 \cos r\varphi + \sum_{r=2}^{\infty}(a_n \sin rn\varphi + b_n \cos rn\varphi) \tag{3.6.8-1}$$

where R_r are the radial measurements from the assumed centre at location r.

For 24 equally spaced measurements:

r = 0, 1, 2, ... 23

φ = 15°; the increment angle of the measurements, and:

$$b_0 = \frac{1}{24}\sum_{r=0}^{r=23} R_\mathrm{r} \tag{3.6.8-2}$$

$$a_1 = \frac{1}{12}\sum_{r=0}^{r=23} R_\mathrm{r} \sin r\varphi \tag{3.6.8-3}$$

$$b_1 = \frac{1}{12}\sum_{r=0}^{r=23} R_\mathrm{r} \cos r\varphi \tag{3.6.8-4}$$

The departure from the main circle at any point r is ε_r,

where:

$$\varepsilon_r = R_\mathrm{r} - b_0 - a_1 \sin\ r\varphi - \cos r\varphi \tag{3.6.8-5}$$

Table 3.6-1 **Values for *G* and *N***

αL	*G*	*N*	αL	*G*	*N*
0	1.000	0	3.2	0.411	1.090
0.2	1.000	0.100	3.4	0.335	1.085
0.4	1.000	0.200	3.6	0.264	1.077
0.6	0.999	0.300	3.8	0.200	1.066
0.8	0.996	0.400	4.0	0.144	1.054
1.0	0.990	0.497	4.2	0.095	1.042
1.2	0.979	0.593	4.4	0.054	1.032
1.4	0.961	0.685	4.6	0.019	1.023
1.6	0.935	0.772	4.7	0.004	1.019
1.8	0.899	0.851	(4.73)	0.000	1.018
2.0	0.852	0.921	4.8	0.000	1.015

[7] See also Enquiry Case 5500/33.

Table 3.6-1 **Values for *G* and *N* (continued)**

αL	G	N	αL	G	N
2.2	0.795	0.979	5.0	0.000	1.009
2.4	0.728	1.025	5.2	0.000	1.005
2.6	0.653	1.058	5.4	0.000	1.001
2.8	0.573	1.078	5.5	0.000	1.000
3.0	0.492	1.088	>5.5	0.000	1.000

Table 3.6-2 **Definitions of cylinder lengths**

Cylinder with light stiffeners	Cylinder with light and heavy stiffeners
For each bay separately, see **3.6.2.1**; $L = (L'_s - w''_1) + 0.4h'$; or $L = L''_s - w'_1 - w''_2$	For each bay separately, see **3.6.2.1**; $L = (L'_s - w''_1) + 0.4h'$; or $L = L''_s - w'_1 - w''_2$; or $L = L'''_s - w'_2$
For each light stiffener separately, see **3.6.2.3.1.1**; $L_s = (L'_s + 0.4h' + L''_s)/2$; or $L_s = (L''_s + L'''_s)/2$	For each light stiffener separately, see **3.6.2.3.1.1**; $L_s = (L'_s + 0.4h' + L''_s)/2$; or $L_s = (L''_s + L'''_s)/2$
For the purpose of evaluating β, see **3.6.2.3.1.1**; $L_c = L_{cyl} + 0.4h' + 0.4h''$	For the purpose of evaluating β, see **3.6.2.3.1.1**; $L_c = L'_c + 0.4h'$; or $L_c = L''_c$; or $L_c = L'''_c$
	For each heavy stiffener separately, see **3.6.2.3.2.1**; $L_c = (L'_c + 0.4h' + L''_c)/2$; or $L_c = (L''_c + L'''_c)/2$

NOTE 1 Where alternative cylinder lengths are given, the greatest value shall be used in the relevant calculations.

NOTE 2 Dimensions h', L'_s, L''_s, etc. are indicated in Figure 3.6-1 and w''_1, w'_1, w''_2, w'_2 are indicated in Figure 3.6-6h).

Table 3.6-3 ***E* values for ferritic and austenitic steels (modulus of elasticity)**

Temperature °C	Ferritic steels in material groups 1 to 6, 7.1, 9 and 11 N/mm^2	Austenitic steels in material group 8 N/mm^2
0	210 000	203 000
20	209 000	200 000
100	205 000	194 000
150	202 000	190 000
200	199 000	185 000
250	195 000	181 000
300	191 000	176 000
350	187 000	172 000
400	181 000	168 000
450	178 000	164 000
500	174 000	159 000
550	168 000	155 000
600	162 000	151 000
650	—	147 000
700	—	142 000
750	—	138 000
800	—	134 000

NOTE Values of E are not suitable for temperatures exceeding those given in Table 2.3-1.

Table 3.6-4 **Values of $(\sigma_e/E)(d/e_w)^2$ for internal flat bar stiffeners**

n_{cyl}	d/R										
	0.01	**0.02**	**0.04**	**0.06**	**0.08**	**0.10**	**0.12**	**0.14**	**0.16**	**0.18**	**0.20**
2	0.0119	0.0236	0.0466	0.0691	0.0913	0.114	0.135	0.157	0.180	0.202	0.225
3	0.0239	0.0461	0.0865	0.123	0.156	0.187	0.217	0.247	0.276	0.305	0.334
4	0.0395	0.0734	0.130	0.176	0.216	0.252	0.286	0.319	0.353	0.386	0.421
5	0.0577	0.103	0.171	0.223	0.266	0.304	0.341	0.378	0.416	0.456	0.498
6	0.0778	0.132	0.208	0.262	0.306	0.347	0.387	0.428	0.472	0.517	0.570
7	0.0981	0.160	0.240	0.294	0.340	0.382	0.427	0.474	0.527	0.580	0.643
8	0.119	0.186	0.268	0.322	0.369	0.415	0.465	0.517	0.580	0.647	0.725
9	0.139	0.210	0.290	0.345	0.394	0.445	0.502	0.565	0.638	0.720	0.812
10	0.158	0.231	0.310	0.365	0.417	0.474	0.536	0.614	0.696	0.792	0.903
11	0.176	0.249	0.328	0.383	0.440	0.502	0.575	0.662	0.758	0.874	1.010
12	0.193	0.266	0.343	0.400	0.461	0.531	0.614	0.715	0.831	0.966	1.121
13	0.209	0.280	0.356	0.416	0.483	0.560	0.657	0.768	0.903	1.058	
14	0.224	0.293	0.368	0.431	0.502	0.594	0.700	0.831	0.981		
15	0.237	0.304	0.379	0.446	0.527	0.628	0.749	0.894	1.068		
16	0.249	0.314	0.389	0.461	0.551	0.662	0.797	0.961			
17	0.260	0.324	0.399	0.476	0.575	0.696	0.850	1.034			
18	0.270	0.332	0.409	0.493	0.599	0.734	0.903	1.106			
19	0.279	0.339	0.418	0.507	0.623	0.773	0.961				
20	0.287	0.346	0.427	0.522	0.652	0.816	1.019				

*NOTE 1 For the maximum permitted value of $(\sigma_e/E)(d/e_w)^2$ see **3.6.2.2**b).*

NOTE 2 For intermediate values of d/R, use logarithmic interpolation.

EXAMPLE For $n_{cyl} = 2$, the value of $(\sigma_e/E)(d/e_w)^2$ is required for d/R = 0.05.

Then:

$$(\sigma_e/E)(d/e_w)^2 = \text{antilog}\left\{\log(0.0466) + [\log(0.0691) - \log(0.0466)]\left[\frac{0.05 - 0.04}{0.06 - 0.04}\right]\right\} = 0.0567.$$

Table 3.6-5 **Values of (σ_e/E) (d/e_w)^2 for external flat bar stiffeners**

n_{cyl}	*d/R*																	
	0.01	**0.011**	**0.012**	**0.015**	**0.02**	**0.025**	**0.03**	**0.04**	**0.045**	**0.05**	**0.06**	**0.08**	**0.10**	**0.12**	**0.14**	**0.16**	**0.18**	**0.20**
2	0.012	0.0132	0.0144	0.0180	0.0241	0.0303	0.0366	0.0492	0.0557	0.0622	0.0755	0.103	0.133	0.164	0.198	0.236	0.277	0.324
3	0.0257	0.0284	0.0311	0.0374	0.0537	0.0687	0.0846	0.119	0.138	0.157	0.201	0.310	0.462	0.695	1.10	1.99[a]		
4	0.0466	0.0517	0.0570	0.0734	0.103	0.137	0.175	0.268	0.326	0.395	0.581	1.44[a]						
5	0.0768	0.086	0.0955	0.126	0.187	0.263	0.361	0.679	0.965	1.46[a]								
6	0.120	0.136	0.153	0.211	0.340	0.537	0.881	1.44[a]										
7	0.183	0.211	0.242	0.356	0.677	1.48[a]												
8	0.279	0.331	0.390	0.648	1.92[a]													
9	0.438	0.541	0.676	1.49[a]														
10	0.736	0.998	1.420[a]															
11	1.49[a]																	

NOTE 1 Buckling cannot occur for $n_{cyl} > 10$, $d/R > 0.01$ under external pressure.

*NOTE 2 For the maximum permitted value of (σ_e/E) (d/e_w)^2 see **3.6.2.2**b).*

NOTE 3 For intermediate values of d/R, use logarithmic interpolation. For an example, see Table 3.6-4.

[a] These values are provided to enable intermediate values to be interpolated.

Table 3.6-6 **Values of L_e/L_s**

$\frac{e^2}{12R^2} \geq 10^{-4}$					
$\frac{L_s}{2\pi R}$	***n***				
	2	**3**	**4**	**5**	**6**
0	1.0980	1.0980	1.0980	1.0980	1.0980
0.01	1.0823	1.0823	1.0663	1.0663	1.0504
0.02	1.0663	1.0504	1.0265	0.9947	0.9629
0.03	1.0504	1.0027	0.9549	0.9019	0.8435
0.04	0.9907	0.9231	0.8515	0.7838	0.7082
0.05	0.8976	0.8276	0.7512	0.6716	0.5952
0.06	0.7921	0.7298	0.6609	0.5871	0.5143
0.07	0.6866	0.6321	0.5707	0.5025	0.4343
0.08	0.6111	0.5630	0.5088	0.4480	0.3877
0.09	0.5355	0.4940	0.4470	0.3935	0.3410
0.1	0.4600	0.4249	0.3852	0.3390	0.2944
$\frac{e^2}{12R^2} = 10^{-5}$					
$\frac{L_s}{2\pi R}$	*n*				
	2	**3**	**4**	**5**	**6**
0	1.0980	1.0980	1.0980	1.0980	1.0980
0.01	1.0823	1.0823	1.0663	1.0663	1.0504
0.02	1.0345	1.0186	0.9947	0.9629	0.9311
0.03	0.9019	0.8807	0.8541	0.8117	0.7639
0.04	0.7242	0.7003	0.6724	0.6326	0.5929
0.05	0.5602	0.5411	0.5220	0.4934	0.4647
0.06	0.4483	0.4350	0.4218	0.4005	0.3793
0.07	0.3752	0.3661	0.3547	0.3388	0.3206
0.08	0.3263	0.3163	0.3084	0.2964	0.2805
0.09	0.2920	0.2847	0.2775	0.2660	0.2525
0.1	0.2578	0.2531	0.2467	0.2355	0.2244

Table 3.6-6 **Values of L_e/L_s continued**

$$\frac{e^2}{12R^2} = 10^{-6}$$

$\frac{L_s}{2\pi R}$	n = 2	n = 3	n = 4	n = 5	n = 6
0	1.0980	1.0980	1.0980	1.0980	1.0980
0.01	1.0663	1.0504	1.0504	1.0504	1.0345
0.02	0.8276	0.8196	0.8037	0.7878	0.7719
0.03	0.5252	0.5199	0.5146	0.5040	0.4934
0.04	0.3740	0.3700	0.3661	0.3621	0.3541
0.05	0.2960	0.2928	0.2897	0.2865	0.2801
0.06	0.2661	0.2632	0.2604	0.2575	0.2521
0.07	0.2362	0.2336	0.2311	0.2285	0.2241
0.08	0.2063	0.2040	0.2018	0.1996	0.1961
0.09	0.1763	0.1744	0.1725	0.1706	0.1681
0.1	0.1464	0.1448	0.1432	0.1416	0.1401

$$\frac{e^2}{12R^2} = 10^{-7}$$

$\frac{L_s}{2\pi R}$	n = 2	n = 3	n = 4	n = 5	n = 6
0	1.0980	1.0980	1.0980	1.0980	1.0980
0.01	0.9072	0.9072	0.8913	0.8913	0.8913
0.02	0.4297	0.4297	0.4218	0.4218	0.4218
0.03	0.2759	0.2759	0.2759	0.2759	0.2759
0.04	0.2207	0.2207	0.2207	0.2191	0.2191
0.05	0.1655	0.1655	0.1655	0.1623	0.1623
0.06	0.1490	0.1487	0.1487	0.1461	0.1461
0.07	0.1324	0.1318	0.1318	0.1299	0.1299
0.08	0.1159	0.1149	0.1149	0.1136	0.1136
0.09	0.0993	0.0980	0.0980	0.0974	0.0974
0.1	0.0828	0.0812	0.0812	0.0812	0.0812

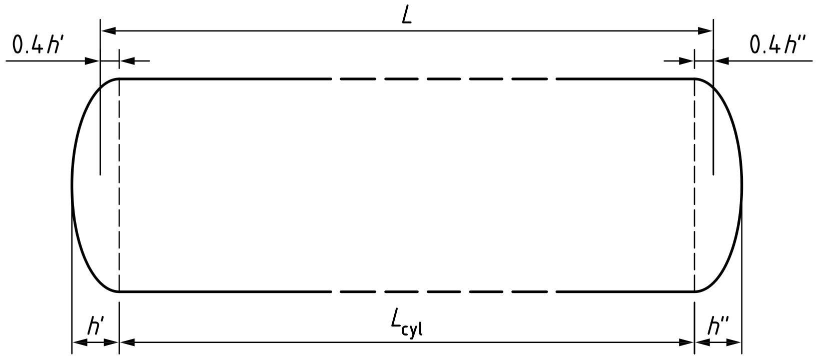
NOTE 1 For intermediate values of $\frac{e^2}{12R^2}$ use logarithmic interpolation for constant values of n.

NOTE 2 For intermediate values of $\frac{L_s}{2\pi R}$ use linear interpolation.

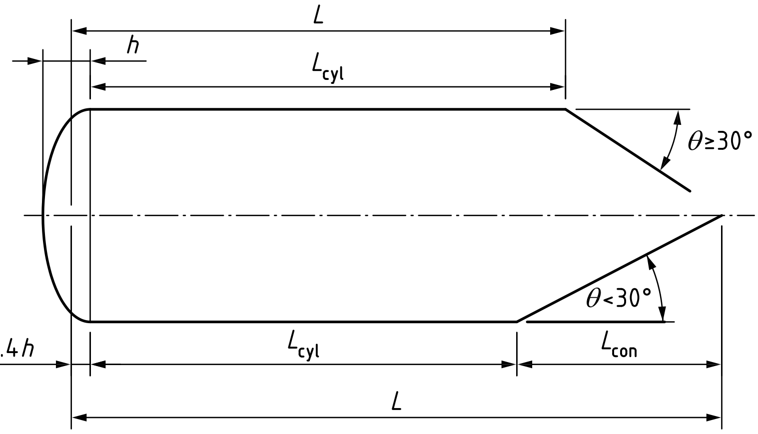
Table 3.6-7 **Values of Z'**

$\frac{e^2}{12R^2}$	n = 2	n = 3	n = 4	n = 5	n = 6
$\geq 10^{-4}$	0.273	0.257	0.235	0.207	0.180
10^{-5}	0.159	0.157	0.154	0.147	0.140
10^{-6}	0.091	0.090	0.090	0.089	0.087
10^{-7}	0.051	0.051	0.051	0.051	0.051

Figure 3.6-1 **Effective lengths of cylinder**



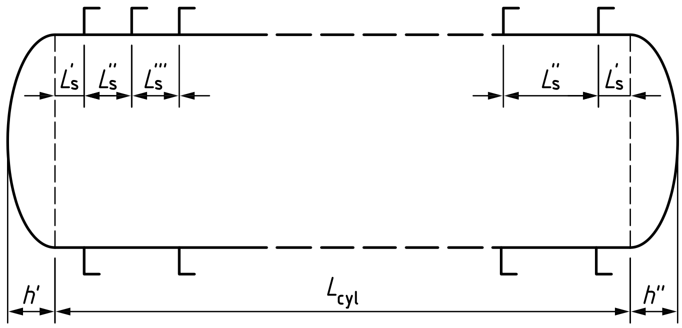
a) Cylinders with dished ends



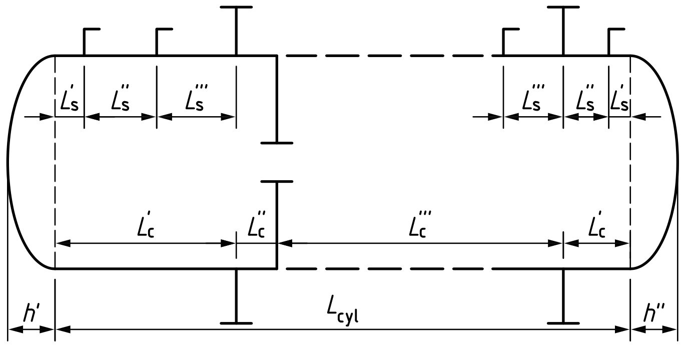
NOTE For cone/cylinder intersections see ***3.6.3.3***

b) Cylinder with dished end and cone

Figure 3.6-1 **Effective lengths of cylinder *(continued)***



c) Cylinder with light stiffeners



d) Cylinder with light and heavy stiffeners

Figure 3.6-2 **Values of ε**

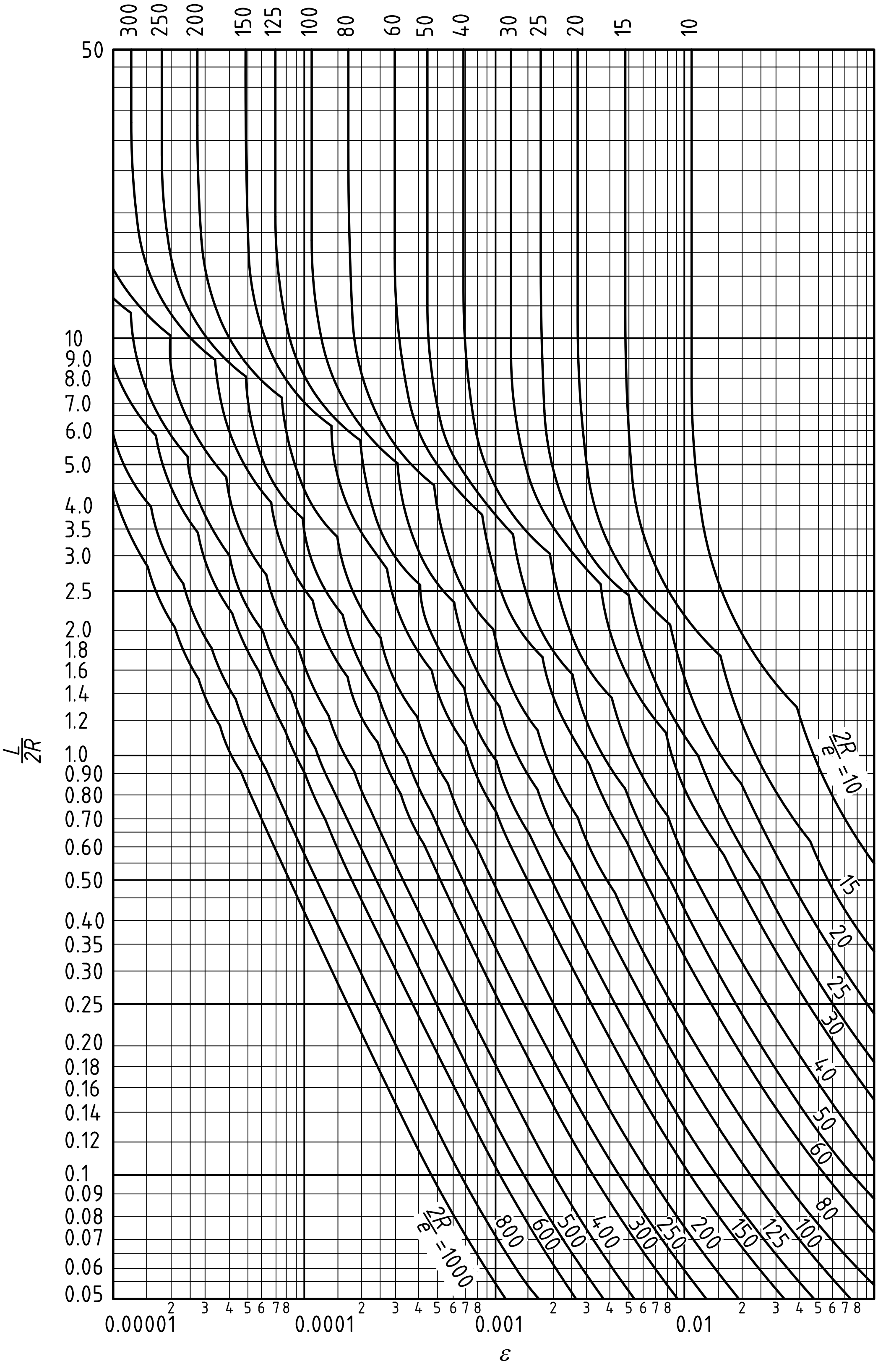
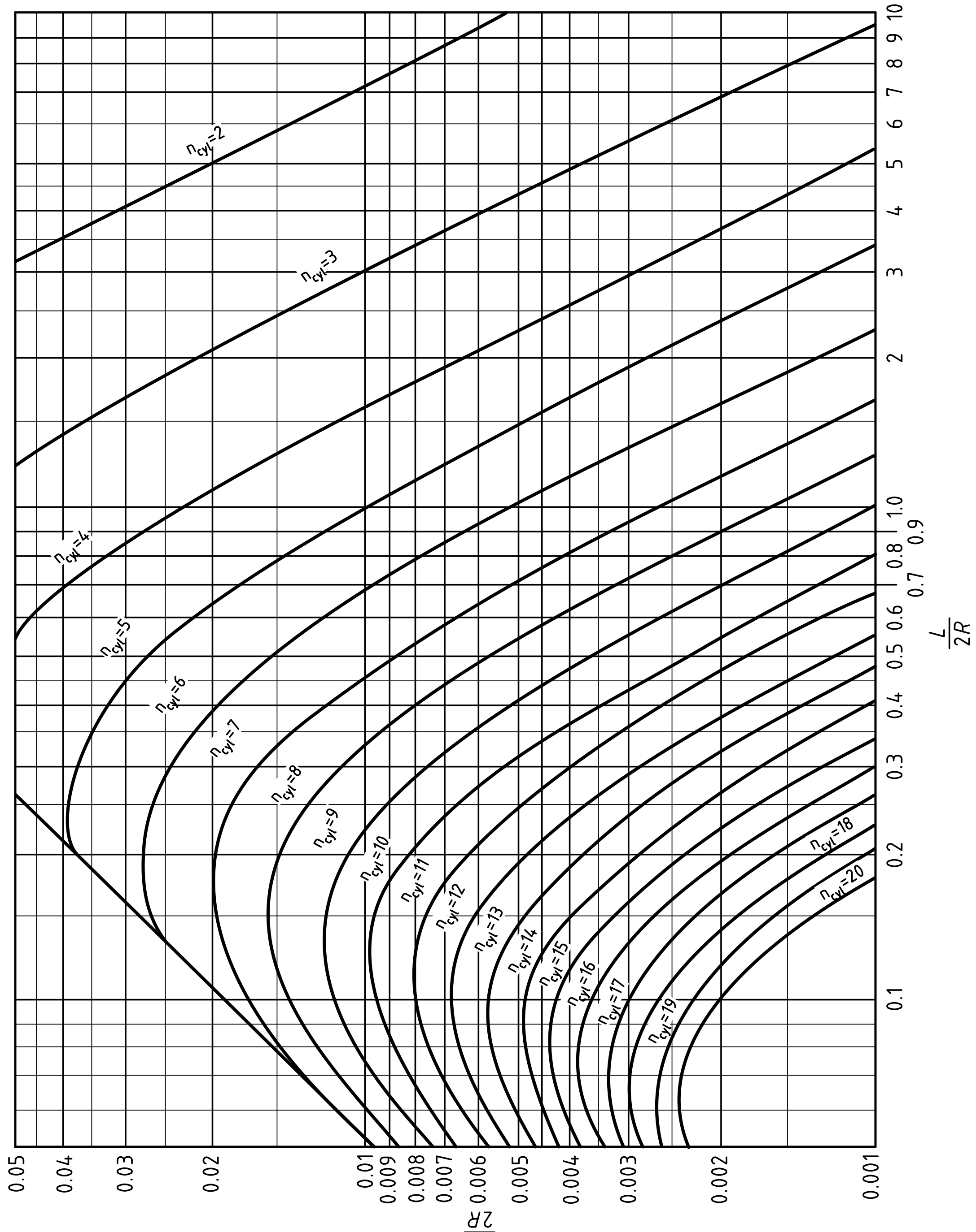


Figure 3.6-3 **Values of n_{cyl}**



NOTE The value of n_{cyl} corresponding to the closest line should be used and, in cases of doubt, both values of n_{cyl} should be considered.

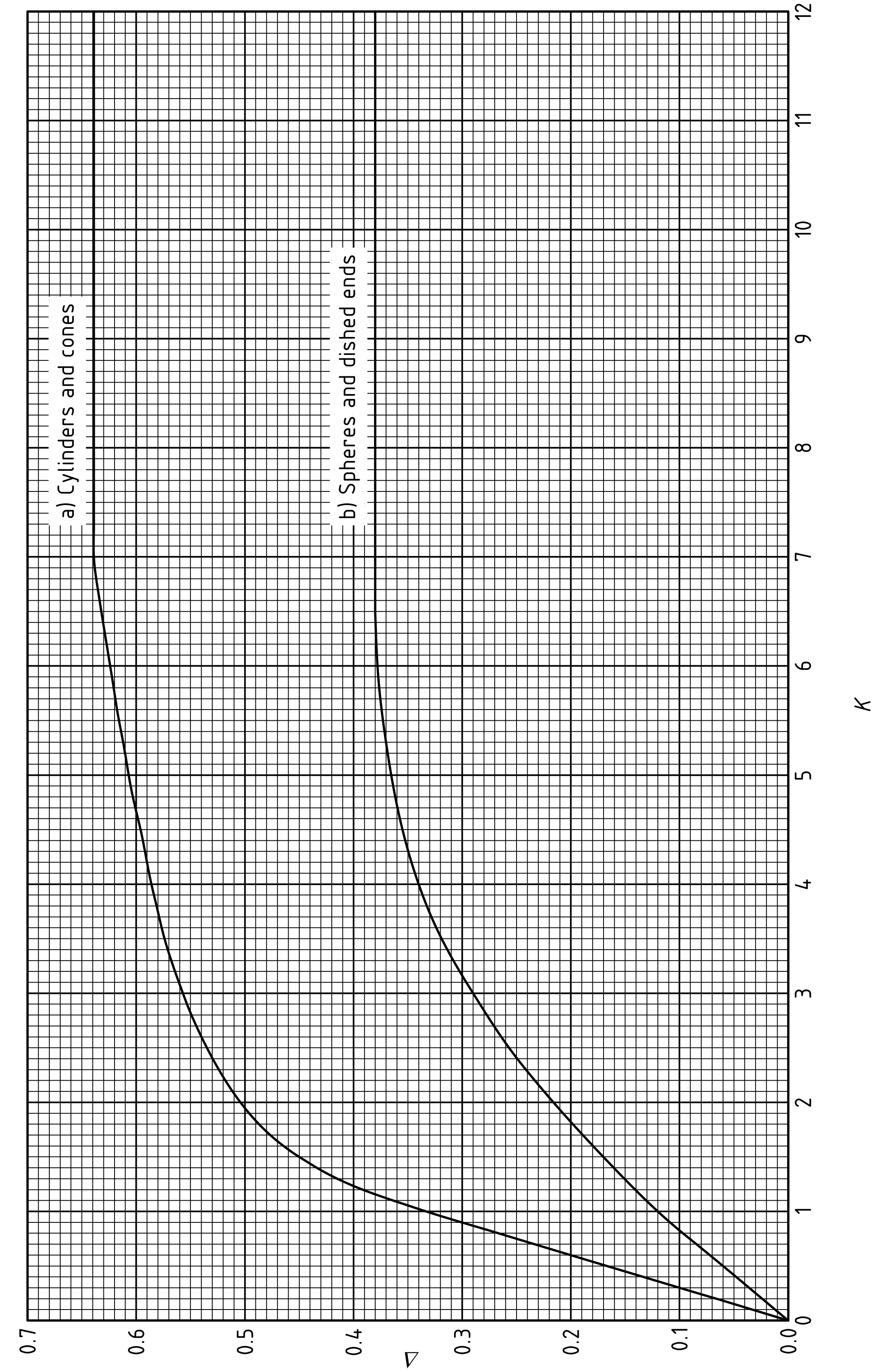


Figure 3.6-4 **Values of Δ**

Figure 3.6-4 **Values of *Δ* (continued)**

Key to symbols *K* and *Δ*

Application	***Δ***	***K***
Cylindrical shells (**3.6.2.1**)	p_a/p_y	p_m/p_y
Conical shells (**3.6.3.1**)	p_a/p_{yc}	p_{mc}/p_{yc}
Spherical shells (**3.6.4**)	p_a/p_{yss}	p_e/p_{yss}

a) Cylinders and cones (hoop stress governing)

K	*0.25*	*0.5*	*0.75*	*1.0*	*1.25*	*1.5*	*1.75*	*2.0*	*2.25*	*2.5*	*2.75*	*3.0*	*3.25*	*3.5*
Δ	0.083	0.167	0.25	0.333	0.403	0.453	0.480	0.503	0.520	0.535	0.548	0.557	0.566	0.574
K	3.75	4.0	4.25	4.5	4.75	5.0	5.25	5.5	5.75	6.0	6.25	6.5	6.75	7.0 and above
Δ	0.580	0.586	0.591	0.597	0.603	0.609	0.611	0.615	0.619	0.623	0.627	0.631	0.635	0.639

b) Spheres and dished ends, cylinders and cones (longitudinal stress governing)

K	0	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5 and above
Δ	0	0.06	0.12	0.17	0.216	0.257	0.29	0.319	0.340	0.355	0.365	0.373	0.378	0.38

Figure 3.6-5 **Stiffening ring with unsupported section**

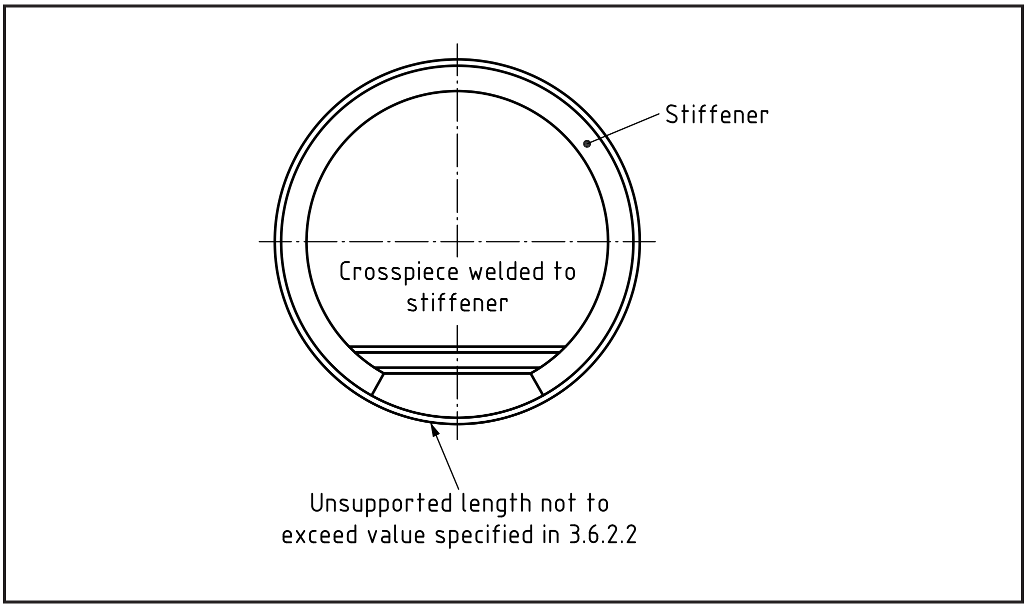
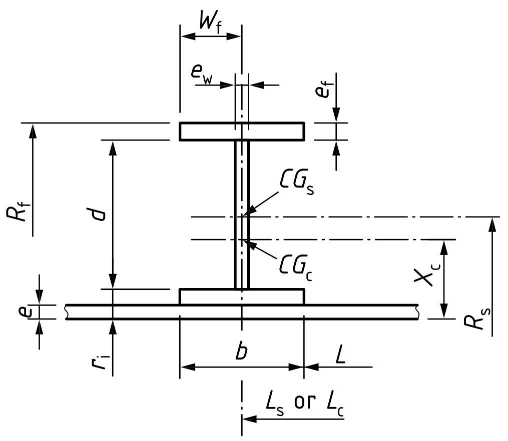
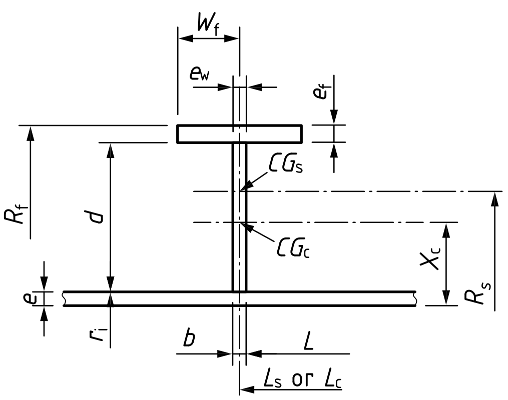


Figure 3.6-6 **Stiffening ring details**

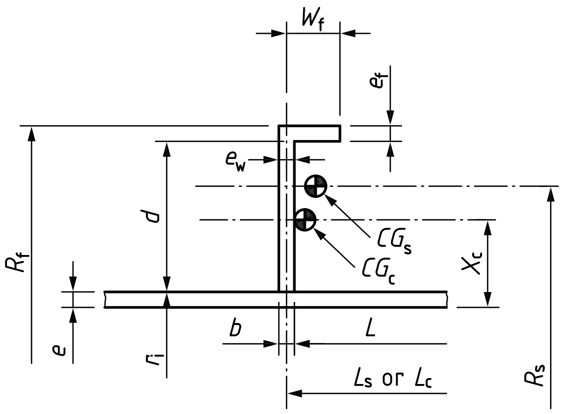


a) External stiffener with symmetrical flanges at both ends of web.

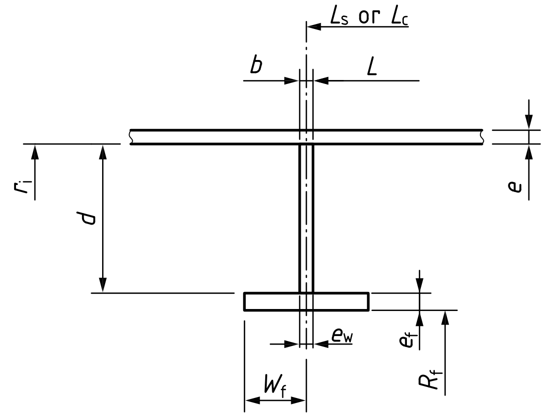


b) External stiffener with symmetrical flange at one end of web.

Figure 3.6-6 **Stiffening ring details *(continued)***



c) External stiffener with asymmetrical flange at one end of web.



d) Internal stiffener with symmetrical flange at one end of web.

For stiffeners a), b) and d) $$C = \frac{de_w^3 + 8e_f w_f^3}{r_i\left[6d^2 e_w + 12e_f w_f(2d + e_f)\right]}$$

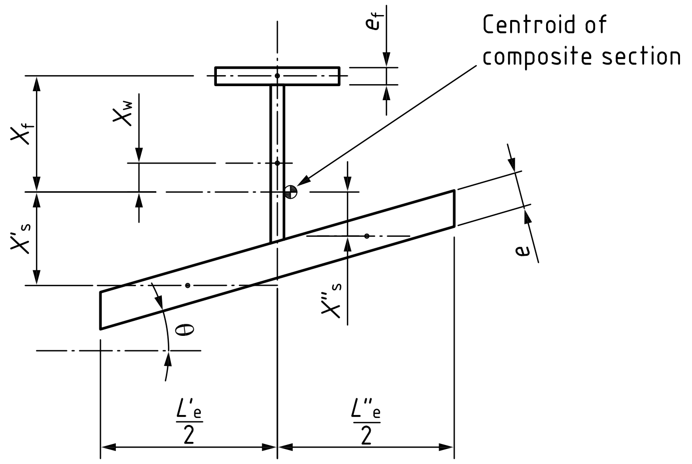
For stiffener c) $$C = \frac{e_f w_f^3}{r_i\left[6d^2 e_w + 6e_f w_f(2d + e_f)\right]} \times \left[\frac{4de_w + 3w_f e_f}{de_w + 3w_f e_f}\right]$$

CG_s: centroid of stiffener

CG_c: centroid of stiffener plus effective length of shell – L_e

For unequal spacing of stiffeners, L_e shall be taken as the average value using Table 3.6-1 for the two adjacent bays.

Figure 3.6-6 **Stiffening ring details *(continued)***



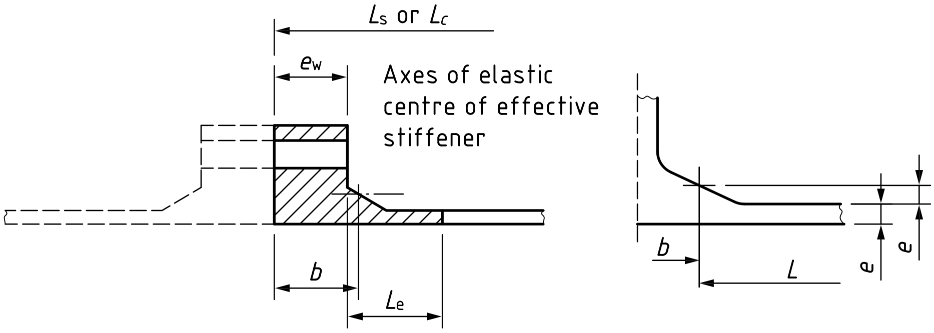
e) Structural members

$$I_c = A_f X_f^2 + A_w X_w^2 + \left(\frac{eL'_e}{2}\right)(X'_s)^2 + \left(\frac{eL''_e}{2}\right)(X''_s)^2 + I_f + I_w + \frac{e}{12}\sin^2\theta\left[\left(\frac{L'_e}{2}\right)^3 + \left(\frac{L''_e}{2}\right)^3\right] + \frac{e^3}{12}\cos^2\theta\left(\frac{L'_e}{2} + \frac{L''_e}{2}\right)$$

where

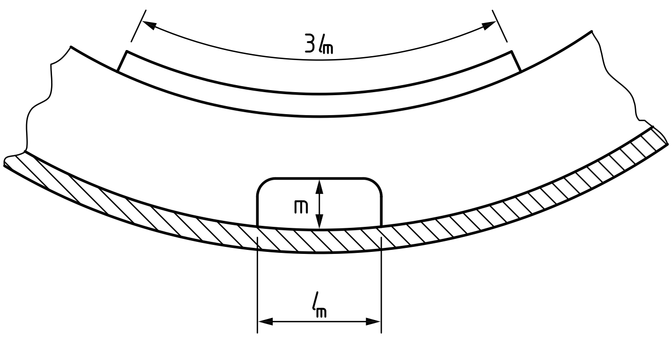
A_f	is the area of flange;
A_w	is the area of web;
I_f	is the second moment of area of flange about its own centroid;
I_w	is the second moment of area of web about its own centroid.

Figure 3.6-6 **Stiffening ring details *(continued)***



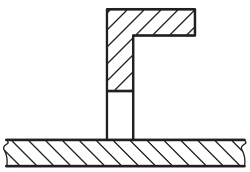
*NOTE A_s of one flange to be taken as the shaded area minus $e(e_w + L_e)$. Combined A_s and I_c of both flanges shall be taken when evaluating their adequacy as stiffeners, in accordance with **3.6.3.2**.*

f) Bolted flanges



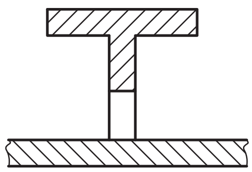
g) Stiffening ring with drainhole (mousehole)

NOTE Remaining notation as a).



i) Asymmetric flange

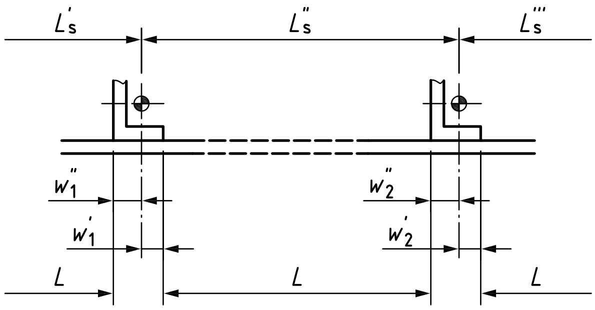
$$C = \frac{e_f w_f^3}{r_i \left[6e_w\left(d^2 - m^2\right) + 6e_f w_f (2d + e_f)\right]} \times \left[\frac{4(d-m)e_w + 3w_f e_f}{(d-m)e_w + 3w_f e_f}\right]$$



ii) Symmetric flange

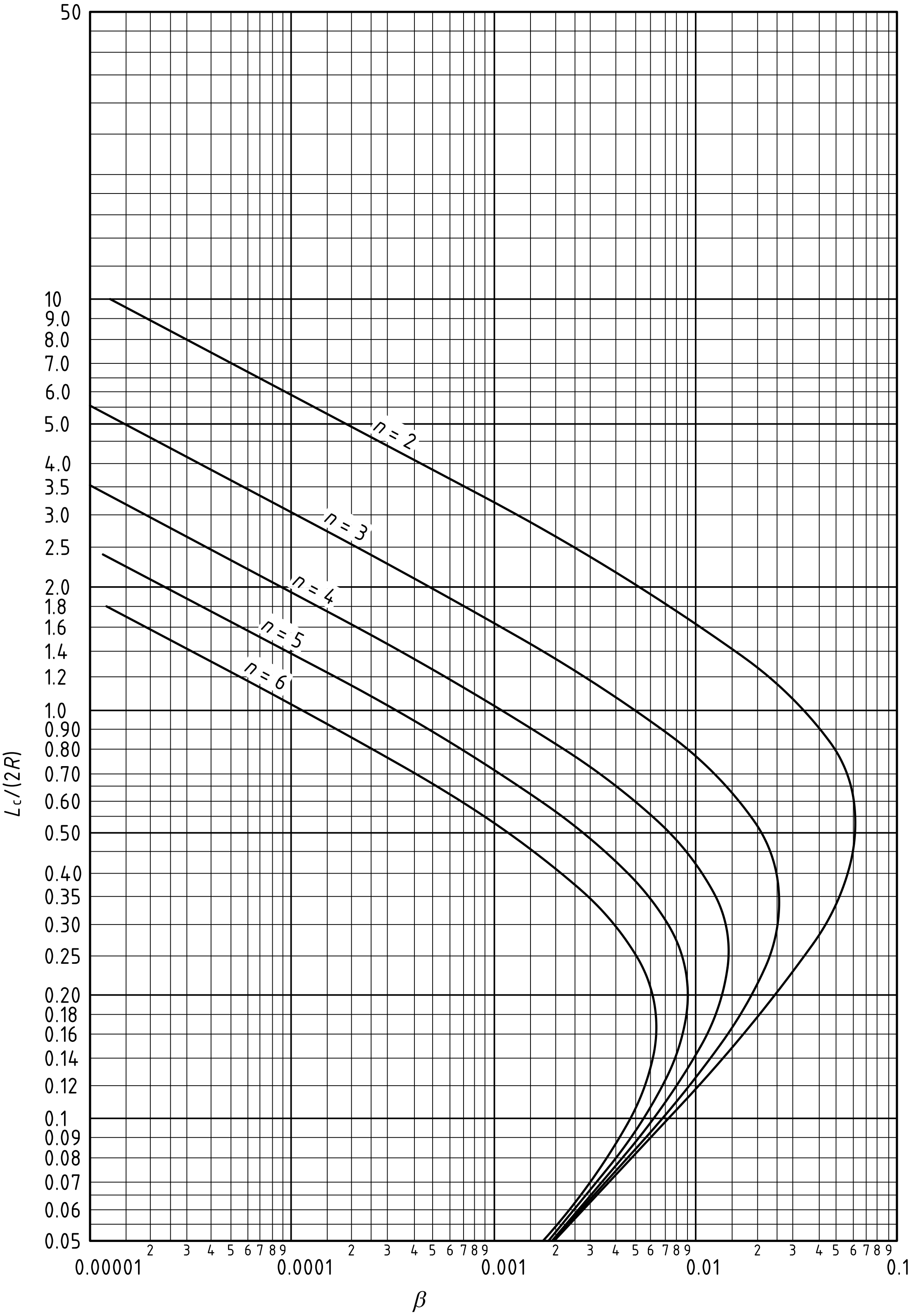
$$C = \frac{e_w^3 (d - m) + 8e_f w_f^3}{r_i \left[6e_w\left(d^2 - m^2\right) + 12e_f w_f (2d + e_f)\right]}$$

Figure 3.6-6 **Stiffening ring details *(continued)***



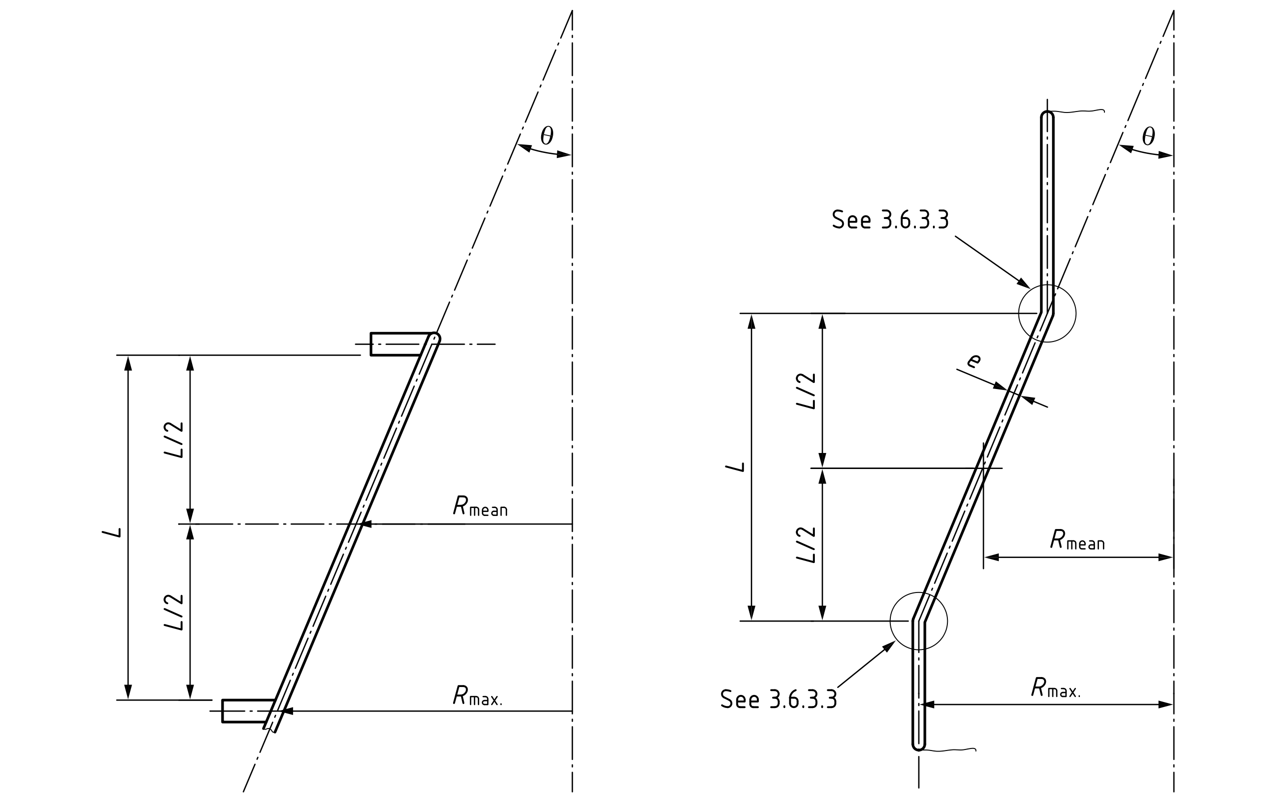
h) Stiffener with flange attached to cylinder

Figure 3.6-7 **Values of β**



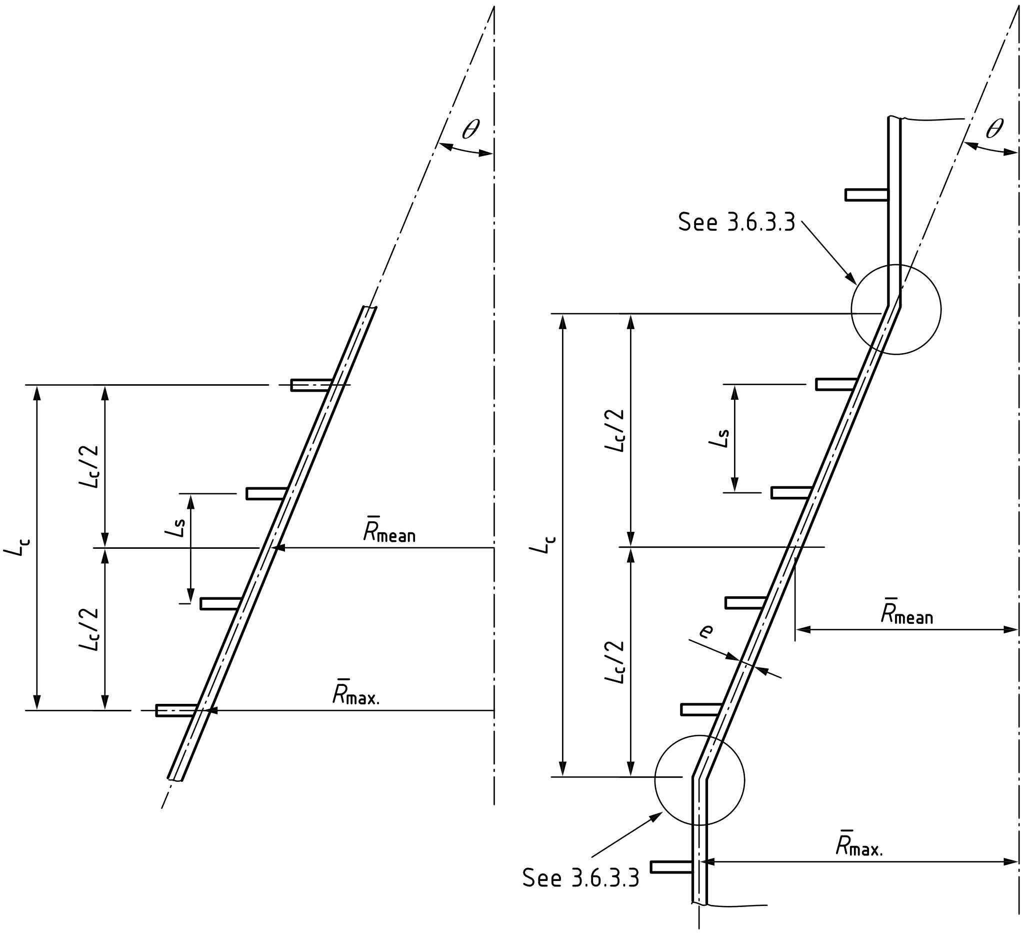
$$\beta = \frac{1}{\left\{n^2 - 1 + 0.5\left(\frac{\pi R}{L_c}\right)^2\right\}\left\{n^2\left(\frac{L_c}{\pi R}\right)^2 + 1\right\}^2}$$

Figure 3.6-8 **Conical sections: typical stiffeners**



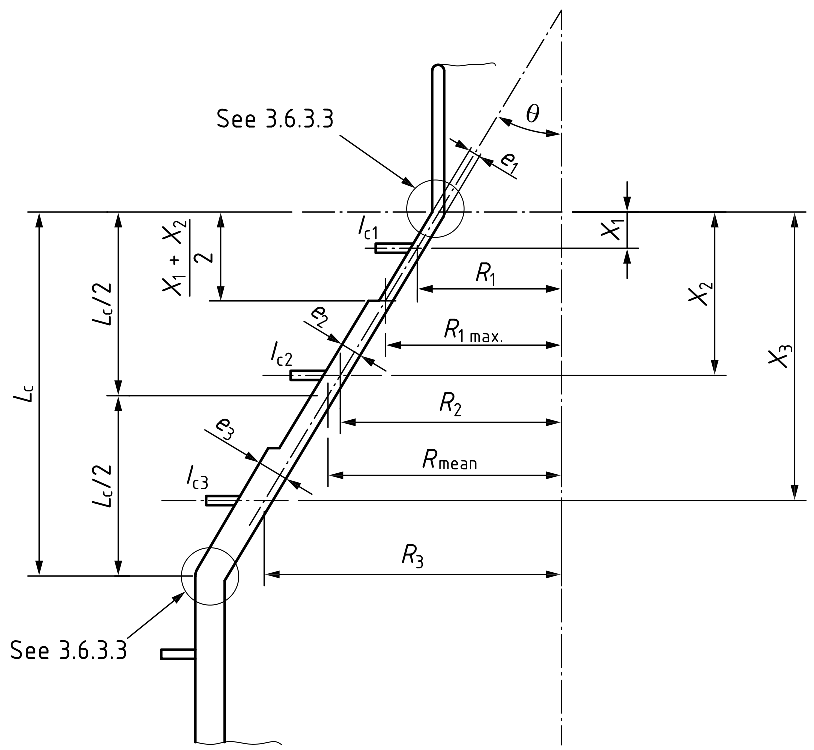
a) Unstiffened lengths (see **3.6.3.1**)

Figure 3.6-8 **Conical sections: typical stiffeners *(continued)***



b) Stiffened conical shells (see **3.6.3.2.1**)

Figure 3.6-8 **Conical sections: typical stiffeners *(continued)***



c) Stiffened conical shells with varying thickness and stiffener pitch (see **3.6.3.2.2**)

Suggested working form 3.6-1

Cylindrical shell external pressure. Simplified hand calculation. Method A for light stiffeners to 3.6.2

Units:

Shell check Material:

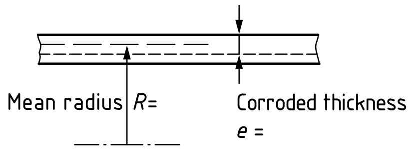
Modulus of elasticity $E =$

Design stress $f =$

Stress factor $s =$

(1.4 for ferritic steels, 1.1 for austenitic steels)

Poisson's ratio $\nu = 0.3$



With no stiffeners (Figure 3.6-1 a) and b))

Unsupported shell length $L =$

For stiffeners (Figure 3.6-1 c) and d))

$\lambda =$ (= +1 for internal stiffeners or = –1 for external stiffeners)

Maximum distance between stiffeners

$L_s =$

Allowable external pressure Assuming $\gamma = 0$ and $L = L_s$ Required $f =$

Assumed corroded thickness $e =$			
$L/(2R) =$			
$2R/e =$			
From Figure 3.6-2 $\varepsilon =$			
$p_y = sfe/R =$			
$p_m = Ee\varepsilon/R =$			
p_a/p_y (from Figure 3.6-4) $=$			
$\therefore$ (allowable) $p_a =$			

Stiffener check Material:

$f_s =$ $k =$

$e^2/(12R^2) =$ $L_s/(2\pi R) =$

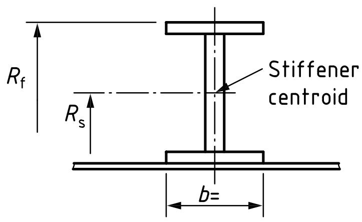
$\alpha = 1.28/\sqrt{Re} =$ $\alpha L =$

$N =$ from Table 3.6-1

If $L_s/(2\pi R) \le 0.1$ with $n = 2$

from Table 3.6-6 $L_e/L_s =$

Hence $L_e =$



If $L_s/(2\pi R) > 0.1$ with $n = 2$

from Table 3.6-7 $Z' =$

NOTE Use logarithmic interpolation, see Table W.1-3

$L_e = Z'R =$

Cross-sectional area	A_s		
Stiffener stress	f_s		
Radius (see Figure 3.6-6)	R_s		
Flange radius (see Figure 3.6-6)	R_f		
2nd moment of area	I_s		
Modified area $= R^2A_s/R_s^2$	A		

$A_c = A_s + eL_e$	=		
$X_c = \{0.5e^2L_e + A_s[0.5e + \lambda(R - R_s)]\} / A_c$	=		
$I_c = e^3L_e/3 + I_s + A_s[0.5e + \lambda(R - R_s)]^2 - A_cX_c^2$	=		
$p_{ys} = \dfrac{sf_s eR_f}{R^2(1-\nu/2)}\left[1+\dfrac{A}{be+2Ne/\alpha}\right]$	=		
$p_n = 3EI_c / (R^3L_s)$ *NOTE p_n to be $> kp$*	=		
d = greater of $\lambda(R - R_f) - X_c + e/2$ and X_c	=		
$\sigma_s = \dfrac{kpsf_s}{p_{ys}} + \dfrac{E\bar{d}}{R}\left[\dfrac{0.015p}{p_n - kp}\right]$	=		
check $sf_s \ge \sigma_s > 0$			

Check stiffener proportions comply with **3.6.2.2**. If $p_a < kp$ or $\sigma_s > sf_s$ see **3.6.2.3.1**

Date Calculation by Checked by

3.7 Supports, attachments and internal structures

3.7.1 General

Supports, attachments and internal structures shall be designed to withstand all loadings likely to be imposed in service due to pressure, weight of vessel and contents, machinery and piping loads, wind, earthquake, etc.

NOTE Deformations likely to occur under such loads and rapid changes in temperature can give rise to significant stresses in supports and attachments and will require particular consideration.

For vessels designed to withstand external pressure, the support arrangements should distribute loadings as evenly as practicable and should avoid points of high load concentration.

The effects on the shell of a pressure vessel of local forces and moments which may come from typical attachments and supports are covered in some detail in Annex G. Criteria for the assessment of the stresses caused are given in that annex and more generally in Annex A.

It is permissible to weld or stud bolt supports, etc. to the shell of a pressure vessel. Weld design shall comply with **3.10**. In the design of stud connections, particular attention shall be given to fatigue loading and to the specification of attachment methods which will consistently meet the design duty. Where significant tensile stresses are likely to be developed through the thickness of a shell plate as a result of a local attachment, suitable tests shall be specified by the manufacturer at the design stage (to be carried out as in **4.2.2.6**) to check that the shell material is locally suitable for such loads.

The materials for attachments welded directly to a pressure component shall comply with **2.1** and **4.3.5.1**. The welding of all attachments shall be carried out by welders and procedures approved in accordance with Section 5.

3.7.2 Supports

3.7.2.1 Design

The design of supports shall enable inspection and maintenance to be carried out during the life of the vessel. Care shall be taken that the temperature gradients in external structures immediately adjacent to the shell do not produce stresses in excess of those laid down as permissible. If necessary, lagging shall be applied to limit the temperature gradient to a value producing acceptable stresses. Loads arising from differential thermal expansion of the shell and the supporting structure in general shall not produce stresses in either in excess of those permitted by the appropriate specification.

NOTE External stays or internal framing which support internal parts may be used to provide a stiffening effect on the shell where external supports are attached. Steel supporting structures that do not form part of the vessel should comply with BS EN 1993-1-1. When such supports are to be constructed in reinforced concrete, BS EN 1992-1-1 should be consulted.

In cases where the design strength is time dependent, components designed in accordance with this clause should be reviewed to ensure that creep deformation (local or general) will be acceptable throughout the agreed design lifetime.

3.7.2.2 Vertical vessels

3.7.2.2.1 Bracket support

Where vertical vessels are supported on lugs or brackets attached to the shell, the supporting members under the bearing attachments shall be as close to the shell as clearance for insulation will permit.

NOTE The choice between a number of brackets and a ring girder will depend upon the condition for each individual vessel.

3.7.2.2.2 Column support

Vertical vessels supported on a number of posts or columns shall, if necessary, be provided with backing or stiffening by means of a ring girder, internal partitions or similar devices in order to resist the forces tending to buckle the vessel wall.

3.7.2.2.3 Skirt support

Skirt supports (for typical details see Annex G) shall be not less than 6 mm thick. Openings shall be made in the side of the skirt to permit inspection of the bottom of the vessel, if it is not readily visible through the supporting framework. All such openings shall be reinforced if necessary. Where the product of skirt diameter (in millimetres), thickness (in millimetres), and temperature at the top of the skirt above ambient (in °C) exceeds 1.6×10^7 (in $mm^2 \cdot °C$), account shall be taken of the discontinuity stresses in both skirt and vessel induced by the temperature gradient in the upper section of the skirt.

NOTE It is recommended that these stresses should be calculated by the methods of references (1) and (2)[8)] and assessed by the criteria of Annex A.

3.7.2.3 Horizontal vessels

Where practicable, only two supports shall be provided for horizontal vessels.

NOTE Horizontal vessels may be supported by means of saddles, equivalent leg supports or ring supports (see Annex G). For thin-walled vessels where excessive distortion due to the weight of the vessel may be expected, ring supports as shown in Figure G.3.2-3 are recommended. Vessels designed to withstand external pressure should be supported close to the ends or alternatively at stiffeners.

Horizontal cylindrical vessels that are provided with vertical external tower-like extensions shall, where necessary, have the extensions supported independently of the vessel with suitable provision to ensure that loads imposed on the vessel due to thermal expansion or contraction are acceptable.

3.7.2.4 Internal structures

3.7.2.4.1 As far as practicable, internal structures and fittings shall be arranged to avoid imposing local concentrated loads on the walls of the vessel, consideration being given to the necessity for a corrosion allowance and avoidance of crevices where corrosion may start.

3.7.2.4.2 Where possible, local loads from internal structures, or from vessel contents, shall be carried by means of appropriate stiffeners and/or spacers, directly to the vessel supports and thus to the foundations without stressing the vessel walls or ends.

[8)] (1) Weil, N.A. and Murphy J.J. Design and analysis of welded pressure vessel supports. *Trans. ASME J. Eng. for Ind.* 1960, February: 1.
(2) Bergman, D.J. Temperature gradients for skirt supports of hot vessels. *Trans. ASME J. Eng. for Ind.* 1963, May: 219.

3.8 Bolted flanged connections

3.8.1 General

*NOTE 1 Suggested working forms with sketches covering the following types of flanges are provided at the end of **3.8.4**. The sketches show the loads and dimensions as defined in **3.8.2**.*

1) Narrow-face flange design: smooth bore;
2) Narrow-face flange design: stepped bore;
3) Narrow-face flange design: slip-on hubbed type;
4) Lap-type joint: loose flange with hub;
5) Lap-type joint: loose flange without hub;
6) Narrow-face flange design: smooth bore (external pressure case);
7) Narrow-face flange design: stepped bore (external pressure case);
8) Narrow-face flange design: slip-on hubbed type (external pressure case);
9) Reverse narrow-face flange design: flange diameter = shell outside diameter;
10) Reverse narrow-face flange design: slip-in type;
11) Reverse narrow-face flange design: flange diameter = shell outside diameter (external pressure case);
12) Reverse narrow-face flange design: slip-in type (external pressure case);
13) Full-face flange design with soft ring type gasket;
14) Reverse full-face flange design to **3.8.7.2**;
15) Reverse full-face flange design to **3.8.7.3**;
16) Full-face flange design with metal to metal contact.

Circular bolted flanged connections used in the construction of vessels to this specification shall either:

a) comply with the requirements for bolted flanged connections as specified in **3.8**; or
b) comply with the appropriate British Standard for pipework flanges, BS EN 1092 or BS EN 1759, and be of an appropriate rating; or
c) comply with ASME B16.5 for pipework flanges and be of an appropriate rating and meet the following provisions:
    1) the material complies with the appropriate standard specification of ASME B16.5 or a comparable British Standard;
    2) Clause **2.1.2.2.2.1** is satisfied;
    3) in the case of service at sub-zero temperatures, Annex D is applied.

Fillet welded pipework flanges shall not be used for design temperatures above 370 °C.

*NOTE 2 The recommendations for the surface finish of the gasket contact surface given in the note to **3.8.1.6** apply to all body flanges and flanges fitted with covers, whether standard or special.*

Where a standard pipework flange mates with a piping flange the surface finish shall be the same as that specified for the mating pipework flange.

NOTE 3 A flange is attached to and supported by a nozzle neck, pipe, or vessel wall, which will be referred to as the shell.

NOTE 4 The design rules have been derived from considerations of strength. Where operation for long periods of time at high temperature is required, without the need for bolt retightening, special consideration may be needed in the design, taking into account the possibility of reduction in gasket load due to creep of the bolts and the flanges. In the design of large diameter flanges special consideration should be given to the choice of gasket, size and pitch of bolts and sequence of bolt tightening when closing the joint. Where operation requires a specific degree of leak-tightness, this should be given by the purchaser in the purchase specification. Although rules for leak-tightness cannot be given in this specification, the manufacturer and gasket supplier need to accommodate any relevant requirements of the purchaser. Special consideration should also be given to applications where flanges are subject to significant additional loading.

3.8.1.1 Bolting-up condition

The bolting-up condition shall apply when the gasket or joint contact surface is seated during assembly of the joint at ambient temperature and with the only loading coming from the bolts.

NOTE The minimum bolt loading to achieve a satisfactory joint is a function of the gasket and the effective gasket area to be seated.

3.8.1.2 Operating condition

The operating condition shall apply when the hydrostatic end force due to the design pressure tends to part the joint and the bolt load has to maintain sufficient pressure on the gasket to ensure a tight joint.

NOTE The minimum bolt load under this condition is a function of design pressure, gasket material and the effective gasket contact area to be kept tight under pressure. More than one operating condition may require consideration. In the case of external pressure there is no minimum bolt load but flange stresses still require consideration.

3.8.1.3 Classification

For the purposes of **3.8**, flange connections shall be classified as follows.

a) *Narrow-faced flanges*

These are flanges where all the face contact area lies inside the circle enclosed by the bolts. Narrow-faced flanges with ring-type gaskets shall comply with **3.8.3** and those with ungasketted seal welded flanges with **3.8.5**.

b) *Full-faced flanges*

These are flanges where the face contact area, either directly or via a gasket or spacer, extends outside the circle enclosing the bolts. Full-faced flanges with soft ring-type gaskets shall comply with **3.8.4** and full-faced flanges with metal to metal contact shall comply with **3.8.8**.

c) *Reverse flanges*

These are flanges where the shell is attached at the outer edge, rather than the inner edge, of the flange. Narrow-face reverse flanges with gaskets shall comply with **3.8.6**.

Full-face reverse flanges with soft ring-type gaskets shall comply with **3.8.7**.

3.8.1.4 General requirements for bolting

If steel bolts or studs smaller than 12 mm are to be used, the bolting material shall have a design stress at 50 °C as given in Table 3.8-1 of more than 160 N/mm^2.

Table 3.8-1 gives recommended bolt stresses for determining the minimum bolt area in **3.8.3.2**. The values in Table 3.8-1 may be increased by 20% if controlled bolt tensioning is used. Bolt design stress may be multiplied by 1.5 for test conditions. Higher values than those given in Table 3.8-1 may be used, based upon known operating experience or more rigorous analysis, when agreed between manufacturer, purchaser and Inspecting Authority.

NOTE 1 In the case of small diameter bolts it may be necessary to give consideration to the use of torque spanners or other means for preventing the application of excessive load on the bolt.

NOTE 2 These stresses are nominal insofar as they may have to be exceeded in practice to provide against all conditions that tend to produce a leaking joint. However there is sufficient margin to provide a satisfactory closure without having to overload or repeatedly tighten the bolts.

NOTE 3 The values for bolting design stress in Table 3.8-1 have been developed from experience and bear some similarity with values quoted for ASME VIII Division 1. Where flange bolting materials are not listed in Table 3.8-1 or covered by a relevant Enquiry Case, it is considered that the following criteria should give an acceptable design stress value. Carbon and other non-austenitic steels in material groups 1 to 7, 9 and 11, the lesser of $R_{p0.2}/3$ at design temperature and $R_m/4$ at ambient temperature. Austenitic steels in material group 8, $R_m/4$ at design temperature.

Special means are required to ensure that an adequate preload is obtained on tightening large diameter bolts and this aspect shall be considered when the nominal bolt diameter is greater than 38 mm.

Bolt dimensions shall be in accordance with either BS 3643-1:2007 for metric series bolting or BS 1580-1:2007 for inch series bolting. For threaded portions of bolt, root areas for use in the calculation of A_b shall be determined as follows:

a) for metric bolting to BS 3643 the root area is based on the minor diameter d_3 as defined in **A.3** of BS 3643-1:2007;

b) for inch series bolting to BS 1580 the root area is the "section at minor diameter" as tabulated for unified coarse thread series (UNC) in column 8 of Table 15 and for unified 8-thread series (8 UN) in column 8 of Table 20 of BS 1580-1:2007.

NOTE 4 Table 3.8-2 gives bolt root areas for some commonly used bolt sizes.

Table 3.8-1 **Recommended design stress values for flange bolting materials[a]**

Material	BS references[c]	Diameter	Recommended design stress (N/mm^2) for design metal temperatures (°C) not exceeding:																			
		mm	50	100	200	250	300	350	400	425	450	475	500	525	550	575	600	625	650	675	700	725	750
Mild and carbon steel	(BS 3692 grade 8.8)	≤ 68	192	174	139	129	119																
	(BS 4190 grade 4.6)	≤ 68	120	113	92	81	71																
1% chromium molybdenum steel	(BS 4882 grade B7, L7)	≤ 63	193	181	167	158	154	148	140														
		> 63 ≤ 100	174	163	152	145	141	134	127														
	(BS 4882 grade B7A)	≤ 100	193	181	167	165	163	158	154	146	118												
12% chromium steel	(BS 4882 grade B6)	≤ 100	190	179	170	168	165	160	146	134													
1% chromium molybdenum vanadium steel	(BS 4882 grade B16)	≤ 100	193	187	183	176	169	165	157	153	144	130	115	93									
1% chromium molybdenum vanadium boron steel	(BS 4882 grade B16A)	≤ 100	174	168	163	159	152	149	143	140	135	130	115	93	61								
Austenitic chromium nickel 18/8 type steel	(BS 4882 grade B8)	All	126	106	89	83	80	77	75	72	71	70	69	68	66	64							
	(BS 4882 grade B8X)	≤ 19	200	169	142	128	117	106	94	87	80	75											
Stabilized austenitic chromium	(BS 4882 grade B8T)	All	126	108	90	83	79	76	75	74	73	73	72	72	68	58							
nickel 18/8 type steel	(BS 4882 grade B8TX)	≤ 19	201	172	144	130	119	107	96	90	84	79											
	(BS 4882 grade B8C)	All	128	118	104	97	92	90	88	88	87	87	87	86	85	74							
	(BS 4882 grade B8CX)	≤ 19	209	189	166	153	139	126	113	106	99	93											
Austenitic chromium nickel	(BS 4882 grade B8M)	All	129	109	94	87	83	79	78	77	76	75	74	74	73	72							
molybdenum steel	(BS 4882 grade B8MX)	≤ 19	207	174	148	136	124	112	100	93	87	81											

Table 3.8-1 **Recommended design stress values for flange bolting materials[a]** *(continued)*

Material	BS references[c]	Diameter	Recommended design stress (N/mm^2) for design metal temperatures (°C) not exceeding:																				
		mm	50	100	200	250	300	350	400	425	450	475	500	525	550	575	600	625	650	675	700	725	750
Precipitation hardening austenitic nickel chromium steel	(BS 4882 grade B17B)	All	145	142	141	139	138	138	137	137	136	136	136	135	134	133	115	90	69				
Precipitation hardening nickel chromium titanium aluminium alloy[b]	(BS 4882 grade B80A)	All	207	206	205	204	204	203	202	200	198	198	198	197	196	194	187	174	152	131	110	90	69

NOTE See Enquiry Case 5500/82 for design stresses for ASTM A193.

[a] For high temperature applications see **3.8.1**, Note 4.

[b] Material liable to exhibit stress relaxation.

[c] BS 4882:1990 is current but partially replaced by BS EN 1515-1:2000 and BS EN 1515-3:2005.

Table 3.8-2 **Bolt root areas**

Metric bolting in accordance with BS 3643-1:2007			
Nominal size mm	**Root area mm^2**	**Nominal size mm**	**Root area mm^2**
M10 × 1.5	52.3	M42 × 3	1 153
M12 × 1.75	76.25	M45 × 4	1 262
M14 × 2	104.7	M48 × 4	1 458
M16 × 2	144.1	M52 × 4	1 742
M18 × 2.5	175.1	M56 × 4	2 050
M20 × 2.5	225.2	M64 × 4	2 743
M22 × 2.5	281.5	M70 × 4	3 328
M24 × 3	324.3	M72 × 4	3 535
M27 × 3	427.1	M76 × 4	3 969
M30 × 3	544	M82 × 4	4 668
M33 × 3	675.1	M90 × 4	5 687
M36 × 3	820.4	M95 × 4	6 375
M39 × 3	979.7	M100 × 4	7 102
UNC bolting in accordance with BS 1580-1:2007			
Nominal size in	**Root area mm^2**	**Nominal size in**	**Root area mm^2**
½	83.3	⅞	276.8
⅝	133.5	1	363.2
¾	199.4		
8 UN bolting in accordance with BS 1580-1:2007			
1	363.2	1⅞	1 503
1⅛	478	2	1 729
1¼	609	2¼	2 226
1⅜	756	2½	2 787
1½	919	2¾	3 419
1⅝	1097	3	4 103
1¾	1290		

3.8.1.5 Where flanges are constructed by welding, weld dimensions shall be in accordance with Annex E. Flange construction shall be of one of the following forms as applicable:

a) face and back welded flange [see Figure E.31a)];

b) bore and back welded flange [see Figure E.31b)];

c) welded neck flange (or taper hub flange) [see Figure E.32a)] or parallel hub (long forged weld neck) type;

d) welding neck flange fabricated from plate [see Figure E.32b)];

e) lapped type [see Figure E.32c)];

NOTE 1 This form is known as a lap-joint. The bolt load is transmitted indirectly through a loose backing flange to a narrow lap or stub flange. The loose flange may have a hub. The stub flange incorporates the gasket contact face. It may be attached to the shell by any of the arrangements permitted for other flange constructions, not just that shown in Figure E.32c).

f) slip-on hubbed flange [see Figure E.33a)];

g) fillet welded flange [see Figure E.33b)].

NOTE 2 For design purposes a distinction is made between the flanges listed in a) to d), in which the bore of the flange coincides with the bore of the shell, and those with a fillet weld at the end of the shell and in which the two bores are different. They are known as smooth bore and stepped bore flanges respectively.

Any fillet radius between flange and hub or shell shall be not less than $0.25g_o$ and not less than 5 mm.

Hub flanges shall not be made by machining the hub directly from plate material without agreement between the manufacturer, purchaser and Inspecting Authority. This agreement shall take into account the stresses in the hub and potential risk of laminations in the hub.

Fillet welds shall not be used for design temperatures above 370 °C.

3.8.1.6 Machining

The bearing surface for the nuts shall be parallel to the flange face to within 1°. Any back facing or spot facing to accomplish this shall not reduce the flange thickness nor hub thickness below design values. The diameter of a spot facing shall be not less than the dimension across corners of the nut plus 3 mm. The radius between the back of the flange and the hub or shell shall be maintained.

NOTE The surface finish of the gasket contact face should be in accordance with the gasket manufacturer's recommendations if any, or should be based on experience or should follow the recommendations given in Table 3.8-3. The flatness of the flange faces should also be in accordance with the gasket manufacturer's recommendations or based on experience.

Table 3.8-3 **Recommended surface finish on gasket contact faces for body flanges and flanges fitted with covers**

<table>
<tr><th rowspan="2">Type of gaskets</th><th colspan="2">Required surface texture range</th><th rowspan="2">Machining details</th></tr>
<tr><th>R_a [a]
μm</th><th>R_z [a]
μm</th></tr>
<tr><td>Compressed asbestos fibre (CAF)
Fibrous substitutes for CAF
Polytetrafluoroethylene (PTFE)
Exfoliated graphite sheet
Rubber and reinforced rubber sheet</td><td>12.5 to 6.3</td><td>50 to 25</td><td>Continuous spiral groove or concentric groove finish</td></tr>
<tr><td>Exfoliated graphite sheet
Spiral wound filled with:
CAF (R); or
PTFE (S)
Rubber and reinforced rubber sheet</td><td>6.3 to 3.2</td><td>25 to 12.5</td><td>Continuous spiral groove or concentric groove finish</td></tr>
<tr><td>Flat metal jacketted asbestos filled (R)</td><td>3.2 to 1.6</td><td>12.5 to 6.3</td><td rowspan="3">Produced by a variety of tool shapes showing no definite tool markings to the eye</td></tr>
<tr><td>Solid flat metal ring (S)
Octagonal or oval metal ring (R)</td><td>1.6 to 0.8</td><td>6.3 to 3.2</td></tr>
<tr><td>Metallic solid or hollow “O” rings including Willis type rings (R)
Fully trapped rubber “O” rings of rectangular section</td><td>0.8 to 0.4</td><td>3.2 to 1.6</td></tr>
</table>

NOTE (R) or (S) indicates a preference for the rougher or smoother end of the range respectively.

[a] R_a and R_z are defined in BS EN ISO 4287.

3.8.2 Notation

For the purposes of **3.8.3** the following symbols apply. All dimensions are in the corroded condition (see **3.1.5**).

NOTE 1 Further and modified notation is given in subsequent subclauses.

A is the outside diameter of the flange or, where slotted holes extend to outside of flange, the diameter to bottom of slots

A_2 is the outside diameter of the contact face between loose and stub flanges in a lap-joint

A_b is the actual total cross-sectional area of bolts at the section of least diameter under load

A_m is the total required cross-sectional area of bolts, taken as the greater of A_{m1} and A_{m2}

A_{m1} is the total cross-sectional area of bolts required for operating conditions, $= W_{m1}/S_b$

A_{m2} is the total cross-sectional area of bolts required for gasket seating, $= W_{m2}/S_a$

B is the inside diameter of flange

B_2 is the inside diameter of the contact face between loose and stub flanges in a lap-joint

b_o is the basic gasket or joint seating width, $= N/2$ with the exception of the ring-joint for which $b_o = N/8$

b is the effective gasket or joint seating width:
$b = b_o$ when $b_o < 6.3$ mm
$b = 2.52\sqrt{b_o}$ when $b_o > 6.3$ mm
(this expression is valid only with dimensions expressed in millimetres)

C is the bolt circle diameter

C_F is the bolt pitch correction factor,

$$= \sqrt{\frac{\text{bolt spacing}}{2d_b + 6t\ /\ (m + 0.5)}}$$

where "bolt spacing" is the distance between bolt centre lines (if calculated value < 1, $C_F = 1$)

D is the inside diameter of shell

d is a factor; for integral method flange design

$$= \frac{U}{V} h_o g_o^{\,2};$$

for loose method flange design

$$= \frac{U}{V_L} h_o g_o^{\,2};$$

d_b is the bolt outside diameter

d_h is the diameter of the bolt holes

e is a factor; for integral method flange design

$$= \frac{F}{h_o};$$

for loose method flange design

$$= \frac{F_L}{h_o};$$

F is a factor for integral method flange design (from Figure 3.8-6)

F_L is a factor for loose hubbed flanges (from Figure 3.8-8)

f is the hub stress correction factor for integral method flange design from Figure 3.8-10 (for values below limit of figure use $f = 1$)

G	is the assumed diameter of gasket load reaction. When $b_o \leq 6.3$ mm, G = mean diameter of gasket contact face, when $b_o > 6.3$ mm, G = outside diameter of gasket contact face less $2b$ (see Figure 3.8-4)
G_1	is the diameter of location of load reaction between loose and stub flanges in a lap-joint, normally assumed to be the mean diameter of the contact face between them
g_o	is the analysis thickness of hub at small end
g_1	is the analysis thickness of hub at back of flange
H	is the total hydrostatic end force = $\pi G^2 p/4$
H_D	is the hydrostatic end force applied via shell to flange = $\pi B^2 p/4$
H_T	is the hydrostatic end force due to pressure on flange face = $H - H_D$
H_G	is the compression load on gasket to ensure tight joint = $2b \times \pi Gmp$
h	is the hub length
h_o	$= \sqrt{Bg_o}$;
h_D	is the radial distance from bolt circle to circle on which H_D acts = $(C - B - g_1)/2$ except for slip-on hubbed and stepped bore flanges for which $h_D = (C - B)/2$
h_G	is the radial distance from gasket load reaction to bolt circle = $(C - G)/2$
h_L	is the radial distance from bolt circle to circle on which load reaction acts for the loose flange in a lap-joint = $(C - G_1)/2$
h_T	is the radial distance from bolt circle to circle on which H_T acts = $(2C - B - G)/4$

NOTE For the stub flange in a lap joint C is replaced by G_1 in the definitions of h_D, h_G and h_T.

K	= A/B except for reverse flanges where $K = B/A$
k	is a design stress factor for narrow-faced gasketted flanges (see **3.8.3.4.2**)
M	= $M_{atm}C_F/B$ (bolting-up condition), or
	= $M_{op}C_F/B$ (operating condition)
M_{atm}	is the total moment acting upon flange for bolting-up condition
M_{op}	is the total moment acting upon flange for operating condition
m	is the gasket factor given in Table 3.8-4
N	is the contact width of gasket, as limited by gasket width and flange facing
p	is the design pressure (this shall be taken as a positive value for use in the equations – see **3.2.3**)
p_e	is the external design pressure (this shall be taken as a positive value for use in the equations – see **3.2.3**)
S_a	is the bolt design stress at atmospheric temperature (see Table 3.8-1)
S_b	is the bolt design stress at design temperature (see Table 3.8-1)
S_{FA}	is the design stress of flange material at atmospheric temperature
S_{FO}	is the design stress of flange material at design temperature
S_{HA}	is the design stress of the hub at atmospheric temperature (see **3.8.3.4.2**)
S_{HO}	is the design stress of the hub at design temperature (see **3.8.3.4.2**)
S_H	is the calculated longitudinal stress in hub
S_R	is the calculated radial stress in flange
S_T	is the calculated tangential stress in flange
T	is a factor from Figure 3.8-5 or Table 3.8-5
t	is the minimum allowable flange thickness, measured at the thinnest section
U	is a factor from Figure 3.8-5 or Table 3.8-5
V	is a factor for the integral method, from Figure 3.8-7
V_L	is a factor for the loose hubbed flanges, from Figure 3.8-9
W_{m1}	is the minimum required bolt load for operating conditions = $H_G + H$
W_{m2}	is the minimum required bolt load for gasket seating = πbGy

W	is the flange design bolt load = $0.5(A_m + A_b)S_a$
X	is the nominal gap between the shell and the loose flange in a lap-joint
Y	is a factor from Figure 3.8-5 or Table 3.8-5
y	is the gasket material minimum design seating stress given in Table 3.8-4;
Z	is a factor from Figure 3.8-5 or Table 3.8-5
λ	is a factor

$$= \left[\frac{te+1}{T} + \frac{t^3}{d}\right]$$

3.8.3 Narrow-faced gasketted flanges

3.8.3.1 General

One of the three following design methods shall be applied to circular narrow face flanges with ring type gaskets or joints under internal pressure, taking account of the exceptions given.

a) Integral method (in which account is taken of support from the shell and the stresses in the shell are evaluated and compared with allowable stresses). The integral method shall not be applied to the slip-on hubbed flange [see Figure E.33a)] or to the loose flange in a lap joint [see suggested working forms (1) and (2)].

b) Loose method (in which the flange is assumed to get no support in bending from the shell and correspondingly imposes no bending stresses on it). The loose method shall only be applied, except for loose flanges in lap joints [see Figure E.32c)], if all of the following requirements are met:

1) $g_o \leq 16$ mm;

2) $\dfrac{B}{g_o} \leq 300$;

3) $p \leq 2$ N/mm^2;

4) operating temperature ≤ 370 °C;

[see suggested working forms (1), (2) and (5)].

c) Loose hubbed flange method, which shall be applied to the slip-on type of hubbed flange and the loose hubbed flange in a lap joint [see suggested working forms (3) and (4)].

The face and back welded flange, bore and back welded flange, parallel hub flange, welding neck flange and fillet welded flange may all be designed by either of the loose or integral methods (see Figure E.31, Figure E.32 and Figure E.33 for these types of flanges). The design methods allow for a taper hub, which may be a weld; the hub assumed for purposes of calculation shall have a slope of not more than 1:1, i.e. $g_1 \leq h + g_0$.

If the hub slope exceeds 1:4 then limitations apply regarding the proximity of the weld to the small end of the hub [see Figure E.32a)].

NOTE 1 In more unusual shapes of hub it may be necessary to choose values of g_1 and h defining a simple taper hub fitting within the profile of the actual assembly.

NOTE 2 There is no minimum value of h for a slip-on hubbed flange.

NOTE 3 The rule for calculating the moment M is independent of the method being used.

3.8.3.2 Bolt loads and areas

Bolt loads and areas shall be calculated for both the bolting-up and operating conditions.

a) *Bolting-up condition*

The minimum bolt load, W_{m2}, shall be πbGy.

b) *Operating condition*

The minimum bolt load, W_{m1}, shall be $H + H_G$.

The required bolt area A_m shall be the greater of A_{m1} and A_{m2}.

The actual bolt area, A_b, shall be not less than A_m.

NOTE Recommended values for the gasket factor, m, and the gasket seating pressure, y, are given in Table 3.8-4 for various gaskets.

3.8.3.3 Flange moments

Flange moments shall be calculated for both the bolting-up and operating conditions.

a) *Bolting-up condition*

The total flange moment shall be:

$$\mathrm{M}_{atm} = Wh_G$$

b) *Operating condition*

The total flange moment shall be:

$$M_{op} = H_D h_D + H_T h_T + H_G h_G$$

For flange pairs having different design conditions, as for example when they trap a tubesheet, bolt loads shall be calculated at bolting-up and operating conditions for each flange/gasket combination separately. W_{m1} and W_{m2} shall then be taken as the greater of the two calculated values. For the flange on which W_{m1} was the lower calculated value, the value of H_G shall be increased as follows:

$$H_G = W_{m1} - H$$

3.8.3.4 Flange stresses and stress limits

3.8.3.4.1 Flange stresses

Flange stresses shall be determined for both bolting-up and operating conditions from the moment, M, as follows, where:

$M = M_{atm}\dfrac{C_F}{B}$ and $M = M_{op}\dfrac{C_F}{B}$ respectively

a) Integral method

longitudinal hub stress $S_H = \dfrac{fM}{\lambda {g_1}^2}$

radial flange stress $S_R = \dfrac{(1.333te + 1)M}{\lambda t^2}$

tangential flange stress $S_T = \dfrac{YM}{t^2} - ZS_R$

b) Loose method

tangential flange stress $S_T = \dfrac{YM}{t^2}$

$S_R = S_H = 0$

c) Loose hubbed flange method

longitudinal hub stress $S_H = \dfrac{M}{\lambda g_1^{\,2}}$

radial flange stress $S_R = \dfrac{(1.333te + 1)M}{\lambda t^2}$

tangential flange stress $S_T = \dfrac{YM}{t^2} - ZS_R$

3.8.3.4.2 Stress limits

For flanges designed to the integral method [see **3.8.3.1**a)] the hub design stress S_{HA} and S_{HO} shall be the lower of the design stresses of the hub material or the connected shell material. For flanges designed to the loose method or loose hubbed method [see **3.8.3.1**b) or **3.8.3.1**c)] S_{HA} and S_{HO} shall be the design stresses of the hub material.

Flange design stresses S_{FA}, S_{FO}, S_{HA}, and S_{HO} for narrow-faced gasketted flanges shall be divided by a factor k if $D > 1\,000$ mm.

$k = (1 + D/2\,000) \times 2/3$ if $1\,000 < D < 2\,000$ mm.

$k = 4/3$ if $2\,000$ mm $< D$.

NOTE 1 The effect of the rule is that for D > 2 000 mm the flange design stress will normally be yield/2.

The flange stresses as calculated in **3.8.3.4.1** shall not exceed the following values, using design stresses at ambient temperature for the bolting-up condition and design stresses at design temperature for the operating condition:

$S_H \le$ the smaller of $1.5S_{HO}$ or $1.5S_{FO}$, or $S_H \le$ the smaller of $1.5S_{HA}$ or $1.5S_{FA}$;

S_R and $S_T \le S_{FO}$ or S_{FA};

$0.5(S_H + S_R) \le S_{FO}$ or S_{FA};

$0.5(S_H + S_T) \le S_{FO}$ or S_{FA}.

NOTE 2 S_{HO} and S_{HA}, the hub design stresses, are the design stresses of the shell material except for the case of welding neck or slip-on hubbed construction.

3.8.3.5 Narrow-face flanges subject to external pressure [see suggested working forms (6), (7) and (8)]

If the flange is subject to both internal and external pressure it shall be designed for both conditions, except that external pressure need not be considered where the external design pressure p_e is less than the internal design pressure p.

The design of flanges for external pressure shall be in accordance with **3.8.3** except that:

p_e replaces p;

$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$;

$W_{m1} = A_{m1} = 0$.

Where the flange for external pressure is one of a flange pair having different design conditions, W_{m1} shall be that calculated for the other member of the pair and M_{op} shall be the greater of M_{op} as calculated in **3.8.3.5** and $W_{m1}h_G$.

3.8.3.6 Lap-joints [see suggested working forms (4) and (5)]

The stub flange may take any of the forms listed in **3.8.1.5** and either the narrow-faced (see **3.8.3**) or full-faced (see **3.8.4**) method shall be applied. Separate calculations shall be carried out for the stresses in the loose and stub flanges.

NOTE 1 The two alternative methods for the stub flange make different conservative assumptions about the way the flange carries the load on it and therefore give different results. The narrow faced method assumes resistance to rotation comes from the flange itself, therefore it tends to give a thicker flange. The full face method assumes resistance to rotation comes from the flange acting as a simple lever, the necessary reaction at the edge of the flange has to be balanced by an increased bolt load.

Bolt loads and areas shall meet the requirements of **3.8.3.2** or **3.8.4.2** as appropriate.

Unless the designer chooses an alternative load reaction position, the diameter of this reaction between stub and loose flanges shall be taken as:

$$G_1 = (A_2 + B_2)/2$$

NOTE 2 An alternative reaction point may increase the loose flange thickness whilst reducing the stub flange thickness, or vice versa.

The bearing stress at the contact face between the stub and loose flanges shall be determined for both the bolting-up and operating conditions using the following equation.

$$\text{bearing stress} = \frac{(W_{m1} \text{ or } W)}{(\pi/2) \times \min\left[(A_2 - X)^2 - G_1{}^2; G_1{}^2 - (B_2 + X)^2\right]}$$

If the diameters A_2 and B_2 are defined by the same component, as shown in Figure 3.8-12, then $X = 0$.

The bearing stress shall not exceed 1.5 times the lower design stress of the two flanges, using the design stress at ambient temperature for the bolting-up condition and the design stress at design temperature for the operating condition.

The stub flange shall meet the requirements for a flange loaded directly by the bolts as given in **3.8.3.4** or **3.8.4**, except that the bolt load is assumed to be imposed at diameter G_1, which therefore replaces C in the calculations. The diameter of the bolt holes, d_h, required in **3.8.4**, shall be zero.

The moment arm on the loose flange for all components of load shall be h_L where $h_L = (C - G_1)/2$ such that:

$$M_{op} = W_{m1} \times h_L; \text{ and}$$

$$M_{atm} = W_{m2} \times h_L.$$

The loose flange stresses and stress limits shall meet the requirements of **3.8.3.4**.

NOTE 3 The option to use integral or loose design method applies to the stub flange.

3.8.3.7 Split ring flanges

It is permissible to split the loose flange in a lap-joint across the diameter to make it readily removable from the nozzle neck or vessel. The design shall be in accordance with **3.8.3.6** modified as follows.

a) When the flange consists of a single split ring, it shall be designed as if it were a solid flange (without splits), using 200% of the moment M required in **3.8.3.6**.

b) When the flange consists of two split rings, each ring shall be designed as if it were a solid flange (without splits), using 75% of the moment M required in **3.8.3.6**. The pair of rings shall be assembled so that the splits in one ring are 90° from the splits in the other ring.

c) The splits shall be located midway between bolt holes.

d) Where the loose split flange is keyed into the back of the mating component, as shown in Figure 3.8-1, the following design method shall be used. The following symbols are in addition to, or modify, those given in **3.8.2**.

D_b is the outside diameter of the contact face;

f_{bo} is the bearing stress; $\leq 1.5f$ (f for the weaker of flange or mating component);

f_{so} is the shear stress; $\leq 0.5f$ (f for the flange material);

H_h is the radial component of the contact face force = $W \tan \alpha$;

H_v is the contact face force = W;

h_h is the lever arm for H_h (see Figure 3.8-2);

NOTE h_h may be negative if the line of action of H_h lies above the centroid of the flange cross-section.

h_v is the lever arm for H_v (see Figure 3.8-2);

h_w is the lever arm for W (see Figure 3.8-2);

t is the flange thickness at the outer diameter;

t_1 is the minimum flange thickness (see Figure 3.8-2);

α is the key slope (see Figure 3.8-2).

$M_{op} = W_{m1}(h_w + h_v + h_h \tan \alpha)$;

$M_{atm} = 0.5(A_m + A_b)S_a(h_w + h_v + h_h \tan \alpha)$;

M_o = greater of M_{op} or $M_{atm} \times S_{FO}/S_{FA}$.

For the purpose of determining lever arms:

$b = N_{min}/3$;

$N_{min} = W/\{\pi(D_b - N) f_{bo}\}$ and $N \geq N_{min}$.

The centroid position is based on the area of the total flange ring.

Determine flange stresses using the equations given in **3.8.3.4.1**b) with:

$$M = \frac{2M_o C_F}{B} \text{ for a single split flange;}$$

$$M = \frac{M_o C_F}{B} \text{ if flange is not split;}$$

where C_F may be taken = 1.0;

$B = A - 2g_o$ (see Figure 3.8-2).

For the purpose of determining factor Y (see Figure 3.8-6 and Table 3.8-5, K shall be taken as $A/(A - 2g_o)$ (see Figure 3.8-2).

In no case shall the dimension t_1 be less than the greater of the values given by the following equations:

$$t_1 = \frac{W'}{\pi(A - 2g_o)f_{so}}$$

$$t_1 = \sqrt{\frac{1.91M''}{S_{FO}(A - 2g_o)}}$$

where

W' = the greater of W_{ml} or $(A_m + A_b)S_a S_{FO}/2S_{FA}$;

M'' = $W'\{(A + 2b - D_b - 2g_o)(1 + \tan^2 \alpha) + t_1 \tan \alpha\}/2$.

3.8.3.8 Flanges with swing bolts or clamp bolts

The design methods of **3.8.3.1** to **3.8.3.6** shall be used to design or check the bolts and solid portion of the flange ring, of flanges fitted with swing bolts or clamp bolts. In addition the following provisions shall be met.

a) Where the clamp is a proprietary item the supplier's data shall be checked to ensure that the clamp size and quantity are capable of transmitting the flange loads W_{m1} and W_{m2}.

b) The load at the bolts shall be assumed to act along the bolt centre line and not along a resolved thrust line.

c) Provision shall be made to ensure equal spacing of loose flange clamps.

d) Special consideration is required for cyclic conditions or attachment to rotating equipment.

e) A lip, or equivalent, shall be provided around the outer edge or the flange to prevent the swing bolts or clamps slipping off under pressurized conditions. A typical lip arrangement is shown in Figure 3.8-3.

f) Swing bolts shall be produced from a single forging.

g) Where slotted holes are provided in the flange or bolted end for the swing bolts, the outside diameter A as defined in **3.5.6.1** and **3.8.2** shall be taken as the diameter to the bottom of the slots.

h) The shear stresses in the swing bolts, the pins and the lugs shall be checked for the bolting-up and operating conditions and shall not exceed half the relevant design stress (see **A.3.3.3**).

i) The bearing stresses in the swing bolts, the pins and the lugs shall be checked for the bolting-up and operating conditions and shall not exceed 1.5 times the relevant design stress (see **A.3.3.3**).

j) The lugs shall be designed for the bolting-up and operating conditions using the methods used in structural engineering.

k) The lug attachment welds should follow the recommendations given in Figure E.34, where t_c is the nominal thickness of the lug (see **1.6**) and t_s is the nominal thickness of the shell or ring to which the lug is attached.

Figure 3.8-1 **Loose keyed flange with mating components**

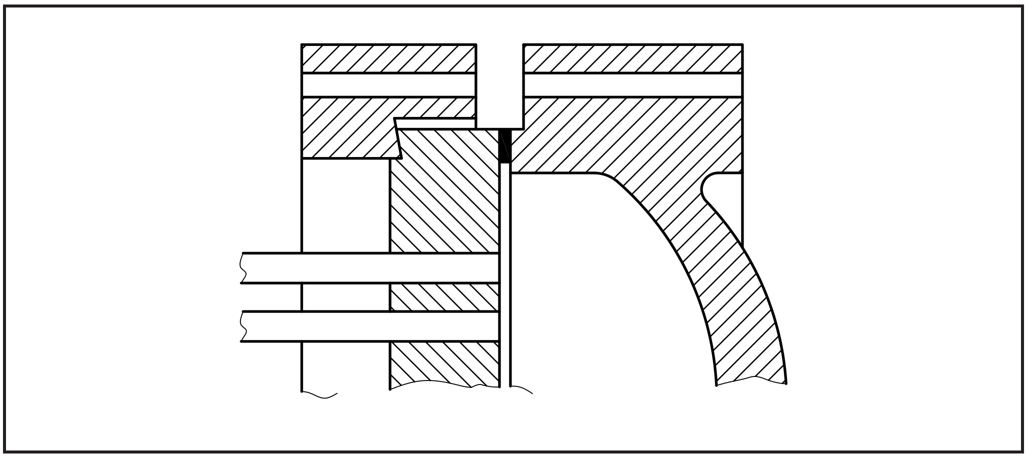


Figure 3.8-2 **Forces and lever arms on loose keyed flange**

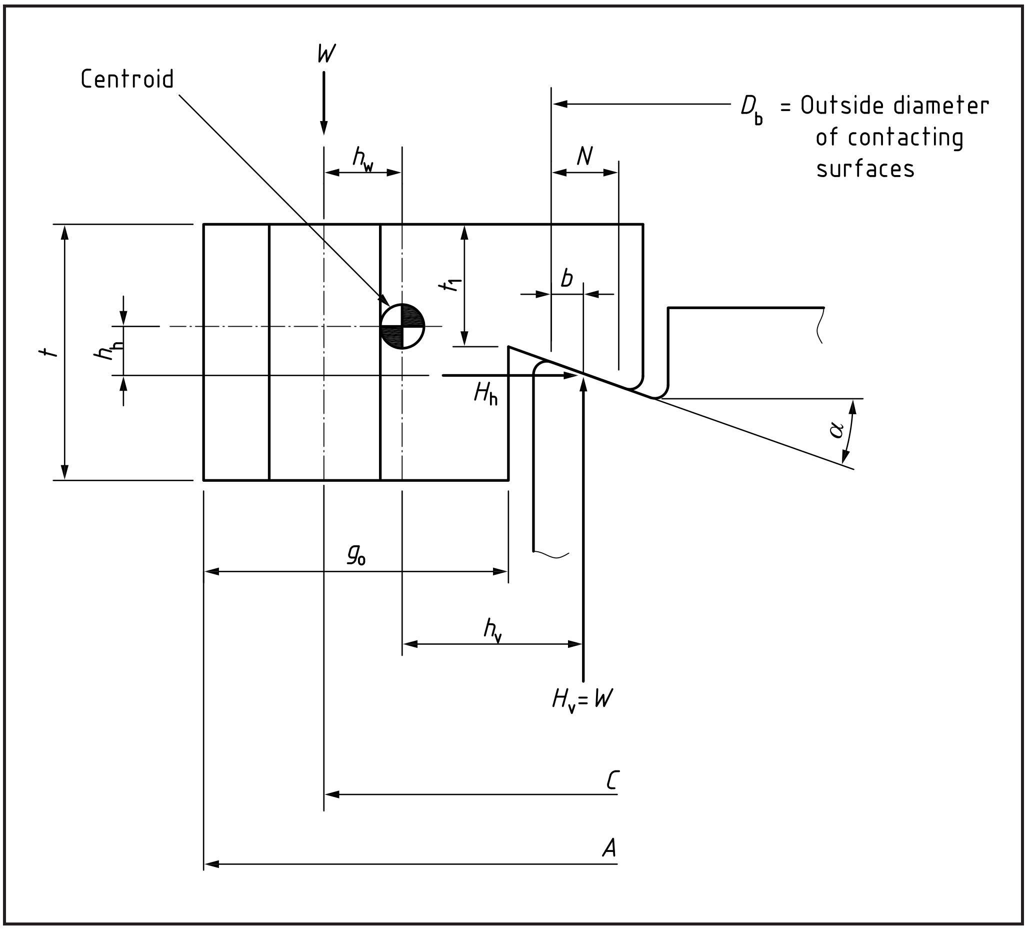
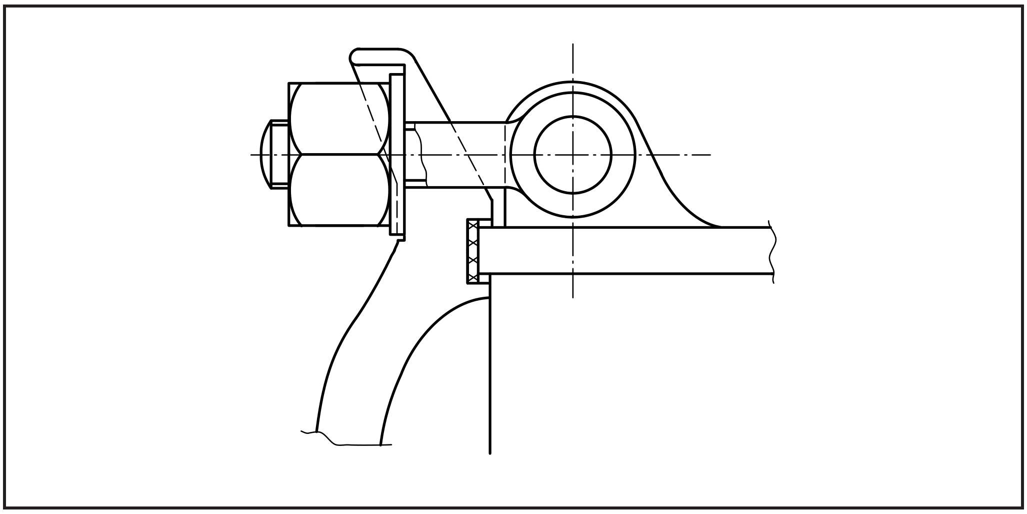


Figure 3.8-3 **Typical lip arrangement for swing bolted flange**



3.8.4 Full-faced flanges with soft ring type gaskets [see suggested working form (13)]

Full-faced flanges with non-metallic gaskets not less than 1.5 mm thick and extending beyond the circle enclosing the bolt holes shall be in accordance with the requirements of **3.8.4**.

For stepped bore and slip-on hubbed full-faced flanges the dimensions B, g_1 and g_0 shall be as shown in suggested working forms (2) and (3) respectively.

3.8.4.1 Additional and modified notation for 3.8.4

For the purposes of **3.8.4** the following symbols are in addition to or modify those given in **3.8.3**.

A_1 is the inside diameter of gasket or inside diameter of flange face, whichever is greater;

b'_o is the basic gasket seating width effective under initial tightening up = $G_o - C$;

b' is the effective gasket seating width $= 4\sqrt{b'_o}$;

NOTE This expression is valid only with dimensions expressed in millimetres.

$2b''$ is the effective gasket pressure width, taken as 5 mm;

E is modulus of elasticity of flange material at design temperature from Table 3.6-3;

G is the diameter at location of gasket load reaction = $C - (d_h + 2b'')$;

G_o is outside diameter of gasket or outside diameter of flange, whichever is less;

H is the total hydrostatic end force = $\pi(C - d_h)^2 p/4$;

H_G is compression load on gasket to ensure tight joint = $2b'' \times \pi Gmp$;

H_R is the balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle;

h_G is radial distance from bolt circle to circle on which H_G acts = $(d_h + 2b'')/2$;

h_R is radial distance from bolt circle to circle on which H_R acts = $(G_o - C + d_h)/4$;

h_T is radial distance from bolt circle to circle on which H_T acts = $(C + d_h + 2b'' - B)/4$;

M is balancing radial moment in flange along line of bolt holes;

n is number of bolts.

3.8.4.2 Bolt loads and areas

Bolt loads shall be calculated in accordance with **3.8.3.2**, taking:

$$W_{m1} = H + H_G + H_R$$

where

$$H_R = \frac{H_D h_D + H_T h_T + H_G h_G}{h_R}$$

$$W_{m2} = \pi C b' y$$

3.8.4.3 Flange design

The flange thickness shall be not less than the value of t from the following equation:

$$t = \sqrt{\frac{6M}{S_{FO}(\pi C - n d_h)}}$$

where

$M = H_R h_R$

The bolt spacing shall not exceed:

$2d_b + (E/200\ 000)^{0.25} \times 6t/(m + 0.5)$

where E is expressed in N/mm^2. If necessary the flange thickness shall be increased to enable this requirement to be met.

The minimum spacing shall be determined by consideration of the space necessary to apply a spanner to the nuts and possible interference from gussets and other obstructions.

3.8.5 Ungasketted seal welded flanges

Ungasketted seal welded flanges (see Figure 3.8-11) shall be designed with **3.8.3**, except that:

a) only the operating condition is to be considered;

b) $G = D_L$ where D_L is outside diameter of seal weld lip;

c) $H_G = 0$.

3.8.6 Reverse narrow-face flanges

3.8.6.1 Reverse narrow-face flanges under internal pressure [see suggested working forms (9) and (10)]

Reverse flanges with narrow-face gaskets under internal pressure, shall be designed in accordance with **3.8.3** except that:

a) the limits on g_o and B/g_o to the application of the loose flange option do not apply;

b) A is the inside diameter of the flange;

c) B is the outside diameter of the flange;

d) $H_D = \pi p D^2/4$;

e) $H_T = H_D - H$ where H_T is the net pressure load on the flange faces;

f) $h_T = (2C - G - D)/4$;

g) $h_D = (B - C - g_1)/2$.If the flange is slipped into the shell with a fillet weld on the outside, so that ($B \approx D$), h_D becomes instead:

$h_D = (B - C)/2$;

h) $M_{op} = H_T h_T + H_D h_D$;

i) $K = B/A$;

j) $M = (M_{atm}$ or $M_{op})C_F/A$.

NOTE 1 The sign of h_T, which may be negative, has to be respected.

NOTE 2 The moment due to gasket reaction is taken as 0 for the operating condition since this assumption gives higher stresses.

3.8.6.2 Reverse narrow-face flanges under external pressure [see suggested working forms (11) and (12)]

Reverse narrow-face flanges under external pressure shall be designed according to the rules of **3.8.6.1** together with the modifications of **3.8.3.5** except that the formula for M_{op} in **3.8.3.5** shall be replaced by $M_{op} = H_D(h_D + h_G) + H_T(h_T - h_G)$.

3.8.7 Reverse full-face flanges [see suggested working forms (14) and (15)]

3.8.7.1 General

For internal pressure, the design method shall be in accordance with **3.8.7.2** or **3.8.7.3** as appropriate. For both design methods, the bolting loads at the ambient condition and the gaskets shall be in accordance with **3.8.4**. The design rules of **3.8.7.3** are not applicable to a full-face flange that is attached to a conventional non-reverse full-face flange.

*NOTE Two alternative design methods are provided for reverse full-face flanges. The first follows the approach of **3.8.3** at the operating condition and assumes resistance to rotation comes from the flange itself; the second follows **3.8.4** and requires a larger bolt area.*

For external pressure, because of the balance of forces, bending moments and thus flange stresses are minimal. However the bolt spacing criteria of **3.8.4.3** shall still apply and this may be used to obtain a minimum flange thickness.

3.8.7.2 Design following method of 3.8.3

Design for the operating condition shall be in accordance with **3.8.3** with the following modifications.

a) A is inside diameter of flange;

b) A_1 is inside diameter of gasket or contact face, whichever is greater;

c) B is outside diameter of flange;

d) D is the shell internal diameter;

e) $N = (C - A_1)/2$;

f) G_o is the outside diameter of the gasket or contact face whichever is the lesser;

g) $H = \pi p(C - d_h)^2/4$;

h) $H_G = 2b \times \pi Cmp$;

i) $H_D = \pi p D^2/4$;

j) $H_T' = H_D - \pi p {A_1}^2/4$;

k) $H_T = 0.5(H - H_D + H'_T)$;

l) $h_D = (B - g_1 - C)/2$; except for the slip-in type flange ($B \approx D$), for which: $h_D = (B - C)/2$;

m) $h'_T = (2C - D - A_1)/4$;

n) $h_T = (2C + d_h - 2A_1)/6$;

o) $M_{op} = H_D h_D + H'_T h'_T - H_T h_T$;

p) $K = B/A$;

q) $M = M_{op} C_F / A$.

NOTE 1 The sign of h'_T, which may be negative has to be respected.

NOTE 2 The moment due to gasket reaction is taken as 0 for the operating condition since this assumption gives higher stresses.

3.8.7.3 Design following method of 3.8.4

Design for the operating condition shall be in accordance with **3.8.4** with the following modifications.

a) A is inside diameter of flange;

b) A_1 is inside diameter of gasket or contact face, whichever is the greater;

c) B is outside diameter of flange;

d) G_o is outside diameter of gasket or contact face, whichever is the lesser;

e) $H_D = \pi p D^2/4$;

f) $H_C = H_D - \pi p C^2/4$; where H_C is the hydrostatic force on the flange-face outside the bolt circle diameter;

g) $h_D = (B - C - g_1)/2$;

h) $h_c = (D - C)/4$;

i) $M_1 = H_D h_D - H_C h_C$;

j) $M = M_1$;

k) W_{m1} shall be calculated as follows:

$h_R = (C - A_1 + d_h)/4$;

$H_R = M/h_R$;

$W_{m1} = H_R + H_D - H_C$.

NOTE The moment due to gasket reactions is taken as 0 for the operating condition since this assumption gives higher stresses.

3.8.8 Full-faced flanges with metal to metal contact [see suggested working form (16)]

These rules shall be applied when there is metal to metal contact both inside and outside the bolt circle before the bolts are tightened with more than a small amount of preload and the seal is provided by an O-ring or equivalent. The rules shall also be applied when the flange is bolted to a flat cover.

Manufacturing procedures and tolerances shall ensure that the flange is not dished so as to give initial contact outside the bolt circle.

NOTE 1 The rules are conservative where initial contact is at the bore.

NOTE 2 It is assumed that a self-sealing gasket is used approximately in line with the wall of the attached pipe or vessel and that the gasket seating load and any axial load from the seal may be neglected.

*NOTE 3 With relevant agreement, **3.2.2** permits the use of alternative requirements or sets of rules. For full face flanges with metal to metal contact a suitable alternative would be Appendix Y of ASME VIII division 1. The requirements of this specification neglect the support against rotation the flange gets from the cylinder. In ASME this support is taken into account and the stresses in the cylinder are calculated and assessed. Normally the thicknesses calculated to ASME are slightly less than those calculated to this specification, but at the expense of a much more complicated calculation.*

3.8.8.1 Additional and modified notation

For the purposes of **3.8.8** the following symbols are in addition to or modify those given in **3.8.3**:

G is the mean diameter of gasket;

H_R is the balancing reaction force outside the bolt circle in opposition to moments due to loads inside the bolt circle;

h_R is the distance from the bolt circle to the circle on which H_R acts = $(A - C)/2$;

M is the balancing radial moment in the flange along line of bolt holes;

n is the number of bolts.

3.8.8.2 Design

Bolt loads shall be calculated in accordance with **3.8.3.2** taking:

$W_{m1} = H + H_R$

where

$H_R = M/h_R$

and

$M = H_D h_D + H_T h_T$

$W_{m2} = 0$

The flange thickness shall be not less than:

$$t = \sqrt{\frac{6M}{S_{FO}(\pi C - n d_h)}}$$

Where two flanges of different internal diameters, both designed in accordance with this section, are to be bolted together to make a joint, the following additional requirements apply:

a) the value of M to be used for both flanges shall be that calculated for the smaller internal diameter;

b) the thickness of the flange with the smaller bore shall be not less than:

$$t = \sqrt{\frac{3(M_1 - M_2)(A + B)}{\pi S_{FO} B(A - B)}}$$

where M_1 and M_2 are the values of M calculated for the two flanges.

Table 3.8-4 **Gasket materials and contact facings: gasket factors (*m*) for operating conditions and minimum design seating stress (*y*)**

NOTE This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using the methods of ***3.8. The design values and other details given in this table are suggested only and are not mandatory.***

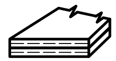
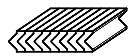
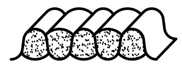
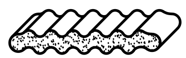
Gasket material		**Gasket factor *m***	**Min. design seating stress *y*** N/mm^2	**Sketches**	**Dimension *N* (min.) mm**
Rubber without fabric or a high percentage of asbestos[a] fibre:					10
[b]below 75° BS and IRH		0.50	0		
75° BS and IRH or higher		1.00	1.4		
Asbestos[a] with a suitable binder for the operating conditions	3.2 mm thick	2.0	11.0		
	1.6 mm thick	2.75	25.5		
	0.8 mm thick	3.50	44.8		
Rubber with cotton fabric insertion		1.25	2.8		
Rubber with asbestos[a] fabric	3-ply	2.25	15.2		
	2-ply	2.50	20.0		
	1-ply	2.75	25.5		
Vegetable fibre		1.75	7.6		
Spiral-wound metal, asbestos[a] filled	Carbon Stainless or monel	2.50 3.00	To suit application[c]		
Corrugated metal, asbestos[a] inserted or Corrugated metal, jacketted asbestos[a] filled	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4 to 6% chrome Stainless steels	2.50 2.75 3.00 3.25 3.50	20.0 25.5 31.0 37.9 44.8	 	

Table 3.8-4 **Gasket materials and contact facings: gasket factors (*m*) for operating conditions and minimum design seating stress (*y*)** *(continued)*

NOTE This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using the methods of ***3.8. The design values and other details given in this table are suggested only and are not mandatory.***

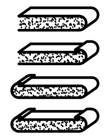
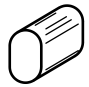
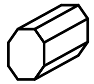
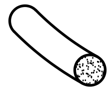
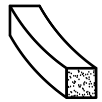
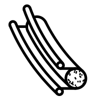
Gasket material		**Gasket factor *m***	**Min. design seating stress *y* N/mm²**	**Sketches**	**Dimension *N* (min.) mm**
Corrugated metal	Soft aluminium	2.75	25.5		10
	Soft copper or brass	3.00	31.0		
	Iron or soft steel	3.25	37.9		
	Monel or 4 to 6% chrome	3.50	44.8		
	Stainless steels	3.75	52.4		
Flat metal jacketted asbestos[a] filled	Soft aluminium	3.25	37.9		
	Soft copper or brass	3.50	44.8		
	Iron or soft steel	3.75	52.4		
	Monel	3.50	55.1		
	4 to 6% chrome	3.75	62.0		
	Stainless steels	3.75	62.0		
Grooved metal	Soft aluminium	3.25	37.9		10
	Soft copper or brass	3.50	44.8		
	Iron or soft steel	3.75	52.4		
	Monel or 4 to 6% chrome	3.75	62.0		
	Stainless steels	4.25	69.5		

Table 3.8-4 **Gasket materials and contact facings: gasket factors (*m*) for operating conditions and minimum design seating stress (*y*)** *(continued)*

NOTE This table gives a list of many commonly used gasket materials and contact facings with suggested design values of m and y that have generally proved satisfactory in actual service when using the methods of ***3.8. The design values and other details given in this table are suggested only and are not mandatory.***

Gasket material		Gasket factor m	Min. design seating stress y N/mm^2	Sketches	Dimension N (min.) mm
Solid flat metal	Soft aluminium Soft copper or brass Iron or soft steel Monel or 4 to 6% chrome Stainless steels	4.00 4.75 5.50 6.00 6.50	60.6 89.5 124 150 179		6
Ring joint[d]	Iron or soft steel Monel or 4 to 6% chrome Stainless steels	5.50 6.00 6.50	124 150 179	 	
Rubber O-rings: below 75° BS between 75° and 85° BS and IRH		0 to 0.25	 0.7 1.4		
Rubber square section rings: below 75° BS and IRH between 75° and 85° BS and IRH		0 to 0.25	 1.0 2.8[e]		
Rubber T-section rings: below 75° BS and IRH between 75° and 85° BS and IRH		0 to 0.25	 1.0 2.8		

NOTE Advice should be sought from the gasket manufacturer on the ability of a gasket to resist the maximum load resulting from bolt load and possibly vacuum load.

[a] New non-asbestos bonded fibre sheet gaskets are not necessarily direct substitutes for asbestos based materials. In particular pressure, temperature and bolt load limitations may be applied. Use within the manufacturer's current recommendations.

[b] See BS ISO 48-2.

[c] See BS 3381. Advice should be sought from the gasket manufacturer on design seating stress.

[d] $b = N/8$.

[e] This value has been calculated.

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*)**

K	*T*	*Z*	*Y*	*U*
1.001	1.91	1000.50	1899.43	2078.85
1.002	1.91	500.50	951.81	1052.80
1.003	1.91	333.83	637.56	700.70
1.004	1.91	250.50	478.04	525.45
1.005	1.91	200.50	383.67	421.72
1.006	1.91	167.17	319.71	351.42
1.007	1.91	143.36	274.11	301.30
1.008	1.91	125.50	239.95	263.75
1.009	1.91	111.61	213.42	234.42
1.010	1.91	100.50	192.19	211.19
1.011	1.91	91.41	174.83	192.13
1.012	1.91	83.84	160.38	176.25
1.013	1.91	77.43	148.06	162.81
1.014	1.91	71.93	137.69	151.30
1.015	1.91	67.17	128.61	141.33
1.016	1.90	63.00	120.56	132.49
1.017	1.90	59.33	111.98	124.81
1.018	1.90	56.06	107.36	118.00
1.019	1.90	53.14	101.72	111.78
1.020	1.90	50.51	96.73	106.30
1.021	1.90	48.12	92.21	101.33
1.022	1.90	45.96	88.04	96.75
1.023	1.90	43.98	84.30	92.64
1.024	1.90	42.17	80.81	88.81
1.025	1.90	40.51	77.61	85.29
1.026	1.90	38.97	74.70	82.09
1.027	1.90	37.54	71.97	79.08
1.028	1.90	36.22	69.43	76.30
1.029	1.90	34.99	67.11	73.75
1.030	1.90	33.84	64.91	71.33
1.031	1.90	32.76	62.85	69.06
1.032	1.90	31.76	60.92	66.94
1.033	1.90	30.81	59.11	63.95
1.034	1.90	29.92	57.41	63.08
1.035	1.90	29.08	55.80	61.32
1.036	1.90	28.29	54.29	59.66
1.037	1.90	27.54	52.85	58.08
1.038	1.90	26.83	51.50	56.59
1.039	1.90	26.15	50.21	55.17
1.040	1.90	25.51	48.97	53.82

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.041	1.90	24.90	47.81	53.10
1.042	1.90	24.32	46.71	51.33
1.043	1.90	23.77	45.64	50.15
1.044	1.90	23.23	44.64	49.05
1.045	1.90	22.74	43.69	48.02
1.046	1.90	22.05	42.75	46.99
1.047	1.90	21.79	41.87	46.03
1.048	1.90	21.35	41.02	45.09
1.049	1.90	20.92	40.21	44.21
1.050	1.89	20.51	39.43	43.34
1.051	1.89	20.12	38.68	42.51
1.052	1.89	19.74	37.96	41.73
1.053	1.89	19.38	37.27	40.96
1.054	1.89	19.03	36.60	40.23
1.055	1.89	18.69	35.96	39.64
1.056	1.89	18.38	35.34	38.84
1.057	1.89	18.06	34.74	38.19
1.058	1.89	17.76	34.17	37.56
1.059	1.89	17.47	33.62	36.95
1.060	1.89	17.18	33.04	36.34
1.061	1.89	16.91	32.55	35.78
1.062	1.89	16.64	32.04	35.21
1.063	1.89	16.40	31.55	34.68
1.064	1.89	16.15	31.08	34.17
1.065	1.89	15.90	30.61	33.65
1.066	1.89	15.67	30.17	33.17
1.067	1.89	15.45	29.74	32.69
1.068	1.89	15.22	29.32	32.22
1.069	1.89	15.02	28.91	31.79
1.070	1.89	14.80	28.51	31.34
1.071	1.89	14.61	28.13	30.92
1.072	1.89	14.41	27.76	30.51
1.073	1.89	14.22	27.39	30.11
1.074	1.88	14.04	27.04	29.72
1.075	1.88	13.85	26.69	29.34
1.076	1.88	13.68	26.36	28.98
1.077	1.88	13.56	26.03	28.69
1.078	1.88	13.35	25.72	28.27
1.079	1.88	13.18	25.40	27.92
1.080	1.88	13.02	25.10	27.59

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.081	1.88	12.87	24.81	27.27
1.082	1.88	12.72	24.52	26.95
1.083	1.88	12.57	24.24	26.65
1.084	1.88	12.43	24.00	26.34
1.085	1.88	12.29	23.69	26.05
1.086	1.88	12.15	23.44	25.57
1.087	1.88	12.02	23.18	25.48
1.088	1.88	11.89	22.93	25.20
1.089	1.88	11.76	22.68	24.93
1.090	1.88	11.63	22.44	24.66
1.091	1.88	11.52	22.22	24.41
1.092	1.88	11.40	21.99	24.16
1.093	1.88	11.28	21.76	23.91
1.094	1.88	11.16	21.54	23.67
1.095	1.88	11.05	21.32	23.44
1.096	1.88	10.94	21.11	23.20
1.097	1.88	10.83	20.91	22.97
1.098	1.88	10.73	20.71	22.75
1.099	1.88	10.62	20.51	22.39
1.100	1.88	10.52	20.31	22.18
1.101	1.88	10.43	20.15	22.12
1.102	1.88	10.33	19.94	21.92
1.103	1.88	10.23	19.76	21.72
1.104	1.88	10.14	19.58	21.52
1.105	1.88	10.05	19.38	21.30
1.106	1.88	9.96	19.33	21.14
1.107	1.87	9.87	19.07	20.69
1.108	1.87	9.78	18.90	20.77
1.109	1.87	9.70	18.74	20.59
1.110	1.87	9.62	18.55	20.38
1.111	1.87	9.54	18.42	20.25
1.112	1.87	9.46	18.27	20.08
1.113	1.87	9.38	18.13	19.91
1.114	1.87	9.30	17.97	19.75
1.115	1.87	9.22	17.81	19.55
1.116	1.87	9.15	17.68	19.43
1.117	1.87	9.07	17.54	19.27
1.118	1.87	9.00	17.40	19.12
1.119	1.87	8.94	17.27	18.98
1.120	1.87	8.86	17.13	18.80

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.121	1.87	8.79	17.00	18.68
1.122	1.87	8.72	16.87	18.54
1.123	1.87	8.66	16.74	18.40
1.124	1.87	8.59	16.62	18.26
1.125	1.87	8.53	16.49	18.11
1.126	1.87	8.47	16.37	17.99
1.127	1.87	8.40	16.25	17.86
1.128	1.87	8.34	16.14	17.73
1.129	1.87	8.28	16.02	17.60
1.130	1.87	8.22	15.91	17.48
1.131	1.87	8.16	15.79	17.35
1.132	1.87	8.11	15.68	17.24
1.133	1.86	8.05	15.57	17.11
1.134	1.86	7.99	15.46	16.99
1.135	1.86	7.94	15.36	16.90
1.136	1.86	7.88	15.26	16.77
1.137	1.86	7.83	15.15	16.66
1.138	1.86	7.78	15.05	16.54
1.139	1.86	7.73	14.95	16.43
1.140	1.86	7.68	14.86	16.35
1.141	1.86	7.62	14.76	16.22
1.142	1.86	7.57	14.66	16.11
1.143	1.86	7.53	14.57	16.01
1.144	1.86	7.48	14.48	15.91
1.145	1.86	7.43	14.39	15.83
1.146	1.86	7.38	14.29	15.71
1.147	1.86	7.34	14.20	15.61
1.148	1.86	7.29	14.12	15.51
1.149	1.86	7.25	14.03	15.42
1.150	1.86	7.20	13.95	15.34
1.151	1.86	7.16	13.86	15.23
1.152	1.86	7.11	13.77	15.14
1.153	1.86	7.07	13.69	15.05
1.154	1.86	7.03	13.61	14.96
1.155	1.86	6.99	13.54	14.87
1.156	1.86	6.95	13.45	14.78
1.157	1.86	6.91	13.37	14.70
1.158	1.86	6.87	13.30	14.61
1.159	1.86	6.83	13.22	14.53
1.160	1.86	6.79	13.15	14.45

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.161	1.85	6.75	13.07	14.36
1.162	1.85	6.71	13.00	14.28
1.163	1.85	6.67	12.92	14.20
1.164	1.85	6.64	12.85	14.12
1.165	1.85	6.60	12.78	14.04
1.166	1.85	6.56	12.71	13.97
1.167	1.85	6.53	12.64	13.89
1.168	1.85	6.49	12.58	13.82
1.169	1.85	6.46	12.51	13.74
1.170	1.85	6.42	12.43	13.66
1.171	1.85	6.39	12.38	13.60
1.172	1.85	6.35	12.31	13.53
1.173	1.85	6.32	12.25	13.46
1.174	1.85	6.29	12.18	13.39
1.175	1.85	6.25	12.10	13.30
1.176	1.85	6.22	12.06	13.25
1.177	1.85	6.19	12.00	13.18
1.178	1.85	6.16	11.93	13.11
1.179	1.85	6.13	11.87	13.05
1.180	1.85	6.10	11.79	12.96
1.181	1.85	6.07	11.76	12.92
1.182	1.85	6.04	11.70	12.86
1.183	1.85	6.01	11.64	12.79
1.184	1.85	5.98	11.58	12.73
1.185	1.85	5.95	11.50	12.64
1.186	1.85	5.92	11.47	12.61
1.187	1.85	5.89	11.42	12.54
1.188	1.85	5.86	11.36	12.49
1.189	1.85	5.83	11.31	12.43
1.190	1.84	5.81	11.26	12.37
1.191	1.84	5.78	11.20	12.31
1.192	1.84	5.75	11.15	12.25
1.193	1.84	5.73	11.10	12.20
1.194	1.84	5.70	11.05	12.14
1.195	1.84	5.67	11.00	12.08
1.196	1.84	5.65	10.95	12.03
1.197	1.84	5.62	10.90	11.97
1.198	1.84	5.60	10.85	11.92
1.199	1.84	5.57	10.80	11.87
1.200	1.84	5.55	10.75	11.81

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.201	1.84	5.52	10.70	11.76
1.202	1.84	5.50	10.65	11.71
1.203	1.84	5.47	10.61	11.66
1.204	1.84	5.45	10.56	11.61
1.205	1.84	5.42	10.52	11.56
1.206	1.84	5.40	10.47	11.51
1.207	1.84	5.38	10.43	11.46
1.208	1.84	5.35	10.38	11.41
1.209	1.84	5.33	10.34	11.36
1.210	1.84	5.31	10.30	11.32
1.211	1.83	5.29	10.25	11.27
1.212	1.83	5.27	10.21	11.22
1.213	1.83	5.24	10.16	11.17
1.214	1.83	5.22	10.12	11.12
1.215	1.83	5.20	10.09	11.09
1.216	1.83	5.18	10.04	11.03
1.217	1.83	5.16	10.00	10.99
1.218	1.83	5.14	9.96	10.94
1.219	1.83	5.12	9.92	10.90
1.220	1.83	5.10	9.89	10.87
1.221	1.83	5.07	9.84	10.81
1.222	1.83	5.05	9.80	10.77
1.223	1.83	5.03	9.76	10.73
1.224	1.83	5.01	9.72	10.68
1.225	1.83	5.00	9.69	10.65
1.226	1.83	4.98	9.65	10.60
1.227	1.83	4.96	9.61	10.56
1.228	1.83	4.94	9.57	10.52
1.229	1.83	4.92	9.53	10.48
1.230	1.83	4.90	9.50	10.44
1.231	1.83	4.88	9.46	10.40
1.232	1.83	4.86	9.43	10.36
1.233	1.83	4.84	9.39	10.32
1.234	1.83	4.83	9.36	10.28
1.235	1.83	4.81	9.32	10.24
1.236	1.82	4.79	9.29	10.20
1.237	1.82	4.77	9.25	10.17
1.238	1.82	4.76	9.22	10.13
1.239	1.82	4.74	9.18	10.09
1.240	1.82	4.72	9.15	10.05

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.241	1.82	4.70	9.12	10.02
1.242	1.82	4.69	9.08	9.98
1.243	1.82	4.67	9.05	9.95
1.244	1.82	4.65	9.02	9.91
1.245	1.82	4.64	8.99	9.87
1.246	1.82	4.62	8.95	9.84
1.247	1.82	4.60	8.92	9.81
1.248	1.82	4.59	8.89	9.77
1.249	1.82	4.57	8.86	9.74
1.250	1.82	4.56	8.83	9.70
1.251	1.82	4.54	8.80	9.67
1.252	1.82	4.52	8.77	9.64
1.253	1.82	4.51	8.74	9.60
1.254	1.82	4.49	8.71	9.57
1.255	1.82	4.48	8.68	9.54
1.256	1.82	4.46	8.65	9.51
1.257	1.82	4.45	8.62	9.47
1.258	1.81	4.43	8.59	9.44
1.259	1.81	4.42	8.56	9.41
1.260	1.81	4.40	8.53	9.38
1.261	1.81	4.39	8.51	9.35
1.262	1.81	4.37	8.49	9.32
1.263	1.81	4.36	8.45	9.28
1.264	1.81	4.35	8.42	9.25
1.265	1.81	4.33	8.39	9.23
1.266	1.81	4.32	8.37	9.19
1.267	1.81	4.30	8.34	9.16
1.268	1.81	4.29	8.31	9.14
1.269	1.81	4.28	8.29	9.11
1.270	1.81	4.26	8.26	9.08
1.271	1.81	4.25	8.23	9.05
1.272	1.81	4.24	8.21	9.02
1.273	1.81	4.22	8.18	8.99
1.274	1.81	4.21	8.15	8.96
1.275	1.81	4.20	8.13	8.93
1.276	1.81	4.18	8.11	8.91
1.277	1.81	4.17	8.08	8.88
1.278	1.81	4.16	8.05	8.85
1.279	1.81	4.15	8.03	8.82
1.280	1.81	4.13	8.01	8.79

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.281	1.81	4.12	7.98	8.77
1.282	1.81	4.11	7.96	8.74
1.283	1.80	4.10	7.93	8.71
1.284	1.80	4.08	7.91	8.69
1.285	1.80	4.07	7.89	8.66
1.286	1.80	4.06	7.86	8.64
1.287	1.80	4.05	7.84	8.61
1.288	1.80	4.04	7.81	8.59
1.289	1.80	4.02	7.79	8.56
1.290	1.80	4.01	7.77	8.53
1.291	1.80	4.00	7.75	8.51
1.292	1.80	3.99	7.72	8.48
1.293	1.80	3.98	7.70	8.46
1.294	1.80	3.97	7.68	8.43
1.295	1.80	3.95	7.66	8.41
1.296	1.80	3.94	7.63	8.39
1.297	1.80	3.93	7.61	8.36
1.298	1.80	3.92	7.59	8.33
1.299	1.80	3.91	7.57	8.31
1.300	1.80	3.90	7.55	8.29
1.301	1.80	3.89	7.53	8.27
1.302	1.80	3.88	7.50	8.24
1.303	1.80	3.87	7.48	8.22
1.304	1.80	3.86	7.46	8.20
1.305	1.80	3.84	7.44	8.18
1.306	1.80	3.83	7.42	8.16
1.307	1.80	3.82	7.40	8.13
1.308	1.79	3.81	7.38	8.11
1.309	1.79	3.80	7.36	8.09
1.310	1.79	3.79	7.34	8.07
1.311	1.79	3.78	7.32	8.05
1.312	1.79	3.77	7.30	8.02
1.313	1.79	3.76	7.28	8.00
1.314	1.79	3.75	7.26	7.98
1.315	1.79	3.74	7.24	7.96
1.316	1.79	3.73	7.22	7.94
1.317	1.79	3.72	7.20	7.92
1.318	1.79	3.71	7.18	7.89
1.319	1.79	3.70	7.16	7.87
1.320	1.79	3.69	7.14	7.85

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.321	1.79	3.68	7.12	7.83
1.322	1.79	3.67	7.10	7.81
1.323	1.79	3.67	7.09	7.79
1.324	1.79	3.66	7.07	7.77
1.325	1.79	3.65	7.05	7.75
1.326	1.79	3.64	7.03	7.73
1.327	1.79	3.63	7.01	7.71
1.328	1.78	3.62	7.00	7.69
1.329	1.78	3.61	6.98	7.67
1.330	1.78	3.60	6.96	7.65
1.331	1.78	3.59	6.94	7.63
1.332	1.78	3.58	6.92	7.61
1.333	1.78	3.57	6.91	7.59
1.334	1.78	3.57	6.89	7.57
1.335	1.78	3.56	6.87	7.55
1.336	1.78	3.55	6.85	7.53
1.337	1.78	3.54	6.84	7.51
1.338	1.78	3.53	6.82	7.50
1.339	1.78	3.52	6.81	7.48
1.340	1.78	3.51	6.79	7.46
1.341	1.78	3.51	6.77	7.44
1.342	1.78	3.50	6.76	7.42
1.343	1.78	3.49	6.74	7.41
1.344	1.78	3.48	6.72	7.39
1.345	1.78	3.47	6.71	7.37
1.346	1.78	3.46	6.69	7.35
1.347	1.78	3.45	6.68	7.33
1.348	1.78	3.45	6.66	7.32
1.349	1.78	3.44	6.65	7.30
1.350	1.78	3.43	6.63	7.28
1.351	1.78	3.42	6.61	7.27
1.352	1.78	3.42	6.60	7.25
1.353	1.77	3.41	6.58	7.23
1.354	1.77	3.40	6.57	7.21
1.355	1.77	3.39	6.55	7.19
1.356	1.77	3.38	6.53	7.17
1.357	1.77	3.38	6.52	7.16
1.358	1.77	3.37	6.50	7.14
1.359	1.77	3.36	6.49	7.12
1.360	1.77	3.35	6.47	7.11

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.361	1.77	3.35	6.45	7.09
1.362	1.77	3.34	6.44	7.08
1.363	1.77	3.33	6.42	7.06
1.364	1.77	3.32	6.41	7.04
1.365	1.77	3.32	6.39	7.03
1.366	1.77	3.31	6.38	7.01
1.367	1.77	3.30	6.37	7.00
1.368	1.77	3.30	6.35	6.98
1.369	1.77	3.29	6.34	6.97
1.370	1.77	3.28	6.32	6.95
1.371	1.77	3.27	6.31	6.93
1.372	1.77	3.27	6.30	6.91
1.373	1.77	3.26	6.28	6.90
1.374	1.77	3.25	6.27	6.89
1.375	1.77	3.25	6.25	6.87
1.376	1.77	3.24	6.24	6.86
1.377	1.77	3.23	6.22	6.84
1.378	1.76	3.22	6.21	6.82
1.379	1.76	3.22	6.19	6.81
1.380	1.76	3.21	6.18	6.80
1.381	1.76	3.20	6.17	6.79
1.382	1.76	3.20	6.16	6.77
1.383	1.76	3.19	6.14	6.75
1.384	1.76	3.18	6.13	6.74
1.385	1.76	3.18	6.12	6.73
1.386	1.76	3.17	6.11	6.72
1.387	1.76	3.16	6.10	6.70
1.388	1.76	3.16	6.08	6.68
1.389	1.76	3.15	6.07	6.67
1.390	1.76	3.15	6.06	6.66
1.391	1.76	3.14	6.05	6.64
1.392	1.76	3.13	6.04	6.63
1.393	1.76	3.13	6.02	6.61
1.394	1.76	3.12	6.01	6.60
1.395	1.76	3.11	6.00	6.59
1.396	1.76	3.11	5.99	6.58
1.397	1.76	3.10	5.98	6.56
1.398	1.75	3.10	5.96	6.55
1.399	1.75	3.09	5.95	6.53
1.400	1.75	3.08	5.94	6.52

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.401	1.75	3.08	5.93	6.50
1.402	1.75	3.07	5.92	6.49
1.403	1.75	3.07	5.90	6.47
1.404	1.75	3.06	5.89	6.46
1.405	1.75	3.05	5.88	6.45
1.406	1.75	3.05	5.87	6.44
1.407	1.75	3.04	5.86	6.43
1.408	1.75	3.04	5.84	6.41
1.409	1.75	3.03	5.83	6.40
1.410	1.75	3.02	5.82	6.39
1.411	1.75	3.02	5.81	6.38
1.412	1.75	3.01	5.80	6.37
1.413	1.75	3.01	5.78	6.35
1.414	1.75	3.00	5.77	6.34
1.415	1.75	3.00	5.76	6.33
1.416	1.75	2.99	5.75	6.32
1.417	1.75	2.98	5.74	6.31
1.418	1.75	2.98	5.72	6.29
1.419	1.75	2.97	5.71	6.28
1.420	1.75	2.97	5.70	6.27
1.421	1.75	2.96	5.69	6.26
1.422	1.75	2.96	5.68	6.25
1.423	1.75	2.95	5.67	6.23
1.424	1.74	2.95	5.66	6.22
1.425	1.74	2.94	5.65	6.21
1.426	1.74	2.94	5.64	6.20
1.427	1.74	2.93	5.63	6.19
1.428	1.74	2.92	5.62	6.17
1.429	1.74	2.92	5.61	6.16
1.430	1.74	2.91	5.60	6.15
1.431	1.74	2.91	5.59	6.14
1.432	1.74	2.90	5.58	6.13
1.433	1.74	2.90	5.57	6.11
1.434	1.74	2.89	5.56	6.10
1.435	1.74	2.89	5.55	6.09
1.436	1.74	2.88	5.54	6.08
1.437	1.74	2.88	5.53	6.07
1.438	1.74	2.87	5.52	6.05
1.439	1.74	2.87	5.51	6.04
1.440	1.74	2.86	5.50	6.03

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.441	1.74	2.86	5.49	6.02
1.442	1.74	2.85	5.48	6.01
1.443	1.74	2.85	5.47	6.00
1.444	1.74	2.84	5.46	5.99
1.445	1.74	2.84	5.45	5.98
1.446	1.74	2.83	5.44	5.97
1.447	1.73	2.83	5.43	5.96
1.448	1.73	2.82	5.42	5.95
1.449	1.73	2.82	5.41	5.94
1.450	1.73	2.81	5.40	5.93
1.451	1.73	2.81	5.39	5.92
1.452	1.73	2.80	5.38	5.91
1.453	1.73	2.80	5.37	5.90
1.454	1.73	2.80	5.36	5.89
1.455	1.73	2.79	5.35	5.88
1.456	1.73	2.79	5.34	5.87
1.457	1.73	2.78	5.33	5.86
1.458	1.73	2.78	5.32	5.85
1.459	1.73	2.77	5.31	5.84
1.460	1.73	2.77	5.30	5.83
1.461	1.73	2.76	5.29	5.82
1.462	1.73	2.76	5.28	5.80
1.463	1.73	2.75	5.27	5.79
1.464	1.73	2.75	5.26	5.78
1.465	1.73	2.74	5.25	5.77
1.466	1.73	2.74	5.24	5.76
1.467	1.73	2.74	5.23	5.74
1.468	1.72	2.73	5.22	5.73
1.469	1.72	2.73	5.21	5.72
1.470	1.72	2.72	5.20	5.71
1.471	1.72	2.72	5.19	5.70
1.472	1.72	2.71	5.18	5.69
1.473	1.72	2.71	5.18	5.68
1.474	1.72	2.71	5.17	5.67
1.475	1.72	2.70	5.16	5.66
1.476	1.72	2.70	5.15	5.65
1.477	1.72	2.69	5.14	5.64
1.478	1.72	2.69	5.14	5.63
1.479	1.72	2.68	5.13	5.62
1.480	1.72	2.68	5.12	5.61

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.481	1.72	2.68	5.11	5.60
1.482	1.72	2.67	5.10	5.59
1.483	1.72	2.67	5.10	5.59
1.484	1.72	2.66	5.09	5.58
1.485	1.72	2.66	5.08	5.57
1.486	1.72	2.66	5.07	5.56
1.487	1.72	2.65	5.06	5.55
1.488	1.72	2.65	5.06	5.55
1.489	1.72	2.64	5.05	5.54
1.490	1.72	2.64	5.04	5.53
1.491	1.72	2.64	5.03	5.52
1.492	1.72	2.63	5.02	5.51
1.493	1.71	2.63	5.02	5.51
1.494	1.71	2.62	5.01	5.50
1.495	1.71	2.62	5.00	5.49
1.496	1.71	2.62	4.99	5.48
1.497	1.71	2.61	4.98	5.47
1.498	1.71	2.61	4.98	5.47
1.499	1.71	2.60	4.97	5.46
1.500	1.71	2.60	4.96	5.45
1.501	1.71	2.60	4.95	5.44
1.502	1.71	2.59	4.94	5.43
1.503	1.71	2.59	4.94	5.43
1.504	1.71	2.58	4.93	5.42
1.505	1.71	2.58	4.92	5.41
1.506	1.71	2.58	4.91	5.40
1.507	1.71	2.57	4.90	5.39
1.508	1.71	2.57	4.90	5.39
1.509	1.71	2.57	4.89	5.38
1.510	1.71	2.56	4.88	5.37
1.511	1.71	2.56	4.87	5.36
1.512	1.71	2.56	4.86	5.35
1.513	1.71	2.55	4.86	5.35
1.514	1.71	2.55	4.85	5.34
1.515	1.71	2.54	4.84	5.33
1.516	1.71	2.54	4.83	5.32
1.517	1.71	2.54	4.82	5.31
1.518	1.71	2.53	4.82	5.31
1.519	1.70	2.53	4.81	5.30
1.520	1.70	2.53	4.80	5.29

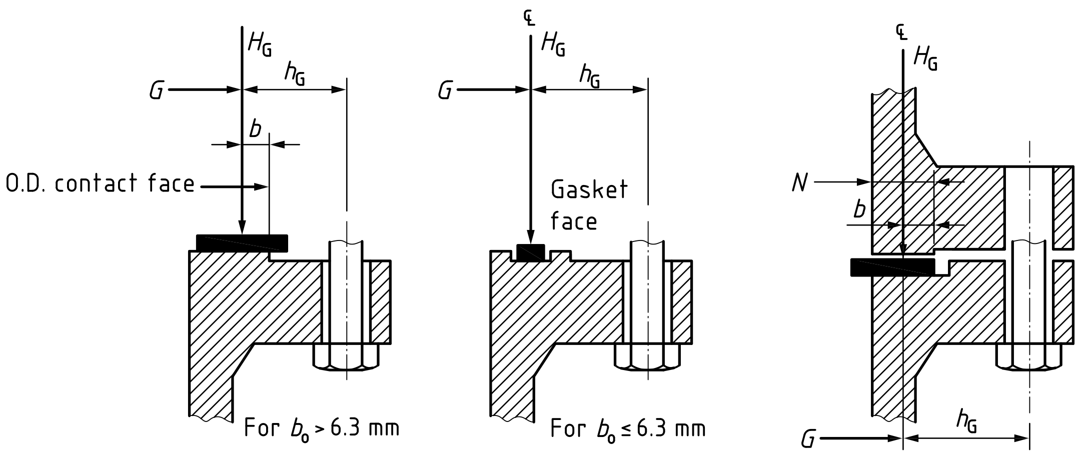
Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.521	1.70	2.52	4.79	5.28
1.522	1.70	2.52	4.79	5.27
1.523	1.70	2.52	4.78	5.27
1.524	1.70	2.51	4.78	5.26
1.525	1.70	2.51	4.77	5.25
1.526	1.70	2.51	4.77	5.24
1.527	1.70	2.50	4.76	5.23
1.528	1.70	2.50	4.76	5.23
1.529	1.70	2.49	4.75	5.22
1.530	1.70	2.49	4.74	5.21
1.531	1.70	2.49	4.73	5.20
1.532	1.70	2.48	4.72	5.19
1.533	1.70	2.48	4.72	5.19
1.534	1.70	2.48	4.71	5.17
1.535	1.70	2.47	4.70	5.17
1.536	1.70	2.47	4.69	5.16
1.537	1.70	2.47	4.68	5.15
1.538	1.69	2.46	4.68	5.15
1.539	1.69	2.46	4.67	5.14
1.540	1.69	2.46	4.66	5.13
1.541	1.69	2.45	4.66	5.12
1.542	1.69	2.45	4.65	5.11
1.543	1.69	2.45	4.64	5.11
1.544	1.69	2.45	4.64	5.10
1.545	1.69	2.44	4.63	5.09
1.546	1.69	2.44	4.63	5.08
1.547	1.69	2.44	4.62	5.07
1.548	1.69	2.43	4.62	5.07
1.549	1.69	2.43	4.61	5.06
1.550	1.69	2.43	4.60	5.05
1.551	1.69	2.42	4.60	5.05
1.552	1.69	2.42	4.59	5.04
1.553	1.69	2.42	4.58	5.03
1.554	1.69	2.41	4.58	5.03
1.555	1.69	2.41	4.57	5.02
1.556	1.69	2.41	4.57	5.02
1.557	1.69	2.40	4.56	5.01
1.558	1.69	2.40	4.56	5.00
1.559	1.69	2.40	4.55	4.99
1.560	1.69	2.40	4.54	4.99

Table 3.8-5 **Values of *T, Z, Y* and *U* (factors involving *K*) *(continued)***

K	*T*	*Z*	*Y*	*U*
1.561	1.69	2.39	4.54	4.98
1.562	1.69	2.39	4.53	4.97
1.563	1.68	2.39	4.52	4.97
1.564	1.68	2.38	4.51	4.96
1.565	1.68	2.38	4.51	4.95
1.566	1.68	2.38	4.50	4.95
1.567	1.68	2.37	4.50	4.94
1.568	1.68	2.37	4.49	4.93
1.569	1.68	2.37	4.48	4.92
1.570	1.68	2.37	4.48	4.92
1.571	1.68	2.36	4.47	4.91
1.572	1.68	2.36	4.47	4.91
1.573	1.68	2.36	4.46	4.90
1.574	1.68	2.35	4.46	4.89
1.575	1.68	2.35	4.45	4.89
1.576	1.68	2.35	4.44	4.88
1.577	1.68	2.35	4.44	4.88
1.578	1.68	2.34	4.43	4.87
1.579	1.68	2.34	4.42	4.86
1.580	1.68	2.34	4.42	4.86

Figure 3.8-4 **Location of gasket load reaction**



NOTE The gasket seating width factors b_o and b shown apply only to flange joints in which the gasket is contained entirely within the inner edges of the bolt holes.

Figure 3.8-5 **Values of *T, U, Y* and *Z***

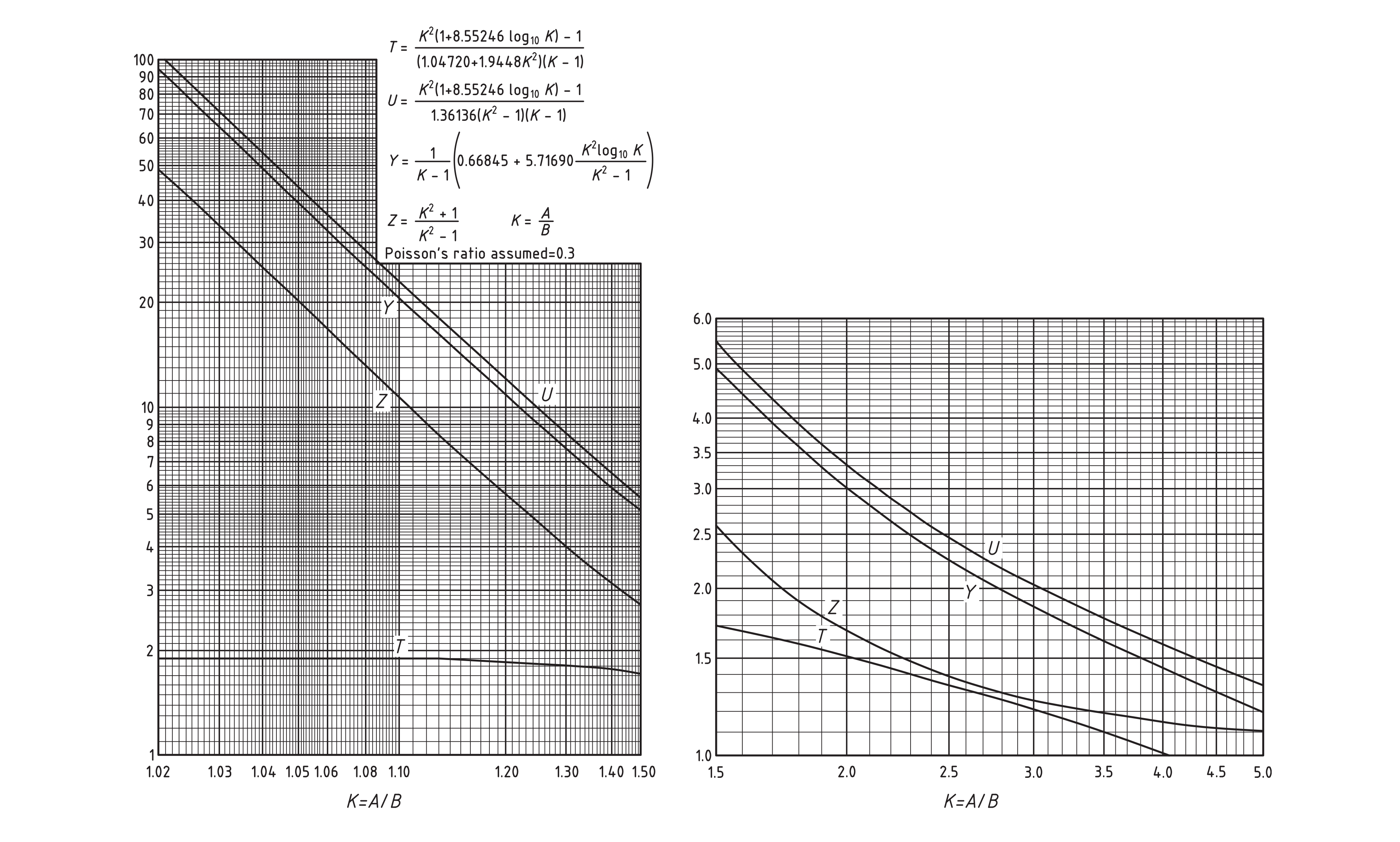


Figure 3.8-6 **Values of *F* (integral method factors)**

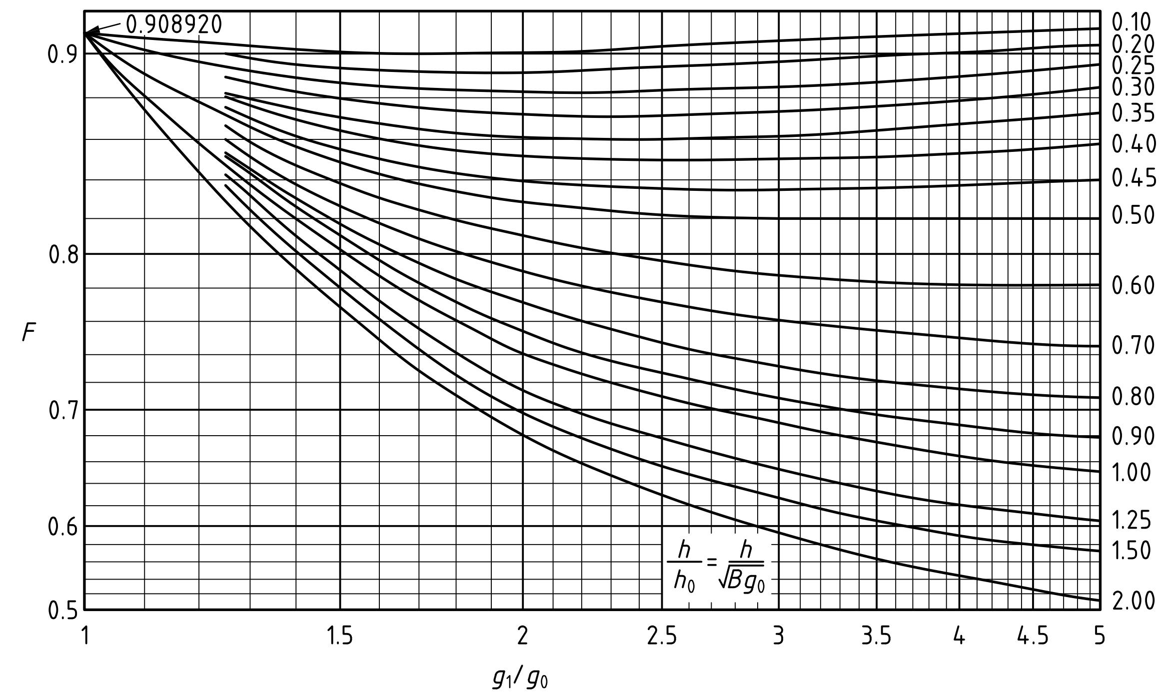


Figure 3.8-7 **Values of *V* (integral method factors)**

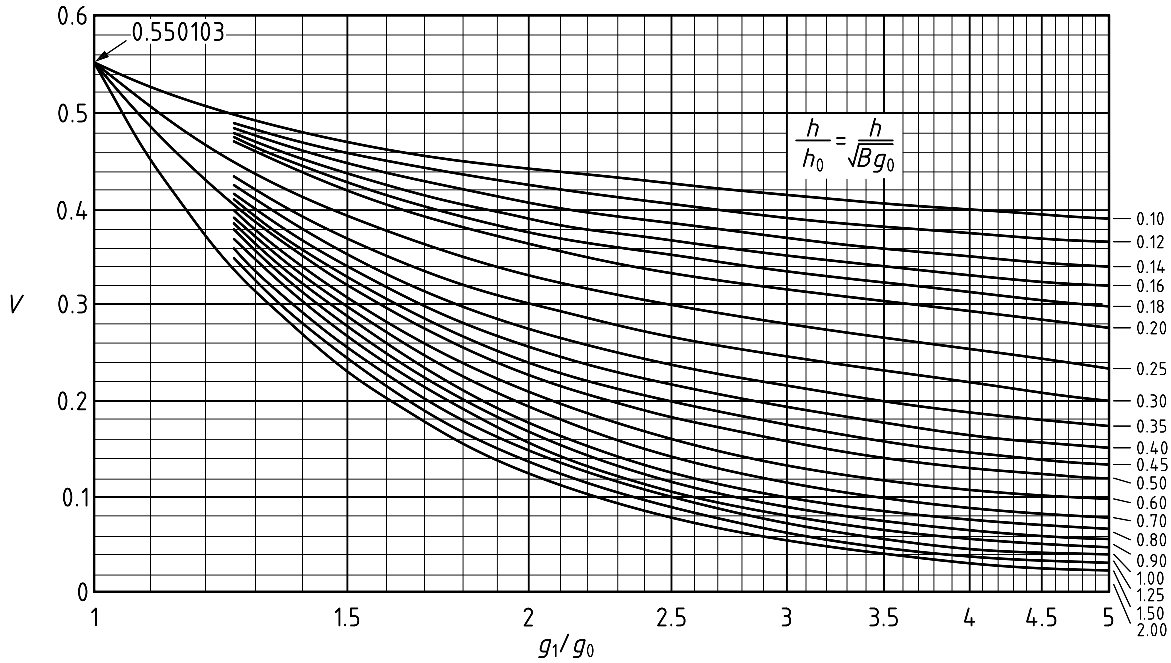


Figure 3.8-8 **Values of F_L (loose hub flange factors)**

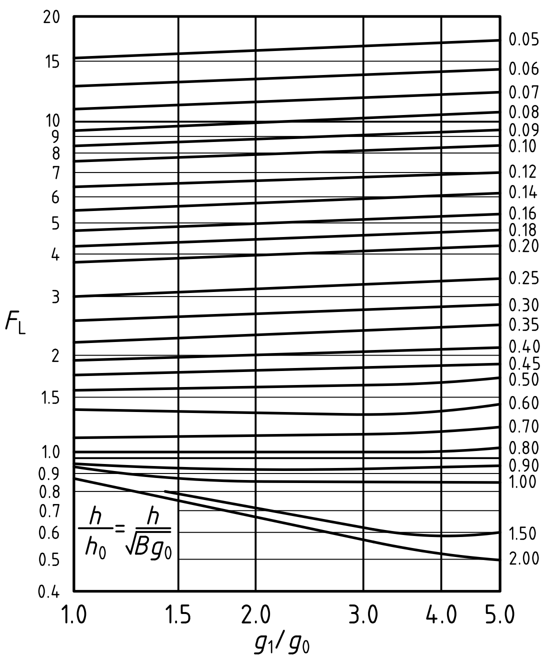


Figure 3.8-9 **Values of V_L (loose hub flange factors)**

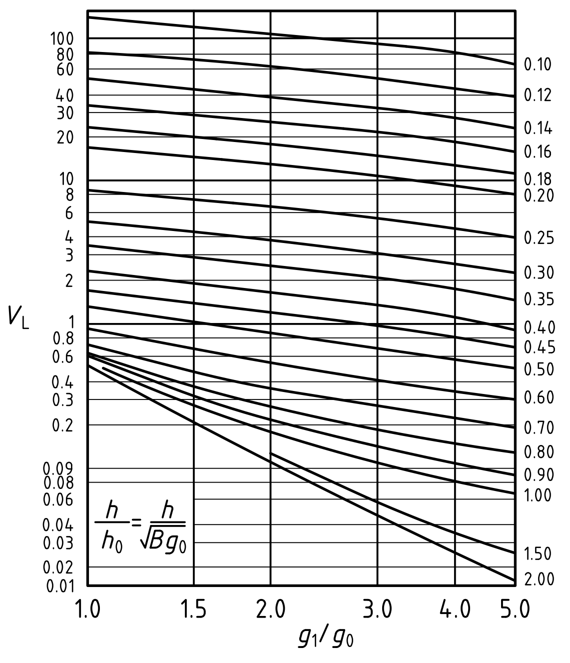


Figure 3.8-10 **Values of *f* (hub stress correction factors)**

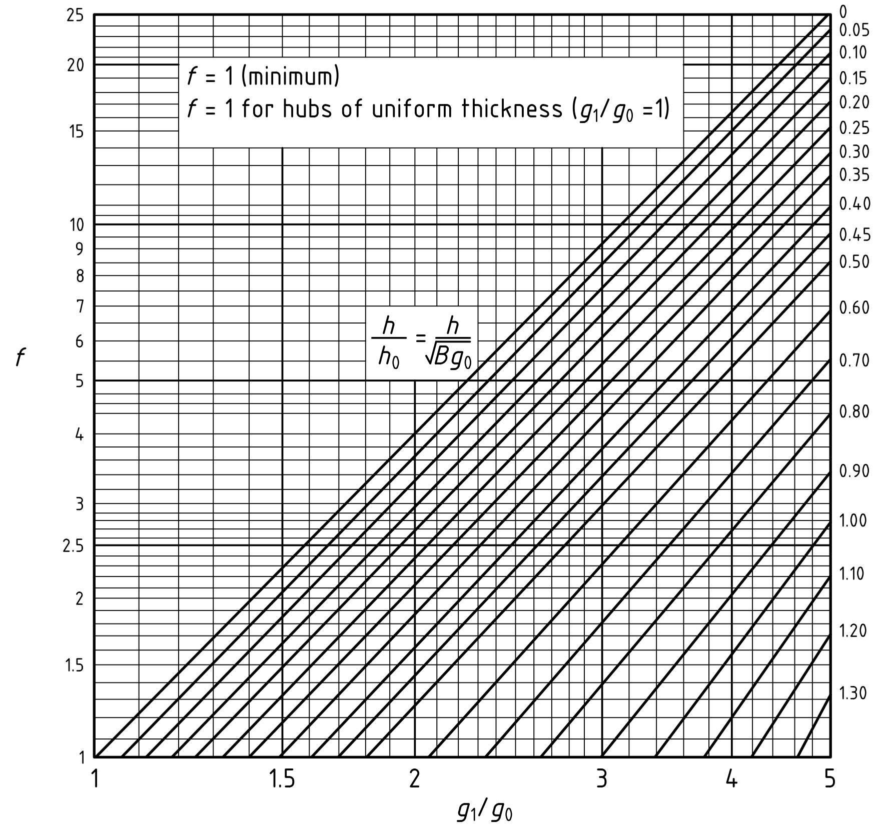


Figure 3.8-11 **Ungasketted, seal-welded-type flanges**

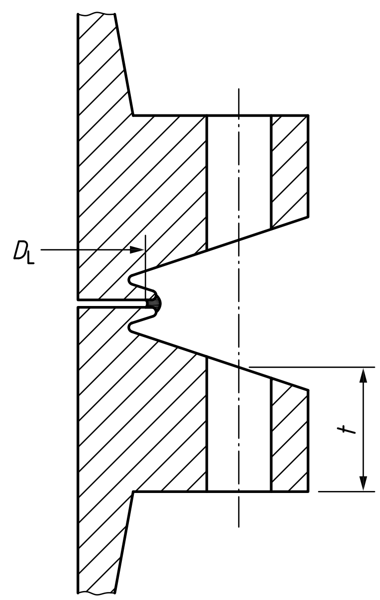
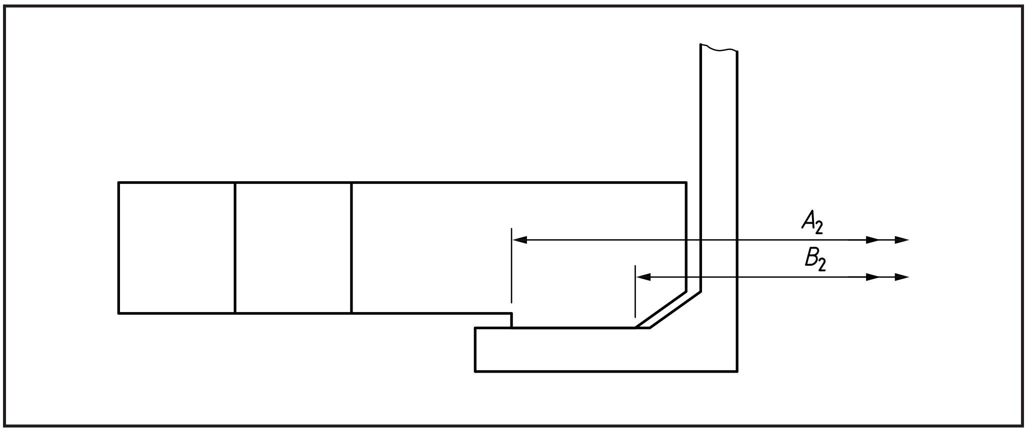
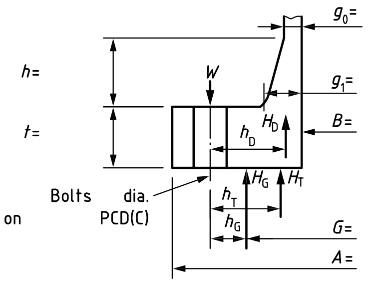


Figure 3.8-12 **Contact face between loose and stub flanges in a lap joint where diameters A_2 and B_2 are defined by the same component**



Suggested working form (1)		
Narrow-face flange design: smooth bore		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Hub/shell material:		
Bolting material:		
Gasket type:		

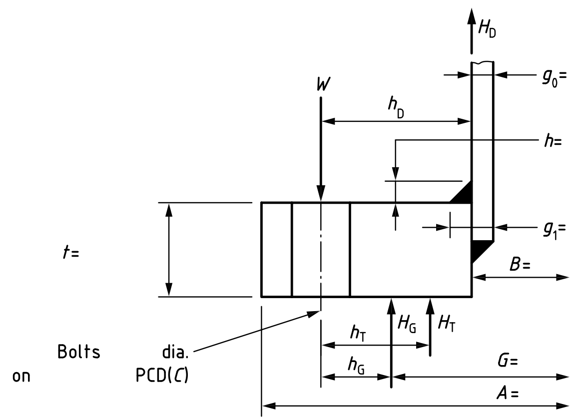


NOTE This configuration is also applicable to parallel hub flanges (f = 1.0).

Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Hub/shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$H = \pi G^2 p/4$ =	$H_G = 2b \times \pi Gmp$ =	
$W_{m1} = H_G + H$ =	$W_{m2} = \pi bGy$ =	
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =	
$A_m = \max(A_{m1}\ ;\ A_{m2})$ =	A_b = number of bolts × root area =	
$W = (A_b + A_m)S_a/2$ =		

Moment arms and moments		
$H_D = \pi B^2 p/4$ =	$h_D = (C - B - g_1)/2$ =	
$H_G = 2b \times \pi Gmp$ =	$h_G = (C - G)/2$ =	
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =	
$H_D h_D$ =		
$H_G h_G$ =		
$H_T h_T$ =		
$M_{op} = H_D h_D + H_G h_G + H_T h_T$ =	$M_{atm} = W h_G$ =	
Loose method criteria		
16 mm > g_0 =	300 > B/g_0 =	
2 N/mm^2 > p =	370 °C > design temperature =	
Shape constants		
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =	
$K = A/B$ =	g_1/g_0 =	
T =	F =	
Z =	V =	
Y =	f =	
U =	$e = F/h_0$ =	
$d = (U/V)h_0 g_0^{\,2}$ =		
Assumed t =		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	$(C_F \geq 1)$	
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = fM/(\lambda g_1^{\,2})$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (2)		
Narrow-face flange design: stepped bore		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Shell material:		
Bolting material:		
Gasket type:		



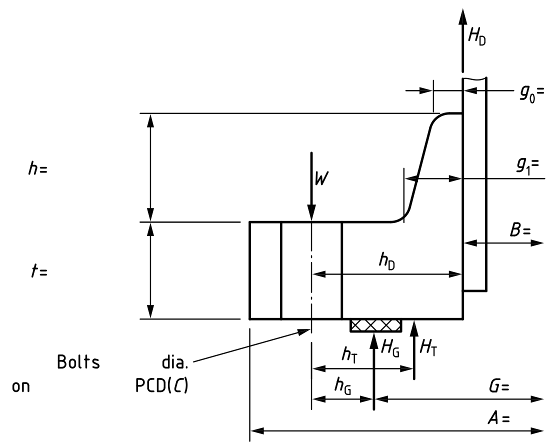
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =

Gasket details (dimensions in millimetres)	
Gasket contact face o.d. =	i.d. =
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =
G = o.d. – $2b$ =	y =
Loads and bolting	
$H = \pi G^2 p/4$ =	$H_G = 2b \times \pi Gmp$ =
$W_{m1} = H_G + H$ =	$W_{m2} = \pi bGy$ =
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =
$A_m = \max(A_{m1}\ ;\ A_{m2})$ =	A_b = number of bolts × root area =
$W = (A_b + A_m)S_a/2$ =	

Moment arms and moments	
$H_D = \pi B^2 p/4$ =	$h_D = (C - B)/2$ =
$H_G = 2b \times \pi Gmp$ =	$h_G = (C - G)/2$ =
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =
$H_D h_D$ =	
$H_G h_G$ =	
$H_T h_T$ =	
$M_{op} = H_D h_D + H_G h_G + H_T h_T$ =	$M_{atm} = Wh_G$ =
Loose method criteria	
16 mm > g_0 =	300 > B/g_0 =
2 N/mm^2 > p =	370 °C > design temperature =
Shape constants	
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =
$K = A/B$ =	g_1/g_0 =
T =	F =
Z =	V =
Y =	f =
U =	$e = F/h_0$ =
$d = (U/V)h_0 g_0^2$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	($C_F \geq 1$)
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	

Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (3)		
Narrow-face flange design: slip-on hubbed type		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Bolting material:		
Gasket type:		



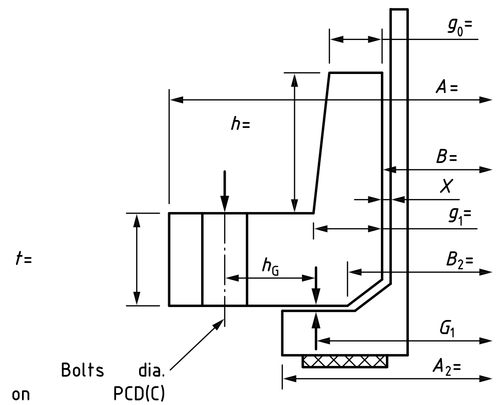
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Bolts	S_b =	S_a =

Gasket details (dimensions in millimetres)	
Gasket contact face o.d. =	i.d. =
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =
G = o.d. – $2b$ =	y =
Loads and bolting	
$H = \pi G^2 p/4$ =	$H_G = 2b \times \pi Gmp$ =
$W_{m1} = H_G + H$ =	$W_{m2} = \pi bGy$ =
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =
$A_m = \max(A_{m1}\ ;\ A_{m2})$ =	A_b = number of bolts × root area =
$W = (A_b + A_m)S_a/2$ =	

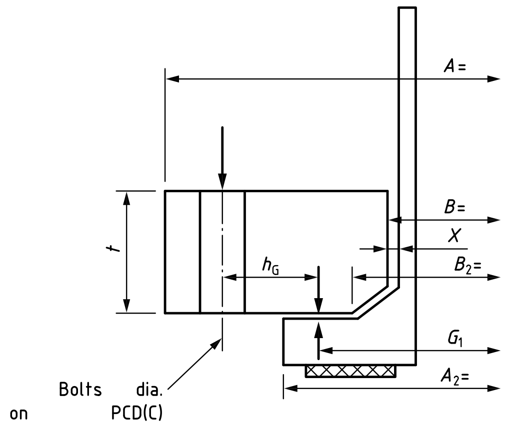
Moment arms and moments	
$H_D = \pi B^2 p/4$ =	$h_D = (C - B)/2$ =
$H_G = 2b \times \pi Gmp$ =	$h_G = (C - G)/2$ =
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =
$H_D h_D$ =	
$H_G h_G$ =	
$H_T h_T$ =	
$M_{op} = H_D h_D + H_G h_G + H_T h_T$ =	$M_{atm} = W h_G$ =
Shape constants	
$h_0 = \sqrt{B g_0}$ =	h/h_0 =
$K = A/B$ =	g_1/g_0 =
T =	F_L =
Z =	V_L =
Y =	$f = 1$
U =	$e = F_L/h_0$ =
$d = (U/V_L) h_0 g_0^2$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m+0.5)}}$ =	($C_F \geq 1$)
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	
Stress calculation	

	Operating condition	Bolting-up condition
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = M/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =		

Conclusion:

Suggested working form (4)		
Lap-type joint: loose flange with hub		
The calculation requires two working forms. This one is for the loose flange. The stub flange is covered by a working form for non-lap jointed flanges, in which C is to be replaced by G_1 taken from this form. The stub flange working form covers design conditions, gasket and bolting matters and provides values of W_{m1} and W for use in this form.		
Corrosion allowance =		
Flange material:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Contact face		
X = gap between shell and loose flange =		
Bearing surface area = $(\pi/2) \times \min[(A_2 - X)^2 - G_1^2; G_1^2 - (B_2 + X)^2]$ =		
Bolt load	W_{m1} =	W =
Bearing stress	=	=
Allowable bearing stress: (1.5 × lower of design stresses for lap and stub)	=	=
$G_1 = (A_2 + B_2)/2$	=	

Moment arms and moments		
$h_G = (C - G_1)/2$ =		
W_{m1} =	W =	
$M_{op} = W_{m1}h_G$ =	$M_{atm} = Wh_G$ =	
Shape constants		
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =	
$K = A/B$ =	g_1/g_0 =	
T =	F_L =	
Z =	V_L =	
Y =	$f = 1$	
U =	$e = F_L/h_0$ =	
$d = (U/V_L)h_0g_0^2$ =		
$C_F = 1$		
Assumed t =		
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = M/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R ; S_T)]$ =		
Conclusion:		

Suggested working form (5)		
Lap-type joint: loose flange without hub		
The calculation requires two working forms. This one is for the loose flange. The stub flange is covered by a working form for non-lap jointed flanges, in which C is to be replaced by G_1 taken from this form. The stub flange working form covers design conditions, gasket and bolting matters and provides values of W_{m1} and W for use in this form.		
Corrosion allowance =		
Flange material:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Contact face		
X = gap between shell and loose flange =		
Bearing surface area = $(\pi/2) \times \min[(A_2 - X)^2 - G_1^2; G_1^2 - (B_2 + X)^2]$ =		
Bolt load	W_{m1} =	W =
Bearing stress	=	=
Allowable bearing stress: (1.5 × lower of design stresses for lap and stub)	=	=
$G_1 = (A_2 + B_2)/2$	=	

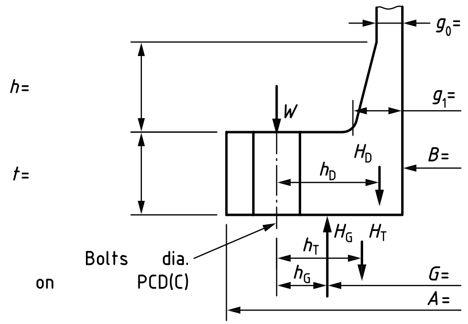
Moment arms and moments	
$h_G = (C - G_1)/2$ =	
W_{m1} =	W =
$M_{op} = W_{m1} h_G$ =	$M_{atm} = W h_G$ =
Shape constants	
$K = A/B$ =	Y =
$C_F = 1$	
Thickness calculation	
$M = M_{op} C_F / B$ =	$M = M_{atm} C_F / B$ =
YM/S_{FO} =	YM/S_{FA} =

$$t = \sqrt{\max\left(\frac{YM}{S_{FO}}, \frac{YM}{S_{FA}}\right)} =$$

where S_{FO} and S_{FA} are the design stresses of the flange material at the design temperature and at atmospheric temperature respectively, divided by the factor k, as defined in **3.8.3.4.2**, if the inside diameter of the shell D > 1 000 mm.

Conclusion:

Suggested working form (6)		
Narrow-face flange design: smooth bore (external pressure case)		
Design external pressure p_e =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Hub/shell material:		
Bolting material:		
Gasket type:		

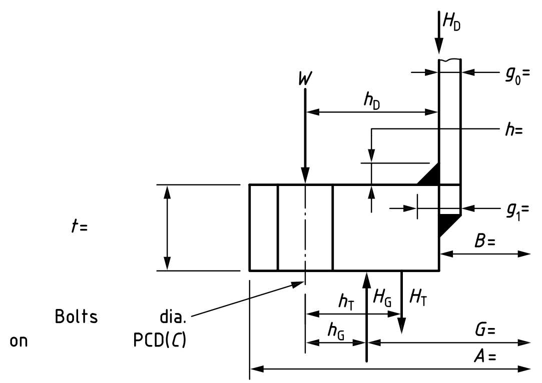


NOTE This configuration is also applicable to parallel hub flanges (f = 1.0).

Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Hub/shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\,;\, 2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$W_{m2} = \pi bGy$ =		
$A_m = A_{m2} = W_{m2}/S_a$ =		
A_b = number of bolts × root area =		
$W = (A_b + A_m)S_a/2$ =		

Moment arms and moments		
$H_D = \pi B^2 p_e/4$ =	$h_D = (C - B - g_1)/2$ =	
$H = \pi G^2 p_e/4$ =	$h_G = (C - G)/2$ =	
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =	
$H_D(h_D - h_G)$ =		
$H_T(h_T - h_G)$ =		
$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$ =	$M_{atm} = Wh_G$ =	
Loose method criteria		
16 mm > g_0 =	300 > B/g_0 =	
370 °C > design temperature =		
Shape constants		
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =	
$K = A/B$ =	g_1/g_0 =	
T =	F =	
Z =	V =	
Y =	f =	
U =	$e = F/h_0$ =	
$d = (U/V)h_0 g_0^2$ =		
Assumed t =		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m+0.5)}}$ =	$(C_F \geq 1)$	
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\,;\,S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (7)		
Narrow-face flange design: stepped bore (external pressure case)		
Design external pressure p_e =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Shell material:		
Bolting material:		
Gasket type:		



Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =

Gasket details (dimensions in millimetres)	
Gasket contact face o.d. =	i.d. =
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =
G = o.d. – $2b$ =	y =

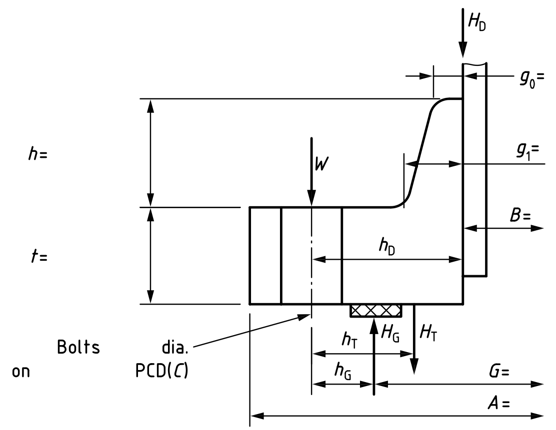
Loads and bolting
$W_{m2} = \pi bGy$ =
$A_m = A_{m2} = W_{m2}/S_a$ =
A_b = number of bolts × root area =
$W = (A_b + A_m)S_a/2$ =

Moment arms and moments	
$H_D = \pi B^2 p_e/4$ =	$h_D = (C - B)/2$ =
$H = \pi G^2 p_e/4$ =	$h_G = (C - G)/2$ =
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =
$H_D(h_D - h_G)$ =	
$H_T(h_T - h_G)$ =	
$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$ =	$M_{atm} = Wh_G$ =
Loose method criteria	
16 mm > g_0 =	$300 > B/g_0$ =
370 °C > design temperature =	
Shape constants	
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =
$K = A/B$ =	g_1/g_0 =
T =	F =
Z =	V =
Y =	f =
U =	$e = F/h_0$ =
$d = (U/V)h_0 g_0^2$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m+0.5)}}$ =	$(C_F \geq 1)$
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	

Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =		
(loose option) YM/t^2 =		

Conclusion:

Suggested working form (8)		
Narrow-face flange design: slip-on hubbed type (external pressure case)		
Design external pressure p_e =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Bolting material:		
Gasket type:		



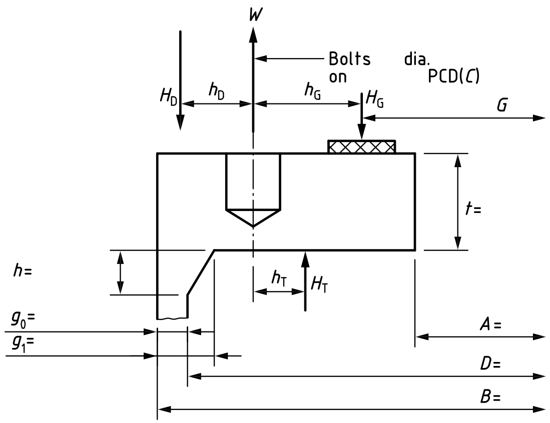
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$W_{m2} = \pi bGy$ =		
$A_m = A_{m2} = W_{m2}/S_a$ =		
A_b = number of bolts × root area =		
$W = (A_b + A_m)S_a/2$ =		

Moment arms and moments	
$H_D = \pi B^2 p_e/4$ =	$h_D = (C - B)/2$ =
$H = \pi G^2 p_e/4$ =	$h_G = (C - G)/2$ =
$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =
$H_D(h_D - h_G)$ =	
$H_T(h_T - h_G)$ =	
$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$ =	$M_{atm} = Wh_G$ =
Shape constants	
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =
$K = A/B$ =	g_1/g_0 =
T =	F_L =
Z =	V_L =
Y =	$f = 1$
U =	$e = F_L/h_0$ =
$d = (U/V_L)h_0 g_0^2$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	($C_F \geq 1$)
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	

Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/B$ or $M_{atm}C_F/B$		
$S_H = M/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =		

Conclusion:

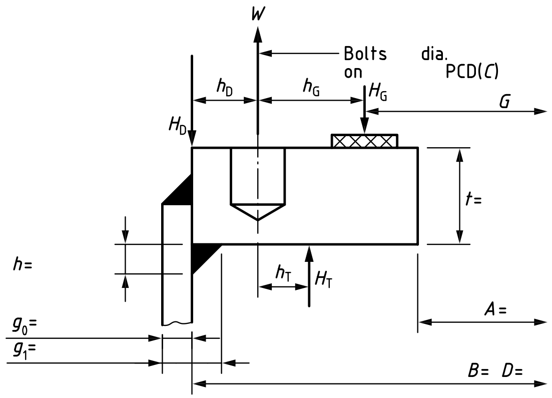
Suggested working form (9)		
Reverse narrow-face flange design: flange diameter = shell outside diameter		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Hub/shell material:		
Bolting material:		
Gasket type:		



Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Hub/shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o ; 2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$H = \pi G^2 p/4$ =	$H_G = 2b \times \pi G m p$ =	
$W_{m1} = H_G + H$ =	$W_{m2} = \pi b G y$ =	
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =	
$A_m = \max(A_{m1} ; A_{m2})$ =	A_b = number of bolts × root area =	
$W = (A_b + A_m)S_a/2$ =		

Moment arms and moments		
$H_D = \pi D^2 p/4$ =	$h_D = (B - C - g_1)/2$ =	
	$h_G = (C - G)/2$ =	
$H_T = H_D - H$ =	$h_T = (2C - D - G)/4$ =	
$H_D h_D$ =		
$H_T h_T$ =		
$M_{op} = H_D h_D + H_T h_T$ =	$M_{atm} = W h_G$ =	
Loose method criteria		
2 N/mm^2 > p =	370 °C > design temperature =	
Shape constants		
$h_0 = \sqrt{B g_0}$ =	h/h_0 =	
$K = B/A$ =	g_1/g_0 =	
T =	F =	
Z =	V =	
Y =	f =	
U =	$e = F/h_0$ =	
$d = (U/V) h_0 g_0^2$ =		
Assumed t =		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	$(C_F \geq 1)$	
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/A$ or $M_{atm}C_F/A$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R ; S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (10)		
Reverse narrow-face flange design: slip-in type		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Shell material:		
Bolting material:		
Gasket type:		



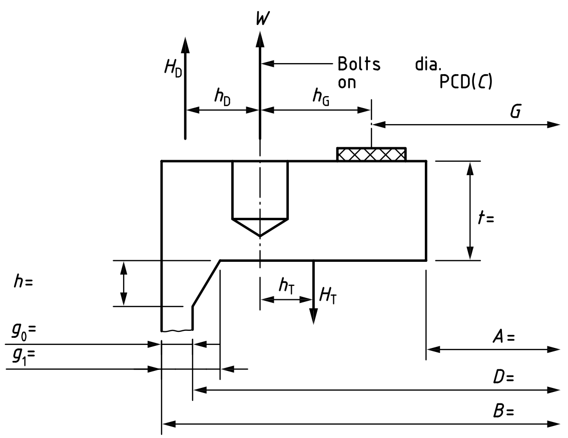
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =

Gasket details (dimensions in millimetres)	
Gasket contact face o.d. =	i.d. =
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =
$b = \min(b_o\,;\,2.52\sqrt{b_o})$ =	m =
G = o.d. – $2b$ =	y =
Loads and bolting	
$H = \pi G^2 p/4$ =	$H_G = 2b \times \pi G m p$ =
$W_{m1} = H_G + H$ =	$W_{m2} = \pi b G y$ =
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =
$A_m = \max(A_{m1}\,;\,A_{m2})$ =	A_b = number of bolts × root area =
$W = (A_b + A_m)S_a/2$ =	

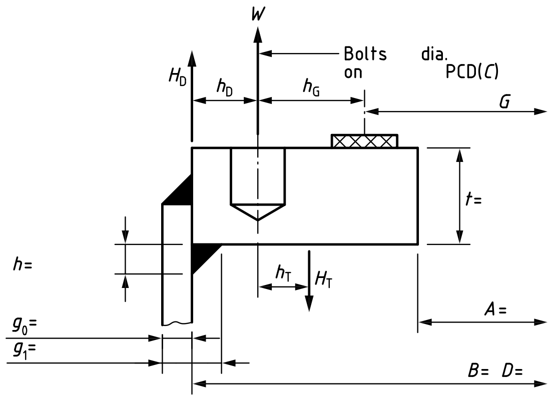
Moment arms and moments	
$H_D = \pi D^2 p/4$ =	$h_D = (B - C)/2$ =
	$h_G = (C - G)/2$ =
$H_T = H_D - H$ =	$h_T = (2C - D - G)/4$ =
$H_D h_D$ =	
$H_T h_T$ =	
$M_{op} = H_D h_D + H_T h_T$ =	$M_{atm} = W h_G$ =
Loose method criteria	
2 N/mm^2 > p =	370 °C > design temperature =
Shape constants	
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =
$K = B/A$ =	g_1/g_0 =
T =	F =
Z =	V =
Y =	f =
U =	$e = F/h_0$ =
$d = (U/V)h_0 g_0^{\,2}$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m+0.5)}}$ =	($C_F \geq 1$)
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	

Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/A$ or $M_{atm}C_F/A$		
$S_H = fM/(\lambda g_1^{\,2})$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\,;\,S_T)]$ =		
(loose option) YM/t^2 =		

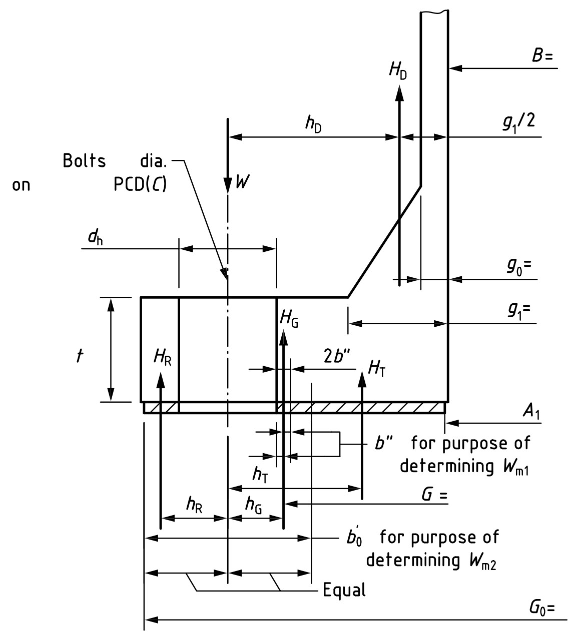
Conclusion:

Suggested working form (11)		
Reverse narrow-face flange design: flange diameter = shell outside diameter (external pressure case)		
Design external pressure p_e =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Hub/shell material:		
Bolting material:		
Gasket type:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Hub/shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\,;\,2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$W_{m2} = \pi bGy$ =	$A_m = A_{m2} = W_{m2}/S_a$ =	
A_b = number of bolts × root area =		
$W = (A_b + A_m)S_a/2$ =		

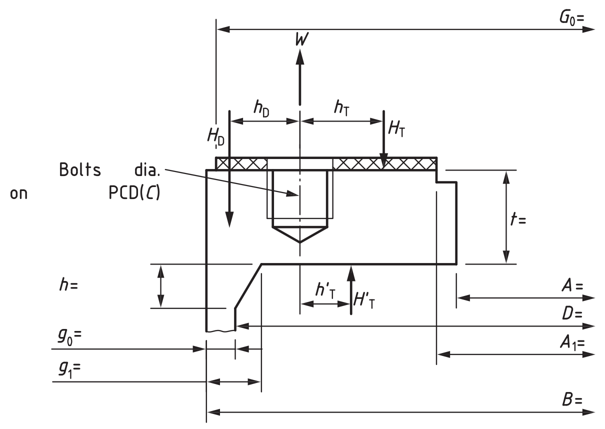
Moment arms and moments		
$H_D = \pi D^2 p_e/4$ =	$h_D = (B - C - g_1)/2$ =	
$H = \pi G^2 p_e/4$ =	$h_G = (C - G)/2$ =	
$H_T = H_D - H$ =	$h_T = (2C - D - G)/4$ =	
$H_D(h_D - h_G)$ =		
$H_T(h_T - h_G)$ =		
$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$ =	$M_{atm} = Wh_G$ =	
Loose method criteria		
370 °C $>$ design temperature =		
Shape constants		
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =	
$K = B/A$ =	g_1/g_0 =	
T =	F =	
Z =	V =	
Y =	f =	
U =	$e = F/h_0$ =	
$d = (U/V)h_0 g_0^2$ =		
Assumed t =		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	$(C_F \geq 1)$	
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/A$ or $M_{atm}C_F/A$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\,;\,S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (12)		
Reverse narrow-face flange design: slip-in type (external pressure case)		
Design external pressure p_e =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Shell material:		
Bolting material:		
Gasket type:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	S_{FA} =
Shell material	S_{HO} =	S_{HA} =
Bolts	S_b =	S_a =
Gasket details (dimensions in millimetres)		
Gasket contact face o.d. =	i.d. =	
N = (o.d. – i.d.)/2 =	$b_o = N/2$ =	
$b = \min(b_o\,;\,2.52\sqrt{b_o})$ =	m =	
G = o.d. – $2b$ =	y =	
Loads and bolting		
$W_{m2} = \pi bGy$ =		
$A_m = A_{m2} = W_{m2}/S_a$ =		
A_b = number of bolts × root area =		
$W = (A_b + A_m)S_a/2$ =		

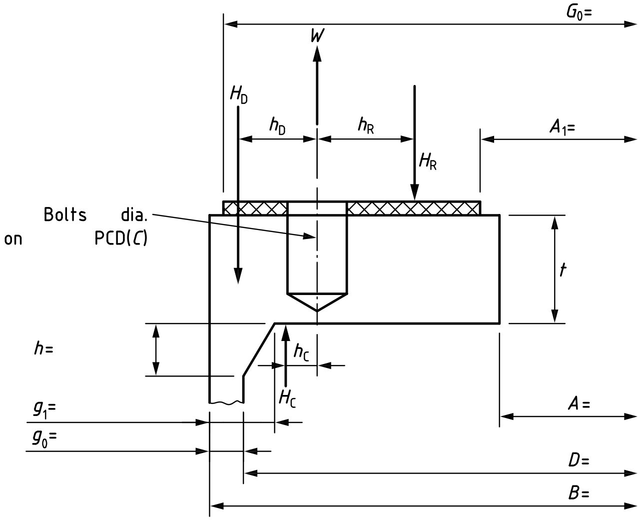
Moment arms and moments		
$H_D = \pi D^2 p_e/4$ =	$h_D = (B - C)/2$ =	
$H = \pi G^2 p_e/4$ =	$h_G = (C - G)/2$ =	
$H_T = H_D - H$ =	$h_T = (2C - D - G)/4$ =	
$H_D(h_D - h_G)$ =		
$H_T(h_T - h_G)$ =		
$M_{op} = H_D(h_D - h_G) + H_T(h_T - h_G)$ =	$M_{atm} = Wh_G$ =	
Loose method criteria		
370 °C > design temperature =		
Shape constants		
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =	
$K = B/A$ =	g_1/g_0 =	
T =	F =	
Z =	V =	
Y =	f =	
U =	$e = F/h_0$ =	
$d = (U/V)h_0 g_0^2$ =		
Assumed t =		
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{(2 \times \text{bolt outside diameter}) + 6t/(m + 0.5)}}$ =	$(C_F \geq 1)$	
$\beta = 1.333te + 1$ =		
$\lambda = (te + 1)/T + t^3/d$ =		
Stress calculation		
	Operating condition	**Bolting-up condition**
$M = M_{op}C_F/A$ or $M_{atm}C_F/A$		
$S_H = fM/(\lambda g_1^2)$ =		
$S_R = \beta M/(\lambda t^2)$ =		
$S_T = YM/t^2 - ZS_R$ =		
$0.5[S_H + \max(S_R\,;\,S_T)]$ =		
(loose option) YM/t^2 =		
Conclusion:		

Suggested working form (13)		
Full-face flange design with soft ring type gaskets		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Bolting material:		
Gasket type:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	
Bolts	S_b =	S_a =
Elastic modulus for flange	E =	

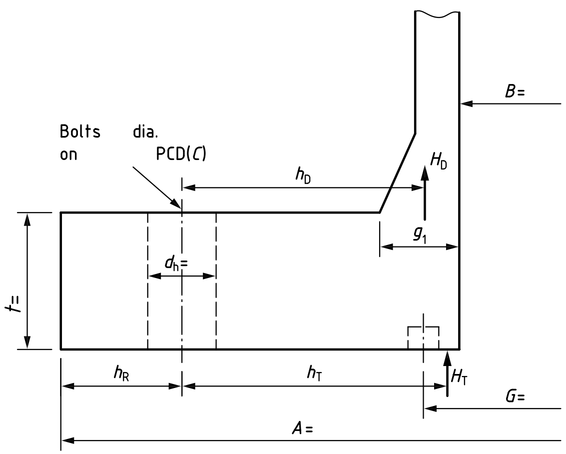
Gasket details (dimensions in millimetres)	
b'_o = lesser of $(G_o - C)$ and $(C - A_1)$ =	$2b''$ = 5 mm
$b' = 4\sqrt{b'_o}$ =	m =
$G = C - d_h - 2b''$	y =
Loads, moment arms and moments	
$H = \pi(C - d_h)^2 p/4$ =	
$H_D = \pi B^2 p/4$ =	$h_D = (C - B - g_1)/2$ =
$H_G = 2b'' \times \pi G m p$ =	$h_G = (d_h + 2b'')/2$ =
$H_T = H - H_D$ =	$h_T = (C + d_h + 2b'' - B)/4$ =
$H_D h_D$ =	
$H_G h_G$ =	
$H_T h_T$ =	
$M = H_D h_D + H_G h_G + H_T h_T$ =	$h_R = (G_o - C + d_h)/4$ =
$H_R = M/h_R$ =	
Bolting	
$W_{m1} = H_G + H + H_R$ =	$W_{m2} = \pi C b' y$ =
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =
$A_m = \max(A_{m1}\,;\,A_{m2})$ =	A_b = number of bolts × root area =
Bolt spacing =	Bolt o.d. d_b =
Thickness calculation	
Pressure thickness: $t = \sqrt{\dfrac{6M}{S_{FO}\left(\pi C - n d_h\right)}} =$ Bolt spacing criterion: $t = \dfrac{\left(\text{Bolt spacing} - 2d_b\right)\times\left(m + 0.5\right)}{6\times\left(E/200\,000\right)^{0.25}} =$	
Conclusion:	

Suggested working form (14)		
Reverse full-face flange to 3.8.7 using the design method of 3.8.3		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Hub/shell material:		
Bolting material:		
Gasket type:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	
Hub/shell material	S_{HO} =	
Bolts	S_b =	S_a =
Elastic modulus for flange	E =	
Gasket details (dimensions in millimetres)		
$N = (C - A_1)/2$ =	$b_o = N/2$ =	
$b = \min(b_o\ ;\ 2.52\sqrt{b_o})$ =	m =	y =
$b'_o = G_o - C$ =	$b' = 4\sqrt{b'_o}$ =	
Loads and bolting		
$H = \pi p(C - d_h)^2/4$ =	$H_G = 2b \times \pi Cmp$ =	
$W_{m1} = H_G + H$ =	$W_{m2} = \pi Cb'y$ =	
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =	
$A_m = \max(A_{m1}\ ;\ A_{m2})$ =	A_b = number of bolts × root area =	
Bolt spacing =	Bolt o.d. d_b =	

Moment arms and moments	
$H_D = \pi pD^2/4$ =	$h_D = (B - C - g_1)/2$ =
$H'_T = H_D - \pi pA_1^2/4$ =	$h'_T = (2C - D - A_1)/4$ =
$H_T = (H - H_D + H'_T)/2$ =	$h_T = (2C + d_h - 2A_1)/6$ =
$H_D h_D$ =	
$H'_T h'_T$ =	
$H_T h_T$ =	
$M_{op} = H_D h_D + H'_T h'_T - H_T h_T$ =	
Loose method criteria	
2 N/mm^2 > p =	370 °C > design temperature =
Shape constants	
$h_0 = \sqrt{Bg_0}$ =	h/h_0 =
$K = B/A$ =	g_1/g_0 =
T =	F =
Z =	V =
Y =	f =
U =	$e = F/h_0$ =
$d = (U/V)h_0 g_0^2$ =	
Assumed t =	
$C_F = \sqrt{\dfrac{\text{bolt spacing}}{\left(2 \times \text{bolt outside diameter}\right) + 6t/\left(m + 0.5\right)}}$ =	($C_F \geq 1$)
$\beta = 1.333te + 1$ =	
$\lambda = (te + 1)/T + t^3/d$ =	
Stress calculation	
$M = M_{op}C_F/A$ =	
$S_H = fM/(\lambda g_1^2)$ =	
$S_R = \beta M/(\lambda t^2)$ =	
$S_T = YM/t^2 - ZS_R$ =	
$0.5[S_H + \max(S_R\ ;\ S_T)]$ =	
(loose option) YM/t^2 =	
Conclusion:	

Suggested working form (15)		
Reverse full-face flange design to 3.8.7 using the design method of 3.8.4		
Design pressure p =		
Design temperature =		
Corrosion allowance =		
Flange material:		
Bolting material:		
Gasket type:		
		
Design stresses	Operating	Ambient
Flange material	S_{FO} =	
Bolts	S_b =	S_a =
Elastic modulus for flange	E =	
Gasket details (dimensions in millimetres)		
$b'_o = G_o - C$ =		$2b''$ = 5 mm
$b' = 4\sqrt{b'_o}$		m =
		y =

Loads, moment arms and moments	
$H_D = \pi p D^2/4$ =	$h_D = (B - C - g_1)/2$ =
$H_C = H_D - \pi p C^2/4$ =	$H_C = (D - C)/4$ =
$H_D h_D$ =	
$H_C h_C$ =	
$M = M_1 = H_D h_D - H_C h_C$ =	
$h_R = (C - A_1 + d_h)/4$ =	
$H_R = M/h_R$ =	
$W_{m1} = H_R + H_D - H_C$ =	
Bolting	
W_{m1} =	$W_{m2} = \pi C b' y$ =
$A_{m1} = W_{m1}/S_b$ =	$A_{m2} = W_{m2}/S_a$ =
$A_m = \max(A_{m1}\,;\,A_{m2})$ =	A_b = number of bolts × root area =
Bolt spacing =	Bolt o.d. d_b =
Thickness calculation	
Pressure thickness: $t = \sqrt{\dfrac{6M}{S_{FO}\left(\pi C - n d_h\right)}} =$ Bolt spacing criterion: $t = \dfrac{\left(\text{Bolt spacing} - 2d_b\right)\times\left(m + 0.5\right)}{6\times\left(E/200\,000\right)^{0.25}} =$	
Conclusion:	

Suggested working form (16)			Loads, moment arms and moments	
Full-face flange with metal/metal contact			*NOTE If the mating flange is of a smaller diameter and designed in accordance with this method, M is the value for this other flange.*	
Design pressure p =				
Design temperature =			$H = \pi G^2 p/4$ =	
Corrosion allowance =			$H_D = \pi B^2 p/4$ =	$h_D = (C - B - g_1)/2$ =
Flange material:			$H_T = H - H_D$ =	$h_T = (2C - B - G)/4$ =
Bolting material:			$H_D h_D$ =	
Gasket type:			$H_T h_T$ =	
			$M = H_D h_D + H_T h_T$ =	
			$h_R = (A - C)/2$ =	
			$H_R = M/h_R$ =	
			Bolting	
			$W_{m1} = H + H_R$ =	
			$A_m = A_{m1} = W_{m1}/S_b$ =	
			A_b = number of bolts × root area =	
			Thickness calculation	
			For any flange: $t = \sqrt{\dfrac{6M}{S_{FO}\left(\pi C - n d_h\right)}} =$ If the mating flange is of larger diameter and designed in accordance with this method: M for other flange = M_2 = $t \geq \sqrt{\dfrac{3\left(M - M_2\right)\left(A + B\right)}{\pi S_{FO} B\left(A - B\right)}} =$	
Design stresses	Operating	Ambient	**Conclusion:**	
Flange material	S_{FO} =			
Bolts	S_b =	S_a =		
Gasket details				
Mean diameter G =				

3.9 Flat heat exchanger tubesheets

The minimum thickness of flat heat exchanger tubesheets shall be calculated in accordance with **3.9.1** to **3.9.6**, the analyses used to obtain the equations being based on the following assumptions.

a) The tubes are of uniform size.

b) Where the exchanger has a pair of tubesheets, they are both of the same thickness.

c) The tubesheet is of constant thickness across the specified diameter.

d) The tubed area is uniformly perforated and nominally circular (untubed partition lanes in multipass units are accepted).

e) Any untubed annular ring is sufficiently narrow to be treated as a ring whose cross-section rotates without appreciable distortion (i.e. D_1 and $D_2 \leq D_o + 6e$).

f) The tubesheet analysis thickness is not less than:

 0.75 times tube o.d. for tube o.d. ≤ 25 mm;

 22 mm for 25 mm < tube o.d. ≤ 30 mm;

 25 mm for 30 mm < tube o.d. ≤ 40 mm;

 30 mm for 40 mm < tube o.d. ≤ 50 mm;

 0.6 times tube o.d. for tube o.d. > 50 mm.

g) The tubesheet thickness shall not be less than that required in the tubed area except that:

 1) where there is a local reduction of thickness in the untubed area for a gasket groove or for a relief groove at the junction with the shell/channel, the analysis thickness of this reduced area shall be at least 0.8 times the analysis thickness of the tubed area;

 2) where there is a flanged extension to the tubesheet its minimum thickness shall conform to **3.9.7**.

NOTE The derivation of these rules is given in Part 4 of PD 6550, the Explanatory Supplement to BS 5500.

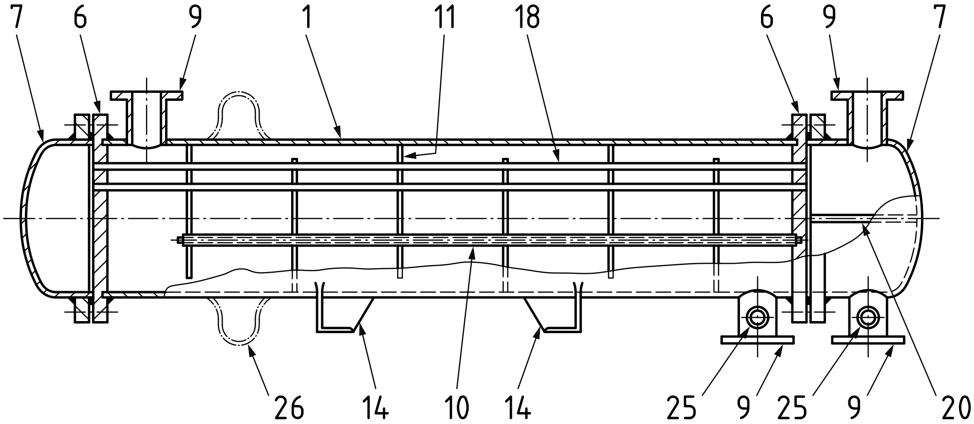
Tubesheets not based upon assumptions a) to g) shall be in accordance with an alternative design method as permitted in **3.2.2**b).

When tubes are expanded into the tubesheet and not welded, the analysis thickness of the tubesheet in the area of the expansion shall be not less than the tube o.d. Where leakage cannot be tolerated, the minimum thickness of tubesheets with expanded only tubejoints shall be 35 mm, unless satisfactory performance has been demonstrated with thinner tubesheets. The nominal tubesheet thickness, shall in no case be less than 19 mm.

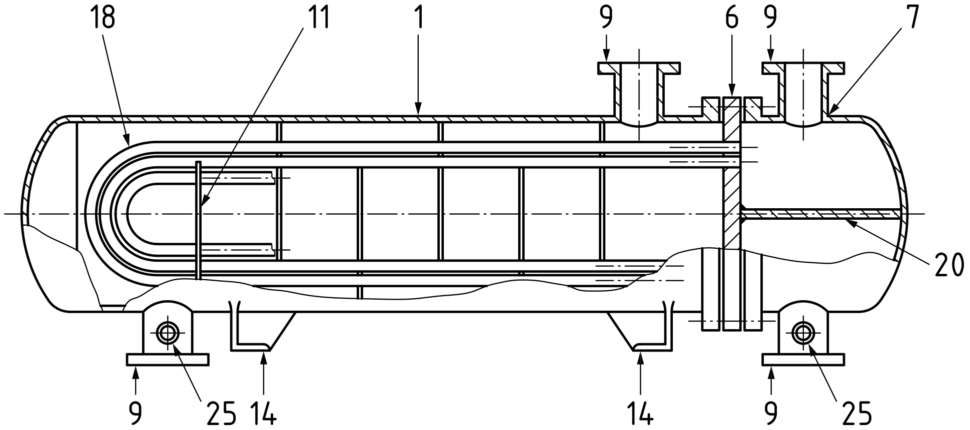
Typical examples of heat exchangers are shown in Figure 3.9-1. The following terminology is used, the item numbers correspond to those shown in Figure 3.9-1. The term “tube bundle“ (item 19) includes items 6, 10, 11, 12, 13 and 18, with the addition of item 4 in floating head heat exchangers.

Item No.	1	Shell
	2	Shell cover
	3	Floating head cover
	4	Floating tubeplate or tubesheet
	5	Clamp ring
	6	Stationary tubeplate or tubesheet
	7	Channel
	8	Channel cover
	9	Nozzle or branch
	10	Tie rod and spacer
	11	Transverse baffle or tube support plate
	12	Impingement plate
	13	Longitudinal baffle
	14	Saddle support
	15	Floating head support
	16	Weir plate
	17	Split ring
	18	Tube (straight or U-tube)
	19	Tube bundle
	20	Pass partition plate
	21	Floating head gland
	22	Floating head gland ring
	23	Vent connection
	24	Drain connection
	25	Instrument or test connection
	26	Expansion joint or bellows
	27	Lifting ring or lug

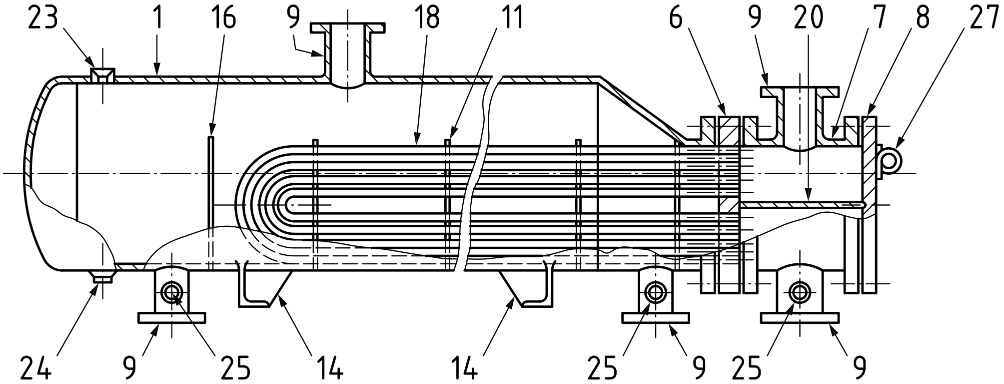
Figure 3.9-1 **Shell and tube heat exchangers**



a) Fixed tubesheet exchanger

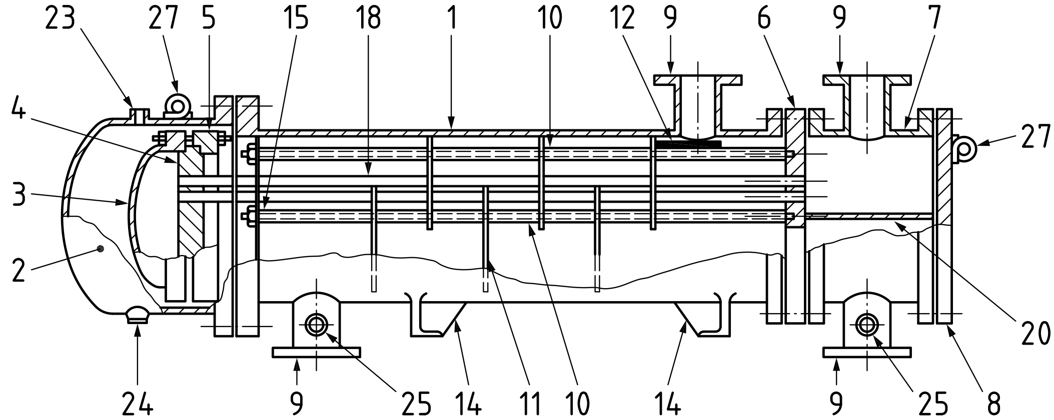


b) U-tube exchanger

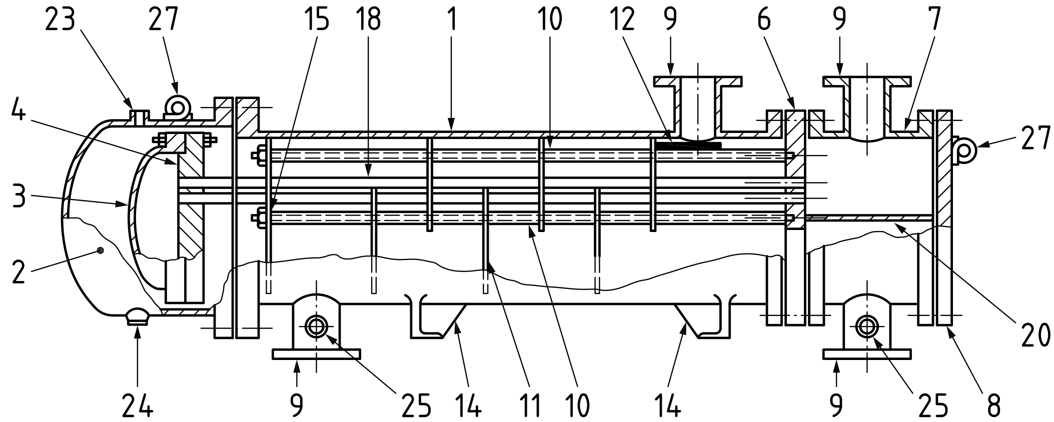


c) U-tube reboiler or evaporator

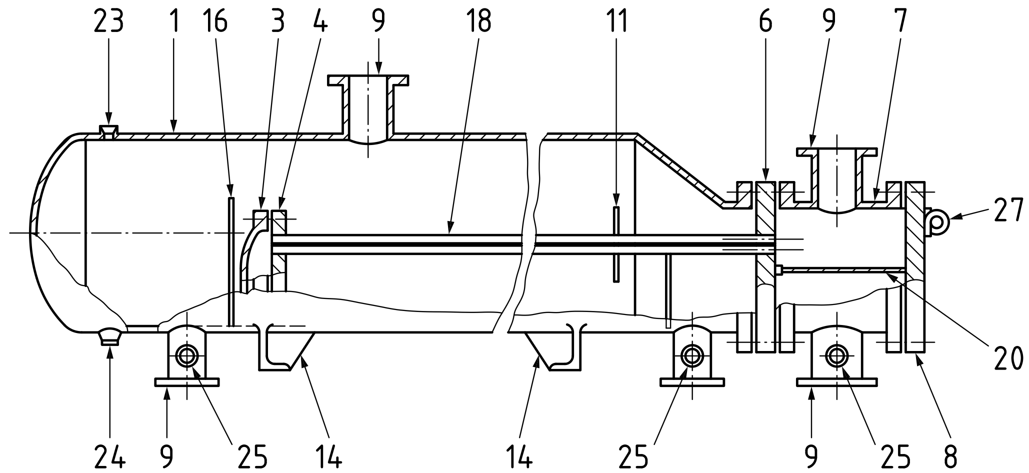
Figure 3.9-1 **Shell and tube heat exchangers (continued)**



d) Floating head exchanger with clamp ring

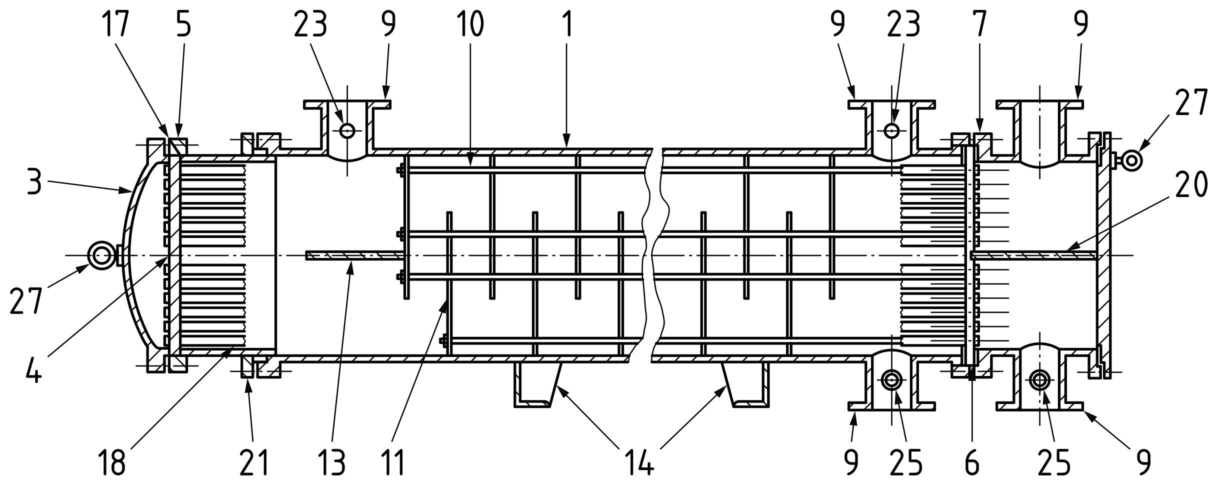


e) Floating head exchanger without clamp ring



f) Floating head reboiler or evaporator without clamp ring

Figure 3.9-1 **Shell and tube heat exchangers (continued)**



g) Floating head exchanger external packed

3.9.1 Notation

For the purposes of **3.9.2**, **3.9.3** and **3.9.4** the following major symbols apply. All dimensions are in the corroded condition (see **3.1.5**), except where otherwise indicated.

C = $C_o + \Delta C$ design factor to be derived from Figure 3.9-4 in conjunction with Table 3.9-1;

C_o is the basic design factor to be derived from Figure 3.9-4 as a function of U/V, or from Table 3.9-1 for U-tubesheets ($U/V = 0$);

ΔC is the corrective design factor to be derived from Table 3.9-1 as a function of actual value of F_s and R;

d is the outside diameter of tubes;

d_h is the tube hole diameter in tubesheet;

d^*_h is the effective tube hole diameter;

D is the outside diameter of shell;

D_o is the diameter of outer tube limit circle;

D_1 is the diameter to which shell fluid pressure is exerted;

D_2 is the diameter to which tube fluid pressure is exerted;

D_j is the effective pressurized diameter of expansion joint bellows as determined by bellows manufacturer or otherwise agreed;

D^* is the flexural rigidity of the tubesheet [see Equation (3.9.4-5)];

e is the minimum tubesheet thickness within the tubed area;

e_c is the channel analysis thickness (including any corrosion allowance) for a minimum distance of $1.8\sqrt{D_2 e_c}$;

e_s is the shell analysis thickness (including any corrosion allowance) for a minimum distance of $1.8\sqrt{D_1 e_s}$;

e_t is the tube thickness (nominal);

E is the elastic modulus of tubesheet material at design temperature from Table 3.6-3;

E_c is the elastic modulus of channel material at design temperature from Table 3.6-3;

E_s is the elastic modulus of shell material at mean metal temperature from Table 3.6-3;

E_t is the elastic modulus of tube material at mean metal temperature from Table 3.6-3;

f is the design strength of the tubesheet;

NOTE 1 In cases where f is time dependent, components designed by the procedure specified in this clause should be reviewed to ensure that creep deformation (local or general) will be acceptable throughout the agreed design lifetime.

f' is the reduced design strength to calculate revised tube stresses in floating head exchangers using nominal tubesheet thickness;

f_t is the design strength of the tubes;

F_q is the tubesheet factor given in Figure 3.9-12;

F_o is the factor for outer tube load given by Figure 3.9-5 and Figure 3.9-6;

F_i is the factor for inner tube load given by Figure 3.9-7 and Figure 3.9-8;

F_s is the effective “solidity” of perforated tubesheet, value between x_1 and x_2 depending on estimated effect of tube wall thickness: unless experimental results are available, a value equal to $(x_1 + x_2)/2$ should be used;

H is the tubesheet factor given in Figure 3.9-13 or Figure 3.9-14;

J is the expansion joint strain factor, = 1.00 for shell without expansion joint,

$$= \frac{1}{1 + (\pi D E_s e_s s)/L} \text{ for shell with bellows joint}$$

(where bellows stiffness is known),
= 0 for thin wall bellows joint;

k is the axial modulus of the tube bundle (full length) [see Equation (3.9.4-6)];

K is the mean strain ratio, tube bundle/shell given [see Equation (3.9.4-13)];

K_c is the edge moment required to rotate the channel through unit angle as given in **3.9.4.2**;

K_s is the edge moment required to rotate the shell through unit angle as given in **3.9.4.2**;

K_θ represents the combined edge restraint due to the channel and shell as given in **3.9.4.2**;

l is the length of expansion, or explosion bonding, within the tubesheet;

L is the tube length between inner faces of tubesheets;

N is the number of tube holes in tubesheet;

P is the tube pitch (spacing between centres);

P^* is the effective tube pitch;

p is the tubesheet design pressure (see **3.9.3.1**);

p_1 is the shell side design pressure;

p'_1 is the effective shell side design pressure for fixed tubesheets [see Equation (3.9.4-14)];

p_2 is the tube side design pressure;

p'_2 is the effective tube side design pressure for fixed tubesheets [see Equations (3.9.4-17) and (3.9.4-18)];

p_d is the effective differential design pressure [see Equation (3.9.4-22)];

NOTE 2 All these design pressures are gauge pressures and algebraic signs should be observed.

p_e is the effective pressure due to restrained differential thermal expansion [see Equation (3.9.4-21)];

p_{Bt} is the equivalent bolting pressure for operating condition [see Equation 3.9.4-23)];

p_{Bs} is the equivalent bolting pressure for bolting-up condition [see Equation 3.9.4-24)];

r_o is the radius to outermost tube hole centre (see Figure 3.9-2);

R = D_1/D_o when $p_1 > p_2$, = D_2/D_o when $p_2 > p_1$, = the greater of D_1/D_o and D_2/D_o when $p_1 = p_2$;

s is the spring rate for bellows-deflection/unit force;

NOTE 3 Calculations for bellows spring rate for thin walled bellows are given in BS EN 13445-3 Section 14 and BS EN 14917. Guidance on the evaluation of bellows spring rate for thick walled bellows using finite element analysis is given in the Standards of the Tubular Exchanger Manufacturers Association (TEMA).

S is the total untubed area of the tubesheet (see Figure 3.9-3);

U $= \left[1.35\frac{\delta E_t D_o}{\eta E L}\right]^{1/3}$ factor for use in Figure 3.9-4, Figure 3.9-5, Figure 3.9-6, Figure 3.9-7 and Figure 3.9-8;

U_L is the centre-to-centre distance between adjacent tube rows (see Figure 3.9-2);

V $= \sqrt{\frac{p}{\Omega \mu f}}$ factor for use in Figure 3.9-4, Figure 3.9-5, Figure 3.9-6, Figure 3.9-7 and Figure 3.9-8;

W_{ti} is the maximum effective tube stress for inner tube [see Equation (3.9.3-5)];

W_{to} is the maximum effective tube stress for outer tube [see Equation (3.9.3-4)];

x_1 $= 1 - N\left[\frac{d}{D_o}\right]^2$

x_2 $= 1 - N\left[\frac{d - 2e_t}{D_o}\right]^2$

x_a is the factor, calculated in **3.9.4.2**, which quantifies the elastic characteristic of the bundle and tubesheet;

z is the factor, given in **3.9.4.2**, dependent on the edge restraint due to both channel and shell;

α_s is the thermal expansion coefficient of shell material at mean metal temperature;

α_t is the thermal expansion coefficient of tube material at mean metal temperature;

β is a tubebundle factor, given in **3.9.4.2**, dependent on the tubesheet flexural rigidity and the tubebundle axial modulus;

δ $= (x_2 - x_1) = 4Ne_t(d - e_t)/D_o^2$;

η is the flexural efficiency of tubesheet and tube walls given by Figure 3.9-10 or Figure 3.9-11;

θ_s is the mean shell metal temperature less 10 °C;

θ_t is the mean tube wall metal temperature less 10 °C;

λ is the ligament efficiency of tubesheet in shear given by equations in **3.9.2.1**;

μ is the ligament efficiency of tubesheet and tube walls in bending given by equations in **3.9.2.1**;

ν is Poisson's ratio for unperforated plate;

ν_c is the Poisson's ratio for the channel;

ν_s is the Poisson's ratio for the shell;

Ω is the design stress factor = 2. This factor allows for the fact that the stress calculated using these requirements is the average bending stress across the ligament at the surface of the plate, and the permissible value is higher than the normal design stress, f;

τ is the design stress for shear; in absence of definition of design stress for shear in Section 2, τ should be taken as 0.5f.

Table 3.9-1 **Values of ΔC as a function of F_s and R for all tubesheets, and C_o for U-tubesheets only (for other types of tubesheets C_o is obtained from Figure 3.9-4)**

R	Stationary tubesheet — clamped		Stationary tubesheet — simply supported			
	C_o	ΔC	C_o	ΔC		
				F_s = 0.45	F_s = 0.60	F_s = 0.80
1.0	0.433	0	0.560	0	0	0
1.05	0.433	0	0.576	−0.002	0	+0.002
1.10	0.433	0	0.592	−0.010	0	+0.010
1.20	0.433	0	0.625	−0.025	0	+0.025
1.30	0.433	0	0.660	−0.040	0	+0.040

NOTE Figure 3.9-9b) shows a simply supported U-tubesheet and Figure 3.9-9e) shows a clamped U-tubesheet.

3.9.2 Characteristics of perforated plates

3.9.2.1 Ligament efficiency

The ligament efficiency for shear calculations shall be calculated from:

$$\lambda = \frac{P - d_h}{P} \tag{3.9.2-1}$$

The ligament efficiency for bending calculations shall be calculated from:

$$\mu = \frac{P^* - d_h^*}{P^*} \tag{3.9.2-2}$$

where P^*, the effective pitch is given by:

$$P^* = \frac{P}{\sqrt{1 - \frac{4\min[(S);(4D_oP)]}{\pi D_o^2}}} \tag{3.9.2-3}$$

and where d^*_h the effective tube hole diameter is given by:

$$d_h^* = \max\left\{\left[d - 2e_t\left(\frac{E_t}{E}\right)\left(\frac{f_t}{f}\right)\left(\frac{l}{e}\right)\right];[d - 2e_t]\right\} \tag{3.9.2-4}$$

The untubed area S and diameter of outer tube limit circle D_o are shown in Figure 3.9-2 and Figure 3.9-3. The depth of expansion l is zero when the tube fixing is welded only and/or lightly expanded.

NOTE 1 Light expansion gives 2% to 3% reduction in tube wall thickness.

NOTE 2 Where there is no untubed diametral land (S = 0) then P = P and where there is only a single untubed diametral lane*

$$P^* = \frac{P}{\sqrt{1 - \frac{4U_L}{\pi D_o}}} \tag{3.9.2-5}$$

Figure 3.9-2 **Tubesheet layout**

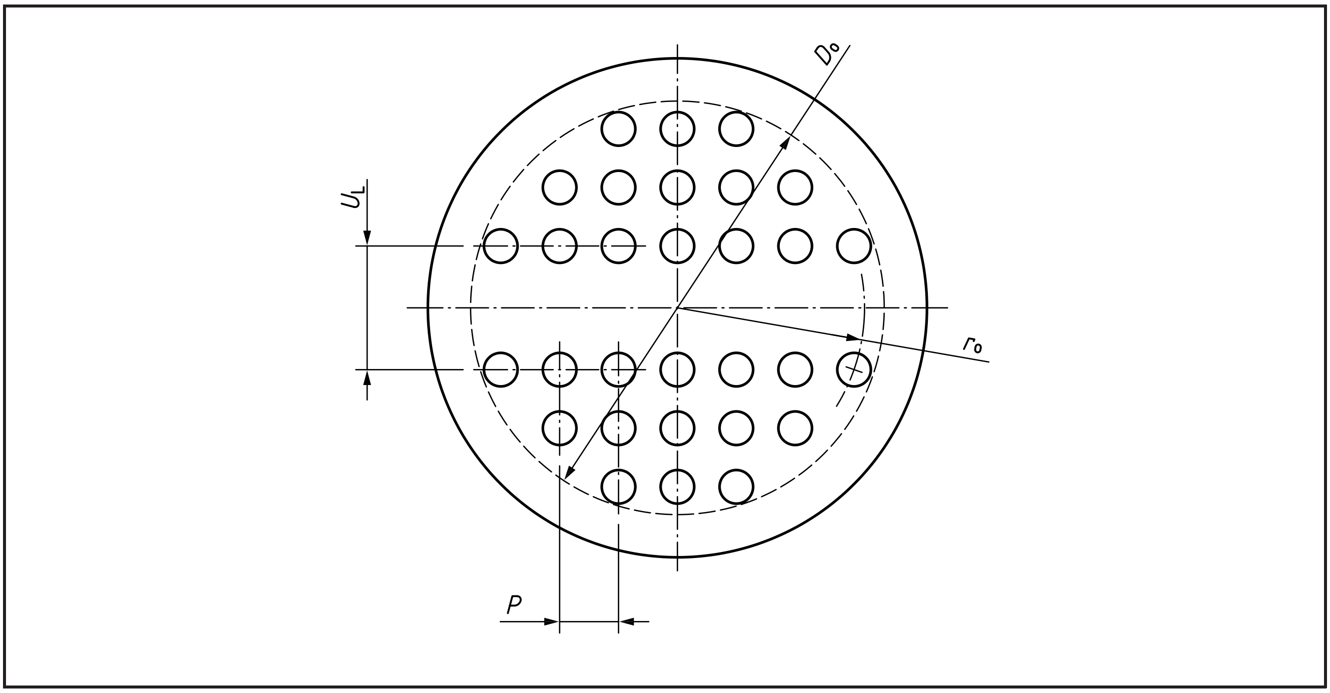
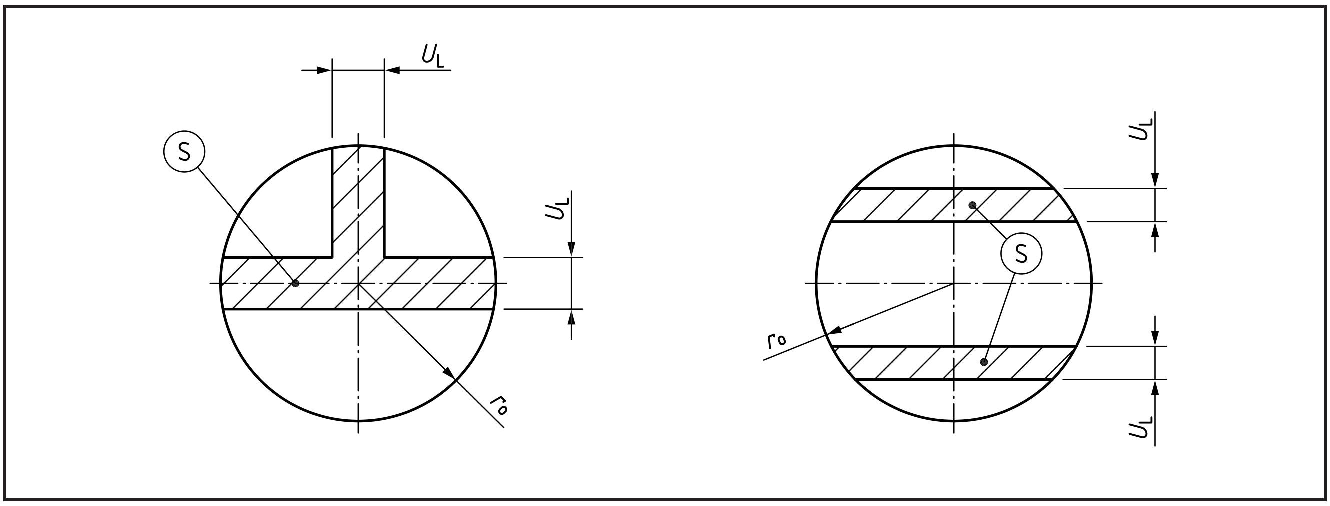


Figure 3.9-3 **Determination of area *S***



3.9.2.2 Flexural efficiency

The flexural efficiency for the tubesheet, η, shall be taken from Figure 3.9-10 or Figure 3.9-11 as follows:

a) for triangular tube layout use Figure 3.9-10;

b) for square tube layout use Figure 3.9-11.

3.9.3 Tubesheets of exchangers with floating heads or U-tubes

For the purposes of **3.9.3.1**, floating heads are denoted as those completely immersed in the shell side fluid; for such heat-exchangers, both tubesheets shall have the same thickness.

3.9.3.1 Design equations

The tubesheet design pressure shall be derived giving due consideration to loss of either pressure:

$$p = |p_2 - p_1| \tag{3.9.3-1}$$

The minimum thickness of a tubesheet within the outer tube limit circle shall be the greater of the values given by the following equations:

$$e = CD_o\sqrt{\frac{p}{\Omega\mu f}}\ \text{(bending)} \tag{3.9.3-2}$$

$$e = \frac{0.155 D_o p}{\lambda\tau}\ \text{(shear)} \tag{3.9.3-3}$$

C is dependent on clamped or simply supported edge conditions for the tubesheet, see Figure 3.9-9 for typical edge conditions.

The maximum effective tube stresses for an inner, W_{ti}, and an outer, W_{to}, tube, as given by the following equations, shall be checked in accordance with **3.9.5**, where a positive value denotes tension and a negative value compression. The two equations will usually, but not necessarily, give values of opposite sign, and both shall be considered in assessing the possibility of loss of tube staying action:

$$W_{to} = +\frac{{D_o}^2}{4Ne_t(d - e_t)}\left[p_2(F_o - x_2) - p_1(F_o - x_1)\right] \tag{3.9.3-4}$$

$$W_{ti} = -\frac{{D_o}^2}{4Ne_t(d - e_t)}\left[p_2(F_i + x_2) - p_1(F_i + x_1)\right] \tag{3.9.3-5}$$

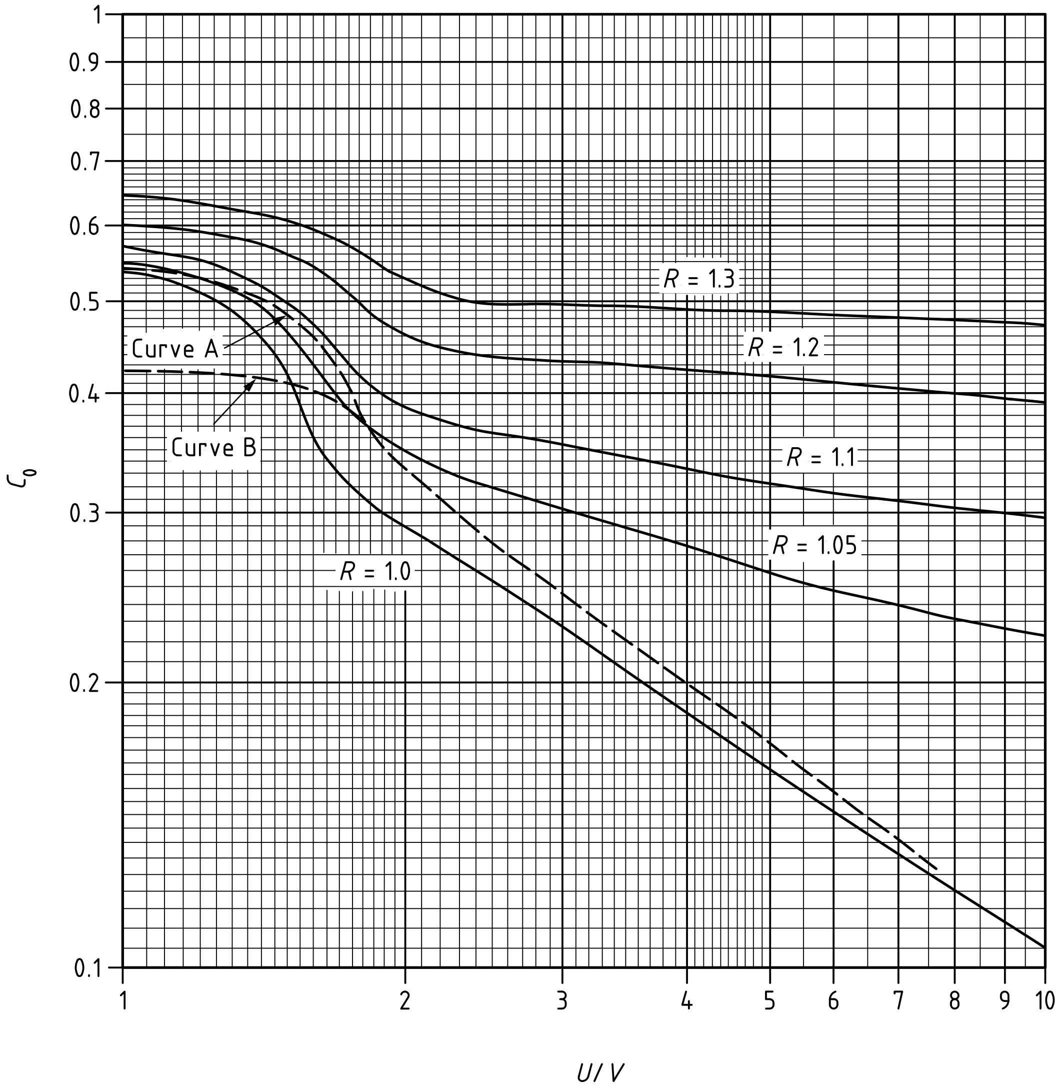
The maximum absolute value of the tube end joint load shall be checked against that permitted in **3.9.6**.

NOTE Where the nominal tubesheet is thicker than the minimum required, account can be taken when calculating the tube longitudinal stresses. The tubesheet design stress f can be lowered to f′ where:

$$f' = f \times \left(\frac{\text{minimum required tubesheet thickness}}{\text{tubesheet nominal thickness}}\right)^2$$

*V, as defined in **3.9.1**, should then be recalculated using f′ and the new factors obtained from Figure 3.9-5, Figure 3.9-6, Figure 3.9-7 and Figure 3.9-8. From this, the revised tube longitudinal stresses may be calculated.*

Figure 3.9-4 **Design curves: determination of C_0**



$C = C_0 + \Delta C$ (see Table 3.9-1)

Δ and $\square$ pitch

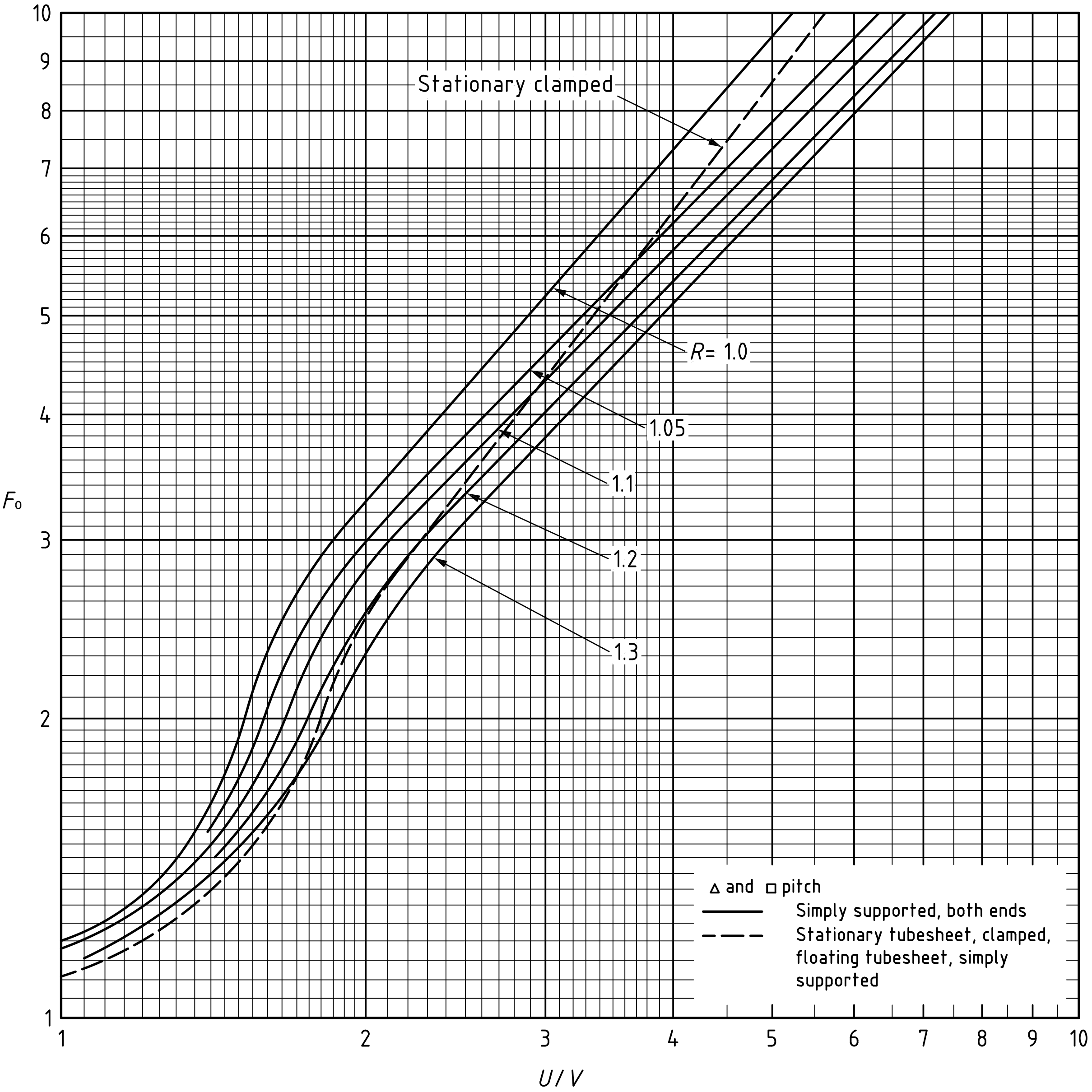
— Stationary tubesheet, simply supported[1)]

--- Stationary tubesheet clamped. When $U/V < 2$ use curve A for simply supported floating tubesheet or curve B for clamped floating tubesheet

NOTE Solid lines apply to construction b) + a) and b) + c) as shown in Figure 3.9-9. Broken line applies to construction d) + a) and d) + c) as shown in Figure 3.9-9.

[1)] For floating tubesheets simply supported or clamped, no distinction is made.

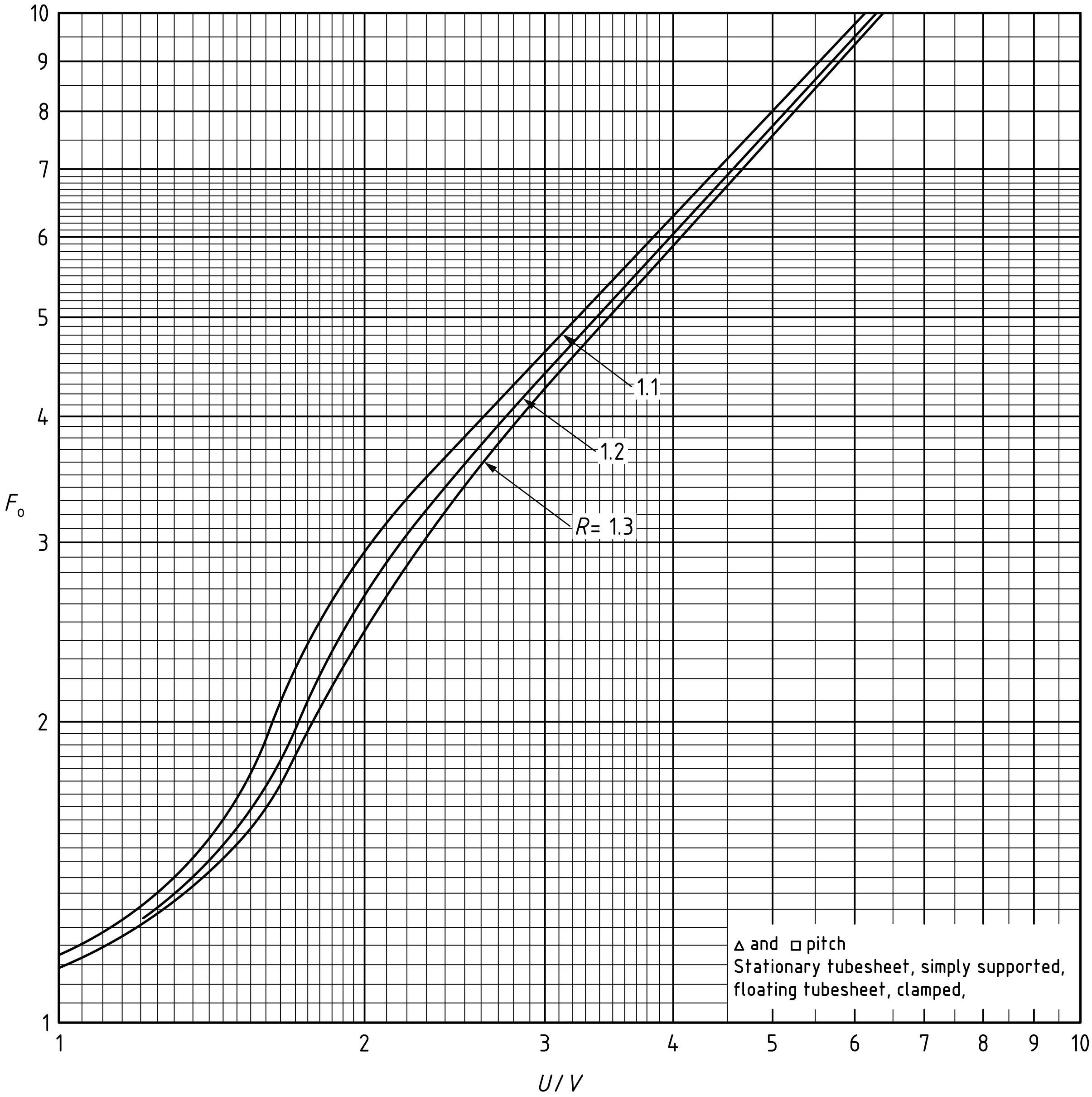
Figure 3.9-5 **Design curves: determination of F_o**



NOTE 1 For U-tubesheets, F_o = 1.

NOTE 2 Solid lines apply to construction b) + a) as shown in Figure 3.9-9. Broken line applies to construction d) + a) as shown in Figure 3.9-9 and also to constructions d) + c) or e) + c) as shown in Figure 3.9-9.

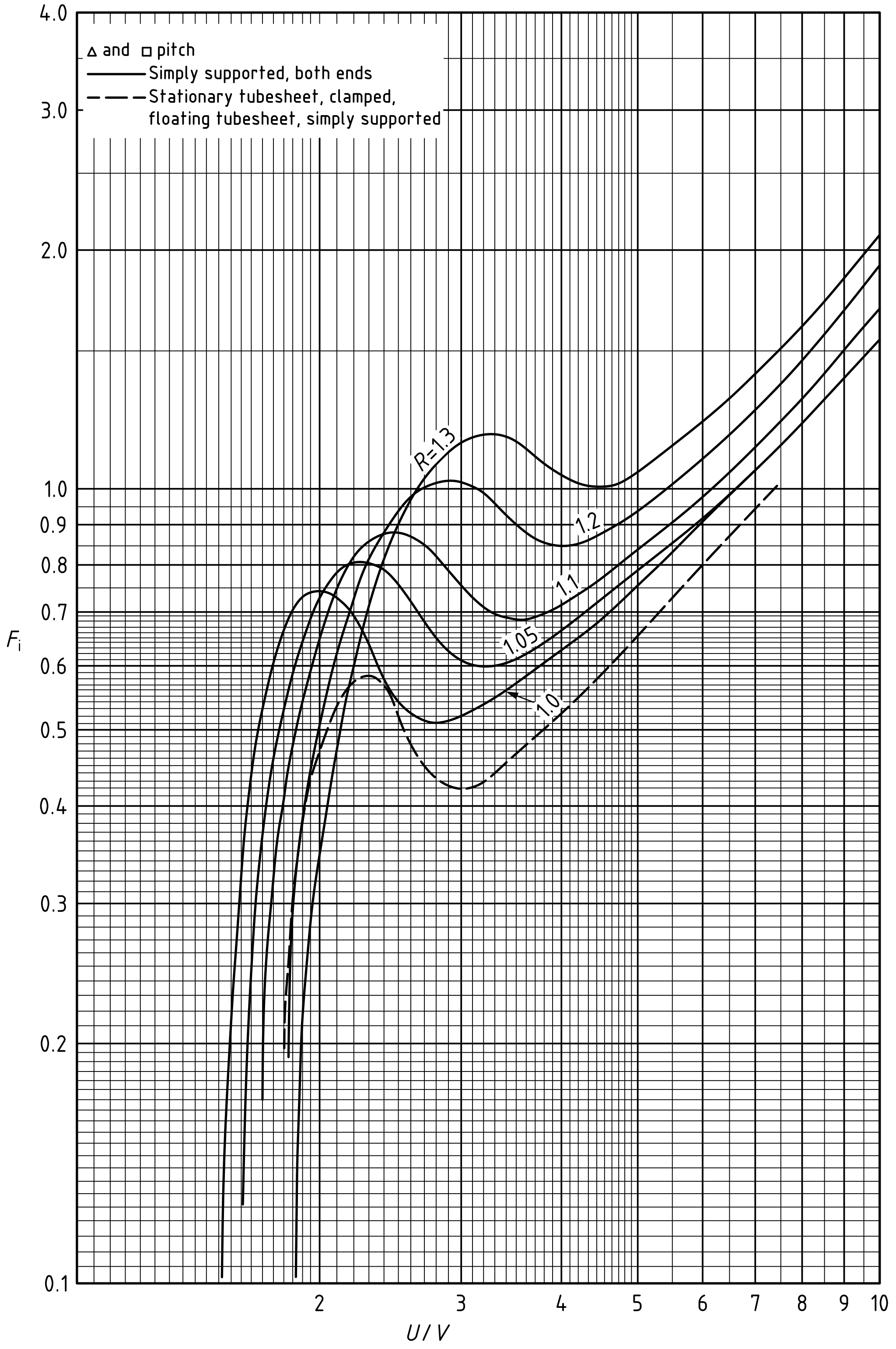
Figure 3.9-6 **Design curves: determination of F_o**



NOTE 1 For U-tubesheets, F_o = 1.

NOTE 2 Applies to construction b) + c) as shown in Figure 3.9-9.

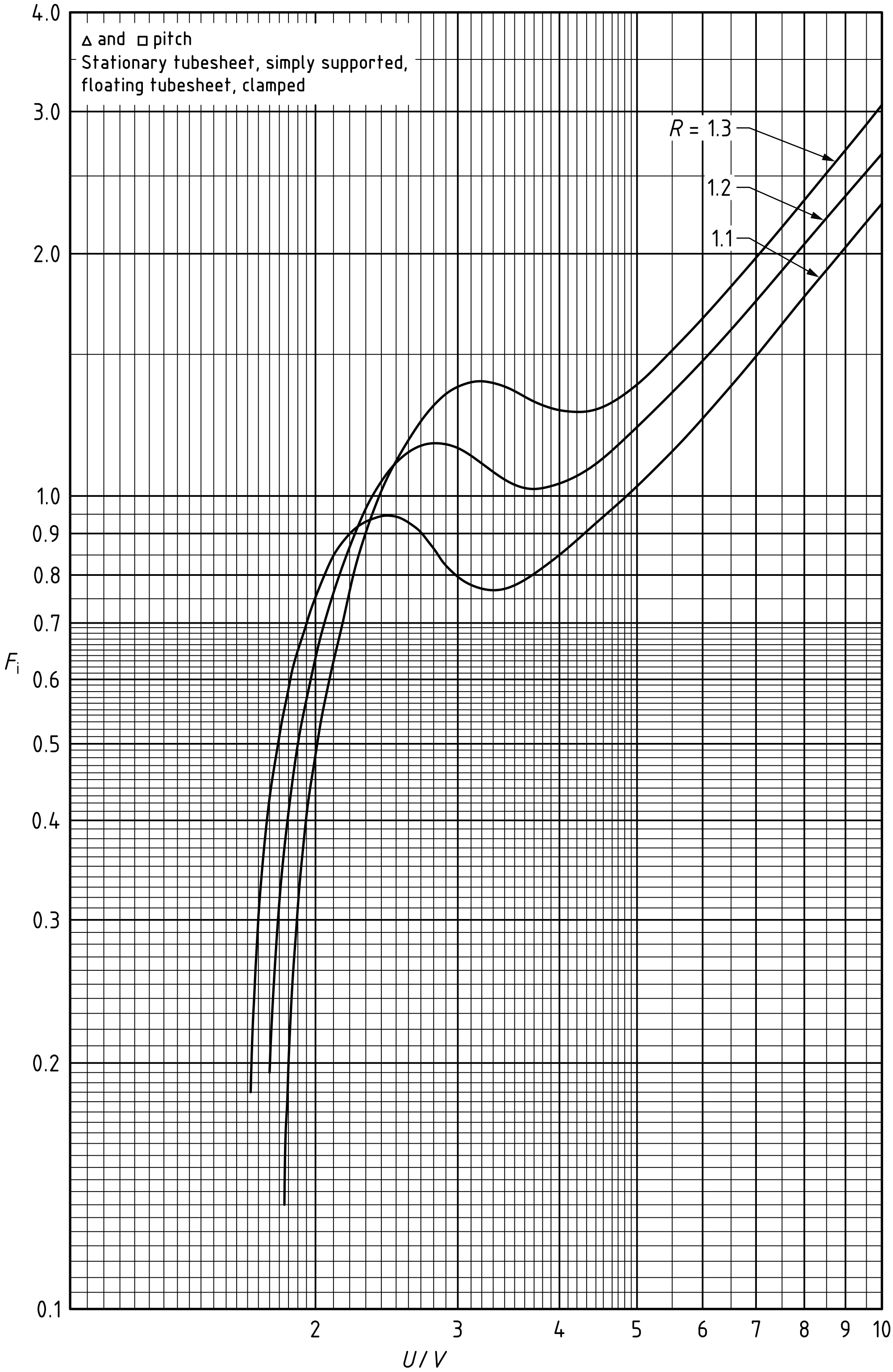
Figure 3.9-7 **Design curves: determination of F_i**



NOTE 1 For U-tubesheets, F_i = –1.

NOTE 2 Solid lines apply to construction b) + a) as shown in Figure 3.9-9. Broken line applies to construction d) + a) as shown in Figure 3.9-9.

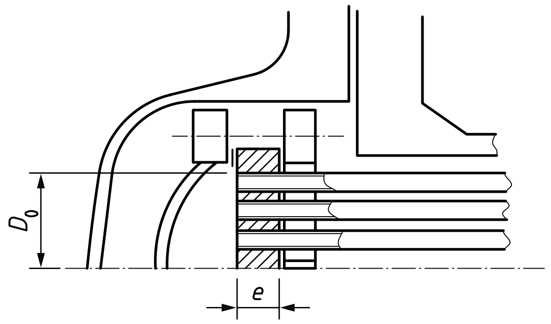
Figure 3.9-8 **Design curves: determination of F_i**



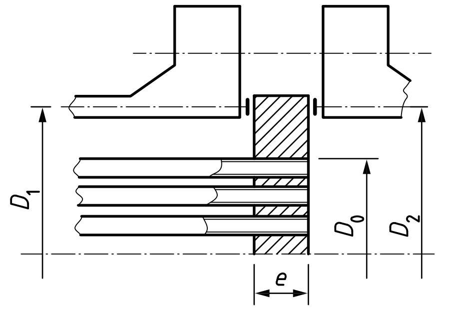
NOTE 1 For U-tubesheets, $F_i = -1$.

NOTE 2 Applies to construction b) + c) as shown in Figure 3.9-9 and also to constructions d) + c) or e) + c) as shown in Figure 3.9-9.

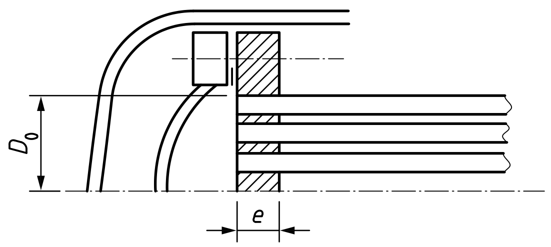
Figure 3.9-9 **Typical clamped and simply supported configurations for floating head or U-tubesheets**



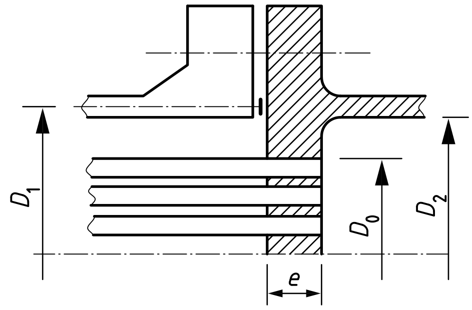
a) Simply supported



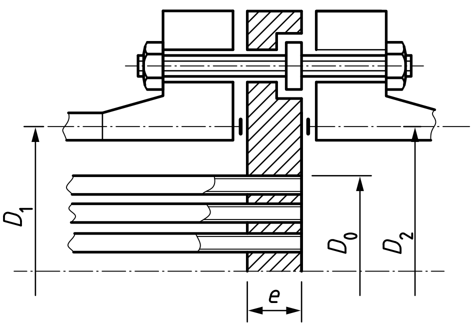
b) Simply supported



c) Clamped



d) See Note 2 below



e) Clamped

NOTE 1 Where a full faced gasket is used the tubesheet is clamped.

NOTE 2 If either shell or channel is welded to a U-tubesheet then:-

U-tubesheet is simply supported if $\frac{4.3D^* + 0.65D_1K_\theta}{1.3D^* + 0.5D_1K_\theta} \geq 2.3$

U-tubesheet is clamped if $\frac{4.3D^* + 0.65D_1K_\theta}{1.3D^* + 0.5D_1K_\theta} < 2.3$

where D^* and K_θ are calculated in accordance with **3.9.4.2**. For K_θ, K_c is zero if the channel is gasketted to the tubesheet and K_s is zero if the shell is gasketted to the tubesheet.

Figure 3.9-10 **Flexural efficiency: triangular layout**

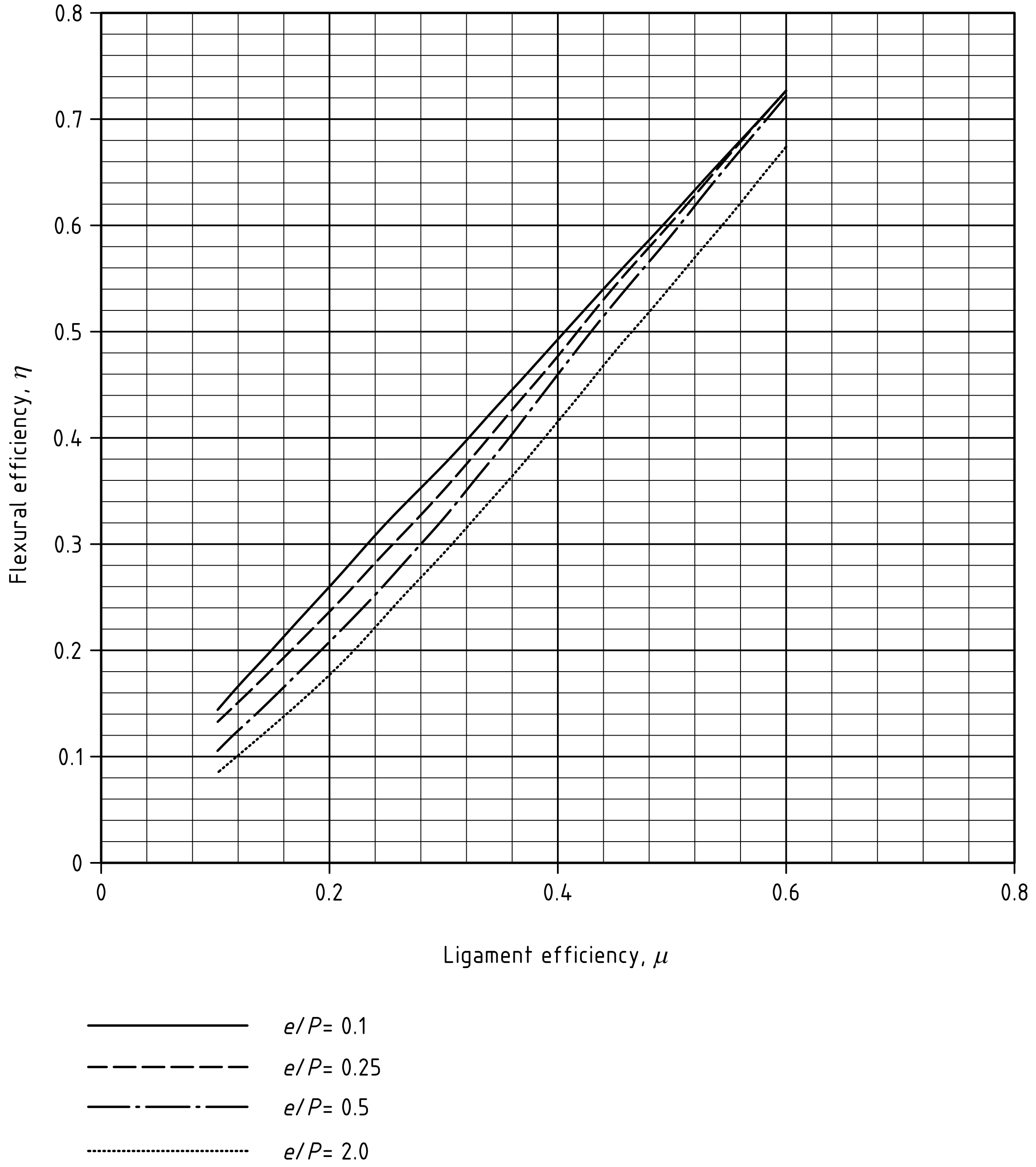
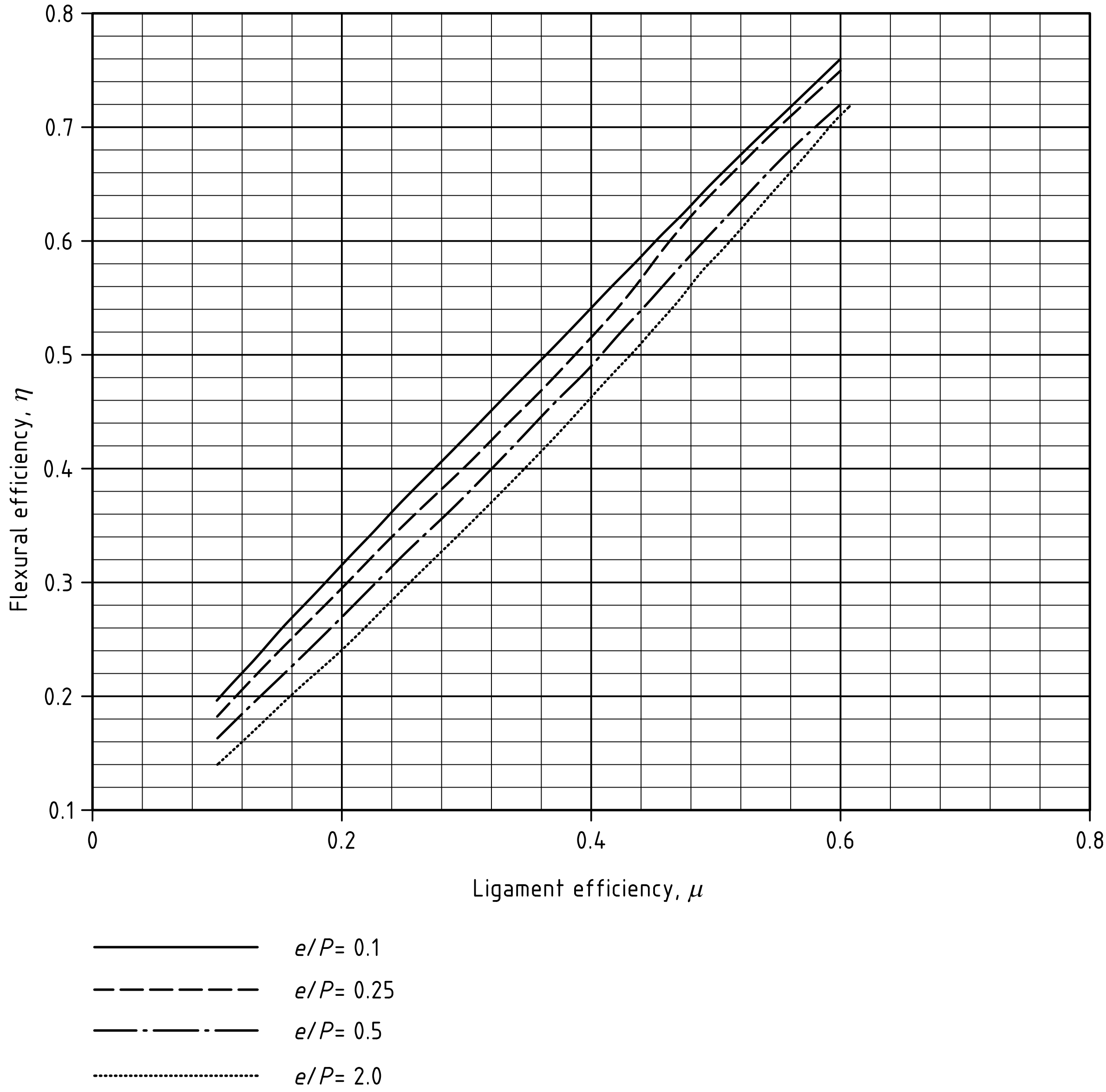


Figure 3.9-11 **Flexural efficiency: square layout**



3.9.4 Tubesheets of fixed tubesheet exchangers

For the purposes of **3.9.4.1**, **3.9.4.2**, **3.9.4.3** and **3.9.4.4** fixed tubesheet heat exchangers shall be considered as those having tubesheets fixed to both ends of the shell, with or without a shell expansion joint except as limited by **3.9.5**. Both tubesheets in a fixed tubesheet exchanger shall have the same thickness.

3.9.4.1 Design considerations

The thickness of the tubesheets shall be the greater of the values given by the equations in **3.9.4.2**. While this thickness will be adequate for the tubesheets it is possible that the temperature differential between tubes and shell may result in overstressing of the shells, tubes or tube-to-tubesheet joints. This shall be checked in accordance with **3.9.4.4** and **3.9.6**. Where necessary, suitable provision shall be made for expansion and/or contraction.

NOTE Design rules for bellows expansion joints are given in BS EN 13445-3 Section 14 and BS EN 14917. Guidance on the design of thick walled bellows using finite element analysis is given in the Standards of the Tubular Exchanger Manufacturers Association (TEMA).

The minimum thickness of the tubesheet shall be calculated using the analysis thicknesses of the shell and channel. Because of the increased constraint from the shell and channel when new, the assumption of analysis thicknesses for the shell and channel might be less conservative than the as-new case. The effect is generally not significant, but when specified in the purchase specification the tubesheet design shall additionally be checked for the as-new condition. The analysis thicknesses of the tubesheet and shell shall be used to calculate the shell and tube longitudinal stresses in accordance with **3.9.4.4**. All of the design rules of **3.9.4** are based upon the nominal thickness of the tubes.

If the tubesheet has a large unpierced annular gap between the tube bundle and the shell, its thickness shall be checked in accordance with **3.5.5.3.1**.

3.9.4.2 Design equations

The minimum thickness of the tubesheet shall be the greater of the values given by the following equations:

$$e = \frac{D_1}{\sqrt{4H}}\sqrt{\frac{p_1'}{\Omega\mu f}} \text{ or } \frac{D_2}{\sqrt{4H}}\sqrt{\frac{p_2'}{\Omega\mu f}} \text{ (bending)} \tag{3.9.4-1}$$

$$e = \frac{0.155 D_o p_1'}{\lambda\tau} \text{ or } \frac{0.155 D_o p_2'}{\lambda\tau} \text{ (shear)} \tag{3.9.4-2}$$

where p'_1 and p'_2 are the effective shell and tube design pressure determined in accordance with **3.9.4.3.1** and **3.9.4.3.2**;

or, where design on the basis only of simultaneous action of both shell and tube side pressure is required, (see **3.9.4.3.4**)

$$e = \frac{D_1}{\sqrt{4H}}\sqrt{\frac{p_d}{\Omega\mu f}} \text{ or } \frac{D_2}{\sqrt{4H}}\sqrt{\frac{p_d}{\Omega\mu f}} \text{ (bending)} \tag{3.9.4-3}$$

$$e = \frac{0.155 D_o p_d}{\lambda\tau} \text{ (shear)} \tag{3.9.4-4}$$

where

p_d is the effective differential design pressure determined in accordance with **3.9.4.3.4**;

$$D^* = \frac{\eta E e^3}{12(1-\nu^2)} \tag{3.9.4-5}$$

$$k = \frac{4NE_t e_t (d - e_t)}{L D_1^2} \tag{3.9.4-6}$$

$$K_c = \frac{E_c (e_c)^{2.5}}{[12(1-\nu_c^2)]^{0.75}(D_2 + e_c)^{0.5}} \tag{3.9.4-7}$$

NOTE 1 K_c is zero when the channel is gasketted to the tubesheet.

$$K_s = \frac{E_s (e_s)^{2.5}}{[12(1-\nu_s^2)]^{0.75}(D_1 + e_s)^{0.5}} \tag{3.9.4-8}$$

$$K_\theta = K_c + K_s \tag{3.9.4-9}$$

$$x_a = \frac{\beta D_1}{2} \tag{3.9.4-10}$$

$$z = \frac{2K_\theta}{\beta D^*} \tag{3.9.4-11}$$

NOTE 2 In a given design, the minimum tubesheet thickness is obtained when z = 0.5, and this may be achieved by altering either e_s or e_c.

$$\beta = \sqrt[4]{\frac{2k}{D^*}} \tag{3.9.4-12}$$

$$K = \frac{E_s e_s (D - e_s)}{E_t e_t N (d - e_t)} \tag{3.9.4-13}$$

In calculating p'_1, p'_2 and p_d for use in the shear Equations (3.9.4-2) and (3.9.4-4), the equivalent bolting pressures p_{Bs} and p_{Bt} shall be taken as zero.

The calculation is an iterative one. A value shall be assumed for e and the calculation made. If e calculated is less than e assumed, it is permissible to make the tubesheet of thickness e assumed. For minimum tubesheet thickness, the iteration should be repeated until:

0.985e assumed < e calculated < 1.000e assumed.

3.9.4.3 Effective shell and tube design pressure

3.9.4.3.1 The effective shell side design pressure, p'_1, shall be calculated from:

$$p'_1 = \max\left(\left|0.667\left(p'_s - p_e\right)\right|; \left|p'_s\right|; \left|p_{Bs}\right|; \left|0.667\left(p'_s - p_e - p_{Bs}\right)\right|; \left|0.667\left(p_{Bs} + p_e\right)\right|; \left|\left(p'_s - p_{Bs}\right)\right|\right) \tag{3.9.4-14}$$

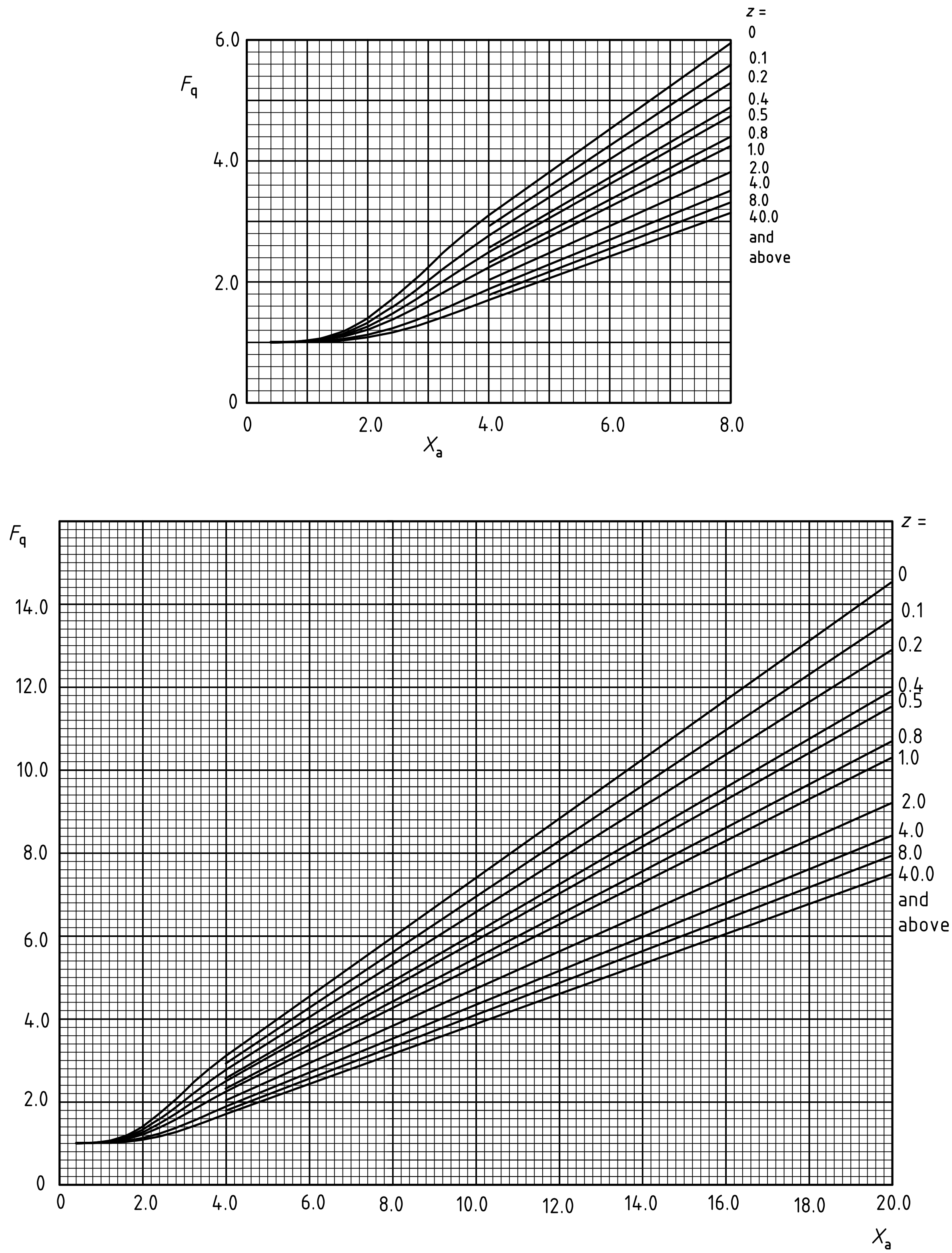
where

$$p'_s = p_1 \left(\frac{0.4J[1.5 + K(1.5 + f_s)] - \left(\frac{1-J}{2}\right)\left(\frac{D_j^2}{D_1^2} - 1\right)}{\left(1 + JKF_q\right)} \right) \tag{3.9.4-15}$$

$$f_s = 1 - N\left(\frac{d}{D_1}\right)^2 \tag{3.9.4-16}$$

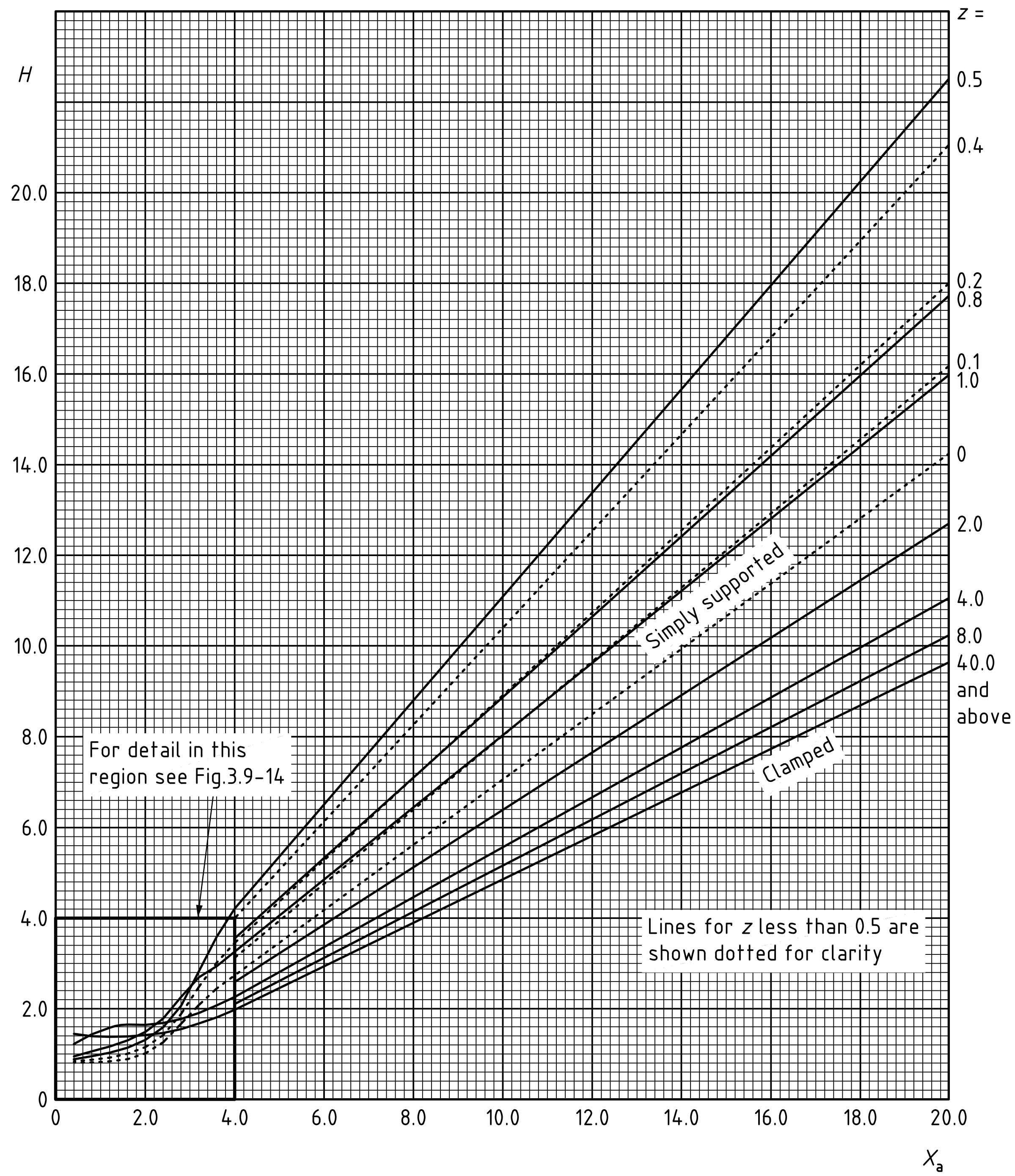
NOTE Equations containing the term p_{Bs} are not applicable for use in the shear equations in ***3.9.4.2****.*

Figure 3.9-12 **Tubesheet: determination of F_q**



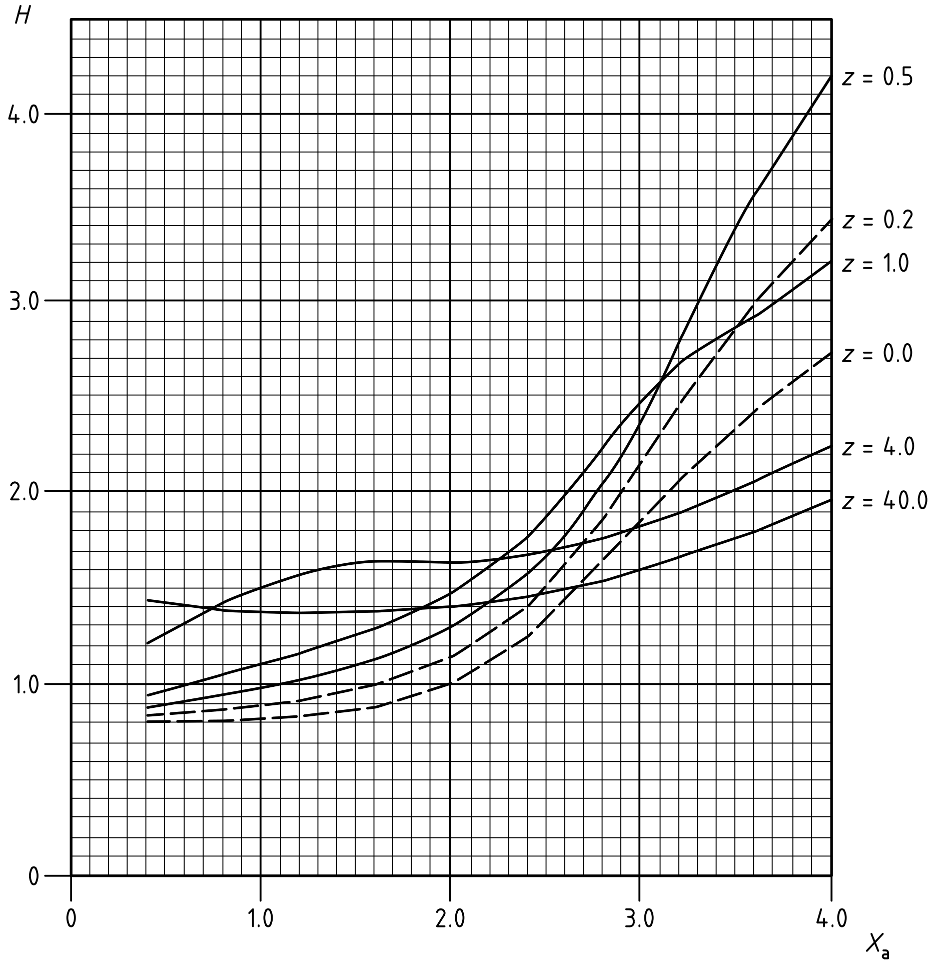
NOTE Linear interpolation should be used for values of z lying between those given.

Figure 3.9-13 **Tubesheet: determination of H for $X_a > 4.0$**



NOTE Linear interpolation should be used for values of z lying between those given.

Figure 3.9-14 **Tubesheet: determination of H for X_a < 4.0**



NOTE Linear interpolation should be used for values of z lying between those given.

3.9.4.3.2 The effective tube side design pressure, p'_2, shall be calculated from Equation (3.9.4-17) or (3.9.4-18):

When p'_s is positive:

$$p'_2 = \max\left(\left|0.667\left(p'_t + p_{Bt} + p_e\right)\right|; \left|p'_t + p_{Bt}\right|\right) \quad (3.9.4\text{-}17)$$

When p'_s is negative:

$$p'_2 = \max\left(\left|0.667\left(p'_t - p'_s + p_{Bt} + p_e\right)\right|; \left|p'_t - p'_s + p_{Bt}\right|\right) \quad (3.9.4\text{-}18)$$

where

$$p'_t = p_2\left[\frac{1 + 0.4JK(1.5 + f_t)}{(1 + JKF_q)}\right] \quad (3.9.4\text{-}19)$$

$$f_t = 1 - N\left[\frac{(d - 2e_t)}{D_2}\right]^2 \quad (3.9.4\text{-}20)$$

3.9.4.3.3 The pressure due to differential thermal expansion, p_e, shall be calculated from:

$$p_e = \frac{4JE_s e_s(\alpha_s\theta_s - \alpha_t\theta_t)}{(D - 3e_s)(1 + JKF_q)} \tag{3.9.4-21}$$

3.9.4.3.4 The effective differential design pressure, p_d, shall be calculated from:

$$p_d = \max\begin{pmatrix} |p'_t - p'_s + p_{Bt}|; |0.667(p'_t - p'_s + p_{Bt} + p_e)|; \\ |p_{Bs}|; |0.667(p_{Bs} + p_e)|; |p'_t - p'_s|; |0.667(p'_t - p'_s + p_e)|; |p_{Bt}| \end{pmatrix} \tag{3.9.4-22}$$

Design based upon p_d is permitted, when it is not possible to have individual and separate pressurization of either the shellside or tubeside in any operating condition and when this simultaneous differential pressure basis is specified in the purchase specification. When it is possible to have vacuum on one side with simultaneous pressure on the other, the purchaser shall specify in the purchase specification that this condition shall be checked using the rules of **3.9.4.3.4**.

NOTE 1 These conditions will require the setting of the tubesheet test pressure to reflect the design differential pressure.

*NOTE 2 It is not permissible to enter the equations in **3.9.4.3.1** with (p_1– p_2) in place of p'_s or the equations in **3.9.4.3.2** with ($p_1 - p_2$) in place of p'_t to determine an effective shell side or tube side design pressure for fixed tubesheets.*

3.9.4.3.5 Equivalent bolting pressures, when fixed tubesheets are extended for bolting to heads with ring type gaskets, shall be calculated from:

$$p_{Bt} = \frac{2\pi M_{op}}{{D_1}^3} \tag{3.9.4-23}$$

$$p_{Bs} = \frac{2\pi M_{atm}}{{D_2}^3} \tag{3.9.4-24}$$

where

M_{op} is the total moment acting upon extension under operating conditions (see **3.8** for the determination of the bolt loads required to calculate this moment);

M_{atm} is the total moment acting upon extension under bolting-up conditions (see **3.8** for the determination of the bolt loads required to calculate this moment).

Where full faced gaskets are fitted $p_{Bt} = p_{Bs} = 0$.

NOTE The load on the gasket trapped between a tubesheet and any pass-partition plate may be neglected in the calculation of bolt load and tube plate thickness.

3.9.4.4 Shell and tube longitudinal stresses

The maximum effective shell and tube stresses calculated as follows shall be checked in accordance with **3.9.5**.

The tensile and compressive longitudinal shell stress shall be calculated from:

$$\frac{(D - e_s){p_s}^*}{4e_s} \tag{3.9.4-25}$$

where

$${p_s}^* = \max\begin{pmatrix} Y(p'_s + p_2 - p'_t - p_e); (p'_s + p_2 - p'_t); \\ (-Yp_e); Y(p'_s - p_e); Y(p_2 - p'_t - p_e); (p_2 - p'_t); p'_s \end{pmatrix} \tag{3.9.4-26}$$

Y = 1.0 if the algebraic sign of p_s* is negative, or

= 0.5 if the algebraic sign of p_s* is positive.

*NOTE 1 When the design is based on simultaneous differential pressure (see **3.9.4.3.4**) only the first three equations for p_s* apply.*

The tensile and compressive longitudinal tube stress shall be calculated from:

$$\frac{F_q p_t^* D_2^2}{4Ne_t(d-e_t)} \tag{3.9.4-27}$$

where

$$p_t^* = \max[Z(p_4+p_e-p_5); Zp_e; (p_4-p_5); Z(p_4+p_e); p_4; -p_5; Z(p_e-p_5)] \tag{3.9.4-28}$$

Z = 1.0 if the algebraic sign of p_t* is negative, or

= 0.5 if the algebraic sign of p_t* is positive.

$$p_4 = \left(p_t^{'} - \frac{f_t}{F_q}p_2\right) \tag{3.9.4-29}$$

$$p_5 = \left(p_s^{'} - \frac{f_s}{F_q}p_1\right) \tag{3.9.4-30}$$

*NOTE 2 When the design is based on simultaneous differential pressure (see **3.9.4.3.4**) only the first three equations for p_t* apply.*

The maximum absolute value of the tube end joint load shall be checked against that permitted in **3.9.6**.

3.9.5 Allowable shell and tube longitudinal stresses

Tensile shell and tube stresses calculated in accordance with **3.9.3.1** and **3.9.4.4** shall be checked to ensure that they do not exceed the allowable values in Section 2.

Compressive shell stresses shall be limited to the allowable value given in **A.3.5**. Compressive tube stresses shall be limited to:

$$\frac{\pi^2 r^2 E_t}{SL_k^2} \text{ when } C \leq \frac{L_k}{r} \tag{3.9.5-1}$$

or

$$\frac{s^{'}f}{S}\left[1-\frac{L_k}{2rC}\right] \text{ when } C > \frac{L_k}{r} \tag{3.9.5-2}$$

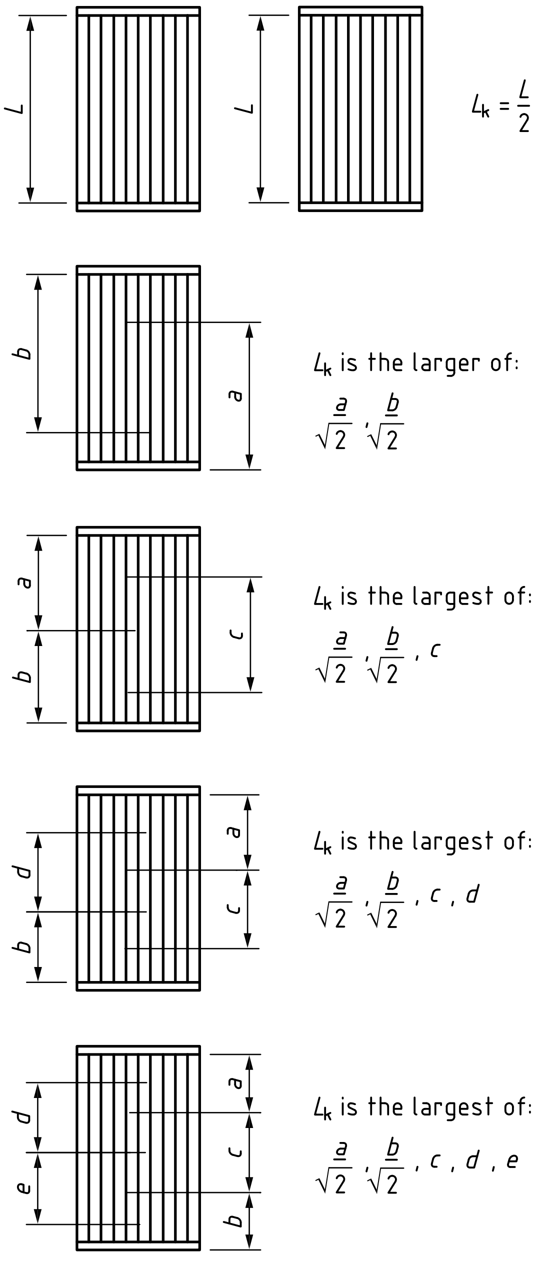
where

r	is the radius of gyration of tube $= 0.25\sqrt{d^2 + (d-2e_t)^2}$;
L_k	is the buckling length (see Figure 3.9-15);
f	is the design stress derived from **2.3.1.1**;
s'	is the factor 1.4 for ferritic steels in material groups 1 to 6, 7.1, 9 and 11, or 1.1 for austenitic steels in material group 8;
E_t	is the elastic modulus of tube material at mean metal temperature from Table 3.6-3;
C	$= \sqrt{\dfrac{2\pi^2 E_t}{s^{'}f}}$

S is the safety factor
$= 3.25 - 0.5F_o$ for floating head and U-tube exchangers, and
$= 3.25 - 0.5F_q$ for fixed tubesheet exchangers.

The safety factor S shall be not less than 1.25 and shall not exceed a value of 2.0.

Figure 3.9-15 **Determination of the buckling length L_k**



NOTE For U-tubes, L is the straight length between tubesheet and bend.

3.9.6 Allowable tube joint end load

The allowable tube joint end load is dependent upon the type of joint, **3.9.6** gives requirements for types listed in Table 3.9-2.

For joints a, b and c the tube joint end load shall be limited to:

tube cross-sectional area × tube design stress × F_r.

For joints d, e and f the tube joint end load shall be limited to:

tube cross-sectional area × tube design stress × $F_e \times F_r \times F_{\bar{y}}$

where

F_r is the reliability factor from Table 3.9-2;

F_e is the expansion factor (not greater than 1.0);
= 1 for grooved holes;
= 1 for explosion expanded/welded tube ends;

$= \dfrac{\text{expanded length}}{\text{tub o.d.}}$ for plain holes;

$F_{\bar{y}}$ is the material factor (not greater than 1.0)

$= \dfrac{\text{tubesheet design stress}}{\text{tube design stress}}$.

When agreed by the manufacturer, purchaser and Inspecting Authority it is permitted to increase the value of F_r given in Table 3.9-2 if the procedure is approved with a push-out test. A recommended push-out test is given in **T.5.6**.

Table 3.9-2 **Values of F_r for typical tube joints**

Joint	F_r
a Welded with minimum weld throat ≥ tube thickness	0.80
b Welded with minimum weld throat < tube thickness	0.55
c Expanded and welded with minimum weld throat ≥ tube thickness	0.80
d Expanded and welded with minimum weld throat < tube thickness	0.55
e Expanded only	0.50
f Explosion expanded/welded	0.80

NOTE Typical examples of arc welded tube to tubeplate joints are given in Annex T.

3.9.7 Tubesheet flanged extension with narrow-face gasket

Where the tubesheet is extended to provide a flange for bolting (as in Figure 3.9-9c), Figure 3.9-9d) or Figure 3.9-9e)), the minimum thickness of the extension between the gasket position and the outside diameter of the tubed portion shall not be less than e_1 as given by Equation (3.5.5-4). The analysis thickness of the tubed portion of the tubesheet shall not be less than e_1.

Suggested working form 3.9-1

Floating head or U-tube tubesheet design

Load case:

Consistent units: dimensions = *: p/f/τ/E =*

	Shellside	Tubeside
Design pressure	p_1	p_2
Corrosion allowance		
Tubesheet design temperature		
Tubesheet material		
Allowable stress at design temperature: bending f		
Allowable stress at design temperature: shear τ		
Design stress factor (**3.9.1**) Ω		
Elastic modulus: tubesheet E		
Elastic modulus: tubes E_t		

Ligament efficiency ref. **3.9.2.1**

For shear $\lambda = (P - d_h)/P$ =

$$P^* = \frac{P}{\sqrt{1 - 4\min\left[(S);(4D_oP)\right]/\left(\pi D_o^2\right)}} =$$

$$d^*_h = \max\left\{\left[d - 2e_t\left(\frac{E_t}{E}\right)\left(\frac{f_t}{f}\right)\left(\frac{l}{e}\right)\right]; \left|d - 2e_t\right|\right\} =$$

For bending $\mu = (P^* - d^*_h)/P^*$ =

Pressure factors ref **3.9.1**

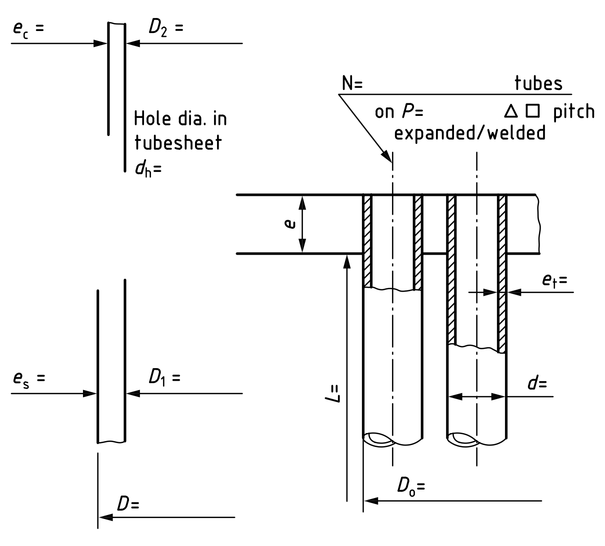
$x_1 = 1 - N(d/D_o)^2$ =

$x_2 = 1 - N[(d - 2e_t)/D_o]^2$ =

$\delta = x_2 - x_1$ =

$F_s = 0.5(x_1 + x_2)$ =

Tubesheet edge support



U-tube tubesheet thickness		**Clause reference**
Pressure acting on		**3.9.3**
$p = \lvert p_2 - p_1 \rvert$		**3.9.3.1**
$V = \sqrt{\dfrac{p}{\Omega\mu f}}$		
$R = D_1/D_o$ if $p_1 > p_2$ $= D_2/D_o$ if $p_2 > p_1$ = max. of above if $p_1 = p_2$		
C_o ΔC $C = C_o + \Delta C$		Table 3.9-1 Table 3.9-1
Bending $e_B = CD_oV$		**3.9.3.1**
Shear $e_S = \dfrac{0.155D_op}{\lambda\tau}$		**3.9.3.1**
Min acceptable thickness = greater of values of e } e		
Tubesheet thickness = e + allowances =		

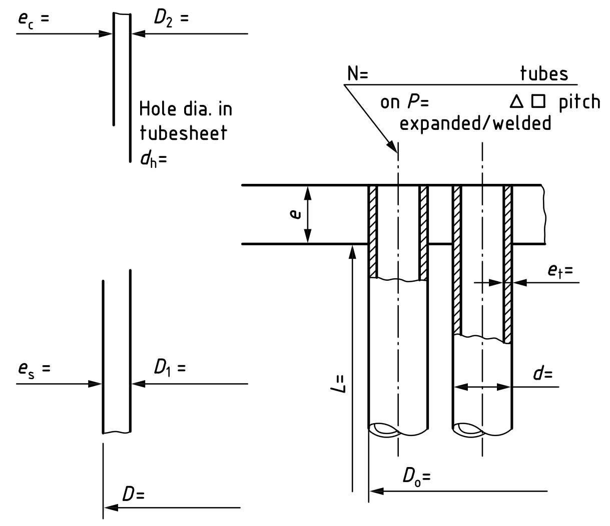
Floating head tubesheet thickness			**3.9.3**
Pressure acting on $\therefore p = \lvert p_2 - p_1 \rvert$ =			**3.9.3.1**
$R = D_1/D_o$ if $p_1 > p_2$ $= D_2/D_o$ if $p_2 > p_1$ = max. of above if $p_1 = p_2$			
Assume $e \geq$ or $< 2P$			
η			Figure 3.9-10 or Figure 3.9-11
$U = \left[\dfrac{1.35\delta E_tD_o}{\eta EL}\right]^{1/3}$			
$V = \sqrt{\dfrac{p}{\Omega\mu f}}$			
U/V			
C_o ΔC $C = C_o + \Delta C$			Figure 3.9-4 Table 3.9-1
Bending $e_B = CD_oV$			**3.9.3.1**
Shear $e_S = \dfrac{0.155D_op}{\lambda\tau}$			**3.9.3.1**
Min acceptable thickness = greater of values of e } e			
Check $e \geq$ or $< 2P$			
Tubesheet thickness = e + allowances =			

Suggested working form 3.9-2							
Fixed tubesheet design							
Load case:							
Consistent units: dimensions =				*: p/f/ τ/E =*			
		Shellside	Tubeside	Assume thickness e			
Design pressure		p_1	p_2	$D^* = \eta E e^3/[12(1-\nu^2)]$			
Corrosion allowance				$\beta = \sqrt[4]{2k/D^*}$			
Tubesheet design temperatures							
Tubesheet material				$x_a = \beta D_1/2$			
Allowable stress at design temperature	bending	f		$z = 2K_\theta/(\beta D^*)$			
	shear	τ		F_q (ref. Figure 3.9-12)			
Design stress factor (**3.9.1**)		Ω		$(1 + JKF_q)$			
	Shell	Tube	Tubesheet	H (ref. Figure 3.9-13 & 14)			
Elastic modulus	E_s	E_t	E	p'_s (ref. **3.9.4.3.1**)			
Expansion coefficient	α_s	α_t	–	p'_t (ref. **3.9.4.3.2**)			
Metal temperature	θ_s	θ_t	–	p_e (ref. **3.9.4.3.3**)			

Ligament efficiency (ref. **3.9.2.1**) For shear $\lambda = (P - d_h)/P =$ $P^* = \dfrac{P}{\sqrt{1 - 4\min\left[(S);(4D_oP)\right]/(\pi D_o^2)}} =$ $d^*_h = \max\left\{\left[d - 2e_t\left(\dfrac{E_t}{E}\right)\left(\dfrac{f_t}{f}\right)\left(\dfrac{l}{e}\right)\right]; \lvert d - 2e_t \rvert\right\} =$ For bending $\mu = (P^* - d^*_h)/P^* =$	**Shellside loading** ref. **3.9.4.2** and **3.9.4.3.1**			
	(a) $0.667(p'_s - p_e)$			
	(b) p'_s			
	(c) p_{Bs}			
	(d) $0.667(p'_s - p_e - p_{Bs})$			
	(e) $0.667(p_{Bs} + p_e)$			
	(f) $(p'_s - p'_{Bs})$			
	p_{1B} = greatest abs. of (a) to (f)			
	p_{1s} = greater abs. of (a) & (b)			
Flexural efficiency (Figure 3.9-10 and Figure 3.9-11) Assume e/P = η =	Bending $e_{1B} = \dfrac{D_1}{\sqrt{4H}}\sqrt{\dfrac{p_{1B}}{\Omega\mu f}}$			
Pressure factors (ref. **3.9.4.3.1** and **3.9.4.3.2**) $f_s = 1 - N[d/D_1]^2 =$ $f_t = 1 - N[(d - 2e_t)/D_2]^2 =$	Shear $e_{1s} = \dfrac{0.155D_o p_{1s}}{\lambda\tau}$			
J factor (ref. **3.9.1**) Shell without bellows $J = 1$ Shell with thin wall bellows $J = 0$ Bellows to shell diameter D = spring rate S = Shell with bellows of know spring rate J =	**Tubeside loading** ref. **3.9.4.2** and **3.9.4.3.2**			
	(a) If p'_s positive, greater abs. of $(p'_t + p_{Bt})$ and $0.667(p'_t + p_{Bt} + p_e)$			
Edge support factor (ref. **3.9.4.2**) $K_c = E_c(e_c)^{2.5}/\{[12(1-\nu_c^2)]^{0.75}(D_2 + e_c)^{0.5}\} =$ $K_s = E_s(e_s)^{2.5}/\{[12(1-\nu_s^2)]^{0.75}(D_1 + e_s)^{0.5}\} =$ $K_\theta = K_c + K_s =$	(b) If p'_s negative, greater abs. of $(p'_t - p'_s + p_{Bt})$ and $0.667(p'_t - p'_s + p_{Bt} + p_e)$			
	p_{2B} = (a) or (b) as applicable			
Strain factor (ref. **3.9.4.2**) $K = E_s e_s(D - e_s)/[E_t e_t N(d - e_t)] =$	p_{2s} = (a) or (b) as applicable *but* OMIT p_{Bs} term			
Tube bundle modulus (ref. **3.9.4.2**) $k = 4NE_t e_t(d - e_t)/(LD_1^2)] =$	Bending $e_{2B} = \dfrac{D_2}{\sqrt{4H}}\sqrt{\dfrac{p_{2B}}{\Omega\mu f}}$			
Equivalent bolting pressure (ref. **3.9.4.3.5**) $p_{Bt} = 2\pi M_{op}/(D_1)^3 =$ $p_{Bt} = 2\pi M_{atm}/(D_2)^3 =$	Shear $e_{2s} = \dfrac{0.155D_o p_{2s}}{\lambda\tau}$			

Suggested working form 3.9-2 *(continued)*

Tubesheet edge support



If agreed, combined loading ref **3.9.4.2** and **3.9.4.3.4**			
(a) $0.667(p'_t - p'_s + p_{Bt})$			
(b) $0.667(p'_t - p'_s + p_{Bt} + p_e)$			
(c) p_{Bs}			
(d) $0.667(p_{Bs} + p_e)$			
(e) $(p'_t - p'_s)$			
(f) $0.667(p'_t - p'_s + p_e)$			
(g) p_{Bt}			
P_{dB} = greatest abs. of (a) to (g)			
P_{ds} = greater abs. of (e) & (f)			
D_d = greater of D_1 and D_2			
Bending $e_{dB} = \frac{D_d}{\sqrt{4H}}\sqrt{\frac{p_{dB}}{\Omega \mu f}}$			
Shear $e_{ds} = \frac{0.155 D_o p_{ds}}{\lambda \tau}$			
Min acceptable thickness = greater of values of e } e			
Check e assumed / e is within 1.5%			
Tubesheet thickness = e + allowances =			

3.10 Design of welds

3.10.1 General

3.10.1.1 Vessels shall be designed with the minimum practical number of seams with adequate access for the deposition and inspection of weld metal to comply with Section 4 and Section 5. As far as possible seams shall be positioned clear of supports, etc., so as to be readily visible in service after removing any insulation. For category 2 components, when openings occur in welded seams or within 12 mm of the toe of any main seam welds additional inspection requirements shall apply [see **5.6.4.2.1** a) 3)].

Where more than two weld seams meet at one point, consideration shall be given to the desirability of intermediate post-weld heat treatment (see **4.1.4**).

3.10.1.2 It is permissible to weld nozzles, pads, branches, pipes and tubes and non-pressure parts to pressure parts, provided that the strength and characteristics of the material of the pressure part are not influenced adversely. Attachments of non-pressure parts by welds which cross, or for which the minimum nominal distance between the edge of the attachment weld and the edge of the existing main welds or nozzle welds, is less than the smaller of twice the nominal thickness of the pressure part or 40 mm, should be avoided; if this is not possible, such welds shall cross the main weld completely rather than stop abruptly near it in order to avoid stress concentration.

Full penetration butt welds shall be used for any radial joints in stiffening rings and in other similar members used for stiffening and support purposes. The soundness of all such welds shall be demonstrated on completion by appropriate radiographic or ultrasonic inspection, unless the attachment of these members to the shell is designed to preclude the possibility of a defect in the radial joint propagating into the shell.

Corner joints with fillet welds only shall not be used unless the plates forming the corner are properly supported independently of such welds.

3.10.2 Welded joints for principal seams

NOTE Typical forms of weld preparation for the principal seams of vessels covered by this specification are indicated in Annex E.

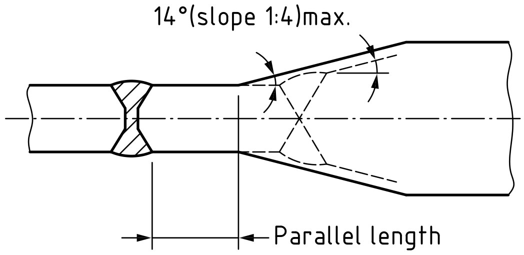
Where practicable, the longitudinal seams of adjacent courses shall be staggered by four times the nominal thickness or 100 mm, whichever is the greater, measured from the toes of the completed welds.

Where plates of different thicknesses are joined by means of butt welding, a tapered transition shall be provided as shown in Figure 3.10-1. It is permissible to include weld metal within the required taper.

Where the design requires intentional offsets of median line, and meets the requirements of **C.3.4.6.4**, special consideration of additional stresses (see **3.2.1**) shall not be required provided the offset of adjacent parts is faired by means of a taper as shown in Figure 3.10-2 and, in the case of longitudinal joints in cylinders or in the case of spherical vessel joints, (or circumferential joints in spherical vessels), the intentional offset does not exceed 10% of the nominal thickness of the thinner plate.

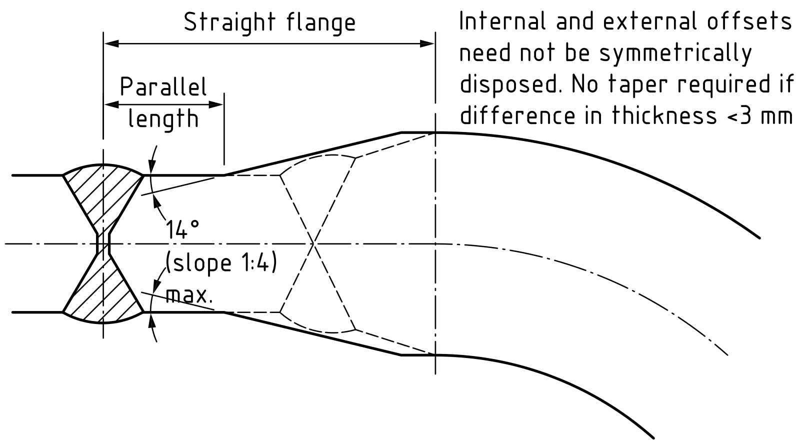
The design of principal seams where the deposited weld metal will have a yield strength (or proof stress) less than either of the materials being joined (see **4.3.2**) shall be the subject of special consideration and, if required, shall be justified by the manufacturer. The ductility of the heat affected zone shall be taken into account as necessary. The location of such seams where they may be subject to high bending stresses shall be avoided.

Figure 3.10-1 **Butt welds in plates of unequal thickness (see Annex E for details of weld preparation)**

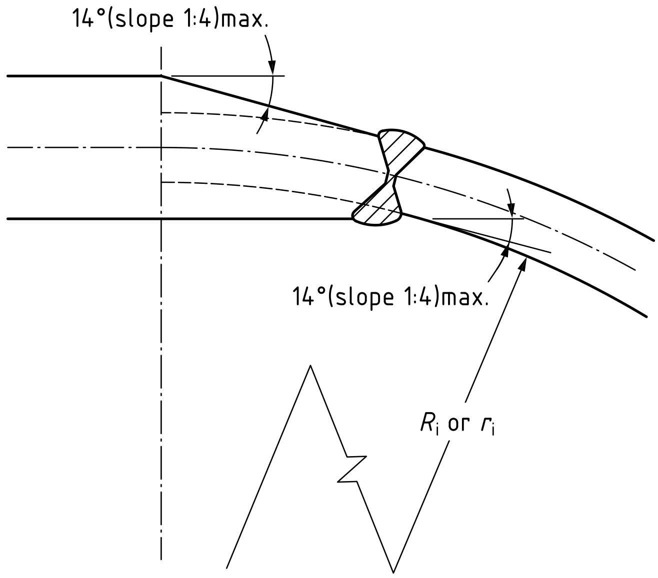


Internal and external offsets need not be symmetrically disposed

a) Plates of unequal thickness

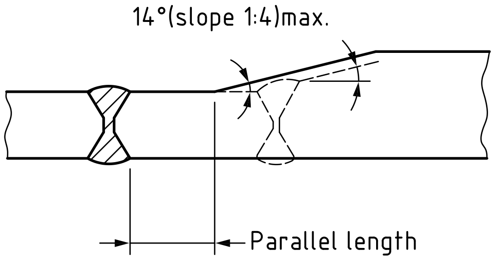


b) End thicker than shell median plane approximate coincident



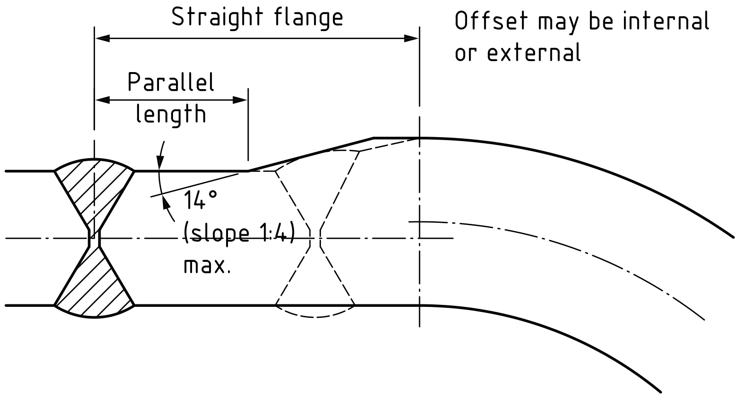
c) End thinner than shell

Figure 3.10-2 **Butt welds with offset of median lines (see 3.10.2)**



Taper may be inside vessel or outside

a) Plates of unequal thickness



b) End thicker than shell median plane offset

3.10.3 Welded joints for other than principal seams

NOTE 1 Recommended forms of weld preparation for branches, studded connections, flanges, jacketted vessels, tube to tubeplate connections, tubeplate to shell connections and flat end connections are detailed in Annex E.

It is important to note that the intention of Annex E is to exemplify sound and commonly accepted practice and not to promote standardization of connections that may be regarded as mandatory, or to restrict development in any way.

Forms of weld preparation in accordance with Annex E shall be acceptable for vessels complying with this specification subject to both:

a) the appropriate requirements of Section 4 and Section 5 being met;

b) the use of established British practice as conveyed by the information in Annex E.

NOTE 2 In the design of weld details, consideration should be given to the non-destructive testing requirements in ***5.6.4****. It is accepted, however, that the most suitable detail for a particular service condition may not necessarily be the most amenable to radiographic and/or ultrasonic inspection. Where the welding of heavy scantlings is involved, details should be selected to minimize the local restraint imposed on the weld during cooling.*[9]

NOTE 3 The recommended shapes of fillet welds, partial penetration welds and full penetration welds are given in the relevant figures of ***E.2****. These weld shapes and dimensions are linked to the thickness of one of the welded components and are based upon sound and commonly accepted practice.*

[9] See Annex F of BS EN 1011-2 for general guidance on the susceptibility of materials to lamellar tearing during fabrication.

NOTE 4 For guidelines on arc welded tube to tubeplate joints see Annex T.

Bolt holes drilled in flange rings that are fabricated from bar or stock plate which is rolled and butt welded to form the ring, should be drilled to avoid the weld joint. Where this is not possible then surface and volumetric non-destructive testing shall be carried out at the weld location in accordance with **5.6.4**.

Stresses in welds subject to fatigue loading shall be assessed in accordance with Annex C.

3.10.4 Welded joints in time dependent applications

In cases where the design strength is time dependent, due consideration shall be given to the importance of achieving adequate long term ductility of the weld material and heat affected zones, as well as that of the parent material.

3.11 Vessels with external jackets or limpet coils

3.11.1 General

Jacketted cylindrical shells (see Figure 3.11-1 and Figure 3.11-2) shall be in accordance with the requirements of **3.11.2** and **3.11.3**. Compensation of any openings through or adjacent to the jacket shall be in accordance with **3.5.4**.

Cylindrical shells with limpet coils (see Figure 3.11-3) shall be in accordance with the requirements of **3.11.4**.

For the weld attachment of the coils to the cylindrical shell see **4.3.5.4**.

NOTE Coils are within the scope of PD 5500 as the coils and their attachment welds are often subject to high thermal stresses and the welds are made onto pressure parts.

3.11.2 Jacketted cylindrical shells

3.11.2.1 General requirements

Jacketted vessels, excluding jacketted troughs, shall be designed in accordance with the requirements for each element stated elsewhere in this specification. The inner vessel shall be designed to resist the full differential pressure that may exist under any operating condition, including accidental vacuum in the inner vessel due to condensation of vapour contents where this circumstance can arise. Particular attention shall be given to the effect of local loads and differential expansion.

Where the inner vessel is to operate under vacuum and the hydraulic test pressure for the jacket is correspondingly increased to test the inner vessel externally, care shall be taken that the jacket shell is designed to withstand this extra pressure (see **5.8**).

NOTE In cases where the design strength is time dependent, components designed by the procedure specified in this section should be reviewed to ensure that creep deformation (local or general) will be acceptable throughout the agreed design lifetime.

3.11.2.2 Stayed jackets

Where jackets are retained with stays they shall be calculated as flat surfaces, unless an alternative design basis can be justified by detailed analysis of local stresses adjacent to the support points. Where leakage past a stay would be dangerous, such as in certain chemical processes, the plates shall not be perforated for the supporting stay.

3.11.2.3 External pressure design for inner jacketted cylinder

The thickness of the cylinder, when subject to external pressure, shall be determined as follows.

a) Where the design conditions for the inner cylinder include external pressure (i.e. jacket pressure) but do not include vacuum, this cylinder shall be checked to **3.6** using a cylindrical length equal to the distance between the jacket blocking rings or sealing rings. These rings shall not be considered as stiffeners for this calculation and are not checked as stiffeners to **3.6.2.2**.

b) Where the design condition for the inner cylinder is a vacuum, the whole cylinder shall be checked to **3.6** and in this case the jacket blocking rings or sealing rings can be considered as stiffeners.

c) Where the design conditions for the inner cylinder include internal vacuum plus external jacket pressure the following procedure shall be used.

   1) Calculate the permissible jacket pressure (p_{all}) acting alone on the inner cylinder in accordance with a) above.

   2) Calculate the permissible vacuum ($p_{e.all}$) acting alone on the inner cylinder in accordance with b) above.

   3) The inner cylinder thickness is adequate when:

$$\frac{p_e}{p_{e.all}} + \frac{p}{p_{all}} \leq 1$$

where

p_e = design vacuum;
p = design pressure in jacket.

3.11.3 Welded jacket connections

3.11.3.1 General

Requirements given in **3.11.3.2** and **3.11.3.3** are applicable to vessels in non-cyclic service and not subject to thermal transients. For vessels in cyclic service or subject to thermal transients the alternative methods of **3.2.2**b) shall be applied.

NOTE Typical recommended forms of the various attachments for types 1 and 2 jackets are illustrated in Annex E, Figure E.34, Figure E.35 and Figure E.36.

3.11.3.2 Notation

For the purposes of **3.11.3.3** and the figures referred to in the Note to **3.11.3.1** the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**).

D is the i.d. of jacket;
d is the o.d. of vessel;
e_c is the minimum thickness of blocking/sealer ring;
e_j is the analysis thickness of jacket;
e_r is the minimum thickness of the vessel at the junction with the blocking/sealing ring of type 2 jackets;
f is the design stress;
p is the design pressure of jacket.

3.11.3.3 Blocking and sealing rings

The thickness of blocking and sealing rings, attached to a cylindrical shell, for jackets of type 1 and type 2 shall be determined as follows. In the case of type 1 jackets, thermal stressing that could arise from differences in metal temperature, expansion coefficients, etc. shall be taken into account, as these are not allowed for in the equations.

a) The thickness of a blocking ring shall not be less than $2e_j$ or the following, whichever is the greater;

for type 1 jackets, $e_c = 0.433(D - d)\sqrt{\frac{p}{f}}$ (3.11.3-1)

for type 2 jackets, $e_c = 0.866\sqrt{\frac{pd(D - d)}{f}}$ (3.11.3-2)

b) The thickness of a sealing ring shall not be less than e_j or the following, whichever is the greater;

for type 1 jackets, $e_c = 0.433(D - d)\sqrt{\frac{p}{f}}$ (3.11.3-3)

for type 2 jackets, $e_c = 0.75\sqrt{\frac{pd(D - d)}{f}}$ (3.11.3-4)

The thickness of sealing and blocking rings attached to the dished end, as shown in Figure 3.11-2a) and with a ring diameter no larger than $d/3$, shall not be less than the wall thickness of the adjacent jacket.

For a blocking or sealing ring of type 2, the vessel cylinder thickness for a distance of $1.4\sqrt{De_r}$ on either side of the junction with the ring shall not be less than e_r where:

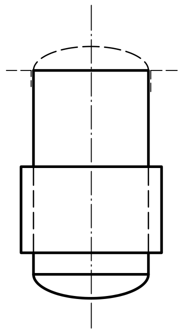
$$e_r = 0.612\sqrt{\frac{pD(D - d)}{f}} \quad (3.11.3\text{-}5)$$

NOTE Outlet branches are designed as follows.

a) Through connections
Typical constructions in which the outlet passes through the jacket space are shown in Figure E.37a) and Figure E.37b).

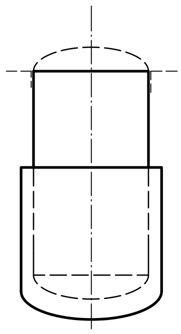
b) Flexible construction
Typical constructions in which the outlet passes through the jacket space are shown in Figure E.37a) and Figure E.37b).

Figure 3.11-1 **Some acceptable types of jacketted vessels**



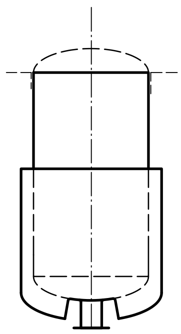
Type 1

Jacket of any length confined entirely to cylindrical shell



Type 2

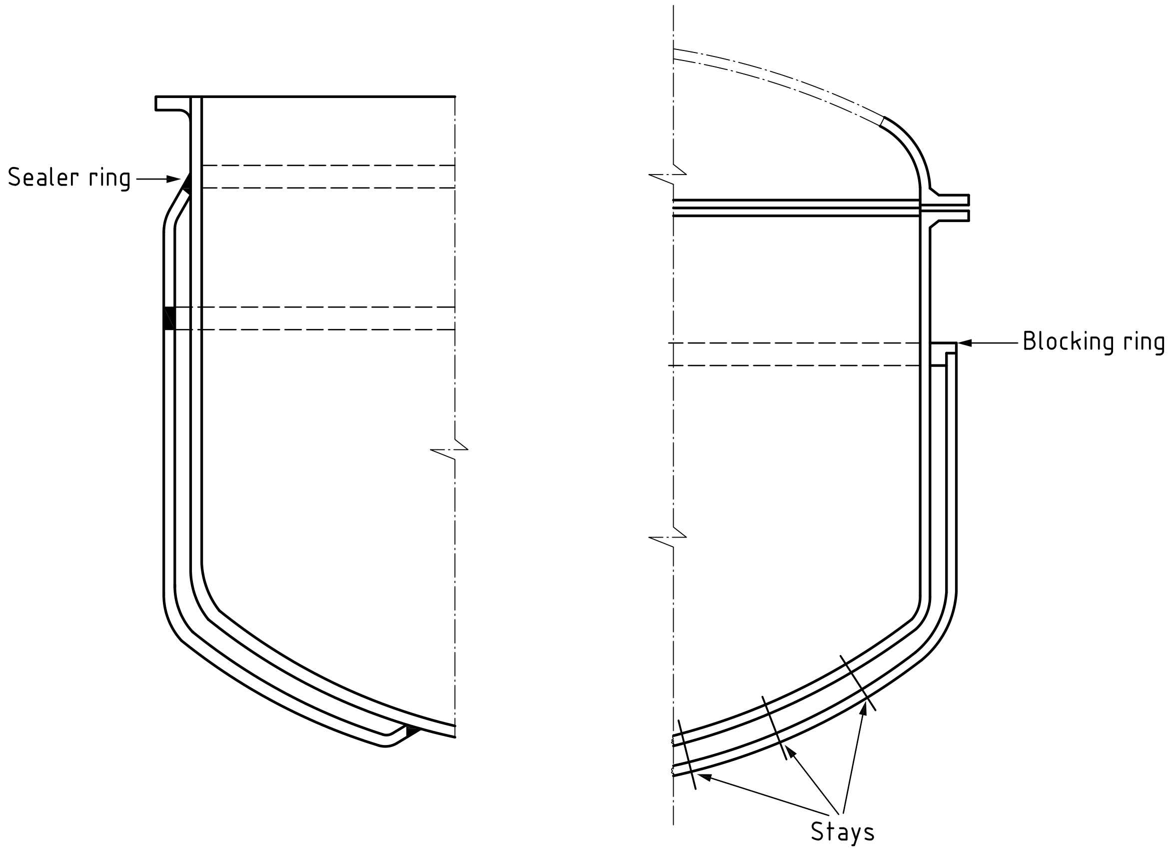
Jacket covering a portion of cylindrical shell and one dished end



Type 2

Jacket covering a portion of cylindrical shell and one dished end, with bottom outlet

Figure 3.11-2 **Typical blocking ring and sealer ring construction**



For thickness of sealer ring see **3.11.3.3**b)

a) Without stays

For thickness of blocking ring see **3.11.3.3**a)
For thickness of jacket section retained with stays, see **3.11.2**

b) Alternative, with stays

NOTE Weld profiles are diagrammatic only.

3.11.4 Cylindrical shells with limpet coils

For the purpose of **3.11.4** the following symbols apply (see Figure 3.11-3 and Figure 3.11-4). All dimensions are in the corroded conditions unless otherwise indicated (see **3.1.5**).

f is the cylinder design stress;

f_c is the limpet coil design stress;

l is the limpet coil attachment length;

L is the span of the coils;

N is the number of coils;

P is the cylinder internal design pressure;

p_c is the limpet coil internal design pressure;

r_c is the limpet coil outside radius.

Minimum limpet coil thickness:

$$e_c = \frac{p_c r_c}{0.75 f_c} \quad (3.11.3\text{-}6)$$

The minimum cylinder thickness shall comply with **3.5.1** and shall be not less than the thickness given by equation (3.11.3-7):

$$e = l\sqrt{\frac{p_c}{3f}} \tag{3.11.3-7}$$

NOTE 1 Formula (3.11.3-7) is based upon the limit load formula for bending in a flat plate of length l, with the bending moment adjusted for the interaction between longitudinal bending and circumferential membrane loads.

Where the cylinder is subject to a vacuum, the coils can be considered to contribute as light stiffeners. The total number of coils, N, shall be split into two or more groups, N_1 to N_n (see Figure 3.11-4), with the effective light stiffener at the centre of each group. The composite cylinder/coil stiffeners shall be checked to **3.6.2.3** using the effective length for each group of coils, $L_e = L_n$ where L_n is the span of the coils + one coil pitch (see Figure 3.11-4). In the calculation of p_{ys} in **3.6.2.3**, the simplification $A = 0$ shall be used.

NOTE 2 In these external pressure calculations for the cylinder, the coil internal pressure can be ignored. Separate consideration is required of any unsupported area that may result from the coils being "stopped" so as to accommodate a nozzle or support.

*The unsupported length of cylinder from the adjacent stiffener shall be checked to **3.6.2.1** using $L = L_a$.*

Figure 3.11-3 **Typical limpet coil**

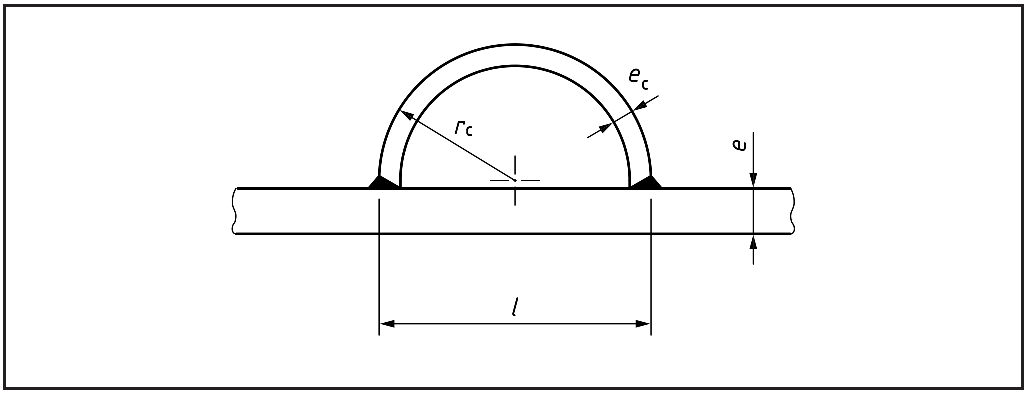
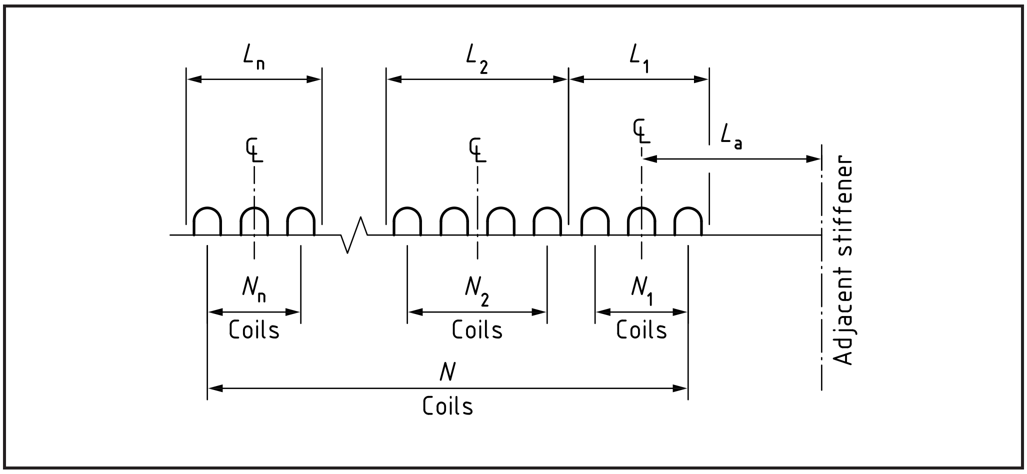


Figure 3.11-4 **Limpet coil arrangements**



3.12 Manholes, inspection openings and quick release openings

Attention is drawn to the UK Confined Spaces Regulations 1997 (SI 1997 No. 1713, ACOP L101) which call for an assessment to be made of the required size of openings. The assessment needs to be taken into account whether or not access is required for rescue purposes using self-contained breathing apparatus. All vessels subject to corrosion shall be provided with inspection and/or access openings so located as to permit a complete visual examination of the interior of the vessel.

NOTE 1 Non-UK applications might have national or regional regulations which require the assessment of openings and their size.

Manholes and inspection openings shall comply with BS 470.

NOTE 2 A range of standardized davits for branch covers of steel pressure vessels is given in BS 5276-1.

Any quick release opening shall be provided with a mechanism to ensure that the vessel cannot be pressurized until the opening is completely sealed and that the vessel is fully depressurized before the opening can be operated.

NOTE 3 BS EN 13445-5:2021, Annex C, gives details that might satisfy these requirements for quick release openings.

3.13 Protective devices for excessive pressure or vacuum

3.13.1 Application

3.13.1.1 Every pressure vessel shall be protected from excessive pressure or vacuum by an appropriate protective device, except as provided for in **3.13.1.2**. Each compartment of a subdivided vessel shall be treated as a separate vessel and be suitably connected to a protective device. Where a vessel is provided with an impervious movable partition, as in a gas loaded hydraulic accumulator, protective devices have to be provided for the spaces on both sides of each partition.

Safety valves and bursting discs shall comply with BS EN ISO 4126. The installation and other safety devices shall conform to BS 1123 or BS EN ISO 4126 as appropriate.

NOTE Other protective devices may be accepted provided they are proved to be suitable for the purpose and reliable. Where these depend on outside sources of energy for their operation, there should be at least two independent sources, and at least two such devices, each having at least 75% of the required discharge capacity, should be provided.

3.13.1.2 When a vessel is fitted with a heating coil or other element whose failure might increase the pressure of the fluid in the vessel above the design level, the designed relieving capacity of the protective device shall normally be adequate to limit this increase to the maximum value specified in **3.13.2**.

However, when the source of pressure (or temperature) is external to the vessel and is such that the pressure cannot exceed the design pressure, it is permissible for a pressure protective device not to be provided on the vessel.

NOTE 1 Examples are the generation of pressure by a compressor or pump whose maximum output pressure cannot exceed the design pressure, or heating by steam or other fluid whose temperature cannot exceed the design temperature.

NOTE 2 Vessels connected together in a system by piping of adequate capacity, free from potential blockages and which does not contain any valve that can isolate any vessel may be considered as a system of vessels for the application of pressure relief.

NOTE 3 The use of a bursting disc as a pressure relieving device may be applicable in the following cases:

a) where pressure rise may be so rapid that the inertia of a relief valve would be a disadvantage;

b) *where even minute leakage of the fluid cannot be tolerated;*

c) *where service conditions may involve heavy deposits or gumming up such as would render a relief valve inoperative.*

NOTE 4 A register of all protective devices fitted to each vessel or system should be maintained by the user. The register should relate the location and service conditions of each device to its individual identification markings.

Where the total capacity of the devices necessary to protect an installation from overpressure requires appropriate account to be taken of operating and fault conditions, the register should also include a record of the relevant calculations.

3.13.1.3 Particular consideration shall be given to vessels contained in casings (e.g. an insulated vessel or vessel jacket where the insulation is contained within sealed cladding). If there is a foreseeable risk of the vessel or jacket failing and causing leakage of the pressurised contents into the casing space then the casing shall be capable of venting to prevent a build up of pressure. Consideration shall be given to the following preventative measures in the design:

a) provision of a vent in the casing;

b) provision of a pressure warning device within the casing;

c) designing the casing with a weak seam which would fail in a safe manner if under minimal pressure;

d) designing the casing to withstand the pressure in the vessel or jacket;

e) fitting a pressure relieving device (such as low pressure bursting disc) into the casing.

3.13.2 Capacity of relief device(s)

The total capacity of the pressure relief device or devices fitted to any vessel or system of vessels shall be sufficient to discharge the maximum quantity of fluid, liquid or gaseous, that can be generated or supplied without occurrence of a rise in vessel pressure of more than 10% above the design pressure.

NOTE 1 The safety valve standards only cover liquid or gaseous fluids. For applications where the valve(s) may be required to discharge a two-phase mixture, the type and capacity of proposed safety valves should be discussed with the valve manufacturer.

NOTE 2 Any requirements for additional safety valve capacity to prevent excessive pressure in the event of fire should be specified by the purchaser in the purchase specification after due consideration of potential fire risks and resulting hazards.

3.13.3 Pressure setting of pressure relieving devices

3.13.3.1 Safety valves shall normally be set to operate at a nominal pressure not exceeding the design pressure of the vessel at the operating temperature.

However, if the capacity is provided by more than one safety valve, it is permissible for only one of the valves to be set to operate in this way and for the additional valve or valves to be set to operate at a pressure not more than 5% in excess of the design pressure at the operating temperature, provided it complies with the overall requirements of **3.13.2**.

3.13.3.2 Bursting discs fitted in place of, or in series with, safety valves shall be rated to burst at a maximum pressure not exceeding the design pressure of the vessel at operating temperature at the temperature of the disc coincident with vessel operating temperature. Where a bursting disc is fitted downstream of a safety valve, the maximum bursting pressure shall also be compatible with the pressure rating of the discharge system and shall be in accordance with BS 2915:1990, Annex C.

NOTE In the case of bursting discs fitted in parallel with valves to protect a vessel against rapid increase of pressure (e.g. see Note 3 to ***3.13.1.2****) the bursting discs should be rated to burst at a maximum pressure not exceeding 1.25 times the design pressure of the vessel at operating temperature at the temperature of the disc coincident with vessel operating temperature.*

Section 4. Manufacture and workmanship

4.1 General aspects of construction

4.1.1 General

Before commencing manufacture, the manufacturer shall submit for approval by the purchaser and Inspecting Authority a fully dimensioned drawing showing the pressure portions of the vessel and carrying the following information (see **1.5.2**).

a) A statement that the vessel is to be constructed in accordance with this specification.

b) Specification(s) with which materials shall comply.

c) Welding procedures to be adopted for all parts of the vessel.

d) Large-scale dimensional details of the weld preparation for the longitudinal and circumferential seams, and details of the joints for branch pipes, seatings, etc., and the position of these seams and other openings.

e) Heat treatment procedure.

f) Non-destructive testing requirements.

g) Test plate requirements.

h) Design pressure(s) and temperature(s) and major structural loadings.

i) Test pressure(s).

j) Amount and location of corrosion allowance.

k) Minimum dished end thickness after forming (see **3.5.2.1**).

By agreement between the purchaser and the manufacturer, it is permissible to commence the manufacture of individual parts of the vessel before approval of the drawings of the complete vessel (see Table 1.4-2 — Items for manufacturer, purchaser and/or Inspecting Authority agreement).

The manufacturer shall not modify the approved drawings except by agreement with the purchaser and Inspecting Authority.

All the thicknesses in Section 4 are nominal (see **1.6**).

4.1.2 Material identification

The manufacturer shall maintain, to the satisfaction of the Inspecting Authority, a positive system of identification for the material used in fabrication in order that all material for pressure parts in the completed work can be traced to its origin. The system shall incorporate appropriate procedures for verifying the identity of material as received from the supplier via the material manufacturer's test certificates and/or appropriate acceptance tests. In laying out and cutting the material, the material identification mark shall be so located as to be clearly visible when the pressure part is completed[1)]. Where the material identification mark is unavoidably cut out during manufacture of a pressure part, it shall be transferred by the pressure part manufacturer to another part of this component. The transfer of the mark shall be witnessed by the manufacturer's inspection department (see Table 5.1-1).

[1)] For vessels required to operate at low temperature, see **D.8.2**.

NOTE For equipment that is subject to the Pressure Equipment Directive (PED) 2014/68/EU or the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019, materials for pressure-bearing parts should be accompanied by inspection documents in accordance with BS EN 10204 and BS EN 764-5.

Records of applicable batches of welding consumables shall be retained.

4.1.3 Order of completion of weld seams

Where any part of a vessel is made in two or more courses, the longitudinal seams shall be completed before commencing the adjoining circumferential seam(s).

4.1.4 Junction of more than two weld seams

Where more than two weld seams meet at one point, consideration shall be given by the manufacturer to the desirability of intermediate stress relief.

4.1.5 Localized thinning

Localized thinning, below the nominal thickness, resulting from cutting, forming, weld preparation or weld dressing shall not result in a thickness less than the minimum thickness plus specified allowances (see **1.6**), except where all the following conditions are fulfilled.

a) Thickness reduction shall not exceed the smaller of $e/20$ and 5 mm.

b) Area of thickness reduction shall fit in a circle of diameter equal to the smaller of e and 60 mm.

c) Any two areas of thickness reduction shall be at least $\sqrt{De}$ apart.
where

D	is the internal diameter;
e	is the nominal thickness, of the component under consideration.

d) The total area of thickness reduction shall be not greater than 2% of the total vessel area.

e) Area of thickness reduction shall not be in the knuckle area of a dished end.

f) Details of thickness reductions shall be recorded in the final documentation (see **1.5.2.2**).

Localized thinning that does not satisfy the above conditions shall be referred to the designer for consideration.

4.1.6 Rectification of departures from tolerance

If rolling, or other method of rectification, is used after final NDT to correct a small departure from the tolerances specified in **4.2.4** and **4.2.5**, an additional NDT shall be applied. The requirement of this additional NDT shall be agreed between the manufacturer, purchaser and the Inspecting Authority.

4.2 Cutting, forming and tolerances

4.2.1 Cutting of material

4.2.1.1 Method

All material shall be cut to size and shape preferably by thermal cutting or machining. However, for plates less than 25 mm thick, it is permissible to use cold shearing provided that the cut edges are dressed back mechanically by not less than 1.5 mm to provide a suitable surface to permit a satisfactory examination of the edges prior to welding.

It is permissible for plates less than 10 mm thick, which are cold sheared, not to be dressed where the cut edges are to be subsequently welded.

NOTE Where preheat is specified for welding the type of material being cut by a thermal process, it may also be necessary to preheat during cutting.

Surfaces which have been thermally cut shall be dressed back by machining or grinding to remove severe notches, slag and scale. Slight oxidation of the cut edges of material groups 1 and 2 produced by machine thermal cutting shall not be regarded as detrimental.

The cut edges of ferritic alloy steel in material groups 4 to 6, 7.1 and 9, which are cut by a thermal process, shall be dressed back by grinding or machining for a distance of 1.5 mm unless the manufacturer can demonstrate to the satisfaction of the Inspecting Authority that the material has not been adversely affected by the cutting process.

4.2.1.2 Examination of cut edges

Before carrying out further work, cut surfaces and heat affected zones shall be examined for defects, including laminations, cracks and slag inclusions. Independent examination by the Inspecting Authority may be required in the case of category 3 components (see Table 5.1-1).

The purchaser shall specify in the purchase specification where supplementary non-destructive testing of cut edges is required in addition to the usual visual methods.

The manufacturer shall notify any major defects, and provide a proposed method of rectification, to the purchaser and Inspecting Authority for their agreement.

Any material damaged in the process of cutting to size and preparation of edges shall be removed by machining, grinding or chipping back to undamaged metal.

4.2.2 Forming of shell sections and plates

4.2.2.1 General

Formed pressure parts may be cold or hot formed and may comprise dished ends, segments and other formed parts. Formed pressure parts may consist of individual sections that are formed subsequently.

Prior to forming, a visual examination of all plates shall be carried out, followed by measurement of the thickness.

As far as practicable, all hot and cold forming shall be done by machine; local heating or hammering shall not be used.

The forming and heat treatment operations shall not significantly alter the material properties from those assumed in the design or this specification.

Heavy scale remaining after any hot forming operation shall be removed by a suitable descaling process which will not impair the quality of the material or have an adverse effect on the corrosion resistance of the exposed surfaces.

NOTE ***D.8.3*** *gives recommendations for forming and heat treatment of vessels in carbon and carbon manganese steels in material groups 1, 2 and 11 designed to operate below 0 °C.*

4.2.2.2 Plates welded prior to hot or cold forming

It is permissible to butt weld plates together prior to forming provided that the joint is non-destructively tested after forming by a method agreed between the manufacturer, purchaser and Inspecting Authority.

Since welds in items subjected to hot forming temperatures, or normalized, will generally suffer significant strength reduction, the manufacturer shall ensure that the filler metal used will satisfy the weld joint design requirements after such heat treatment.

4.2.2.3 Cold forming

4.2.2.3.1 Ferritic steel (material groups 1 to 6, 7.1, 9 and 11)

If the inside radius of curvature of a cold formed cylindrical pressure part is less than 10 times the nominal thickness in the case of carbon and carbon manganese steels in material groups 1, 2 and 11, or 18 times the nominal thickness in the case of all other ferritic steels in material groups 3 to 6, 7.1 and 9, an appropriate post forming heat treatment shall be applied to restore properties to levels which will ensure that the material properties are not significantly altered from those assumed in design.

All dished ends which have been cold formed shall be heat treated for the same purpose unless the manufacturer demonstrates that the cold formed properties are adequate and the material properties are not significantly altered from those assumed in design.

4.2.2.3.2 Austenitic steel (material group 8)

Cold formed austenitic steels do not require a subsequent softening heat treatment (as described in **4.2.2.4.2**) when the minimum design temperature is −196 °C and above and any of the following conditions a), b) or c) are satisfied, unless particular corrosion resistance or other purchase requirements are specified. For a), b) and c), A, the elongation at break, shall be as defined in BS EN ISO 6892-1.

a) When the specified minimum elongation at break $A \geq 30\%$ and a level of 15% cold deformation is not exceeded. Or,

   the residual elongation after cold forming is demonstrated to be ≥ 15%. This can be assumed to be demonstrated in the case of material where the specified minimum value for elongation at break A is less than 30%, but the actual elongation as measured in the material acceptance certificate is ≥ 30%.

b) With levels of cold forming above 15%, proof is provided in individual cases that the residual elongation at break A after cold forming is at least 15%.

c) For cold formed ends (semi-elliptical, torispherical and hemispherical ends) the acceptance certificate for the base material (prior to cold forming) shows the following values for elongation at break A:

   ≥ 40% for wall thickness ≤15 mm;

   ≥ 45% for wall thickness >15 mm.

NOTE It is expected that for such dished end material there will be at least 15% residual elongation after cold forming. This can be checked using the following formula for level of deformation in dished end forms.

For conditions a) and b) the level of deformation shall be determined for shell and cone forms by:

$$\text{deformation \%} = \frac{50e}{R_m}\left(1 - \frac{R_m}{R_{mo}}\right)$$

where

e is the nominal thickness of the initial product;

R_m is the mean radius of the final product;

R_{mo} is the mean radius of the initial product;

and for dished end forms by:

$$\text{deformation \%} = 100 \ln \frac{D_b}{D_e - 2e}$$

where

e is the nominal thickness of the initial product;

D_b is the diameter of the blank or the diameter of the intermediate product;

D_e is the external diameter of the final product;

ln is the natural logarithm.

Where the preceding conditions a), b) or c) do not exist, cold formed austenitic stainless steels shall be softened after cold forming by a softening treatment as described in **4.2.2.4.2**.

4.2.2.4 Hot forming

4.2.2.4.1 Ferritic steel (material groups 1 to 6, 7.1, 9 and 11)

Forming procedures involving plate heating shall be agreed between the manufacturer, purchaser and Inspecting Authority and when required the manufacturer shall provide data to justify his procedure.

The forming procedure shall specify the plate heating rate, the holding temperature, the temperature range and time in which the forming takes place and shall give details of any heat treatment to be given to the formed part.

4.2.2.4.2 Austenitic steel (material group 8)

Austenitic steel plates to be heated for hot working shall be heated uniformly in a neutral or oxidizing atmosphere without flame impingement, to a temperature not exceeding the recommended hot working temperature of the material. Deformation shall not be carried out after the temperature of the materials has fallen below 900 °C. Local heating shall not be applied.

After hot working is completed the material shall be heated to the agreed softening temperature for a period not less than 30 min. The softening temperatures and period for warm worked, high proof material shall be agreed between the manufacturer, purchaser and Inspecting Authority. After softening, the surface shall be descaled.

4.2.2.5 Manufacture of shell plates and ends

Shell plates shall be formed to the correct contour to ensure compliance with tolerances specified in **4.2.3**.

Where practicable, dished ends shall be made from one plate. Dishing and peripheral flanging of end plates shall be done by machine, flanging preferably being done in one operation. Sectional flanging is permitted provided that it is agreed between the manufacturer, purchaser and Inspecting Authority. The flanges shall be cylindrical, of good surface and free from irregularities.

4.2.2.6 Examination of formed plates

All plates, after being formed and before carrying out further work upon them, shall be examined visually and checked for thickness.

Where required by **3.7.1** additional examination by suitable non-destructive testing methods shall also be carried out.

4.2.3 Assembly tolerances

4.2.3.1 Middle line alignments

The root faces of the welding preparations shall be aligned within the tolerances permitted by the welding procedure specification and the components shall be aligned as indicated on the drawings within the following tolerances. The tolerances shall be applied to the intended position of the middle lines of adjacent components whether coincidentally or intentionally offset.

a) For longitudinal joints in cylindrical components and joints in spherical components, the middle lines of adjacent plates shall be aligned within the following tolerances.

For plate thickness e up to and including 10 mm	1 mm.
For plate thickness e over 10 mm up to and including 50 mm	10% of thickness or 3 mm, whichever is the smaller.
For plate thickness e over 50 mm up to 200 mm	e/16 or 10 mm, whichever is the smaller.
For plate thickness e over 200 mm	tolerances are to be agreed between the manufacturer, purchaser and Inspecting Authority.

b) For circumferential joints, the middle lines of adjacent plates shall be in alignment within the following tolerances.

For plate thickness e up to and including 10 mm	1 mm.
For plate thickness e over 10 mm up to and including 60 mm	10% of thickness of thinner part plus 1 mm, or 6 mm, whichever is the smaller.
For plate thickness e over 60 mm up to 200 mm	10% of the thickness of thinner part.
For plate thickness e over 200 mm	tolerances are to be agreed between the manufacturer, purchaser and Inspecting Authority.

4.2.3.2 Surface alignment

The misalignment at the surface of the plates for plate thickness e shall be as specified in a) and b). If this misalignment is exceeded, the surface shall be tapered with a slope of 1:4 over a width that includes the width of the weld, the lower surface being built up with added weld metal, if necessary, to provide the required taper.

a) For longitudinal joints in cylindrical components and joints in spherical components, the surfaces of adjacent plates shall be aligned within the following tolerances.

For plate thickness *e* up to and including 12 mm	*e*/4.
For plate thickness *e* over 12 mm up to and including 50 mm	3 mm.
For plate thickness *e* over 50 mm	the lesser of *e*/16 or 10 mm.

b) For circumferential joints, the surfaces of adjacent plates shall be aligned within the following tolerances.

For plate thickness *e* up to and including 20 mm	*e*/4.
For plate thickness *e* over 20 mm up to and including 40 mm	5 mm.
For plate thickness *e* over 40 mm	the lesser of *e*/8 or 20 mm.

4.2.3.3 Attachments, nozzles and fittings

All pads, reinforcing plates, manhole frames, lugs, brackets, stiffeners, supports and other attachments shall fit closely, and the gap at all exposed edges to be welded shall not exceed 2 mm or one-twentieth of the thickness of the attachment at the point of attachment, whichever is greater.

Except where specific dimensions are shown on the fully dimensioned drawing, the maximum gap between the outside of any branch or shell and the inside edge of the hole of the shell, flange, reinforcing ring or backing ring shall not exceed 1.5 mm for openings up to 300 mm, and 3 mm for openings over 300 mm. To achieve this gap it is permissible to machine over a sufficient length of the outside diameter of the vessel or nozzle to accommodate the attachment to which it is to be welded. This machined length shall not extend beyond the toes or edges of the attachment welds.

4.2.4 Tolerances for vessels subject to internal pressure

4.2.4.1 Tolerances for ends

4.2.4.1.1 Circumference

The external circumference of the completed end shall not depart from the calculated circumference (based upon nominal inside diameter and the actual plate thickness) by more than the amounts shown in Table 4.2-1.

Table 4.2-1 **Circumference**

Outside diameter (nominal inside diameter plus twice actual plate thickness)	Circumferential tolerance
Up to and including 650 mm	±5 mm
Over 650 mm	±0.25% of circumference

4.2.4.1.2 Circularity (out-of-roundness)

The difference between the maximum and minimum inside diameters of the straight flange shall comply with the requirements for shells (see **4.2.4.2.3**).

4.2.4.1.3 Thickness

The thickness shall be taken as the thickness of the end after manufacture and shall be applicable over the whole area of the end. Variations in thickness (thinning) arising during manufacture shall be gradual.

4.2.4.1.4 Profile

The inner depth of dishing, measured along the vessel centre line, from the plane passing through the point where the straight flange joins the knuckle radius, shall in no case be less than the theoretical depth, nor shall this depth be exceeded by more than the values given in Table 4.2-2. Variations of the profile shall not be abrupt but shall merge gradually in the specified shape. The knuckle radius shall not be less than specified, and shall have common tangents with both the straight flange and the dished profile, at each join. The crown radius shall not be greater than that specified in the design.

Table 4.2-2 **Tolerance on depth of dished ends**

Diameter of end	Permissible increase in depth of dishing
Up to and including 3 000 mm	1.25% of diameter
Over 3 000 mm up to and including 7 600 mm	38 mm
Over 7 600 mm	0.5% of diameter

4.2.4.2 Tolerances for cylindrical shells

The shell sections of completed vessels shall comply with **4.2.4.2.1**, **4.2.4.2.2** and **4.2.4.2.3**.

4.2.4.2.1 Circumference

The tolerances on circumference shall comply with **4.2.4.1.1**.

4.2.4.2.2 Straightness

The maximum deviation of the shell from a straight line shall not exceed 0.3% either of the total cylindrical length or of any individual 5 m length of the vessel. Measurements shall be made to the surface of the parent plate and not to a weld, fitting or other raised part.

4.2.4.2.3 Circularity (out-of-roundness and peaking)

The tolerance on the circularity of the shell shall be as follows.

a) The difference between the maximum and minimum internal diameters measured at any one cross-section, expressed as a percentage of the nominal shell outside diameter, D (in mm), shall not exceed:

$$\left(0.5 + \frac{625}{D}\right) \text{\% or 1\% whichever is the smaller.}$$

Measurements shall be made to the surface of the parent plate and not to a weld, fitting or other raised part.

For vessels fabricated from pipe the permissible variation in diameter (measured externally) shall be in accordance with the specification governing the manufacture of the pipe or tube.

At nozzle positions a greater out-of-roundness is permitted if it can be justified by calculation and is agreed between the manufacturer, purchaser and Inspecting Authority.

There shall be no discernible flats. Any local deviation from circularity shall be gradual.

In the case of vessels to be installed in the vertical position, which are to be checked in the horizontal position, the checks shall be repeated after turning the shell through 90° about its long axis. The measurements shall be averaged and the amount of out-of-roundness calculated from the values so determined.

b) Where irregularities in the profile occur at the welded joint and are associated with "flats" adjacent to the weld the irregularity in profile or "peaking" shall not exceed the values given in Table 4.2-3 and Table 4.2-4 except in cases covered by c) below.

The method of measurement (covering peaking and ovality) shall be by means of a 20° profile gauge (or template). The use of such a profile gauge is illustrated in Figure 4.2-1. Two readings shall be taken, P_1 and P_2 on each side of the joint, at any particular location, the maximum peaking is taken as being equivalent to 0.25 (P_1 + P_2).

The inside radius of the gauge shall be equal to the nominal outside radius of the vessel.

Measurements shall be taken at approximately 250 mm intervals on longitudinal seams to determine the location with the maximum peaking value. Other types of gauges, such as bridge gauges or needle gauges, may be used.

The maximum peaking value, when not supported by special fatigue analysis, shall be in accordance with Table 4.2-3.

Table 4.2-3 **Maximum permitted peaking**

Dimensions in mm

Vessel wall thickness e	Maximum permitted peaking
$e < 3$	1.5
$3 \leq e < 6$	2.5
$6 \leq e < 9$	3.0
$9 \leq e$	$e/3$

Peaking values in excess of the above are only permitted when supported by special fatigue analysis but in any event shall not exceed the values specified in Table 4.2-4.

c) Irregularities in profile, for vessels which have been constructed of steel having a specified minimum yield strength, R_e, exceeding 400 N/mm^2, shall not exceed 2% of the gauge length when checked by a 20° gauge.

That is:

$$\delta \leq 0.00111\ \pi D$$

where

δ is the maximum local irregularity;

D is the shell outside diameter.

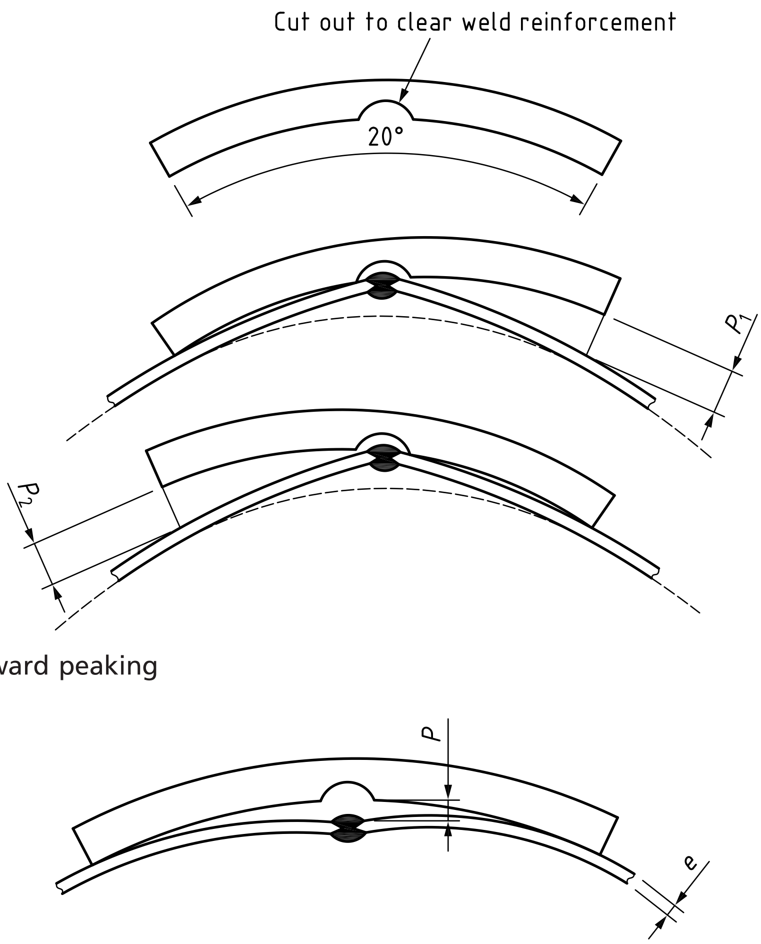
This maximum value may be increased by 25% if the length of the irregularities does not exceed one quarter of the length of the shell part between two circumferential joints with a maximum of 1 m. Greater irregularities require proof by calculation or strain gauge measurement that the stresses are permissible.

Table 4.2-4 **Maximum permitted peaking when special analysis is used**

Dimensions in mm

Vessel ratio wall thickness *e* to diameter *D*	Maximum permitted peaking
$e/D \leq 0.025$	5 or < *e*
$e/D > 0.025$	10 or < *e*

Figure 4.2-1 **Profile gauge details and application**



a) Outward peaking

b) Inward peaking

NOTE Excessive cut-out will lead to an under-estimate of maximum peaking.

4.2.4.3 Tolerances for spherical vessels

The tolerance on circularity, for both out-of-roundness and peaking, of a spherical vessel shall be in accordance with the limits specified in **4.2.4.2.3** for cylindrical vessels.

4.2.5 Tolerances for vessels subject to external pressure

The tolerances for vessels subject to internal pressure, as specified in **4.2.4**, also apply to vessels subject to external pressure. In addition the tolerances specified in **3.6.1** apply together with any allowed deviation from shape or increased tolerance as allowed in **3.6.1**. The relevant drawings and documentation shall specify the tolerances in accordance with **3.6.1**. The tolerances specified in **3.6.1** do not apply within a distance L_s from a nozzle outer wall where L_s is calculated in accordance with **3.5.4**.

It is not necessary to make measurements of radius on every vessel. A permissible procedure is to show from experience and measurement that, under defined conditions, required radial tolerances are achieved provided quality of workmanship is verified on each vessel by measurement of diameter. These measurements having been made, it is then only necessary to obtain true measurements of radius in case of doubt.

NOTE Guidance on the measurement of these tolerances is given in Enquiry Case 5500/33.

4.2.6 Structural tolerances

*NOTE Requirements for tolerances, supplemental to **4.2.3**, **4.2.4** and **4.2.5** may be specified by the purchaser in the purchase specification. For guidance on general structural tolerances, see Annex L.*

4.3 Welded joints

4.3.1 General

No production welding of joints shall be commenced until:

a) the welding procedures proposed have been approved in accordance with **5.2**;

b) welder/operators have been approved in accordance with **5.3**;

c) where stipulated by the purchaser in the purchase specification, production control test plate requirements have been agreed;

d) any examination required by the Inspecting Authority on the assembly of category 3 components has been undertaken (see Table 5.1-1).

4.3.2 Welding consumables

4.3.2.1 Welding consumables (e.g. wire, electrodes, flux, shielding gas) shall be the same type as those used in the welding procedure and shall be within the limits specified in BS EN ISO 15614-1. To ensure that no unacceptable deterioration occurs, the storing and handling of welding consumables shall be controlled in accordance with procedures written on the basis of the makers' information.

The manufacturer of the vessel shall provide evidence that the deposited weld metal is suitable in all respects for the intended duty and has tensile properties derived from the weld procedure tests not less than those specified for the parent material, except in the case of 9% Ni (sub-group 9.3) steels which shall comply with **4.3.2.2**, **4.3.2.3** and **4.3.2.4**.

4.3.2.2 Although ferritic consumables are suitable for certain 9% Ni applications, their selection shall have particular regard to toughness requirements of the weldment. Weld metal properties and thickness limits for welded joints made with a ferritic filler shall be the subject of agreement between the manufacturer, purchaser and Inspecting Authority.

4.3.2.3 For plates of 9% Ni and of thickness 20 mm and above, circular section all-weld metal tensile test pieces shall be used to measure the 0.2% proof strength ($R_{p0.2}$). For plates of 9% Ni and less than 20 mm thick, $R_{p0.2}$ shall be measured from a transverse tensile test piece in accordance with the method given in Annex X.

NOTE 1 Nickel based and some austenitic filler materials will undermatch the parent material yield strength and may also undermatch the parent metal tensile strength. The weld metal properties of these consumables should satisfy a minimum 0.2% proof strength of 360 N/mm^2.

NOTE 2 The tensile strength of the transverse tensiles should meet a minimum value of 655 N/mm^2 (equivalent to 95% of minimum parent metal properties).

4.3.2.4 It is permissible when welding 9% Ni materials to use austenitic stainless steel consumables down to −196 °C, but for temperatures below −101 °C this is only by agreement between the manufacturer, purchaser and Inspecting Authority.

4.3.3 Preparation of plate edges and openings

4.3.3.1 Weld preparations and openings of the required shape shall be formed in accordance with **4.2.1**.

4.3.3.2 The profile of the weld preparation shall be as specified in the approved welding procedure (see **5.2**).

4.3.4 Assembly for welding

4.3.4.1 Joints shall be fitted in accordance with the dimensional tolerances specified in the welding procedure specification and **4.2.3**.

4.3.4.2 It is permissible to use tack welds and incorporate them in the final weld but they shall be sound and have been made to an agreed and approved welding procedure (see **5.2**).

4.3.5 Attachments and the removal of temporary attachments

4.3.5.1 Attachments

Attachments welded directly to a pressure component shall be of the same nominal composition as that of the pressure component immediately adjacent unless for material availability reasons it is not possible to have attachments of the same nominal composition, in which case other materials or dissimilar materials (see **4.3.5.3**) may be agreed by the manufacturer, purchaser and Inspecting Authority. The welding processes and operators shall be approved in accordance with Section 5. Welds of permanent attachments to pressure parts shall be examined by appropriate non-destructive testing methods (see **5.6**).

Temporary attachments welded to the pressure parts shall be kept to a practical minimum.

4.3.5.2 Removal of attachments

Temporary attachments shall be removed prior to the first pressurization unless they have been designed to the same quality as permanent attachments. The removal technique shall be such as to avoid, as far as practicable, impairing the integrity of the pressure containment and shall be by chipping and grinding or thermal cutting followed by chipping or grinding. Any rectification necessary by welding of damaged regions after removal of attachments shall be undertaken in accordance with an approved welding procedure (see **5.2**). The area from which temporary attachments have been removed shall be dressed smooth and examined by appropriate non-destructive testing methods.

*NOTE Attention is also drawn to the requirements of **4.5.3.1** and **4.5.3.2** which apply to vessels subject to post-weld heat treatment.*

4.3.5.3 Attachments of dissimilar metal

It is permissible to attach dissimilar metal attachments to intermediate pieces, in turn connected directly to the shell. Compatible welding materials shall be used for dissimilar metal joints.

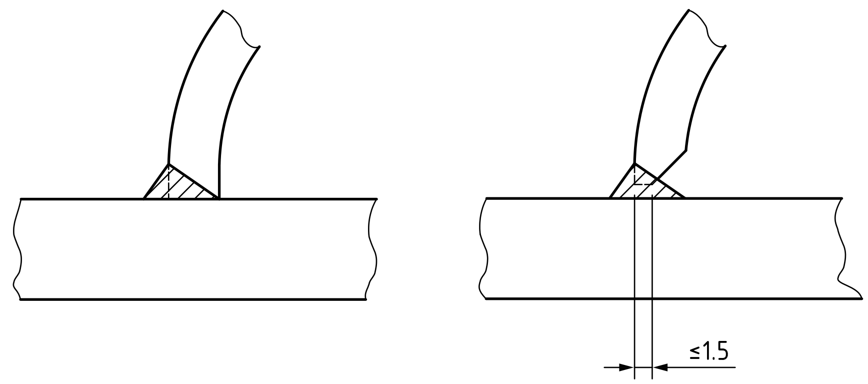
NOTE General recommendations for welding consumables and post-weld heat treatment of dissimilar ferritic steel joints are given in Annex H.

4.3.5.4 Limpet coils

Taking into account the thermal stressing and risk of crevice corrosion or stress corrosion cracking, a weld preparation designed to give full fusion through the thickness of the coil is required with a weld throat thickness not less than the coil thickness. Where the coil is thin, welding without a preparation may be used (see Figure 4.3-1) where it is demonstrated to the satisfaction of the purchaser and Inspecting Authority that it gives full fusion.

A pre-production test piece simulating the proximity of coils and accessibility is required. This test piece shall be subject to visual, surface crack and macro examination. For the whole length of the production attachment, the Inspecting Authority shall witness the set up and carry out visual inspection of the root run (if applicable) and visual inspection of the completed weld (see Table 5.1-1). Surface crack detection shall be applied to the whole length of the welds in accordance with the requirements of **5.6.6**.

Figure 4.3-1 **A thin limpet coil weld detail**



Dimensions in millimetres

4.3.6 Butt joints

4.3.6.1 Butt welds between plates of unequal thickness

Where a butt welded seam is required between plates of different thicknesses, the thicker plate shall be reduced in thickness by one of the methods allowed in **3.10.2**.

4.3.6.2 Backing strips

For construction category 1, with the exception of materials and thicknesses permitted for construction category 2, permanent backing strips shall not be used. In all other cases it is permissible to use permanent backing strips when the second side is inaccessible for welding, subject to agreement between the manufacturer, purchaser and Inspecting Authority, provided that non-destructive testing can be satisfactorily carried out where applicable and provided that the inside of the vessel is not subject to corrosion or fatigue. Only by agreement between the manufacturer, purchaser and Inspecting Authority is it permissible for circumferential butt joints in tubes to be welded with temporary, permanent or consumable backing rings (see Table 1.4-2).

Where a backing strip is to be used, the material shall be such that it will not adversely influence the weld. Unless otherwise agreed between the purchaser and the manufacturer, backing strips shall be carefully removed prior to any special non-destructive tests on the joint (see Table 1.4-2).

4.3.7 Welding: general requirements

4.3.7.1 All surfaces to be welded shall be thoroughly cleaned of oxide scale, oil or other foreign substances to a clean metal surface and for a distance of at least 12 mm from each welding edge.

4.3.7.2 Distortion due to welding shall be minimized by suitable attention to the welding sequence.

4.3.7.3 Each run of weld metal shall be thoroughly cleaned and all slag removed before the next run is deposited.

4.3.7.4 The second side of joints welded from both sides shall be cleaned back to sound metal before depositing weld metal at the second side, unless the agreed welding procedure (see **5.2**) has demonstrated that satisfactory fusion and penetration are obtained.

In the case of category 3 components, an independent examination of the second side of such joints may be required by the Inspecting Authority (see Table 5.1-1).

4.3.7.5 Stray arcing is to be avoided. Where it does occur the area affected shall be dressed by grinding and surface crack detected.

4.3.7.6 Designs required for welds are given in **3.10**.

NOTE The relatively high residual magnetism of 9% Ni steel can disrupt the welding arc. To avoid this potential problem it is generally advisable to ensure that the item to be welded has a low residual magnetism and to avoid practices that will cause magnetic induction, such as the use of magnetic handling devices and magnetic particle inspection.

4.4 Permanent joints other than welding

4.4.1 General

Where non-welded joints are made, procedures shall be qualified for all joints, in a manner similar to that used in qualifying welding procedures. Similarly, operators shall be qualified for carrying out such procedures.

4.4.2 Mechanical roller expansion

Mechanical roller expansion procedures and expansion operator approvals shall be conducted in accordance with BS EN 13445-4:2021, Annex C. Other expanding processes, e.g. hydro-expanding and explosive expanding, are not dealt with in BS EN 13445-4:2021, Annex C (for such processes see **4.4.1** above). Where mechanical roller expansion and welding are used in combination, BS EN 13445-4:2021, Annex C applies if the mechanical strength of the joint is only assured by the roller expansion process.

4.4.3 Brazing

For hand brazing and manual mechanical brazing, brazing procedures and brazer approvals shall be conducted in accordance with BS EN 13134 and BS EN ISO 13585. The rules for fully automated furnace brazing shall be in accordance with BS EN 14276-1:2020.

4.5 Heat treatment

4.5.1 Preheat requirements

4.5.1.1 The manufacturer shall state the proposed preheat temperature to avoid hard zone cracking in the heat affected zone, for each type of weld including those for all attachments and tack welds. No welding shall be carried out when the temperature of the parent metal within 150 mm of the joint is less than 5 °C.

Austenitic steels in material group B do not require preheat for welding.

The preheat temperature shall depend upon the composition and thickness of the metal being welded and upon the weld process and arc energy being used.

NOTE 1 Guidance on the selection of preheat temperature to suit particular combinations of plate composition and thickness for processes with different arc energies and diffusible hydrogen content for carbon and carbon manganese steel in material groups 1, 2 and 11 can be made by reference to:

a) BS EN 1011;

b) "Welding steels without hydrogen cracking" by F R Coe, the Welding Institute,1973.

NOTE 2 For guidance on preheating for arc welded tube to tubeplate joints, see Annex T.

4.5.1.2 The manufacturer shall include any preheat requirements in the proposed welding procedure (see **4.3.1**).

4.5.1.3 The temperature shall be checked during the period of application. The methods to check temperature shall be thermocouples, contact pyrometers or temperature indicating crayons.

4.5.1.4 Where preheat is specified welding shall continue without interruption. If, however, continuity is affected, preheat shall be maintained or the joint shall be slowly cooled under an insulation blanket. Before recommencing welding preheat shall be applied.

4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

4.5.2.1 Hot formed parts of vessels shall receive a normalizing or grain refining heat treatment, either before or after welding, unless the process of hot forming was performed within such a temperature range and followed by cooling in such a manner as would provide this treatment for the material concerned (see **2.1.2.2.2.7**).

4.5.2.2 Where normalizing is undertaken, the parts shall be brought to normalizing temperature at a suitably controlled rate and shall be maintained at the temperature long enough for thorough soaking. Actual heating rates are not critical but shall be controlled to the extent necessary to avoid any possibility of mechanical damage to the parts in question during the heating process. They shall then be uniformly cooled at the appropriate rate.

NOTE This is generally achieved by cooling freely in still air.

Where the geometry of the parts is such that the cooling rate will not be the same throughout, the necessity for a further stress relieving treatment shall be considered with particular attention being paid to a slow rate of cooling.

In the case of alloy steels, the range of cooling rates experienced shall not result in mechanical properties different from those specified.

4.5.3 Post-weld heat treatment

4.5.3.1 Post-weld heat treatment in accordance with **4.5.5** shall be carried out following completion of all welding in the following cases.

a) Ferritic steel (material groups 1 to 6, 9 and 11) vessels designed to operate above 0 °C where the thickness at any welded connection exceeds that listed in Table 4.5-1 unless otherwise agreed between purchaser, manufacturer and Inspecting Authority to permit a greater thickness based upon fracture mechanics analyses in accordance with Annex U.

b) Ferritic steel (material groups 1 to 6, 9 and 11) vessels designed to operate below 0 °C when post-weld heat treatment is necessary in accordance with Annex D.

c) Vessels intended for service with media liable to cause stress corrosion cracking in service, where, following the review required by **3.3.1**, it was felt that this cracking was still a risk with the vessel.

d) Where specified by the purchaser in the purchase specification.

In special circumstances, and by agreement between the manufacturer, purchaser and Inspecting Authority, welding is permitted to be carried out on lightly loaded and non-pressure parts of the vessels previously subjected to heat treatment, without subsequent reheat treatment, provided suitable tests and controls are instituted to establish that the material will not be adversely affected.

Where pressurized components contain fillet welds only the thickness to be used when applying the requirements for post-weld heat treatment in **4.5.3** shall be the same as that specified in **4.5.3.3**a) for two components butt welded together.

NOTE 1 Recommendations for post-weld heat treatment of dissimilar ferritic steels are given in Annex H.

NOTE 2 For guidance on post-weld heat treatment for arc welded tube to tubeplate joints, see Annex T.

4.5.3.2 The heat treatments apply specifically to the final post-weld heat treatment to be carried out on the vessel. In cases where intermediate stress relieving treatments are necessary, consideration shall be given to carrying these out at lower temperatures.

4.5.3.3 Where the vessel contains welded joints connecting parts which differ in thickness, the thickness to be used in applying the requirements for post-weld heat treatment shall be:

a) the thinner of the two parts butt welded together;

b) the thickness of the shell in connection to flanges, tube plates or similar connections;

c) the weld throat thickness of the shell or end plate to nozzle weld in nozzle attachment welds;

d) the base material thickness in material integrally clad with an austenitic or nickel base corrosion resistance material (clad plate);

e) the base material thickness divided by four where an austenitic or nickel based corrosion resistance material is weld deposited on the base material surface.

4.5.3.4 When additional welds or weld repairs have been made to a vessel after post-weld heat treatment, a further heat treatment shall be carried out in accordance with **4.5.4**. The thickness to be used in defining the time required at this further heat treatment temperature shall be the thickness of the weld applied after the original post-weld heat treatment.

4.5.3.5 For austenitic steels the details of any post-weld heat treatment shall be agreed between the manufacturer, purchaser and Inspecting Authority.

4.5.4 Methods of heat treatment

4.5.4.1 Wherever possible, the vessel shall be heat treated by heating as a whole in an enclosed furnace. Where it is impracticable to heat treat the whole vessel in a furnace it is permissible to adopt the methods described in **4.5.4.2**, **4.5.4.3**, **4.5.4.4**, **4.5.4.5** and **4.5.4.6**, but it should be noted that they may not ensure the same degree of immunity from susceptibility to stress corrosion cracking.

4.5.4.2 It is permissible to heat treat the vessel in sections in an enclosed furnace, providing the overlap is at least 1 500 mm or $5\sqrt{Re}$, whichever is the greater. Where this method is used the portion outside the furnace shall be shielded so that the longitudinal temperature gradient is such that the distance between the peak and half peak temperature is not less than $2.5\sqrt{Re}$, where R is the internal radius.

4.5.4.3 It is permissible to heat treat circumferential seams in shells locally by heating a shielded band around the entire circumference. The width of the heated band shall be not less than $5\sqrt{Re}$, the weld being in the centre. Sufficient insulation shall be fitted to ensure that the temperature of the weld and its heat affected zone is not less than that specified and that the temperature at the edge of the heated band is not less than half the peak temperature. In addition, the adjacent portion of the vessel outside the heated zone shall be thermally insulated such that the temperature gradient is not harmful.

NOTE A minimum total insulated band width of $10\sqrt{Re}$ *is recommended for the purpose of complying with this requirement.*

4.5.4.4 It is permissible to heat treat locally branches or other welded attachments by heating a shielded circumferential band around the entire vessel. In such cases either:

a) the requirements of **4.5.4.3** shall apply with the exception that the width of the heated band shall cover a minimum distance of $2.5\sqrt{Re}$ in each direction from the edge of the weld which connects the nozzle or attachment to the vessel; or

b) modifications shall be agreed between the manufacturer, purchaser and Inspecting Authority where the requirements in a) cannot be strictly applied.

4.5.4.5 It is permissible to heat the vessel internally, for which purpose it shall be fully encased with thermal insulating material.

4.5.4.6 It is permissible to post-weld heat treat vessels of different thicknesses (not exceeding a ratio of 2:1) in the same furnace charge according to the heat treatment requirements for the thickest vessel in the charge.

4.5.5 Post-weld heat treatment procedure

4.5.5.1 Post-weld heat treatment temperature and time at temperature shall be as given in Table 4.5-1.

NOTE The use of Table 4.5-1 can result in residual stresses higher than those that would have resulted from the use of the version of Table 4.5-1 published prior to amendment 2 of the 2003 edition of PD 5500. The level of residual stresses can have implications for, for example, any stress corrosion cracking.

In cases where the requirements in Table 4.5-1 cannot be strictly applied, modifications shall be agreed between the manufacturer, purchaser and Inspecting Authority.

For vessels made from materials other than sub-groups 1.1, 1.2 and 1.3, the temperature range is only advisory. The validity of any given case shall be decided by the manufacturer and the requirements modified as necessary. This shall be by agreement between the manufacturer, purchaser and Inspecting Authority.

Table 4.5-1 **Requirements for post-weld heat treatment of ferritic steel vessels**

<table>
<tr><th rowspan="2">Material group[f] in accordance with Table 2.1-1</th><th colspan="2">Steel</th><th rowspan="2">Heat treatment condition[a] of base material</th><th colspan="3">Post-weld heat treatment</th></tr>
<tr><th>Grade or type</th><th>British Standard</th><th>Nominal thickness e_n[b]
mm</th><th>Holding time
min</th><th>Holding tempera-ture
°C</th></tr>
<tr><td rowspan="2">1.1
1.2
1.3</td><td>Unalloyed steels</td><td>BS EN 10028-2
BS EN 10216-1
BS EN 10216-2
BS EN 10216-4
BS EN 10217-1
BS EN 10217-2
BS EN 10217-4
BS EN 10217-5
BS EN 10217-6
BS EN 10222-2</td><td rowspan="2">N or NT</td><td rowspan="2">≤ 35[c]
> 35 ≤ 90
> 90</td><td rowspan="2">30
$e_n - 5$
$40 + 0.5e_n$</td><td rowspan="2">550 to 600</td></tr>
<tr><td>Normalized fine grain steels</td><td>BS EN 10028-3
BS EN 10216-3
BS EN 10217-3
BS EN 10222-4</td></tr>
<tr><td>1.1
1.2</td><td>16Mo3</td><td>BS EN 10028-2
BS EN 10216-2
BS EN 10217-2
BS EN 10217-5
BS EN 10222-2</td><td>N or NT or QT</td><td>≤ 35[c]
> 35 ≤ 90
> 90</td><td>30
$e_n - 5$
$40 + 0.5e_n$</td><td>550 to 620</td></tr>
<tr><td>5.1</td><td>13CrMo4-5[d]</td><td>BS EN 10028-2
BS EN 10216-2
BS EN 10222-2</td><td>N or QT</td><td>≤ 15
> 15 ≤ 60
> 60</td><td>30
$2e_n$
$60 + e_n$</td><td>630 to 680</td></tr>
<tr><td>5.2</td><td>10CrMo9-10[e]
11CrMo9-10[e]</td><td>BS EN 10028-2
BS EN 10216-2
BS EN 10222-2</td><td>NT or QT</td><td colspan="2" rowspan="2">As specified for steel 13CrMo4-5</td><td>670 to 720</td></tr>
<tr><td>5.3</td><td>X16CrMo5-1</td><td>BS EN 10222-2</td><td>NT or A</td><td>700 to 750</td></tr>
<tr><td>5.4</td><td>X11CrMo9-1</td><td>BS EN 10216-2</td><td>NT or A</td><td>≤ 12
> 12 ≤ 60
> 60</td><td>30
$2.5e_n$
$90 + e_n$</td><td>740 to 780</td></tr>
<tr><td>6.4</td><td>X20CrMoNiV11-1</td><td>BS EN 10216-2
BS EN 10222-2</td><td>NT or QT</td><td colspan="2">As specified for steel 1CrMo9-1</td><td>730 to 770</td></tr>
<tr><td>9.1
9.2</td><td>MnNi and Ni steels except X8Ni9</td><td>BS EN 10028-4
BS EN 10216-4
BS EN 10222-3</td><td>N or NT or QT</td><td>≤ 35[c]
> 35 ≤ 90
> 90</td><td>30
$e_n - 5$
$40 + 0.5e_n$</td><td>530 to 580</td></tr>
<tr><td>9.3</td><td>X8Ni9</td><td>BS EN 10028-4
BS EN 10216-4
BS EN 10222-3</td><td>See relevant standard of base material</td><td colspan="3">Normally welded with austenitic filler metal. In view of possible carbon diffusion post-weld heat treatment should be avoided</td></tr>
</table>

[a] Heat treatment conditions of base material: A = annealed; N = normalized; NT = normalized and tempered; QT = quenched and tempered.

[b] Nominal thickness e_n is that required by **4.5.3.3** or **4.5.3.4**.

[c] For thicknesses less than or equal to 35 mm post-weld heat treatment is only necessary in special cases [e.g. to reduce the danger of stress corrosion cracking or hydrogen cracking (sour gas)].

[d] No post-weld heat treatment is required if all the following conditions are fulfilled: tubes have a nominal diameter less than 120 mm and a nominal wall thickness less than 13 mm.

[e] No post-weld heat treatment required if all the following conditions are fulfilled: tubes have a nominal diameter less than 120 mm, a nominal wall thickness less than 13 mm and a design temperature greater than 480 °C.

[f] For ferritic steel vessels fabricated using materials in groups which are not listed, guidance should be taken from Table 10.1-1 in BS EN 13445-4:2021.

4.5.5.2 Furnace post-weld heat treatment of vessels or components shall comply with the following.

a) The temperature of the furnace at the time the vessel or component is placed in it shall not exceed:

1) for ferritic steels in material groups listed in Table 4.5-1, 400 °C for vessels or components of less than 60 mm thickness and not of complex shape. 300 °C for vessels or components of 60 mm thickness or over or of complex shape;

2) for austenitic steels in material group 8, 300 °C.

b) The rate of heating from the temperature in a) for ferritic steels in material groups listed in Table 4.5-1 shall not exceed the following:

1) 220 °C/h for vessel or component thicknesses not exceeding 25 mm;

2) 5 500 °C/h divided by the thickness in millimetres for vessel or component thicknesses greater than 25 mm and not exceeding 100 mm;

3) 55 °C/h for vessel or component thickness greater than 100 mm.

c) The rate of heating from 300 °C for austenitic steels in material group 8 shall not exceed:

1) 220 °C/h for vessel or component thicknesses not exceeding 25 mm;

2) 200 °C/h for vessel or component thicknesses exceeding 25 mm.

d) During the heating and cooling periods, variation in temperature throughout the vessel or component shall not exceed 150 °C within 4 500 mm and the temperature gradient shall be gradual. Above 500 °C, this variation shall not exceed 100 °C.

e) During the heating and holding periods, the furnace atmosphere shall be so controlled as to avoid excessive oxidization of the surface of the vessel or component. There shall be no direct impingement of flame on the vessel or component.

f) When the vessel or component has attained a uniform holding temperature as given in Table 4.5-1 the temperature shall be held for the period given in Table 4.5-1.

g) Vessels or components in ferritic steels in material groups listed in Table 4.5-1 shall be cooled in the furnace to a temperature not exceeding 400 °C at a rate not exceeding the value for heating in b).

NOTE 1 Below 400 °C the component may be cooled in still air.

h) Vessels or components in austenitic steels in material group 8 shall be rapid cooled from the solution treatment temperature.

NOTE 2 Rapid cooled may be in air or quenched. Intergranular corrosion can occur if the cooling rate is not sufficiently rapid to avoid inter-granular chromium carbide precipitation. The same requirement applies to locally solution-treated welds. In these cases inter-granular corrosion is not necessarily readily visible by inspection.

4.5.5.3 Local post-weld heat treatment of vessels or components shall comply with the following.

a) The rate of heating from the temperature given in **4.5.5.2**a) shall not exceed that given in **4.5.5.2**b) or **4.5.5.2**c) as appropriate.

b) The rate of cooling down to 400 °C for ferritic steels in material groups listed in Table 4.5-1 shall not exceed that given in **4.5.5.2**g).

NOTE Below 400 °C lagging may be stripped.

c) The rate of cooling down for austenitic steels in material group 8 shall be the same as **4.5.5.2**h).

4.5.5.4 The temperature specified shall be the actual temperature of any part of the vessel or zone being heat treated, and shall be determined by thermocouples in effective contact with the vessel.

4.5.5.5 A sufficient number of temperatures shall be recorded continuously and automatically. Several thermocouples shall be applied to ensure that the whole vessel, or zone, being treated is within the range specified and additional pyrometers utilized to check that undesirable thermal gradients do not occur.

4.5.6 Mechanical properties after heat treatment

4.5.6.1 Except for material in sub-groups 1.1, 1.2 and 1.3 with treatment times less than three hours, the PWHT shall take into account the effect on the mechanical properties of the base material, including plate, forgings, pipe, and any welds, in accordance with **4.5.6.2** and **4.5.6.3**.

4.5.6.2 Where the time at temperature exceeds three hours for material in sub-groups 1.1, 1.2 and 1.3, the effect of this time at temperature shall be allowed for by either method a) or b) as follows.

a) Demonstrating, from the pre PWHT material test certification that both:

1) the actual yield and tensile strength, are greater than or equal to 15% above the specified minimum value given in the material specification; and

2) the minimum required impact test temperature (RITT), see **D.3.4**, is at least 30 °C above the specified impact test temperature of the purchased material;

or

b) Demonstrating that the material properties after welding and heat treatment as described below meet the minimum specification requirements. This shall be done either on:

1) test materials heat treated with the vessels; or

2) test coupons subjected to a simulated PWHT by the material supplier; or

3) test coupons subjected to a simulated PWHT by the vessel manufacturer.

4.5.6.3 For all other materials, except those covered in **4.5.6.2**, the PWHT temperature shall be at least 30 °C below the maximum tempering temperature, and the time at the PWHT temperature shall not exceed three hours.

When the maximum PWHT temperature is less than 30 °C below the tempering temperature or the time at PWHT exceeds three hours (including when Table 4.5-1 requires a longer time), then the effect of the PWHT on the properties of such material shall be demonstrated by one of the following:

a) on test materials heat treated with the vessels; or

b) on test coupons subjected to a simulated PWHT by the material supplier; or

c) on test coupons subjected to a simulated PWHT by the vessel manufacturer.

NOTE When simulating the effect of longer times at temperature it is permissible to cover multiple heat treatments by one heat treatment with the total summed time at the specified temperature. Alternatively, when multiple heat treatments at slightly different temperatures (±40 °C) and times are to be simulated by one heat treatment, this may be accomplished by reference to the Hollman-Jaffe time/temperature parameter, ensuring that the simulated treatment achieves the same Hollman-Jaffe time/temperature parameter as the sum of the multiple treatments.

4.6 Surface finish

4.6.1 Except where otherwise agreed between the purchaser and the manufacturer, the whole of the internal surface of the vessel shall be cleaned and shall be free from loose scale, grit, oil and grease.

4.6.2 When special types of finish are to be provided, on the inside or outside surface of the vessel, e.g. degree of polish, they shall be specified by the purchaser in the purchase specification.

Section 5. Inspection and testing

5.1 General

Each pressure vessel shall be inspected during construction. Sufficient inspections shall be made to ensure that the materials, construction and testing comply in all respects with this specification. Inspection by the Inspecting Authority shall not absolve the manufacturer from his responsibility to exercise such quality assurance procedures as will ensure that the requirements and intent of this specification are satisfied.

Table 5.1-1 summarizes the inspection stages covered in Section 4 and Section 5 in the course of which the Inspecting Authority is required to check by direct participation in, or witnessing of, particular activities that the manufacturer's quality assurance procedures are effective. Otherwise the manner in which the Inspecting Authority performs its surveillance of the manufacturer and discharges the responsibilities defined under **1.4.3** is a matter on which it shall exercise its discretion in the light of its knowledge and experience with the quality system and associated working procedures used by the manufacturer to comply with this specification.

The other principal inspection stages covered in Section 4 and Section 5 are summarized in Table 5.1-2.

The Inspecting Authority shall have access to the works of the manufacturer at all times during which work is in progress, and shall be at liberty to inspect the manufacture at any stage and to reject any part not complying with this specification. The Inspecting Authority shall have the right to require evidence that the design complies with this specification.

The Inspecting Authority shall notify the manufacturer before construction begins regarding the stages of the construction at which special examinations of materials will be made, and the manufacturer shall give reasonable notice to the Inspecting Authority when such stages will be reached, but this shall not preclude the Inspecting Authority from making examinations at any other stages, or from rejecting material or workmanship whenever they are found to be defective.

All thicknesses in Section 5 are nominal (see **1.6**).

Table 5.1-1 **Inspection stages in the course of which participation by the Inspecting Authority is mandatory** (see **5.1**)

Inspection stage	Clause No.	Remarks
Checking of drawing	**4.1.1**	Approval before manufacture
Correlation of material certificates with materials and check for conformity with material specification	**4.1.2**	The manufacturer is required to make the certificates available to the Inspecting Authority for independent checking
Identification of material and witnessing of transfer of identification marks in manufacturer's works	**4.1.2**	Origin of material to be demonstrated from available records to the satisfaction of the Inspecting Authority. Any transfer of identification marks to be witnessed by the manufacturer's inspection department *NOTE Examination of material at product maker's works, witnessing of acceptance, tests, etc. by the Inspecting Authority is not required unless specified by the purchaser in the purchase specification.*

Table 5.1-1 **Inspection stages in the course of which participation by the Inspecting Authority is mandatory (see 5.1)** *(continued)*

Inspection stage	Clause No.	Remarks
Examination of material cut edges and heat affected zones	**4.2.1.2**	For category 3 components the Inspecting Authority should not normally perform this examination on every joint of each component, but shall exercise its discretion consequent to the results of examination carried out
Approval of weld procedures	**4.3.1** **5.3**	The Inspecting Authority is required to witness tests unless the procedures are already approved
Approval of welders and operators	**4.3.1** **5.3**	The Inspecting Authority is required to witness tests unless the welders and operators are already approved
Examination of set up of seams for welding, including dimensional check, examination of weld preparations, tack welds, etc.	**4.3.1**	For category 3 components the Inspecting Authority should not normally perform this examination on every joint of each component, but shall exercise its discretion consequent to the results of examination carried out
Inspection of set up and welding	**4.3.5.4**	For limpet coils
Inspection of second side of weld preparations after first side is completed and root cleaned	**4.3.7.4**	For category 3 components the Inspecting Authority should not normally perform this examination on every joint of each component, but shall exercise its discretion consequent to the results of examination carried out
Examine non-destructive test reports and check compliance with agreed procedure and acceptability of any defects	**5.6.6.7**	The manufacturer is required to make the reports available to the Inspecting Authority for independent checking
Examine heat treatment records and check compliance with agreed procedure	**4.5.3**	The manufacturer is required to make the records available to the Inspecting Authority for independent checking
Witness the pressure test and where necessary record the amount of permanent set	**5.8**	On all categories
Examine completed vessel before despatch. Check marking	**5.8.9** **5.8.10**	On all categories

Table 5.1-2 **Other principal stages of inspection**

Inspection stage	Clause No.
Visual examination of material *f* or flaws, laminations, etc. thickness checking	**4.2.2.1**
Witnessing of production weld tests (if specified)	**5.4**
Examination of welded joints after forming	**4.2.2.2**
Examination of plates after forming	**4.2.2.6**

5.2 Approval testing of fusion welding procedures

5.2.1 Approval testing of welding procedures shall be conducted, recorded and reported in accordance with BS EN ISO 15614-1:2017+A1, Level 2, or

BS EN ISO 15614-8 as appropriate (see Annex T) as modified by **5.2.3**, **5.2.4**, **5.2.5** and **5.2.6**.

5.2.2 The manufacturer shall supply a list of all the welding procedures required in the fabrication of the vessel, which shall also identify the applicable weld procedure test/approval report or record. Test pieces shall be provided which are representative of the various thicknesses and materials to be used in applying each weld procedure, except that, in cases where the manufacturer can furnish proof of previously authenticated tests and results on the same type of joint and material within the permitted variables of BS EN ISO 15614-1:2017+A1, Level 2, or BS EN ISO 15614-8, he is not required to perform any further tests.

The production and testing of any test pieces shall be witnessed by the purchaser or the Inspecting Authority.

All welding shall be performed in accordance with a welding procedure specification or other work instruction which conforms to BS EN ISO 15609-1.

5.2.3 Additional testing shall be carried out as specified in **5.2.3.1**, **5.2.3.2**, **5.2.3.3** and **5.2.3.4** as appropriate.

5.2.3.1 In addition to the requirements given in Table 2 of BS EN ISO 15614-1:2017+A1 for butt welds in vessel plate over 10 mm thick, one all weld metal tensile test shall be carried out.

The all-weld tensile test shall be carried out in accordance with BS EN ISO 5178. Depending on which parameter the design criteria are based, the tensile and/or yield strength shall be not less than the corresponding specified minimum values for the parent metal. Due account shall be taken of special cases where undermatching weld metal has to be employed. The elongation shall be not less than 0.8 times the specified minimum value for the parent metal.

5.2.3.2 Tests shall be conducted at room temperature except for either of the following applications.

a) Applications where the design temperature exceeds the relevant temperature given in Table 5.2-1. In such cases the all weld tensile test as required by **5.2.3.1** shall be carried out (or be referred to a previous test carried out) at any temperature within the range given in Table 5.2-1. The yield stress value obtained in this test shall be not less than the specified minimum yield stress value for the parent material at the corresponding temperature.

b) Applications operating below 0 °C. Annex D gives details for the impact testing of weld procedure test plates for steels in groups 1, 2 and 4. Subclause **5.2.6** gives requirements for steels in sub-group 9.3.

Table 5.2-1 **Tensile test temperature**

Material	**Design temperature °C**	**Tensile test temperature °C**
C and CMn steels (including groups 1 and 2)	250	250 to 350
1½Cr½Mo	350	350 to 450
2¼Cr½Mo	350	350 to 450
5Cr½Mo	350	350 to 500
Stainless steel	400	400 to 550

5.2.3.3 Either of the following tests shall be carried out on branch connections.

a) A welding procedure test on a branch connection as shown in Figure 4 of BS EN ISO 15614-1:2017+A1 shall only qualify a weld procedure specification for welding a branch connection in accordance with this specification when mechanical properties of the joint have been established by an equivalent butt weld as shown in Figure 1 and Figure 2 of BS EN ISO 15614-1:2017+A1.

b) A weld procedure approval test on a butt joint in plate or pipe as shown in Figure 1 and Figure 2, respectively, of BS EN ISO 15614-1:2017+A1 shall give approval for pipe branch connections and nozzle to shell connections, where:

   1) the joint details and geometry for the branch connections have been accepted by the contracting parties; and

   2) a welded branch connection using the same joint details and geometry has been previously demonstrated as sound in any steel, on the basis of volumetric and surface non-destructive examination.

5.2.3.4 A pre-existing weld procedure test performed in accordance with BS EN 288-3 or BS 4870-1, previously acceptable to an Inspecting Authority, shall remain acceptable providing it satisfies the intent of the technical requirements of BS EN ISO 15614-1. However, the range of approval of such a test shall be in accordance with the ranges in BS EN ISO 15614-1 except as modified by **5.2.3**.

NOTE Existing procedures conforming to BS EN 288-3 or the earlier BS 4870-1 are considered technically equivalent to those specified in BS EN ISO 15614-1 when similar types of tests have been carried out. Thus the bend tests specified in BS EN 288-3 or BS 4870-1 are considered equivalent to those specified in BS EN ISO 15614-1 even though the exact number and bend angle differ. Similarly visual, radiographic, ultrasonic, surface crack detention, transverse tensile, hardness, macro and impact tests are considered equivalent.

Where BS EN ISO 15614-1 calls for a type of test to be performed that has not been carried out on the pre-existing procedure qualification tests, additional tests, as described in the Introduction of BS EN ISO 15614-1, should be carried out. For example, if impact tests have not been carried out on the BS 4870-1 test plate it is only necessary to do an additional set of impact tests on a test piece made in accordance with BS EN ISO 15614-1.

5.2.4 The preheat, interpass temperature, intermediate and post-weld heat treatments of test plates shall be the same as for production welding, except for the following.

a) As permitted within the requirements of BS EN ISO 15614-1:2017+A1, Level 2, or BS EN ISO 15614-8.

b) It is permissible to increase the preheat temperature used during fabrication by up to 100 °C without reapproval.

c) Pre-existing welding procedure qualification test plates that were post-weld heat treated for time and temperatures in accordance with the version of Table 4.5-1 prior to amendment 2 of the 2003 edition of PD 5500, shall qualify for vessel heat treatments in accordance with the lower temperatures and/or shorter times in Table 4.5-1.

*NOTE The time at temperature as applied to a pressure vessel may be increased up to two times that applied to a welding procedure approval test plate (see also **4.5.4**). Conversely, the time at temperature may be reduced from that applied to the welding procedure approval test plate, down to the minimum time allowed for a pressure vessel in accordance with Table 4.5-1.*

5.2.5 For the all weld tensile test, the amount by which the tensile strength or yield stress is permitted to exceed the specified minimum value for the parent metal shall be subject to agreement between the manufacturer, purchaser and Inspecting Authority.

5.2.6 Where 9% Ni steels are concerned, the requirements of **4.3.2** shall apply and additionally those given in Table 5.2-2.

Table 5.2-2 **Weld procedure tests for butt welds in 9% Ni steel**

Operating temperature	**Butt welds: weld procedure mechanical test for joints up to 50 mm thickness**			
	All weld metal (see Note 1) (10 mm to 50 mm)	**Transverse tensile (see Note 1)**	**Bend tests (see Note 2)**	**Impact tests for weld metals (see Note 3)**
All	2	2	to BS EN ISO 15614-1	3 test specimens: 27 J average value

NOTE 1 0.2% proof strength value of the filler metal shall be demonstrated as required in ***4.3.2****.*

NOTE 2 For undermatching strength filler metals, longitudinal bend tests may be used in lieu of root and face or side bend tests.

NOTE 3 Where non nickel-base austenitic filler metals are used, the weld fusion boundary is to be impact tested and is to comply with the same requirements as the weld metal. The location of the Charpy V-notch on the fusion boundary will be dependent upon the weld preparation and welding process and is to be agreed with the Inspecting Authority. Weld procedure test records should indicate their location by means of a sketch.

5.3 Welder and operator approval

5.3.1 Approval testing of welders and operators shall be conducted, recorded and reported in accordance with BS EN ISO 9606-1:2017 or BS EN 287-1, or with BS EN ISO 14732:2013 (welding operators), as appropriate, except for tube to tubeplate joints – see Annex T. Revalidation of welder qualifications to BS EN ISO 9606-1:2017 for pressure vessel purposes shall be in accordance with **9.3 b)** only, and revalidation of welding operator qualifications shall be in accordance with **5.3 b)** or **5.3 c)** of BS EN ISO 14732:2013.

5.3.2 All welders and welding machine operators engaged on the welding of pressure parts of vessels fabricated in accordance with this specification shall pass the welder approval tests which are designed to demonstrate their competence to make sound welds of the types on which each is to be employed.

5.3.3 Welders and welding operators who have passed the specified tests shall be approved for welding on all vessels within the limits of the procedure provided they remain in the employ of the same manufacturer. A welder or welding operator who welds successfully all the test pieces required for a welding procedure test in accordance with **5.2** shall not normally be required to undertake separate welder approval tests. If a welder or welding operator has not been engaged on the fabrication of vessels using the process and equipment appropriate to the procedure for a period of more than 6 months, or if there is any reason to doubt his ability to make satisfactory production welds, the purchaser or the Inspecting Authority may require the welder to retake the whole or part of the approval test.

NOTE 1 The approval tests of a welder or welding operator, when completed to the satisfaction of a recognized Inspection Authority, may be accepted by other Inspecting Authorities, subject to mutual agreement prior to the commencement of welding and unless otherwise stated in the enquiry and order.

*NOTE 2 The welder's qualification should be endorsed by an Inspecting Authority every 2 years in accordance with **9.3b)** of BS EN ISO 9606-1:2017 or **9.3** of BS EN 287-1:2011.*

*NOTE 3 The welding operator's qualification should be endorsed by an Inspecting Authority every 3 years in accordance with **5.3 b)** or **5.3 c)** of BS EN ISO 14732:2013.*

5.3.4 A list of welders and operators, together with records of their approval tests, shall be retained by the manufacturer.

NOTE The manufacturer may be required to submit to the purchaser evidence of approval of any welder or welding machine operator engaged in the fabrication of a vessel.

5.4 Production control test plates

5.4.1 Vessels in materials other than 9% Ni steel

Production control test plates shall not be required unless specified by the purchaser in the purchase specification or as detailed in Annex D. In such cases the number of test plates to be provided and the detailed tests to be made on these, including acceptance criteria, shall be agreed between the manufacturer, purchaser and Inspecting Authority.

NOTE Recommendations covering the preparation and testing of production test plates, when these are required, are given in Annex Q, and in Annex T in the case of arc welded tube to tubeplate joints.

5.4.2 9% Ni steel vessels

Production control test plates shall be provided until such time as the manufacturer has demonstrated that production welding produces satisfactory weld properties. The number of test pieces provided and the detailed tests to be made on these shall be agreed between the manufacturer, purchaser and Inspecting Authority taking account of the special requirements for 9% Ni steel procedure tests specified in **5.2.6**, the acceptance value being in accordance with **4.3.2**.

5.5 Destructive testing

Destructive testing shall not be required.

5.6 Non-destructive testing

5.6.1 General

The non-destructive testing of welded joints for final acceptance purposes (see **5.6.4**) shall depend on the construction category of the component as determined by Table 3.4-1, or as otherwise agreed (see **3.4.1**). Non-destructive testing of parent plate is also required, as appropriate, at the following stages:

a) examination of plate welded prior to hot forming (see **4.2.2.2**);

b) examination of areas subject to significant through thickness tensile stress (see **4.2.2.6** and **E.2.5.9**).

Visual examination shall accompany all non-destructive testing and this examination shall be recorded.

A comprehensive schedule shall be prepared by the manufacturer covering the non-destructive testing requirements for vessels, identifying the following.

1) The stages during the manufacture of the vessel (and its components) at which non-destructive testing as required by this specification will be carried out. This shall include any supplementary non-destructive testing required under the provisions of **4.2.1.2**, **5.6.4.1.2** and **5.6.4.3**.
2) The choice of non-destructive testing method and relevant procedure to be used.
3) The acceptance criteria.

NOTE It is recommended that this schedule should similarly cover any additional non-destructive testing used by the manufacturer as part of his quality control process.

Non-destructive testing personnel shall hold an appropriate certificate of competence (e.g. Personnel Certification in Non-destructive Testing (PCN)[1) which is recognized by the Inspecting Authority; otherwise the Inspecting Authority shall satisfy themselves as to the competence of such personnel.

5.6.2 Parent materials

When the purchaser specifies in the purchase specification that non-destructive testing of the parent materials is required, the procedure to be adopted shall be in accordance with appropriate British Standards as follows.

Castings	BS EN 1369 and BS EN 1371-1
Forgings (penetrant testing)	BS EN ISO 3452-1
Pipes and tubes	Appropriate annex of particular product standard
Plate	BS EN 10160

More comprehensive ultrasonic examination of plate in regions near attachment openings and welds may be necessary (see **5.6.6.2**).

Acceptance standards for flaws revealed by non-destructive testing of unwelded parent materials shall be agreed by the manufacturer and the purchaser, or the Inspecting Authority. Where repairs by welding are authorized, non-destructive testing techniques for the repair and subsequent acceptance standards shall also be agreed by the manufacturer and the purchaser, or the Inspecting Authority.

5.6.3 Components prepared for welding

Where non-destructive testing is specified to supplement the visual examination of fusion faces for welding or of plate edges (see **4.2.1** and **4.3.3.2**), the method shall be either magnetic particle or penetrant inspection.

NOTE Suitable techniques may be selected from BS EN ISO 3452-1 or BS EN ISO 17638, as appropriate.

Particular care shall be taken to ensure that residues from testing materials do not have a deleterious effect on the quality of any subsequent welding.

5.6.4 Non-destructive testing of welded joints

NOTE Guidance on non-destructive testing of arc welded tube to tubeplate joints is given in Annex T.

1) Administered by the Central Certification Board, c/o British Institute of Non-destructive Testing, Midsummer House, Riverside Way, Bedford Road, Northampton NN1 5NX.

5.6.4.1 Components to construction category 1

The final non-destructive testing shall be carried out after completion of any post-weld heat treatment, except when working in materials and thickness permitted for construction category 2 (see Table 3.4-1). Imperfections revealed by non-destructive testing shall be assessed in accordance with **5.7.2.1** and **5.7.2.2**. Where a vessel is made up of a number of category 1 components that have been stress relieved and examined as sub-assemblies and are then assembled to complete the final vessel, the whole again being stress relieved, only the welds that were made to complete the vessel, together with any intersecting weld seams for a distance of three material thicknesses from the point of intersection, shall be examined after the final stress relief of the whole vessel.

Where a further stress relief of the complete fabrication is carried out following the repair of a defect revealed by the final non-destructive testing of the vessel, only the area of the repair shall be re-examined. This examination should include the repaired area, together with a distance of three material thicknesses (not repair weld thicknesses) on either side of the repair, and should include a similar distance along any weld seams intersecting the area of repair.

5.6.4.1.1 Examination for internal flaws

The full length of all Type A welds shall be examined by radiographic or ultrasonic methods. The full length of all welded joints (other than fillet welds) of Type B in or on pressure parts shall be examined by ultrasonic and/or radiographic methods where the thinnest part to be welded exceeds the limits given in Table 5.6-1. Where a branch compensation plate is used, the shell and the compensation plate shall be considered as one component of total thickness equal to the combined thickness of the shell and compensation ring unless:

a) the branch to shell weld is separate from, *or* is completed and inspected before, the branch to compensation ring weld; and

b) the outer compensation ring to shell weld is not completed until the welds referred to in a) have been completed.

Table 5.6-1 **Thickness limits for examination of internal flaws**

Material group	Thickness mm
1, 2 and 8	40
1.2 CMo	30
4	15
Remainder	10

5.6.4.1.2 Examination for surface flaws

The full length of all Type B and all other attachment welds shall be examined by magnetic particle or penetrant methods. When a nozzle or branch is fitted with a compensation ring, the full length of the Type B nozzle to shell weld shall be examined by magnetic particle or dye penetrant methods prior to fitting of the compensation ring. The completed nozzle to compensation ring weld and the fillet weld around the periphery of the compensation ring shall also be examined after fitting of the compensation ring. Type A welds shall be examined by these methods when agreed between the manufacturer, the purchaser and the Inspecting Authority.

5.6.4.2 Components to construction category 2 (see Table 3.4-1)

Category 2 construction shall be subjected to partial non-destructive testing, as specified in **5.6.4.2.1** and **5.6.4.2.2**. Such non-destructive testing shall be employed at as early a stage in the fabrication process as practicable as a measure of quality control and the locations selected for testing shall be representative of all welding procedures and the work of each welder or operator employed. Results of non-destructive testing shall be assessed in accordance with **5.7.2.1** and **5.7.2.3**.

In cases where fabrication procedures require main seams to be welded at site, such seams shall be 100% examined by radiographic and/or ultrasonic methods generally in accordance with **5.6.5.1** and the results interpreted against the acceptance levels specified in **5.7.2.4**.

5.6.4.2.1 Examination for internal flaws

Radiographic and/or ultrasonic methods shall be in accordance with **5.6.5.1**. At each inspection location the minimum length of weld examined shall be 200 mm or the length of the weld whichever is the lesser.

a) *Welded seams other than nozzle and branch attachments*

For the purposes of this clause, a welded seam is considered to be the complete length of a butt joint between two plates forming part of a vessel as illustrated by Type A in Figure 5.6-1. At least 10% of the aggregate length of these seams shall be subject to examination. All the following locations shall be included.

1) At each intersection of longitudinal and circumferential butt joints. Where inclusion of all intersections exceeds the 10% allowance then the higher sum shall be included.

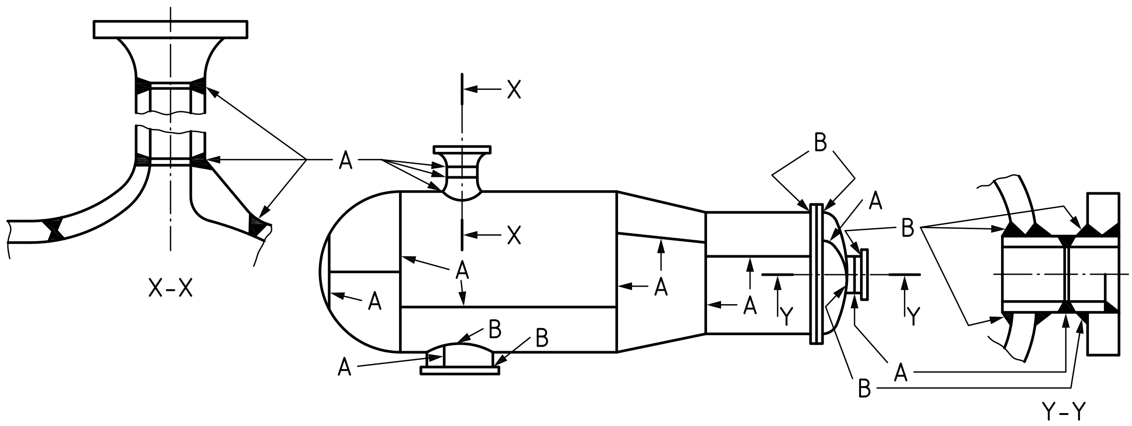
2) If necessary, randomly selected locations on longitudinal and circumferential butt joints in shells and end plates sufficient to make the total amount of examination up to at least 10%.

3) When openings occur in or within 12 mm of welded seams, such seams shall be examined on each side of the opening for a length not less than the diameter of the opening. These shall be included as an addition to a).

b) *Nozzles and branch attachments*

Butt joints as illustrated by Type A in Figure 5.6-1 shall have the total number of nozzles and branches divided into groups of 10 or less. The complete circumferential and longitudinal butt joints of at least one nozzle or branch in each group of 10 or less shall be examined.

Figure 5.6-1 **Illustration of welded joints for non-destructive testing**



Type A

Main seam welded joints within main shells, transitions in diameter, communicating chambers, jackets and nozzles. Main seam welded joints within a flat or formed end or within a sphere.

Butt welds connecting forged branches to main shells and nozzles, such as shown in Figure E.22 and Figure E.23, and in detail X-X above.

Butt welds connecting welding neck flanges, tube sheets or flat ends to main shells, to nozzles and to communicating chambers, such as shown in Figure E.32a), Figure E.35e), Figure E.38b), Figure E.38c), Figure E.41 and Figure E.43b).

Butt welds in stiffening rings and support rings.

Type B

Welded joints connecting flanges other than welding neck type, tube sheets or flat ends to main shells, to nozzles and to communicating chambers, such as shown in Figure E.31 to Figure E.44, except in the cases shown in Figure E.32a), Figure E.35e), Figure E.38b), Figure E.38c), Figure E.41 and Figure 43b).

Welded joins connecting nozzles or communicating chambers to main shells such as set-on and set-in connections shown in Figure E.8 to Figure E.30 and Figure E.37, except in the special cases shown in Figure E.22 and Figure E.23.

Welds attaching compensating plates to shell and end plates. They may be fillet welds or full penetration welds.

Butt welds in a compensating plate.

Butt welds in flange rings and blocking rings which are fabricated from bar or plate stock then rolled and butt welded to form a ring.

NOTE 1 See Figure E.1, Figure E.2, Figure E.3 and Figure E.4 for typical butt weld preparations.

*NOTE 2 Refer to **3.5.3.4**c) for additional non-destructive testing requirements for welded joint between the large end of a cone and a cylinder, without an intermediate knuckle.*

5.6.4.2.2 Examination for surface flaws

Magnetic particle and/or dye-penetrant methods shall be in accordance with **5.6.5.2**. Such examinations shall be conducted on both of the following:

a) the full length of all butt welds within flange rings and welds attaching nozzles, branches and compensating plates, to shell and end plates;

b) at least 10% of the length of all other attachment welds to pressure components.

5.6.4.3 Components to construction category 3 (see Table 3.4-1)

Unless details producing significant through thickness tensile stress (see **E.2.5.9**) are used, non-destructive testing for internal flaws shall not be required. However, subject to agreement between the manufacturer and purchaser, or Inspecting Authority it is permissible to use magnetic particle, or penetrant methods as aids to the required visual examination.

Acceptance criteria for flaws revealed by visual examination, including aided visual examination, shall be in accordance with Table 5.7-3.

Where nozzles and branch attachments are designed using category 1 or category 2 stresses, as permitted by **3.4.2.2**, the attachment weld shall be examined for surface flaws in accordance with **5.6.4.2.2**.

5.6.5 Choice of non-destructive test methods for welds

5.6.5.1 Internal flaws

The choice as to whether radiographic or ultrasonic testing is used to satisfy the requirements of this clause shall be agreed between the purchaser, the manufacturer and the Inspecting Authority.

NOTE Radiographic and ultrasonic methods both have advantages and disadvantages in so far as flaw detection, identification and sizing are concerned. Radiography is particularly suitable for the detection and identification of "volume" defects such as cavities and solid inclusions and incomplete penetration where a gap exists. Ultrasonic flaw detection is very suitable for the detection and sizing of planar defects such as cracks, lack of fusion and "tight" incomplete penetration in ferritic steels in material groups 1 to 6, 7.1, 9 and 11. The choice should be based on the most suitable method to the particular application and material. An important consideration is joint geometry which may have an overriding influence on choice of method. In exceptional cases it may be necessary to employ both methods on the same seam.

5.6.5.2 Surface flaws

NOTE Magnetic particle and penetrant testing do not indicate the depth of surface imperfections and their application is to ensure that no unacceptable surface defects are present.

The choice of method depends on material, magnetic methods being quicker and more economic for ferritic steels in material groups 1 to 6, 7.1, 9 and 11, but unsuitable for austenitic steels in material group 8, where penetrant methods shall be employed.

It is permissible to use alternative methods of non-destructive testing for the assessment of the depth of surface defects by agreement between the manufacturer and the purchaser and/or the Inspecting Authority.

5.6.6 Non-destructive testing techniques for welds

5.6.6.1 Radiographic techniques

Normally radiographic examination shall be in accordance with BS EN ISO 17636-1:2013 Class B, however the maximum area for a single exposure shall conform to the requirements of Class A.

It is permissible to use other techniques by agreement between the manufacturer and the Inspecting Authority provided it can be demonstrated that they will achieve comparable sensitivities.

5.6.6.1.1 Marking and identification of radiographs

Each section of weld radiographed shall have suitable symbols affixed to identify the following:

a) the job or workpiece serial number, order number or similar distinctive reference number;

b) the joint;

c) the section of the joint;

d) arrows, or other symbols, alongside but clear of the outer edges of the weld to clearly identify its position.

NOTE The location of the welded seam may be identified for instance with a letter L for a longitudinal seam, C for a circumferential seam, with the addition of a numeral (1, 2, 3, etc.) to indicate whether the seam was the first, second, third, etc., of that type.

The symbols consisting of lead arrows, letters and/or numerals shall be positioned so that their images appear in the radiograph to ensure unequivocal identification of the section.

Where radiographs are required of the entire length of a welded seam, sufficient overlap shall be provided to ensure that the radiographs cover the whole of the welded seam and each radiograph shall exhibit a number near each end.

Radiographs of repair welds shall be clearly identified R1, R2, etc., for the first repair, second repair, etc.

5.6.6.2 Ultrasonic techniques

Ultrasonic examination shall be in accordance with BS EN ISO 17640:2018 using the following examination levels: A for weld thicknesses less than 40 mm, B for weld thicknesses 40 mm and greater but less than 100 mm and C for weld thicknesses of 100 mm and greater.

5.6.6.3 Magnetic particle techniques

Magnetic particle inspection techniques shall comply in all respects with BS EN ISO 17638 and BS EN ISO 23278. Their use shall be limited to applications where surface flaws are being sought.

Particular care shall be taken to avoid damage to surfaces by misuse of the magnetic equipment employed and if such damage occurs it shall be remedied to the satisfaction of the Inspecting Authority.

5.6.6.4 Penetrant techniques

Dye or fluorescent penetrant examination of welds shall be carried out in accordance with BS EN ISO 3452-1.

5.6.6.5 Surface condition and preparation for non-destructive surface testing

The surface condition and preparation for non-destructive testing shall be as follows.

a) *Radiography*

   Surfaces shall be dressed only where weld ripples or weld surface irregularities will interfere with interpretation of the radiographs.

b) *Ultrasonics*

   The condition of the surfaces that will be in contact with the probe shall be in accordance with BS EN ISO 17640:2018.

NOTE 1 Depending on the profile and surface condition, dressing of the weld area may be necessary even when contact is only to be made with the parent metal.

c) *Magnetic particle method*

The surface shall be free of any foreign matter which would interfere with interpretation of the test and shall, where necessary, be dressed to permit accurate interpretation of indications.

NOTE 2 If non-fluorescent testing media are employed, a suitable contrast medium (e.g. complying with BS EN ISO 9934-2 may be applied after cleaning and prior to magnetization.

d) *Penetrant method*

The surface shall be free of any foreign matter which would interfere with the application and interpretation of the test. Care shall be taken to avoid masking of flaws by distortion of surface layers by any dressing process which may be necessary.

5.6.6.6 Marking, all non-destructive testing methods

Permanent marking of the vessel alongside welds shall be used to provide reference points for the accurate location of the seam with respect to the test report. If the purchaser requires a specific method of marking this shall be detailed in the purchase specification. Stamping shall not be used where it may have a deleterious effect on the material in service (for low temperature applications see **D.5.2**).

5.6.6.7 Reporting of non-destructive testing examinations

5.6.6.7.1 General

The following general information shall be given on reports.

a) The date and time of the examination and report.

b) The name(s) and qualifications (e.g. PCN certificate category and reference number) of the personnel responsible for the examination and the interpretation.

c) Identification of the vessel and seam under examination.

d) Brief description of joint design, material, welding process and heat treatment employed (if any).

e) Cleaning and surface preparation or dressing prior to non-destructive testing.

f) Description and location of all relevant indications of defects, together with all permanent records, e.g. radiographs, photographs, facsimiles, scale drawings or sketches, as appropriate. Corresponding reports of visual examination shall be provided.

5.6.6.7.2 Additional information for specific methods

The following additional information for specific methods shall be given on reports.

a) *Radiography*

The image quality indicator type and image quality achieved in accordance with BS EN ISO 17636-1:2013.

Details of the radiographic technique.

b) *Ultrasonics*

Report on parent metal examination including internal soundness, thickness and surface condition.

Details of the ultrasonic technique and equipment employed.

c) *Magnetic particle method*

Details of the technique(s) employed.

d) *Penetrant method*

Details of the materials and techniques employed.

5.7 Acceptance criteria for weld defects revealed by visual examination and non-destructive testing

5.7.1 General

The assessment of any defects in main constructional welds shall comply with **5.7.2**. Where the vessel is assessed to Annex C, any weld defects in main construction welds shall be assessed additionally to **C.3.4.2**.

NOTE Guidance on the acceptance criteria for arc-weld tube to tubeplate joints is given in Annex T.

5.7.2 Assessment of defects

Defects shall be assessed according to one or other of the alternatives in **5.7.2.1**, **5.7.2.2**, **5.7.2.3** and **5.7.2.4**. Defects that are unacceptable shall be either repaired or deemed not to comply with this specification.

Defects outside the limits given in **5.7.2.1**, **5.7.2.2**, **5.7.2.3** and **5.7.2.4** may be assessed to Annex U when so agreed between the purchaser, manufacturer and Inspecting Authority.

Where flaws repeatedly occur that are acceptable in accordance with this clause but outside the acceptance levels specified in BS EN ISO 9606-1, BS EN 287-1 and BS EN ISO 15614-1 for procedure and welder approval, the reasons for this shall be investigated and appropriate corrective action taken to improve future welding performance.

5.7.2.1 Category 1 and category 2 constructions

If any flaws present do not exceed the levels specified in Table 5.7-1, Table 5.7-2 or Table 5.7-3, the weld shall be accepted without further action.

NOTE Details for vessels intended for operating in the creep range may require special consideration.

5.7.2.2 Category 1 construction

When acceptance levels[2)] different from those given in Table 5.7-1, Table 5.7-2 or Table 5.7-3 have been established for a particular application and are suitably documented, it is permissible for them to be adopted by specific agreement between the purchaser, the manufacturer and the Inspecting Authority.

Similarly particular flaws[2)] in excess of those permitted in Table 5.7-1, Table 5.7-2 or Table 5.7-3 are permitted to be accepted by specific agreement between the purchaser, the manufacturer and the Inspecting Authority after due consideration of material, stress and environmental factors in each case.

[2)] For example see BS 7910.

5.7.2.3 **Category 2 construction** (see Figure 5.7-1)

The locations selected under **5.6.4.2.1** shall be deemed to be representative of the welds on which they are placed. An examination of an intersection shall be representative of two welds. A defect detected on the circumferential seam shall be representative of the whole circumferential seam. A defect detected on the longitudinal seam shall be representative of the whole longitudinal seam. A defect detected on a nozzle or branch weld shall be representative of a group of ten or less nozzle or branch welds.

a) *Table 5.7-1*, *Table 5.7-2* and *Table 5.7-3*. *Planar defects*

If any defects are present in the samples examined, the total length of the welded seam represented by each 10% sample shall be examined by the same non-destructive testing methods and assessed in accordance with **5.7.2.4** which permits some relaxation in non-planar defects.

b) *Table 5.7-1*, *Table 5.7-2* and *Table 5.7-3*. *Non-planar defects*

If there are no planar defects but the sample contains defects in excess of the maximum as given in Table 5.7-1, Table 5.7-2 and Table 5.7-3, two further random checks shall be made on the represented welds. These random checks shall be assessed against Table 5.7-1, Table 5.7-2 and Table 5.7-3.

If these checks indicate that the two additional areas are acceptable then the original sample shall be assessed in accordance with **5.7.2.4**. If outside these requirements, the area shall be repaired, re-examined by the same non-destructive testing methods and reassessed in accordance with **5.7.2.4**.

The route to be followed in the event of various imperfections being found shall be as shown in Figure 5.7-1.

5.7.2.4 Acceptance levels (reassessment of category 2 construction)

The acceptance levels given in Table 5.7-1, Table 5.7-2 and Table 5.7-3, except as modified by Table 5.7-4, and Table 5.7-5, shall be applied.

5.7.3 Repair of welds

No rectification, repair or modification shall be made without the approval of the purchaser and Inspecting Authority.

Unacceptable imperfections shall be either repaired or deemed not to comply with this specification. Repair welds shall be carried out to an approved procedure and subjected to the same acceptance criteria as original work.

Repair welds of vessels subject to fatigue loading shall be assessed in accordance with Annex C.

Table 5.7-1 **Radiographic acceptance levels**

<table>
<tr><th colspan="2">Imperfection type</th><th colspan="3">Permitted maximum</th></tr>
<tr><td rowspan="3">Planar defects</td><td>Cracks and lamellar tears</td><td colspan="3">Not permitted</td></tr>
<tr><td>Lack of root fusion
Lack of side fusion
Lack of inter-run fusion</td><td colspan="3">Not permitted</td></tr>
<tr><td>Lack of root penetration</td><td colspan="3">Not permitted</td></tr>
<tr><td rowspan="6">Cavities</td><td>a) Isolated pores (or individual pores in a group)</td><td colspan="3">$\varphi \leq e/4$ and
φ 3.0 mm for e up to and including 50 mm
φ 4.5 mm for e over 50 mm up to and including 75 mm
φ 6.0 mm for e over 75 mm</td></tr>
<tr><td>b) Uniformly distributed or localized porosity</td><td colspan="3">2% by area[a] for $e \leq 50$ mm and pro rata for greater thicknesses</td></tr>
<tr><td>c) Linear porosity</td><td colspan="3">Unless it can be shown that lack of fusion or lack of penetration is associated with this defect (which is not permitted) it should be treated as for individual pores in a group</td></tr>
<tr><td>d) Wormholes isolated</td><td colspan="3">$l \leq 6$ mm, $w \leq 1.5$ mm</td></tr>
<tr><td>e) Wormholes aligned</td><td colspan="3">As linear porosity</td></tr>
<tr><td>f) Crater pipes</td><td colspan="3">As wormholes isolated</td></tr>
<tr><td rowspan="5">Solid inclusions</td><td rowspan="3">a) Individual and parallel to major weld axis

NOTE Inclusions to be separated on the major weld axis by a distance equal to or greater than the length of the longer and the sum of the lengths of the inclusions shall not exceed the total weld length.</td><td>Main butt welds</td><td>$l = e \leq 100$ mm
$w = e/10 \leq 4$ mm</td><td></td></tr>
<tr><td rowspan="2">Nozzle and branch attachment welds</td><td>Inner half of cross-section</td><td>Outer quarters of cross-section</td></tr>
<tr><td>$w = e/4 \leq 4$ mm

$l = \frac{c}{4} \leq$ 100 mm</td><td>$w = e/8 \leq 4$ mm

$l = \frac{c}{8} \leq$ 100 mm</td></tr>
<tr><td>b) Individual and randomly oriented (not parallel to weld axis)</td><td colspan="3">As isolated pores</td></tr>
<tr><td>c) Non-linear group</td><td colspan="3">As localized porosity</td></tr>
<tr><td colspan="5">Abbreviations used:
e is the parent metal thickness. In the case of dissimilar thicknesses e applies to the thinner component;
w is the width of imperfections;
l is the length of imperfections;
φ is the diameter of imperfections;
c is the mean length of the circumferential weld.</td></tr>
</table>

NOTE 1 The simultaneous presence of more than one type of allowable flaw within a given length of weld is permitted and each type should be individually assessed.

NOTE 2 "Inner half" of cross-section refers to the middle region, the remainder being the "outer quarters".

[a] Area to be considered should be the length of the weld affected by porosity, but not less than 50 mm, multiplied by the maximum width of the weld locally.

Table 5.7-2 **Ultrasonic acceptance levels applicable to ferritic steels and weld metals in material groups 1 to 6, 7.1, 9 and 11, in the thickness range 7 mm to 100 mm inclusive**

<table>
<tr><th>Echo response height</th><th>Type of indication (see Note 1)
mm</th><th colspan="2">Maximum permitted dimensions
mm</th></tr>
<tr><td>Greater than DAC</td><td>All</td><td colspan="2">Nil</td></tr>
<tr><td rowspan="4">50% to 100% DAC
{(DAC – 6dB) to DAC}</td><td>Threadlike (Th)
i.e. $h < 3$</td><td colspan="2">Greater of
$l \leq \frac{e}{2}$ or ≤ 5</td></tr>
<tr><td>Volumetric (Vl)
i.e. $h \geq 3$</td><td colspan="2">w or $l \leq 5$</td></tr>
<tr><td>Planar longitudinal (Pl)
i.e. $h \geq 3$</td><td colspan="2">Lesser of
$l \leq \frac{e}{2}$ or ≤ 5</td></tr>
<tr><td>Nozzle and branch attachment welds volumetric (Vl) and threadlike (Th)</td><td>Inner half of cross-section
$l = \frac{c}{8} \leq 100$</td><td>Outer quarters of cross-section
$l = \frac{c}{6} \leq 100$</td></tr>
<tr><td rowspan="3">20% to 100% DAC
{(DAC – 14 dB) to DAC}</td><td>Planar surface (Ps)
(see Note 2)
i.e. $h \geq 3$</td><td colspan="2">$l \leq 5$</td></tr>
<tr><td>Multiple (M)
(see Note 3)</td><td colspan="2">l, w or $h \leq 5$</td></tr>
<tr><td>Isolated (Is)
i.e. $h < 3$</td><td colspan="2">$l \leq 5$</td></tr>
<tr><td rowspan="5">20% to 50% DAC
{(DAC – 14 dB) to (DAC – 6 dB)}</td><td>Threadlike (Th)
i.e. $h < 3$</td><td colspan="2">$l \leq e$</td></tr>
<tr><td>Volumetric (Vl)
i.e. $h \geq 3$</td><td colspan="2">w or $l \leq e$</td></tr>
<tr><td>Planar longitudinal (Pl)
i.e. $h \geq 3$</td><td colspan="2">$l \leq \frac{e}{2}$</td></tr>
<tr><td>Planar transverse (Pt)
i.e. $h \geq 3$</td><td colspan="2">$l \leq 5$</td></tr>
<tr><td>Nozzle and branch attachment welds volumetric (Vl) and threadlike (Th)</td><td>Inner half of cross-section
$l = \frac{c}{8} \leq 100$</td><td>Outer quarters of cross-section
$l = \frac{c}{8} \leq 100$</td></tr>
<tr><td>Less than 20% of DAC
{less than (DAC – 14 dB)}</td><td>All</td><td colspan="2">No limit</td></tr>
</table>

Table 5.7-2 **Ultrasonic acceptance levels applicable to ferritic steels and weld metals in material groups 1 to 6, 7.1, 9 and 11, in the thickness range 7 mm to 100 mm inclusive** *(continued)*

Echo response height	Type of indication (see Note 1) mm	Maximum permitted dimensions mm
Abbreviations used: *e* is the parent metal thickness. In the case of dissimilar thicknesses, *e* applies to the smaller thickness; *h* is the through-wall dimension of flaw; *w* is the width of flaw; *l* is the length of flaw; *c* is the mean length of the circumferential weld.		

NOTE 1 The following definitions apply to the types of indication covered in Table 5.7-2.

Planar longitudinal (Pl): indication having a planar nature, which lies parallel to, or closely-parallel to, the weld axis (e.g. longitudinal crack, lack of side-wall fusions, lack of inter-run fusion).

Planar transverse (Pt): indication having planar nature, which lies transverse to the weld axis (e.g. transverse crack).

Planar surface (Ps): indication of Pl or Pt, which lies within 25% of e or 6 mm (whichever is the smaller) of the nearest surface where e is the parent metal thickness or, in the case of dissimilar joined thicknesses, the smaller thickness (e.g. longitudinal and transverse cracks, lack of side-wall fusion, lack of root fusion and lack of root penetration).

Multiple (M): group or cluster of indications in which individual indications cannot be resolved at the reference sensitivity (see Note 3) (e.g. group or cluster of cavities or inclusions).

Volumetric (Vl): indications having measurable length and/or width and measurable through-wall dimension, and which cannot be classified as planar (e.g. linear or globular cavity or inclusion).

Threadlike (Th): indication having measurable length but no measurable width or through-wall dimension, and which cannot be classified as planar (e.g. linear inclusion).

Isolated point (Is): indication having no measurable dimension and which can be resolved at the reference sensitivity from neighbouring indications. (It is not possible to define from the ultrasonic information alone whether an isolated point indication is actually a pore, inclusion, short crack or small area of lack of fusion.)

NOTE 2 Indications shall be disregarded only by agreement between the manufacturer and the Inspecting Authority.

NOTE 3 Where adjacent, linearly-aligned inclusions are separated by a distance of less than twice the length of the longest inclusion, they shall be considered as continuous. The total, combined length shall be assessed against the appropriate flaw size criteria in Table 5.7-2.

Figure 5.7-1 **Partial non-destructive testing (NDT) category 2 constructions**

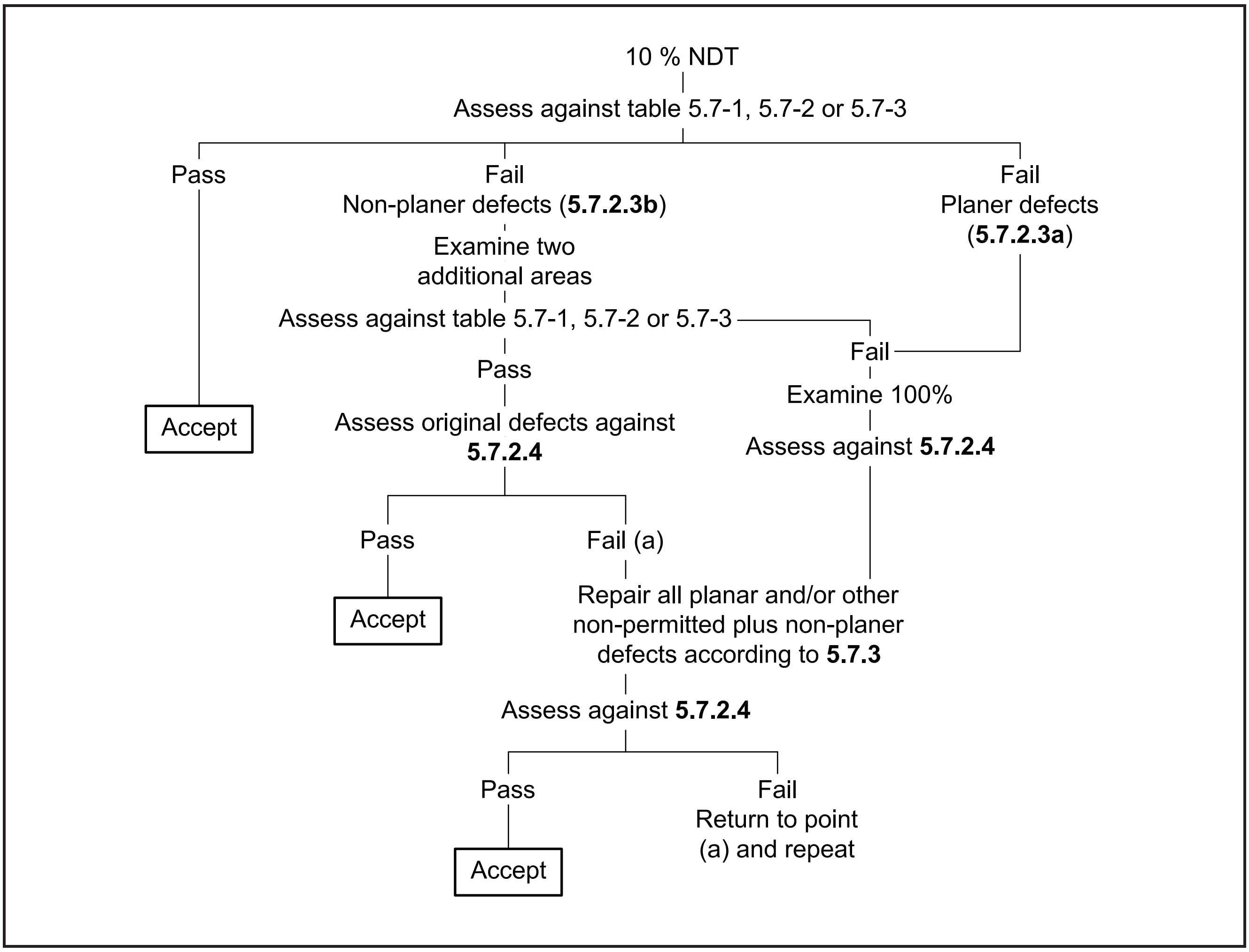


Table 5.7-3 **Visual and crack detection acceptance level**

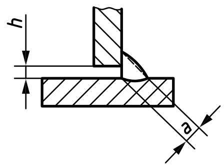
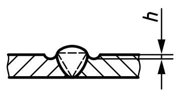
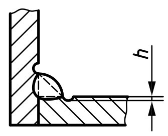
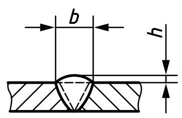
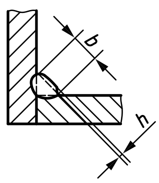
Imperfection designation	Remarks	Limits for imperfections
Planar		Not permitted
Porosity		As cavity type defects in Table 5.7-1
Bad fit-up, fillet welds	An excessive or insufficient gap between the parts to be joined  Gaps exceeding the appropriate limit may in certain cases be compensated for by a corresponding increase in the throat	$h \leq 0.5$ mm + 0.1a, max. 2 mm
Undercut	Smooth transition is required  	Long imperfections: not permitted Short imperfections: $h \leq 1.0$ mm
Excess weld metal	Smooth transition is required 	$h \leq 1$ mm + 0.1b, max. 5 mm
Excessive convexity		$h \leq 1$ mm + 0.10b, maximum 3 mm

Table 5.7-3 **Visual and crack detection acceptance level *(continued)***

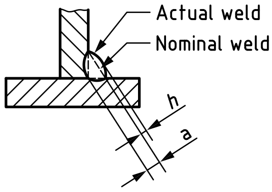
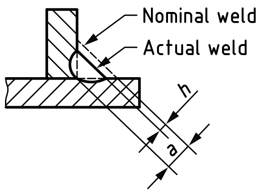
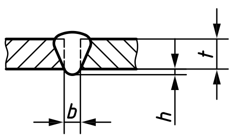
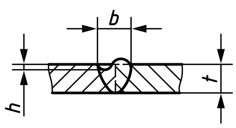
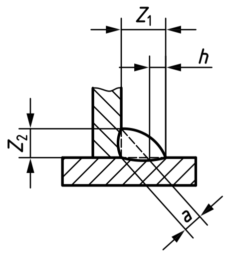
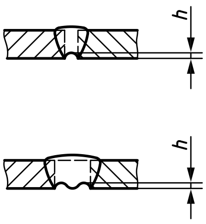
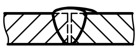
Imperfection designation	Remarks	Limits for imperfections
Fillet weld having a throat thickness greater than the nominal value	For many applications a throat thickness greater than the nominal one may not be cause for rejection 	$h \le 1$ mm + 0.30a, max. 5 mm
Fillet weld having a throat thickness smaller than the nominal value	A fillet weld with an apparent throat thickness smaller than that specified should not be regarded as being imperfect if the actual throat thickness with a compensating greater depth of penetration complies with the specified value 	Long imperfections: not permitted. Short imperfections: $h \le 0.3$ mm + 0.1a, max. 1 mm
Excessive penetration	 *NOTE* b *can be either the actual/design root gap or the width of the excessive penetration*	$h \le 1$ mm + 0.3b, max. 3 mm
Linear misalignment		See **4.2.3**
Incompletely filled groove Sagging	Smooth transition is required 	Long imperfections: not permitted Short imperfections: $h \le 0.1t$, max. 1.5 mm

Table 5.7-3 **Visual and crack detection acceptance level *(continued)***

Imperfection designation	Remarks	Limits for imperfections
Excessive asymmetry of fillet weld	It is assumed that an asymmetric fillet weld has not been expressly specified 	$h \leq 2$ mm + 0.20a
Root concavity Shrinkage groove	Smooth transition is required 	$h \leq 1.5$ mm
Overlap		Not permitted
Poor restart		Not permitted
Stray flash or arc strike		See **4.3.7.5**

NOTE The definitions of short imperfections and long imperfections are as follows.

a) Short imperfections

In cases when the weld is 100 mm long or longer, imperfections are considered to be short imperfections if, in the 100 mm which contains the greatest number of imperfections, their total length is less than 25 mm. In cases when the weld is less than 100 mm long, imperfections are considered to be short imperfections if their total length is less than 25% of the length of the weld.

b) Long imperfections

Long imperfections are those which exceed the limits specified for short imperfections in a) above.

The definition of short imperfections is identical with 3.3 of BS EN ISO 5817:2007. The wording in BS EN ISO 5817:2023, 3.2 and 3.3 has been amended, but the meaning is the same.

Table 5.7-4 **Radiographic acceptance levels (reassessment of category 2 construction)**

<table>
<tr><th>Imperfection type</th><th colspan="3">Permitted maximum</th></tr>
<tr><td>a) Isolated pores (or individual pores in a group)</td><td colspan="3">$\varphi \leq e/4 \leq 6$ mm</td></tr>
<tr><td>b) Uniformly distributed or localized porosity</td><td colspan="3">2% by area[a]</td></tr>
<tr><td rowspan="3">c) Solid inclusion, individual and parallel to major welds axis
NOTE Inclusions to be separated on the major weld axis by a distance equal to or greater than the length of the longer inclusion and aggregate length not to exceed the total length</td><td>Main butt welds</td><td>$l = 2e$</td><td>$w = e/4 \leq 4$ mm</td></tr>
<tr><td rowspan="2">Nozzle and branch attachment welds</td><td>Inner half of cross-section</td><td>Outer quarter of cross-section</td></tr>
<tr><td>$w = e/2 \leq 4$ mm
$l \leq c/2 \leq 100$ mm</td><td>$w = e/4 \leq 4$ mm
$l \leq c/4 \leq 100$ mm</td></tr>
<tr><td>d) Solid inclusions, non-linear group</td><td colspan="3">4% by area[a]</td></tr>
</table>

NOTE The symbols are as defined in Table 5.7-1.

[a] Area to be considered should be the length of the weld affected by porosity, but not less than 50 mm, multiplied by the maximum width of the weld locally.

Table 5.7-5 **Ultrasonic acceptance levels (reassessment of category 2 construction)**

Echo response height	Type of indication mm	Maximum permitted dimension mm	
Greater than DAC	All	Nil	
50% to 100% DAC {(DAC – 6dB) to DAC}	Threadlike (Th) i.e. $h < 3$	l < greater of e or 10	
	Volumetric (Vl) i.e. $h \geq 3$	w or $l \leq 10$	
	Planar longitudinal (Pl) i.e. $h > 3$	$l \leq 5$	
	Nozzle and branch attachment welds volumetric (Vl) and threadlike (Th)	Inner half of cross-section $l = \frac{c}{4} \leq 100$	Outer quarters of cross-section $l = \frac{c}{8} \leq 100$
20% to 100% DAC {(DAC – 14 dB) to DAC}	Planar surface (Ps) i.e. $h > 3$	$l \leq 5$	
	Multiple (M)	l, w or $h \leq 10$	
	Isolated (Is) i.e. $h < 3$	$l \leq 10$	
20% to 50% DAC {(DAC – 14 dB) to (DAC – 6 dB)}	Threadlike (Th) i.e. $h < 3$	$l \leq 2e$	
	Volumetric (Vl) i.e. $h \geq 3$	w or $l \leq 2e$	
	Planar longitudinal (Pl) i.e. $h \geq 3$	$l \leq \frac{e}{2}$	
	Planar transverse (Pt) i.e. $h \geq 3$	$l \leq 5$	
	Nozzle and branch attachment welds volumetric (Vl) and threadlike (Th)	Inner half of cross-section $l = \frac{c}{2} \leq 100$	Outer quarters of cross-section $l = \frac{c}{4} \leq 100$

NOTE The symbols and notes are as defined in Table 5.7-2.

5.8 Pressure tests

5.8.1 General

A pressure test shall be carried out on all vessels constructed in accordance with this specification to demonstrate, as far as it is possible with a test of this nature, the integrity of the finished product. The first pressurization shall be carried out under controlled conditions with appropriate safety precautions. Some permanent dilation of a vessel is likely on first pressurization but this possibility needs special consideration only where fine dimensional tolerances are specified for the finished vessel, in which case the effects of fabrication on the property values assumed for design purposes shall be taken into account where appropriate.

NOTE Additional detailed guidance may be obtained from HSE Guidance Note GS4, Safety in Pressure Testing, 4th edition, August 2012.

5.8.2 Basic requirements

5.8.2.1 Where practicable (see **5.8.2.5**) the finished vessel, i.e. after post-weld heat treatment, if any, shall, in the presence of the Inspecting Authority, withstand satisfactorily such of the following pressure tests as may apply.

a) "Standard" hydraulic test for acceptance where the required thickness of all pressure parts can be calculated. See **5.8.3**.

b) Pneumatic test for acceptance where the required thickness of all pressure parts can be calculated, but where the use of liquid testing media is not practicable. See **5.8.4**.

c) Proof hydraulic test where the required thickness cannot be determined by calculation. See **5.8.6**.

d) Combined hydraulic/pneumatic test. See **5.8.7**.

5.8.2.2 The procedure to be followed shall be agreed beforehand, preferably at the design stage, between the manufacturer, purchaser and Inspecting Authority and shall be such as to minimize the risk to personnel in the event of failure of the vessel during test. Consideration shall be given to factors such as the test fluid, the size and location of the vessel under test and its position relative to other buildings, plant, public roads and areas open to the public and other equipment and structures in the vicinity. This agreed procedure shall define any areas at risk during the test and how these are to be controlled.

5.8.2.3 The pressure in the vessel under test shall be gradually increased to a value of 50% of the specified test pressure; thereafter the pressure shall be increased in stages of approximately 10% of the specified test pressure until this is reached. At no stage shall the vessel be approached for close inspection until the pressure has been positively reduced to a level lower than that previously attained. The pressure(s) at which the vessel will be approached for close inspection shall be specified in the test procedure. Such pressure(s) need not exceed design pressure but, if in excess of this figure, shall not exceed 95% of the pressure already attained and held for at least 15 min.

The required test pressure shall be maintained for not less than 30 min. During the test the vessel shall exhibit no sign of general plastic yielding.

On completion of the hydraulic test, release of the pressure shall be gradual and from the top of the vessel.

Adequate venting shall be ensured before drainage, particularly in the case of large thin vessels, to prevent collapse.

5.8.2.4 If it is considered by the purchaser, manufacturer or the Inspecting Authority that there would be undue risk of brittle fracture in testing at the temperature of the available test fluid, for a vessel which would otherwise appear to be suitable for the specified service, it is permissible to elevate the test temperature to an agreed value (see **5.8.2.2** and **5.8.3.2**). This value shall not be lower than the design reference temperature obtained from Figure D.1 or Figure D.2 as appropriate for the material impact test temperature of the shell material.

5.8.2.5 Where it is not practicable to pressure test a complete vessel due to its size or mode of manufacture, the test procedure for the whole or parts of the pressure vessel shall be subject to agreement between the purchaser, the manufacturer and the Inspecting Authority at the design stage.

5.8.2.6 Each chamber of multi-compartment vessels consisting of two or more separate chambers shall be subject to the "standard" test pressure specified in **5.8.5** without support from pressure in any adjoining chamber. Where the purchaser specifies in the purchase specification that a common dividing wall can be designed for a differential pressure, this pressure shall be shown on the drawings and on the vessel nameplate (see **5.8.9**). The dividing wall shall be tested to a pressure calculated to **5.8.5.1** with *p*, the design pressure, taken as the design differential pressure.

5.8.2.7 When any chamber of a multi-compartment vessel is designed for vacuum conditions, account shall be taken of this in determining the pressure to be applied to the chamber under test.

5.8.2.8 Vessels which have been repaired subsequent to the pressure test shall be re-subjected to the specified pressure test after completion of the repairs and after any heat treatment.

5.8.2.9 All temporary pipes and connections and blanking devices shall be designed to withstand the "standard" test pressure determined in accordance with **5.8.5**.

5.8.2.10 Care shall be taken to ensure that the vessel, its supports and foundations can withstand the total load that will be imposed on them during the test.

5.8.2.11 No vessel undergoing pressure testing shall be subjected to any form of shock loading, e.g. hammer testing.

5.8.2.12 Pressure gauges used in testing vessels shall be indicating pressure gauges and shall be connected directly to the vessel or component, or with a pressure line that does not include intermediate valves.

If the indicating gauge is not readily visible to the operator controlling the pressure applied from a safe location, an additional indicating gauge shall be provided where it will be visible to the operator and inspector throughout the duration of pressurizing, testing and depressurizing or venting of the vessel or component. It is recommended that a recording gauge be used in addition to the indicating gauge, particularly for large vessels and systems when more than one gauge is specified, or required.

Dial indicating pressure gauges used in testing shall be graduated over a range of approximately double the maximum intended test pressure, but in no case shall the range be less than 1.5 times nor more than 4 times the intended test pressure. Digital reading pressure gauges having a wider range may be used, provided the readings give the same or a greater degree of accuracy than obtained with dial pressure gauges.

All indicating and recording type gauges used shall be calibrated against a standard dead weight tester, a calibrated master gauge or a mercury column, and be re-calibrated at least once a year, or at any time there is a reason to believe that they are in error.

NOTE Attention is drawn to the UK Health and Safety Executive Guidance Document GS4 which addresses safe systems of work, safeguarding and maintenance for pressure testing.

5.8.3 Hydraulic testing

5.8.3.1 The "standard" test pressure determined in accordance with **5.8.5** shall be applied.

5.8.3.2 Water shall normally be used as the pressurizing agent.

NOTE 1 To avoid the risk of freezing it is recommended that the temperature of the water during the test should be not less than 7 °C. However if the temperature of the water during the test is expected to be lower than this, special precautions may be necessary to prevent such freezing especially in small diameter branch connections.

NOTE 2 Attention is drawn to the need to control the chloride content of test water in the case of austenitic stainless steel vessels[3].

NOTE 3 Where other liquids are used, additional precautions may be necessary depending on the nature of the liquid.

5.8.3.3 Vessels and connections shall be properly vented before the test pressure is applied to prevent the formation of air pockets.

5.8.4 Pneumatic tests (see also 5.8.7 and 5.8.8)

5.8.4.1 Pneumatic testing is potentially a much more dangerous operation than hydraulic testing and is permitted only to be carried out subject to the following conditions.

a) Either on vessels of such design and construction that it is not practicable for them to be filled with liquid, or on vessels for use in processes that cannot tolerate trace liquids and where the removal of such trace liquids is impracticable.

b) After consultation at the design stage with the Inspecting Authority and other relevant safety authorities on the adequacy of the safety precautions proposed by the manufacturer to ensure that as far as possible no person is exposed to injury should the vessel fail during the test operation, and of any special precautions to minimize the risk of such failure, and with written approval by the Inspecting Authority before the test of the procedure specified in **5.8.2** with particular reference to the following:

   1) the adequacy of blast protection;

   2) the extent of area cleared for test safety purposes;

   3) the degree of confidence in stress analysis of vessel details;

   4) the adequacy of any non-destructive testing carried out before the test;

   5) the resistance of the vessel materials to fast fracture;

   6) the procedure to prevent local chilling during filling and emptying of the vessel;

   7) the extent of remote monitoring provided during test.

5.8.4.2 The "standard" test pressure determined in accordance with **5.8.5** shall be applied.

5.8.4.3 The test arrangement shall be such that the temperature of the gas entering the vessel is not lower than the agreed test temperature.

NOTE 1 Attention is drawn to the fact that if the gas pressure is let down to the vessel under test from high pressure storage, its temperature will fall.

NOTE 2 Attention is also drawn to the possibility of condensation occurring within the vessel.

[3] "Guide Notes on Safe Use of Stainless Steel in Chemical Process Plant", (1978) paragraph 1.4, Institution of Chemical Engineers, George E Davis Building, 165-171 Railway Terrace, Rugby, Warwickshire CV21 3HQ, England.

5.8.4.4 Prior to the pneumatic testing of vessels all welds not non-destructively tested in accordance with **5.6.4.1** shall be tested by magnetic particle and/or dye penetrant methods.

5.8.5 "Standard" test pressure

5.8.5.1 The following procedure to calculate the "standard" test pressure shall be applied to vessels and components.

The test pressure for hydraulic, pneumatic and combined hydraulic/pneumatic tests shall, except when otherwise stated elsewhere in **5.8**, be not less than the "standard" test pressure, p_t, determined as follows for vessels and components (see **3.4.1**).

$$p_t = 1.25\left(p\frac{f_a}{f_t} \times \frac{t}{t-c}\right) \tag{5.8.5-1}$$

where

- p is the design pressure;
- f_a is the nominal design strength value (i.e. category 1 or 2) for the material at test temperature, determined in accordance with **2.3.1.1**;
- f_t is the nominal time-independent design strength value (i.e. category 1 or 2) for the material, determined in accordance with **2.3.1.1** at the design temperature, or at the highest temperature at which time-independent properties are available if this is lower than the design temperature;
- t is the nominal thickness of the section under consideration;
- c is the corrosion allowance.

*NOTE 1 The design pressure p includes the static head of the operating liquid as specified in **3.2.3**.*

*NOTE 2 Pressure vessel components that are designed to meet the UK Pressure Equipment (Safety) Regulations (PER) or the EU Pressure Equipment Directive (PED) are required by clause 40 of PART 6 of the PER and clause 7.4 of Annex 1 of the PED to have a test pressure no lower than 1.43 times the maximum allowable pressure. The PER and the PED define maximum allowable pressure as the maximum pressure for which the equipment is designed. This test pressure might require a design check to be carried out in accordance with **5.8.5.2**.*

In the case where the vessel to be tested comprises a number of non-connected parts (e.g. the shellside and the tubeside of a heat exchanger) each part shall be tested independently with the appropriate "standard" test pressure in each case. Where the vessel comprises a number of interconnected components (e.g. the cylinder, and dished end) with different "standard" test pressures, the test pressure shall be not less than the lower bound test pressure as determined by the following procedure:

a) determine p_t in accordance with Equation (5.8.5-1) for each component of the vessel with a type A welded joint (see Figure 5.6-1) taking into account the static head of liquid applicable for each component;

b) designate the highest and lowest values so determined as p_{tH} and p_{tL} respectively;

c) where $p_{tH} \leq 1.35 \times$ design pressure, the lower bound test pressure = p_{tH};

   where $p_{tH} > 1.35 \times$ design pressure, the lower bound test pressure = $1.35p$ or p_{tL} whichever is higher.

In step c), the design pressure shall be taken as the value for the component with the largest static head.

For vertical vessels being tested in the vertical position the test pressure at the top of the vessel shall be not less than the lower bound test pressure. Due to the effect of the static head of the test liquid the pressure at the bottom of the vessel will be higher than the lower bound test pressure, and it will be necessary to carry out the check specified in **5.8.5.2**.

NOTE 3 It is not intended that the determination of the test pressure should include consideration of non-fabricated components previously pressure tested, e.g. heat exchanger tubes.

5.8.5.2 Where a component is subjected to an internal pressure test that is greater than 1.08 times the "standard" test pressure, as determined in accordance with **5.8.5.1**, then a design check shall be carried out. This design check shall be in accordance with Section 3 using a design pressure equal to the test pressure and using a design stress equal to 90% of the minimum specified yield or proof stress of the component material at test temperature. See **3.8.1.4** for the requirement for design stress for bolting during test conditions.

NOTE These design checks may be required in the following cases,

a) where it is proposed to carry out the test with the vessel in a different orientation to that to which it is designed to operate,

b) where the test pressurizing medium is denser than the design contents,

c) where the vessel is subject to a "system test" of adjacent piping,

d) where the vessel is of such a size that there are additional hydrostatic effects at a hydraulic pressure test.

Where a component is subjected to an external pressure test, a design check shall be carried out in accordance with **3.6** using an external design pressure equal to 0.8 times the external test pressure and a design stress at test temperature. Tolerances shall be in accordance with **3.6**.

5.8.5.3 Where at the time of manufacture the operating conditions of a vessel are not known, e.g. in the case of vessels made for stock, the hydraulic test pressure shall be that pressure which will generate a membrane stress of not less than 85% of the minimum specified yield or proof stress of the material at the test temperature.

5.8.5.4 Normally where a vessel is lined or coated by a process which could impair the integrity of the structure, e.g. glass lining, or weld cladding, the "standard" pressure test shall be performed after completion of this process. Alternatively, for other than weld clad vessels, it is permissible to reduce the "standard" test pressure after completion of lining to not less than 1.1 times design pressure provided that the "standard" test pressure as calculated in accordance with **5.8.5.1** has been applied before lining.

5.8.5.5 Where reasonably practicable, single wall vessels subject to operation under vacuum conditions shall be tested under vacuum or applied external pressure to simulate vacuum conditions. Where practicable, the external pressure on the vessel under test, whether resulting from vacuum in the vessel or from applied external pressure, shall be 1.25 times the design external pressure, but in no case shall it be less than the design external pressure.

Where a test under vacuum or applied external pressure is not reasonably practicable, single wall vessels subject to vacuum shall be given an internal pressure test at a gauge pressure of 1.5 bar[4)] except where the maximum possible vacuum is limited by anti vacuum valves or other suitable means. In the

[4)] 1 bar = 10^5 N/m^2 = 0.1 N/mm^2 = 100 kPa.

latter case the internal test pressure shall be a matter of agreement (see **5.8.2.2**) between the manufacturer, purchaser and the Inspecting Authority.

NOTE In special cases where the vessel designed for vacuum duty would not withstand this internal pressure test without overstrain or where the stability of the vessel under vacuum duty requires to be proven, alternative testing methods should be agreed between the purchaser, the manufacturer and the Inspecting Authority.

5.8.5.6 Where the inner vessel of a jacketted vessel is designed to operate at atmospheric pressure or under vacuum conditions, the test pressure need only be applied to the jacket space. In such cases p shall be taken as the differential design pressure between the jacket and the inner vessel for the purpose of calculating p_t (see also **5.8.5.2**).

5.8.5.7 The applied test pressure shall include the amount of any static head acting at the point under consideration.

5.8.5.8 If leakage or distortion of a gasketted component under test conditions is considered to be of concern, particular relevant design checks shall be made and arrangements made to contain any leakage.

5.8.6 Proof hydraulic test

5.8.6.1 A proof testing procedure to be followed for vessels (or vessel parts) of which the strength cannot satisfactorily be calculated (see **3.2.2**) shall be agreed between the manufacturer, purchaser and the Inspecting Authority (see **5.8.2.2**).

5.8.6.2 The procedure shall specify the method to be used during the test to determine strain and inelastic behaviour.

It is permissible to adopt either of the following methods within the limitations described in a) and b).

a) *Strain gauge technique.*

Before the test is begun or any pressure has been applied to the vessel, strain gauges of electrical resistance or other types shall be affixed to both the inside and outside surfaces of the vessel. The number of gauges, their positions and their directions shall be chosen so that principal strains and stresses can be determined at all points of interest. The type of gauge and the cementing technique shall be chosen so that strains up to 1% can be determined.

b) *Strain indicating coating technique*

The use of strain indicating techniques shall be limited to vessels in carbon or carbon manganese steels in material groups, 1, 2 and 11 of wall thickness not greater than 25 mm and where the thickness is calculated using $2f/3$ in place of f in the equations given in **3.5.1.2**.

1) The vessel shall be subjected to pressure not exceeding:

$$p_t = p \times \frac{f_a}{f_t} \times \frac{t}{t-c} \qquad (5.8.6\text{-}1)$$

(see **5.8.5.1** for nomenclature).

2) After the release of this pressure the outside surface in the areas not covered by the design rules shall be coated with a substance which will indicate the onset of yielding.

Strain indicating coatings shall be of the lime wash type or other types by agreement between the manufacturer, purchaser and the Inspecting

Authority; strain indicating coatings of the brittle lacquer type shall not be used.

A control specimen shall be prepared under simulated test conditions and strained to the onset of yield in order to demonstrate the ability of the coating to indicate first yield under the test conditions. The onset of yield shall be taken as 1 000 microstrain.

The test conditions shall simulate:

i) environmental conditions;

ii) loading rate;

iii) thickness of coating and curing conditions.

NOTE Strain indicating coatings can be used to identify the position of high stress prior to the application of strain gauges.

5.8.6.3 Pressure shall be applied gradually until either the "standard" test pressure for the expected design pressure is reached for strain gauges vessels, or "1.5/1.25" times the "standard" test pressure is reached for vessels with strain indicating coating, or significant yielding of any part of the vessel occurs.

When either of these points is reached, the pressure shall not be further increased.

If the strain gauge technique given in **5.8.6.2**a) is adopted, it is permissible to disregard any indication of localized permanent set provided that there is no evidence of general distortion of the vessel.

If the strain indicating coating technique given in **5.8.6.2**b) is adopted, the onset of yielding on (outside) surfaces shall be considered to indicate significant yielding.

NOTE The apparent difference in criteria is to allow for the fact that the greatest strains normally occur on the inside surface of the vessel.

5.8.6.4 The highest pressure which is applied shall be maintained for the time sufficient to permit inspection in accordance with **5.8.2.3**.

5.8.6.5 Where the strain gauge technique given in **5.8.6.2**a) is adopted, strain readings shall be taken as the pressure is increased. The pressure shall be increased by steps of approximately 10% until the "standard" test pressure, p_t, is reached or until significant general yielding occurs. Strain readings shall be repeated during unloading. Should the plot of strain versus pressure during the application of pressure and unloading show evidence of non-linearity it is permissible for the pressure reached to be reapplied not more than five times until the loading and unloading curves corresponding to two successive pressure cycles substantially coincide. Should coincidence not be attained, the pressure p_y (see **5.8.6.5.2**) shall be taken as the pressure range corresponding to the linear portion of the curve obtained during the final unloading.

*NOTE The term significant general yielding is intended to apply to the type of yielding which occurs when the general stress level in a substantial portion of the vessel under test exceeds the yield point of the material. It is not intended to apply to the type of yielding which occurs during the first application of (test) pressure to a component due to stress redistribution at points of unavoidable stress concentration (e.g. inside crotch of nozzles). Also it is not intended that readings obtained from gauges at such points should be considered in isolation against the requirements of **5.8.6.5**.*

5.8.6.5.1 If the "standard" test pressure, p_t, is reached and a linear pressure/strain relationship obtained, the expected design pressure shall be considered to be confirmed.

5.8.6.5.2 If the final test pressure is limited to a value less than the "standard" test pressure, p_t, or the pressure range corresponding to the linear portion of the pressure/strain record (see **5.8.6.5**) is less than p_t, the design pressure shall be calculated from the following equation:

$$p = \frac{1}{1.25}\left(p_y \frac{t-c}{t}\frac{f_t}{f_a}\right) \quad (5.8.6\text{-}2)$$

where

p	is the design pressure;
p_y	is the pressure at which significant yielding occurs or the pressure range corresponding to linear pressure/strain behaviour of most highly strained part of vessel during final unloading (see **5.8.6.5**);
t, c, f_t and f_a	are as defined in **5.8.5.1**.

5.8.6.6 Where the strain indicating coating technique given in **5.8.6.2**b) is applied to the outside surface of the vessel:

a) if 1.5/1.25 times the "standard" test pressure is reached without significant yielding, the expected design pressure shall be considered to be confirmed;

b) if significant yielding occurs at a pressure less than 1.5/1.25 times the "standard" test pressure, the design pressure shall be calculated from the equation in **5.8.6.5.2**.

5.8.7 Combined hydraulic/pneumatic tests

In cases where it is desired to test a vessel that is partly filled with liquid, the pneumatic pressure shall be applied above the liquid level and at no point of the vessel shall the total pressure applied during the test cause the general membrane stress to exceed 90% of the yield or proof stress of the material.

All the relevant requirements of **5.8.1**, **5.8.2**, **5.8.3**, **5.8.4** and **5.8.5** shall apply to the conduct of combined hydraulic/pneumatic tests.

5.8.8 Leak testing

5.8.8.1 Any requirements for a gas leak test to be carried out before the hydraulic or pneumatic test shall be given by the purchaser in the purchase specification.

NOTE Reference may be made to BS EN 1779:1999 to guide the selection of a suitable leak test method. Bubble emission testing is detailed in BS EN 1593:1999, helium testing in BS EN 13185:2001 or BS EN ISO 20485:2018 and pressure change testing in BS EN 13184:2001. A test for this purpose may be applied to any vessel without observing the requirements applying to pneumatic acceptance tests, providing the test pressure does not exceed 10% of the design pressure.

5.8.8.2 It is permissible to carry out leak testing with air or gas up to 1.1 times the design pressure on any component vessel that has satisfactorily withstood the "standard" hydraulic, pneumatic or combined hydraulic/pneumatic test.

5.8.9 Vessel nameplate

Each pressure vessel shall have a permanently attached nameplate showing:

a) the number, date and amendment (where applicable) of this specification, i.e. PD 5500:wxyz+An[5)] where wxyz is the year of publication of this issue of the specification and n is the amendment number;

b) the name of the manufacturer;

c) the manufacturer's serial number identifying the vessel;

d) the design pressure;

e) the design temperature;

f) the hydraulic or pneumatic test pressure;

g) the date of manufacture;

h) the identifying mark of the Inspecting Authority;

i) any statutory marking required.

A facsimile of this nameplate shall be prepared and submitted to the purchaser in accordance with **1.5.2.2**g).

5.8.10 Final inspection

5.8.10.1 Visual and dimensional inspection

The visual and dimensional inspection shall be performed following completion of all welding and post-weld heat treatments but before application of any coating, irrespective of type, and before the pressure test. If the vessel is to be partially or totally assembled on site, the manufacturer and Inspecting Authority shall determine any elements that can receive protective coating prior to dispatch to site. Adequate provisions shall be made to allow safe access to all areas of the vessel together with any necessary lighting, measurement equipment and dimensional aids in order to conduct these inspections.

The scope of the visual and dimensional inspection shall include, but not be limited to, the following:

a) conformity of construction with the approved vessel drawings including dimensional requirements specified on the drawings and in this specification;

b) the condition of the completed vessel with particular attention to finished weld joints, nozzle connections and attachments in respect of the weld profile and general weld geometry, as complying with the approved vessel drawings and this specification;

c) check of material markings for traceability of material against documented records;

d) check of welders and NDT identification on the vessel against documentation, if applicable.

Any remedial actions resulting from this inspection shall be carried out, re-inspected and cleared before the pressure test.

5.8.10.2 Review of documentation

The scope of the review of documentation shall include test certificates of the welding procedures approval, certificates of the welders approval, certificates for NDT personnel, production test reports, NDT reports and post-weld heat-treatment records.

The extent of the review and all deviations shall be reported and any remedial actions cleared with the Inspecting Authority.

[5] Marking PD 5500:wxyz+An on or in relation to a product represents a manufacturer's declaration of conformity, i.e. a claim by or on behalf of the manufacturer that the product meets the requirements of the specification. The accuracy of the claim is therefore solely the claimant's responsibility. Such a declaration is not to be confused with third party certification of conformity, which may also be desirable.

5.9 Inspection requirements for cast components

The following provisions satisfy the requirements of **3.4.2.3** for the detection and repair of defects in castings with a cast factor of 0.9.

5.9.1 Examination

For carbon, low alloy or high alloy steel castings in material groups 1 to 6, 8, 9 and 11 produced either by static or centrifugal casting, a casting factor of 0.9 can be used provided the castings are examined in accordance with a quality specification agreed between the manufacturer, purchaser and the Inspecting Authority.

NOTE A suitable specification could be based upon Appendix 7 of ASME VIII division 1.

5.9.2 Defects

Where defects are repaired by welding, the completed repair shall be subject to re-examination and such heat treatment as is agreed between purchaser, Inspecting Authority, manufacturer and material supplier.

5.9.3 Identification and marking

In addition to any manufacturer and material marking, castings shall be identified as having a casting factor of 0.9. It is recommended that these castings are painted a colour to differentiate them on the shop floor from castings of factor 0.7.

Annex A Requirements for design where loadings and components are not covered by Section 3

A.1 General

This annex gives design criteria for stress systems resulting from the application of loads and/or the use of components or types of component not covered explicitly by Section 3 (see **3.2.2**) or relevant annexes. The intention is to ensure that in such circumstances the design basis is consistent with that underlying the rules specified in Section 3. Formal analysis in accordance with this annex is only required in the case of significant additional loadings or loadings from components significantly different to those covered in Section 3. Relevant experience of similar designs may be considered in deciding whether an analysis is necessary.

A.2 Notation

For the purposes of this annex the following symbols apply.

e	is the analysis thickness of main vessel section (see **1.6**);
f	is the design stress;
	NOTE If it is required to evaluate the limits in this annex for category 3 components, the value of f may be taken as that permitted for components of categories 1 and 2 provided that there are no welded seams in the vicinity of the point under consideration.
f_b	is the primary bending stress intensity;
f_g	is the secondary stress intensity;
f_L	is the local primary membrane stress intensity;
f_m	is the general primary membrane stress intensity;
f_1, f_2 and f_3	are the principal stresses required to determine stress intensities;
R	is the mean radius of main vessel section;
$R_{e(T)}$	corresponds to the minimum value of R_{eL} or $R_{p0.2}$ ($R_{p1.0}$ for austenitic steels) specified for the grade of material concerned at a temperature T (tested in accordance with BS EN ISO 6892-2);
ν	is Poisson's ratio.

A.3 Non-creep conditions

The criteria in **A.3.1** to **A.3.6** apply for design temperatures at which the design strength is not time dependent (see **2.3.1.3**).

A.3.1 General criteria

A.3.1.1 Gross plastic deformation

There shall be the same theoretical margin against gross plastic deformation for all design details as that provided against gross plastic deformation in major membrane areas. For this purpose the required margin against gross plastic deformation shall be assumed to be $R_{e(T)}/f$ for materials covered in Table 2.1-2 or Table K.1-2 to Table K.1-12. For other materials the value for the nearest equivalent material covered in Table 2.1-2 or Table K.1-2 to Table K.1-12 shall be assumed.

In establishing conformity with this criterion investigations shall take account of plastic behaviour. If the theory of plastic limit analysis is employed, the limit load shall be taken as the load producing gross plastic deformation, although this may be a conservative estimate. It is also safe, though possibly conservative, to assume that a load, which does not change sign and which, on the basis of a shakedown analysis satisfies **A.3.1.2**, will be less or equal to the load for gross plastic deformation.

A list of references dealing with limit analysis of various configurations is given in **A.5**.

Where it is impracticable to perform plastic analysis, elastic analysis shall be employed as detailed in **A.3.2** (covering **A.3.3** and **A.3.4**) to demonstrate compliance with this criterion; alternatively strain measurements shall be made on the actual vessel during pressure and load tests.

A.3.1.2 Incremental collapse

The stress systems imposed should shakedown to elastic action within the first few operating cycles. The operating loads to be considered include pressure and all loadings of the type listed in **3.2.1** where relevant.

In demonstrating conformance with this criterion a shakedown analysis (e.g. see **G.2.6**) shall preferably be employed.

In cases where loads change sign during the cycle it should be demonstrated (e.g. see **A.3.4.2.4**) that the total range of maximum stress due to the range of loads does not exceed twice the yield stress of the material ($2 \times R_{e(T)}$).

Alternatively elastic analysis as detailed in **A.3.2** shall be employed (covering **A.3.3** and **A.3.4**).

A.3.1.3 Buckling

For components or loadings associated with substantial compressive stress buckling shall not occur under a combined load less than twice the design combined load at design temperature. Care shall also be taken to avoid buckling under test conditions. The design and test combined loads are to include pressure and simultaneous loadings of the type listed in **3.2.1** in conjunction with permissible fabrication imperfections. Where significant compressive stresses are present the possibility of buckling shall be investigated to satisfy this criterion and the design modified if necessary. While generally it is not possible to do this by elastic analysis a relevant criterion for cases where compressive stresses are due to highly localized loads is given in **A.3.3**.

For compressive general primary membrane stress, see **A.3.5**.

A.3.1.4 Fatigue

The need or otherwise for a fatigue analysis should be determined by application of Annex C.

A.3.2 Demonstration of design acceptability by stress analysis

Subclauses **A.3.3** and **A.3.4** give alternative criteria for demonstrating the acceptability of design on the basis of stresses estimated by elastic analysis. This elastic analysis shall take account of gross structural discontinuities (e.g. nozzles, changes in shell curvature), but not of local stress concentrations due to changes in profile such as fillet welds.

*NOTE 1 Design by elastic analysis can only provide an approximate solution for the fundamental design criteria given in **A.3.1**. Since the criteria in **A.3.1** have been set to ensure that results are always safe, some applications may give results that are conservative by a significant margin.*

The criteria in **A.3.3** provide stress limits for elastically calculated stresses adjacent to attachments and supports and to nozzles and openings that are subject to the combined effects of pressure and externally applied loads. It is recognised that for such geometries the estimation of primary stresses as required in **A.3.4** is particularly difficult and the concept is therefore avoided. Thermal stresses are not covered.

If the loaded nozzle area or opening is less than $2.5\sqrt{Re}$ from another stress concentrating feature, stresses as calculated in accordance with Annex G become unreliable and some other method of assessing the total stress is required, e.g. finite element stress analysis or proof test.

NOTE 2 The minimum distance between stress concentrating features should be measured from the junction with the shell not the toe of the weld.

The criteria in **A.3.4** are intended for general application to the geometries outside the scope of **A.3.3** and to geometries covered by **A.3.3** when thermal stresses are present (due for example to temperature gradients in the vicinity of a nozzle). In the latter case, **A.3.3** criteria apply to the non-thermal stresses but it is also necessary to apply the primary plus secondary stress category of **A.3.4.2.4** with the thermal stresses included.

A.3.3 Specific criteria for limited application

A.3.3.1 Attachments and supports

For these limits to apply the loaded area has to have a dimension in the circumferential direction not greater than one-third of the shell circumference. The stresses adjacent to the loaded area due to pressure acting in the shell shall be taken as the shell pressure stresses without any concentrating effects due to the attachment.

These limits apply to the maximum stress intensity adjacent to the attachment due to internal pressure and externally applied loads, obtained from **G.2.2** and **G.2.3** for cylindrical shells, or from **G.2.4** for spherical shells.

These limits may also be applied to stresses obtained from finite element stress analysis if the estimation of primary stresses is not possible.

Under the design combined load the following stress limits apply:

a) the membrane stress intensity shall not exceed $1.2f$;

b) the stress intensity due to the sum of membrane and bending stresses shall not exceed $2f$.

A.3.3.2 Nozzles and openings subject to pressure and externally applied loads

For these limits to apply the nozzle or opening has to be reinforced in accordance with **3.5.4**.

These limits apply to the maximum stress intensity adjacent to the nozzle or opening due to internal pressure and externally applied loads, obtained from **G.2.2** and **G.2.3** for cylindrical shells, or from **G.2.5** for spherical shells.

These limits may also be applied to stresses obtained from finite element stress analysis if the estimation of primary stresses is not possible.

Under the design combined load the stress intensity due to the sum of membrane and bending stresses shall not exceed $2.25f$.

A.3.3.3 Additional stress limits for attachments, supports, nozzles and openings

Where significant compressive membrane stresses are present the possibility of buckling shall be investigated and the design modified if necessary (see **A.3.1.3**). In cases where the external load is highly concentrated, an acceptable procedure is to limit the sum of membrane and bending stresses (total compressive stress) in any direction at the point to 0.9 of the minimum yield point of the material at design temperature. (See **2.3.2**, $R_{e(T)}$ for definition of minimum yield point).

Where shear stress is present alone, it shall not exceed 0.5f. The maximum permissible bearing stresses shall not exceed 1.5f.

A.3.4 Specific criteria for general application (except buckling)

The rules of **A.3.4** require the calculated stresses to be grouped into five stress categories (see **A.3.4.2**) and appropriate stress intensities f_m, f_L, f_b and f_g to be determined from the principal stressed f_1, f_2 and f_3 in each category, using the maximum shear theory of failure. Appropriate limits are given for the stress intensities so calculated.

A.3.4.1 Terminology

A.3.4.1.1 Stress intensity

The stress intensity is twice the maximum shear stress, i.e. the difference between the algebraically largest principal stress and the algebraically smallest principal stress at a given point. Tension stresses are considered positive and compression stresses are considered negative.

A.3.4.1.2 Gross structural discontinuity

A gross structural discontinuity is a source of stress or strain intensification that affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole.

Examples of gross structural discontinuities are end to shell and flange to shell junctions, nozzles and junctions between shells of different diameters or thicknesses.

A.3.4.1.3 Local structural discontinuity

A local structural discontinuity is a source of stress or strain intensification that affects a relatively small volume of material and does not have a significant effect on the overall stress or strain pattern or on the structure as a whole.

Examples of local structural discontinuities are small fillet radii, small attachments and partial penetration welds.

A.3.4.1.4 Normal stress

The normal stress is the component of stress normal to the plane of reference; this is also referred to as direct stress.

Usually the distribution of normal stress is not uniform through the thickness of a part, so this stress is considered to be made up in turn of two components one of which is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration, and the other of which varies with the location across the thickness.

A.3.4.1.5 Shear stress

The shear stress is the component of stress acting in the plane of reference.

A.3.4.1.6 Membrane stress

The membrane stress is the component of stress that is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.

A.3.4.1.7 Bending stress

The bending stress is the component of stress that is proportional to the distance from the centre of the wall thickness.

A.3.4.1.8 Primary stress

A primary stress is a stress produced by mechanical loadings only and so distributed in the structure that no redistribution of load occurs as a result of yielding. It is a normal stress, or a shear stress developed by the imposed loading, that is necessary to satisfy the simple laws of equilibrium of external and internal forces and moments. The basic characteristic of this stress is that it is not self-limiting. Primary stresses that considerably exceed the yield strength will result in failure, or at least in gross distortion. A thermal stress is not classified as a primary stress. Primary stress is divided into "general" and "local" categories. The local primary stress is defined in **A.3.4.1.9**.

Examples of general primary stress are:

a) the stress in a circular, cylindrical or spherical shell due to internal pressure or to distributed live loads;

b) the bending stress in the central portion of a flat end due to pressure.

A.3.4.1.9 Primary local membrane stress

Cases arise in which a membrane stress produced by pressure or other mechanical loading and associated with a primary and/or a discontinuity effect produces excessive distortion in the transfer load to other portions of the structure. Conservatism requires that such a stress be classified as a primary local membrane stress even though it has some characteristics of a secondary stress. A stressed region may be considered as local if the distance over which the stress intensity exceeds $1.1f$ does not extend in the meridional direction more than $0.5\sqrt{Re}$, and if it is not closer in the meridional direction than $2.5\sqrt{Re}$ to another region where the limits of general primary membrane stress are exceeded.

An example of a primary local stress is the membrane stress in a shell produced by external load and moment at a permanent support or at a nozzle connection.

A.3.4.1.10 Secondary stress

A secondary stress is a normal stress or a shear stress developed by the constraint of adjacent parts or by self-constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions that cause the stress to occur, and failure from one application of the stress is not to be expected.

An example of secondary stress is the bending stress at a gross structural discontinuity.

A.3.4.1.11 Peak stress

The basic characteristic of a peak stress is that it does not cause any noticeable distortion and is objectionable only as a possible source of a fatigue crack or a brittle fracture. A stress that is not highly localized falls into this category if it is of a type that cannot cause noticeable distortion.

Examples of peak stress are:

a) the thermal stresses in the austenitic steel cladding of a carbon steel vessel;

b) the surface stresses in the wall of a vessel or pipe produced by thermal shock;

c) the stress at a local structural discontinuity.

The current methodology of design against fatigue failure given in Annex C does not require a peak stress to be considered. Where alternative methods of fatigue assessment are used it may be necessary to consider peak stress.

A.3.4.2 Stress categories and stress limits

A calculated stress depending upon the type of loading and/or the distribution of such stress will fall within one of the five basic stress categories defined in **A.3.4.2.1**, **A.3.4.2.2**, **A.3.4.2.3**, **A.3.4.2.4** and **A.3.4.2.5**. For each category, a stress intensity value is derived for a specific condition of design. To satisfy the analysis this stress intensity shall fall within the limit detailed for each category.

A.3.4.2.1 General primary membrane stress category

The stresses falling within the general primary membrane stress category are those defined as general primary stresses in **A.3.4.1.8** and are produced by pressure and other mechanical loads, but excluding all secondary and peak stresses. The value of the membrane stress intensity is obtained by averaging these stresses across the thickness of the section under consideration. The limiting value of this intensity f_m is the allowable stress value f except as permitted in this annex.

A.3.4.2.2 Local primary membrane stress category

The stresses falling within the local primary membrane stress category are those defined in **A.3.4.1.8** and are produced by pressure and other mechanical loads, but excluding all thermal and peak stresses. The stress intensity f_L is the average value of these stresses across the thickness of the section under consideration and is limited to $1.5f$. By definition, the local primary membrane stress category includes f_m in those cases where it is present.

A.3.4.2.3 General or local primary membrane plus primary bending stress category

The stresses falling within the general or local primary membrane plus primary bending stress category are those defined in **A.3.4.1.8**, but the stress intensity value f_b, $(f_m + f_b)$ or $(f_L + f_b)$ is the highest value of those stresses acting across the section under consideration excluding secondary and peak stresses. f_b is the primary bending stress intensity, which means the component of primary stress proportional to the distance from centroid of solid section. The stress intensity f_b, $(f_m + f_b)$ or $(f_L + f_b)$ is not to exceed $1.5f$.

A.3.4.2.4 Primary plus secondary stress category

The stresses falling within the primary plus secondary stress category are those defined in **A.3.4.1.8**, plus those of **A.3.4.1.10**, produced by pressure, other mechanical loads and general thermal effects. The effects of gross structural discontinuities, but not of local structural discontinuities (stress concentrations), shall be included. The stress intensity value $(f_m + f_b + f_g)$ or $(f_L + f_b + f_g)$ is the highest value of these stresses acting across the section under consideration and is to be limited to $3.0f$ (see also Note 1 to Figure A.1).

Figure A.1 and Table A.1 have been included to guide the designer in establishing stress categories for some typical cases and stress intensity limits for combinations of stress categories. There will be instances when reference to definitions of stresses will be necessary to classify a specific stress condition to a stress category. **A.3.4.2.5** explains the reason for separating them into two categories “general” and “secondary” in the case of thermal stresses.

Figure A.1 **Stress categories and limits of stress intensity**

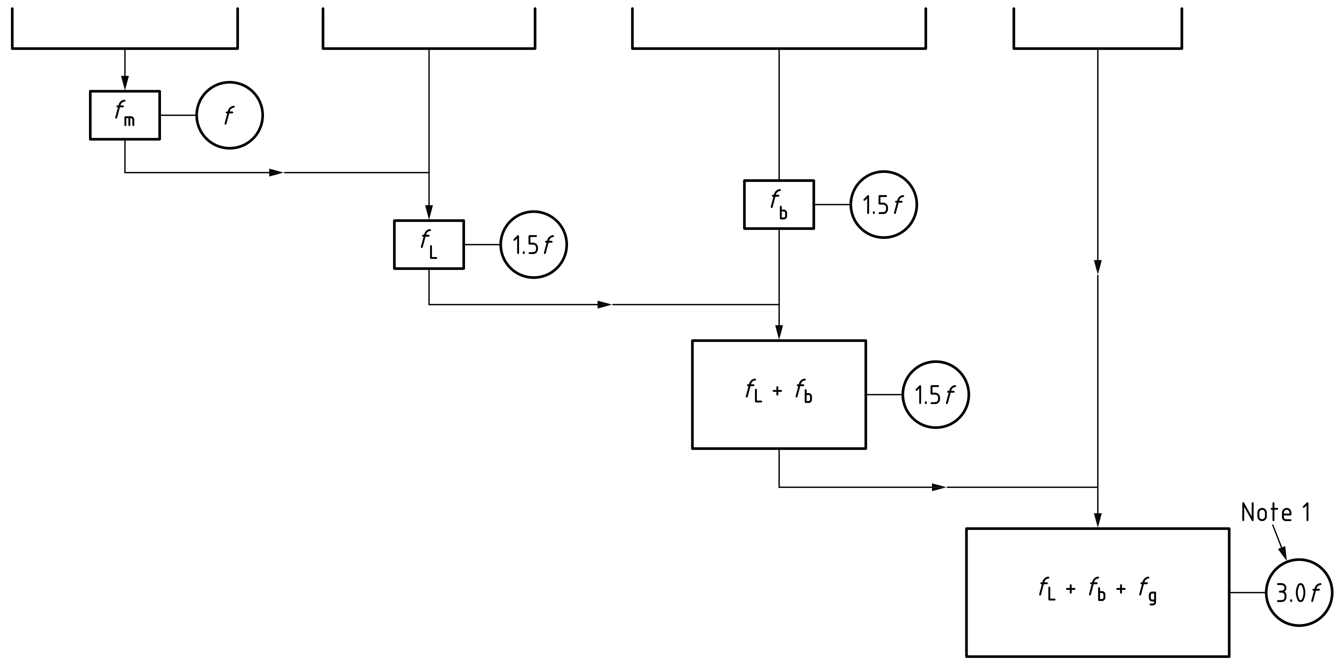
Stress category	**Primary**			**Secondary**
	General	**Local membrane**	**Bending**	
Description (for examples see Table A.1)	Average primary stress across solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Average primary stress across any solid section. Considers discontinuities but not concentrations. Produced only by mechanical loads, by definition includes f_m in those cases where it is present.	Component of primary stress proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by mechanical loads.	Self-equilibrating stress necessary to satisfy continuity of structure. Occurs at structural discontinuities. Can be caused by mechanical load or by differential thermal expansion. Excludes local stress concentrations (see Note 2).
Symbol (see Note 3)	f_m	f_L	f_b	f_g
Combination of stress components and allowable limits of stress intensities				

Figure A.1 **Stress categories and limits of stress intensity** *(continued)*

NOTE This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analysed, the value of f is to be taken as the average of the f values for the highest and the lowest temperature of the metal during the transient. When part or all of the secondary stress is due to mechanical load, the value of f is to be taken as the f value for the highest temperature of the metal during the transient.

NOTE 2 The stresses in category f_g are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc., and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of f_m (or f_L) + f_b + f_g and not f_g alone.

NOTE 3 The symbols f_m, f_L, f_b and f_g do not represent single quantities but rather sets of six quantities representing the six stress components.

Table A.1 **Classification of stresses for some typical cases**

Vessel component	Location	Origin of stress	Type of stress	Classification
Cylindrical or spherical shell	Shell plate remote from discontinuities	Internal pressure	General membrane Gradient through plate thickness	f_m f_g
		Axial thermal gradient	Membrane Bending	f_g f_g
	Junction with end or flange	Internal pressure	Membrane Bending	f_L f_g
Any shell or end	Any section across entire vessel	External load or moment, or internal pressure	General membrane averaged across full section. Stress component perpendicular to cross-section	f_m
		External load or moment	Bending across full section. Stress component perpendicular to cross-section	f_m
	Near nozzle or other opening	External load or moment, or internal pressure	Local membrane Bending Peak (fillet or corner)	f_L f_g [a]
	Any location	Temperature difference between shell and end	Membrane Bending	f_g f_g
Dished end or conical end	Crown	Internal pressure	Membrane Bending	f_m f_b
	Knuckle or junction to shell	Internal pressure	Membrane Bending	f_L[b] f_g
Flat end	Centre region	Internal pressure	Membrane Bending	f_m f_b
	Junction to shell	Internal pressure	Membrane Bending	f_L f_g
Perforated end or shell	Typical ligament in a uniform pattern	Pressure	Membrane (average through cross-section) Bending (average through width of ligament, but gradient through plate) Peak	f_m f_b [a]
	Isolated or a typical ligament	Pressure	Membrane Bending Peak	f_g f_g [a]

Table A.1 **Classification of stresses for some typical cases** *(continued)*

Vessel component	Location	Origin of stress	Type of stress	Classification
Nozzle	Cross-section perpendicular to nozzle axis	Internal pressure or external load or moment	General membrane (average across full section). Stress component perpendicular to section	f_m
		External load or moment	Bending across nozzle section	f_m
	Nozzle wall	Internal pressure	General membrane Local membrane Bending Peak	f_m f_L f_g [a]
		Differential expansion	Membrane Bending Peak	f_g f_g [a]
Cladding	Any	Differential expansion	Membrane (peak) Bending (peak)	[a] [a]
Any	Any	Thermal gradient through plate thickness	Bending (component of stress proportional to distance from centre) Peak (component of stress departing from linear component f_g)	f_g [a]
Any	Any	Any	Stress concentration-peak (notch effect)	[a]

[a] Refer to **A.3.4.1.11**.

[b] Consideration should also be given to the possibility of buckling and excessive deformation in vessels with large diameter-to-thickness ratio.

A.3.4.2.5 Thermal stress

Thermal stress is a self-balancing stress produced by a non-uniform distribution of temperature or by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature.

For the purpose of establishing allowable stresses, the following two types of thermal stress are recognized, depending on the volume or area in which distortion takes place.

a) General thermal stress is associated with distortion of the structure in which it occurs. If a stress of this type, neglecting stress concentrations, exceeds twice the yield strength of the material, the elastic analysis may be invalid and successive thermal cycles may produce incremental distortion. This type is therefore classified as secondary stress in Table A.1 and Figure A.1.

   Examples of general thermal stress are:

   1) the stress produced by an axial thermal gradient in a cylindrical shell;

   2) the stress produced by the temperature difference between a nozzle and the shell to which it is attached.

b) Local thermal stress is associated with almost complete suppression of the differential expansion and thus produces no significant distortion. Such stresses should be considered only from the fatigue standpoint.

Examples of local thermal stresses are:

1) the stress in a small hot spot in a vessel wall;

2) the thermal stress in a cladding material which has a coefficient of expansion different from that of the base metal.

A.3.5 Limit for longitudinal compressive general membrane stress in a vessel

A.3.5.1 Cylindrical sections

A.3.5.1.1 General

It is permitted to use either **A.3.5.1.2** or **A.3.5.1.3** to calculate the permissible longitudinal compressive general membrane stress in a cylindrical section subject to longitudinal compression with or without simultaneous external pressure.

A.3.5.1.2 Limits based upon 3.6

The permissible longitudinal compressive stress at the design temperature $\sigma_{z,allow}$ in the cylindrical section shall be calculated as follows:

$$K = p_e / p_{yss} \tag{A-1}$$

where p_e and p_{yss} are defined in Equations (3.6.4-2) and (3.6.4-1) respectively, in **3.6.4**.

If the design temperature is not greater than 350 °C Δ is obtained from Figure A.2 in terms of K. If the design temperature is greater than 350 °C the use of Figure A.2 is not permitted and Δ is obtained from Figure 3.6-4 curve b), or the alternative method in **A.3.5.1.3** may be used.

$$\sigma_{z,allow} = \Delta sf \tag{A-2}$$

where s and f are defined in **3.6.1.1**.

For load cases LC9, LC11, LC12 and LC13 in Table 3.2-1 the permissible longitudinal compressive stress at ambient temperature $\sigma_{z,allow,a}$ shall be calculated using values of f, p_e and p_{yss} at ambient temperature.

For load case LC10 in Table 3.2-1 the permissible longitudinal compressive stress at the test temperature $\sigma_{z,allow,test}$ shall be calculated using values of f, p_e and p_{yss} at the test temperature.

When the vessel is not under external pressure the longitudinal compressive stress σ_z in the cylindrical section, as defined in **B.3.3**, shall not exceed $\sigma_{z,allow}$, $\sigma_{z,allow,a}$ or $\sigma_{z,allow,test}$, as specified in Table 3.2-1 for the load case under consideration.

NOTE 1 For vessels that are not under external pressure the maximum longitudinal compressive stress will generally occur when the internal pressure is zero.

When the vessel is subject to simultaneous longitudinal compression and external pressure, the following interaction formula shall be used to calculate design values of longitudinal compressive stress and external pressure in the cylindrical section.

$$\frac{\sigma_z}{\sigma_{z,allow}} + \frac{p_{ex}}{p_{ex,allow}} \leq 1 \tag{A-3}$$

where:

σ_z is the longitudinal compressive stress as defined in **B.3.3**, excluding the longitudinal component of the stress due to external pressure;

$\sigma_{z,allow}$ is the permissible longitudinal compressive stress as calculated in Equation (A-2);

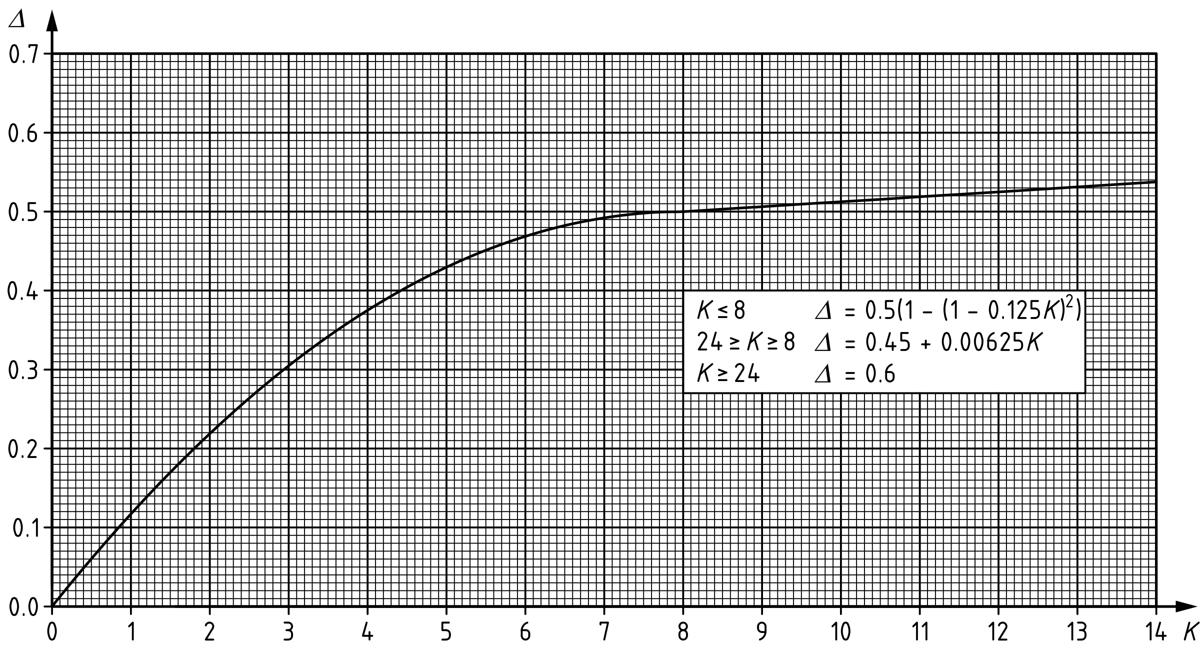
p_{ex} is the design external pressure;

$p_{ex,allow}$ is the allowable external pressure derived from **3.6.2.1**.

NOTE 2 Where the simultaneous longitudinal compression and external pressure result from wind loading and vacuum, and where a low frequency of vacuum occurrence can be demonstrated, the wind loading may be based upon a 2 year figure rather than a 50 year figure.

*NOTE 3 The value of Δ used to calculate the permissible longitudinal compressive stress $\sigma_{z,allow}$ is not the same as the value of Δ obtained from Figure 3.6-4 curve a) which is used in **3.6.2.1** to calculate the allowable external pressure $p_{ex,allow}$.*

Figure A.2 **Curve for the evaluation of Δ**



A.3.5.1.3 Alternative limits based upon ECCS[1)] recommendations

The rules in **A.3.5.1.3** are based on the 4th Edition of the ECCS[1)] recommendations.

NOTE An alternative procedure based on the requirements of EN 1993-1-6 is given in Enquiry Case 5500/141.

These rules are dependent upon the cylinder meeting the following tolerances, which are checked using three templates.

a) A straight rod of length $4\sqrt{Re}$ (but not longer than 95% of the distance between any circumferential and meridional welds).

b) A circular template of the same length as above, bent to the circular shape of the mean cylindrical outer surface.

c) A straight rod of length 25*e*.

[1)] ECCS European Recommendations Buckling of steel Shells 4th Edition, 1988, No. 56, ECCS (European Convention for Constructional Steelwork) Technical Working Group 8.4, Stability of shells.

The three sections, a), b) and c), of Figure A.3 show the use of three templates. Templates a) and b) are held, respectively against any meridian and against any parallel circle, anywhere between welds. Template c) is held across any circular weld.

If all the *w/l* values are less than 0.01 then:

$$\alpha = \frac{0.83}{\sqrt{1.0 + 0.01R/e}} \text{ if } R/e < 212 \quad \text{(A-4)}$$

$$\alpha = \frac{0.7}{\sqrt{0.1 + 0.01R/e}} \text{ if } R/e \geq 212 \quad \text{(A-5)}$$

If any *w/l* values are in the range 0.01 to 0.02 then:

$$\alpha = \frac{0.83(1.5 - 50w/l)}{\sqrt{1.0 + 0.01R/e}} \text{ if } R/e < 212 \quad \text{(A-6)}$$

$$\alpha = \frac{0.7(1.5 - 50w/l)}{\sqrt{0.1 + 0.01R/e}} \text{ if } R/e \geq 212 \quad \text{(A-7)}$$

where w and l are dimensions as shown in Figure A.3.

If any *w/l* values are greater than 0.02 then the rules of **A.3.5.1.3** cannot be used.

$$\Delta = 0.5\alpha K \text{ if } \alpha K < 0.5 \quad \text{(A-8)}$$

$$\Delta = 0.6667 - 0.2749(\alpha K)^{-0.6} \text{ if } \alpha K \geq 0.5 \quad \text{(A-9)}$$

where K is obtained from Equation (A-1).

The permissible longitudinal compressive stress in the cylindrical section at the design temperature $\sigma_{z,allow}$ shall be calculated from Equation (A-2) using Δ obtained from Equation (A-8) or (A-9).

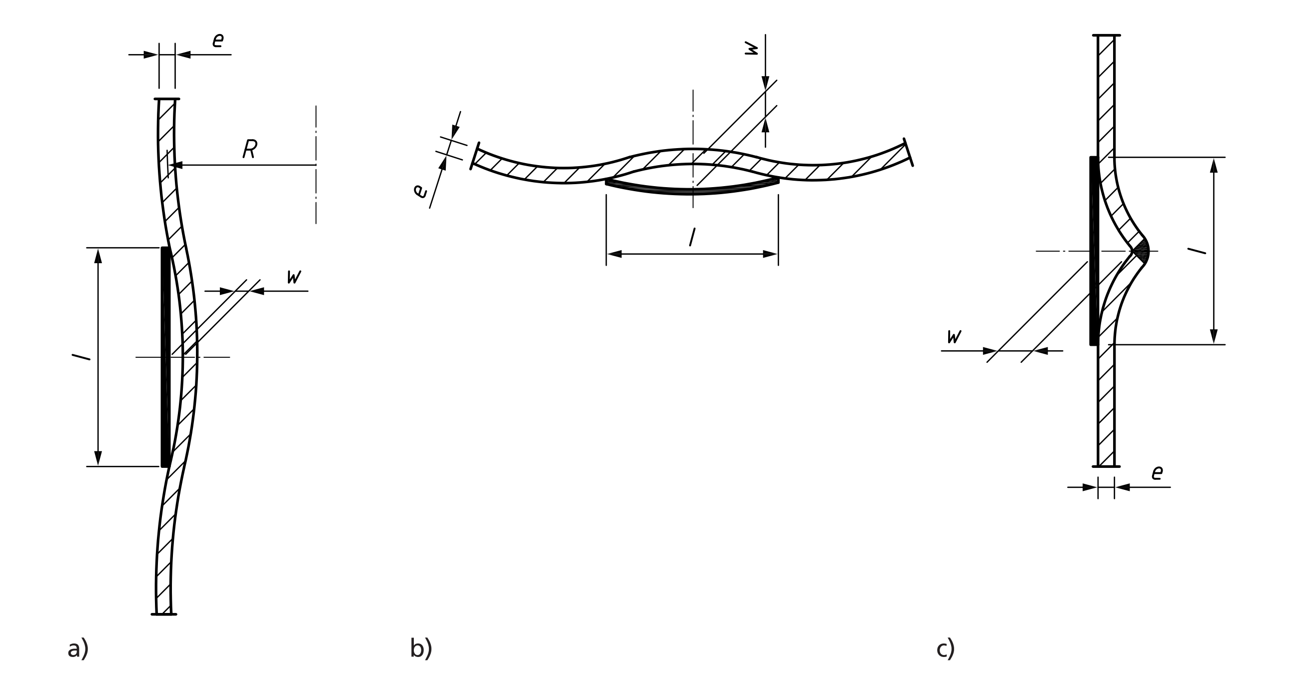
For load cases LC9, LC11, LC12 and LC13 in Table 3.2-1 the permissible longitudinal compressive stress at ambient temperature $\sigma_{z,allow,a}$ shall be calculated using values of f, p_e and p_{yss} at ambient temperature.

For load case LC10 in Table 3.2-1 the permissible longitudinal compressive stress at the test temperature $\sigma_{z,allow,test}$ shall be calculated using values of f, p_e and p_{yss} at the test temperature.

When the vessel is not under external pressure the longitudinal compressive stress σ_z in the cylindrical section, as defined in **B.3.3**, shall not exceed $\sigma_{z,allow}$, $\sigma_{z,allow,a}$ or $\sigma_{z,allow,test}$, as specified in Table 3.2-1 for the load case under consideration.

When the vessel is subject to simultaneous longitudinal compression and external pressure, the interaction formula in Equation (A-3) shall be used to calculate design values of longitudinal compressive stress and external pressure in the cylindrical section.

Figure A.3 **Use of templates to check tolerances (see A.3.5.1.3)**



A.3.5.2 Conical sections

For conical sections the limits for longitudinal compressive general membrane stress in a cylinder given in **A.3.5.1** shall apply with the following modifications. Cylinder radius R in Equations (3.6.4-1) and (3.6.4-2) shall be replaced by $R_c/\cos\theta$.

where:

R_c is the local mean radius of the cone at the point under consideration;

θ is the semi angle of the cone as defined in **3.6.1.1**.

additionally

$p_{ex,\ allow}$ is the allowable external pressure of the cone derived from **3.6.3.1**.

NOTE The requirement for the conical section applies along the full length of the cone but normally the worst case will be at the small end even though the external pressure calculation is based on the large end of the cone.

A.3.6 Wind and earthquake conditions

All allowable tensile stresses and stress intensities (membrane or bending, primary or secondary) may be increased by a factor of 1.2 when wind and earthquake loadings are calculated in accordance with **B.5** and **B.6** and are combined with other loadings in accordance with Table 3.2-1; wind and earthquake loadings need not be assumed to act simultaneously. Limitations on compressive stresses in **A.3.1.3**, **A.3.3.3** and **A.3.5** are not hereby relaxed.

A.4 Creep conditions

In the absence of comprehensive design criteria for components in the creep range, the requirements specified in Section 3 should be applied, but see Note to 3.2.4.

A.5 Bibliography

ERBATUR, F.H., KIRK, A. and GILL, S.S. Plastic behaviour of oblique flush nozzles in cylindrical pressure vessels: an experimental investigation. *Pressure Vessels and Piping*, 1973, **1**, 93-118.

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DINNO, K.S. and GILL, S.S. A method for calculating the lower bound limit pressure for thick shells of revolution with specific reference to cylindrical vessels with torispherical ends. *International Journal of Mech. Sciences*, 1974, **16**, 415-427.

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DINNO, K.S. and GILL, S.S. A lower bound limit pressure analysis for the oblique intersection of a flush cylindrical nozzle and the torus of a cylindrical vessel with a torispherical end. *Journal of Strain Analysis*, 1974, **9** (No. 4), 247-262.

Annex B Requirements for cylindrical, spherical and conical shells under combined loadings, including wind and earthquakes

B.1 General

B.1.1 This annex deals with the determination and maximum permitted values of general primary membrane stress intensity for cylindrical, spherical and conical shells (excluding very flattened cones) subject to combinations of loads in addition to internal pressure.

The loadings considered are a bending moment acting in a plane containing the shell axis (e.g. due to wind loading in the case of a vertical vessel or weight loading in a horizontal vessel), an axial force (e.g. due to weight in a vertical vessel) and a torque about the vessel axis (e.g. form offset piping and wind loads).

The relaxations permitted for wind and earthquake loadings under **A.3.6** (and **3.2.7**) do not apply to the compressive stress limits given in **A.3.5**. The latter limits are also applicable to vessels under external pressure.

B.1.2 The limits given in **B.4** are applicable to regions remote[1)] from shell discontinuities such as changes in curvature, openings, stiffeners, etc., and remote from the points of application of the additional loads (e.g. supports). For the treatment of the stresses local to points of application of load and shell discontinuities under combined loads see **A.3.3** and **A.3.4**.

A method to design for global loads at cylinder to cone joints of the types shown in Figure 3.5-5 and Figure 3.5-6 is given in **B.7**.

B.1.3 The general approach is that, with wind and earthquake loads excluded, the stress intensity according to the maximum shear stress criterion shall nowhere exceed the design stress. For this purpose the compressive stress in the thickness direction (radial stress) is assumed to be 0.5p. An increased level is permitted when wind and earthquake loads are included. Limits for compressive stresses are also included to guard against buckling.

B.1.4 The shell thickness shall never be less than that required for internal pressure in **3.5.1.2**a), **3.5.1.2**b) and **3.5.3.3** for cylindrical, spherical and conical shells respectively.

B.1.5 It is not possible to give explicit equations for thickness under combined loading and a solution by trial and error is necessary. Moreover, it is necessary to determine the location of the maximum equivalent membrane stress and, if buckling is a possibility, the location (which may not be coincident) of the region of maximum buckling hazard.

B.1.6 The calculation shall be performed for the combinations of load expected in service (see Table 3.2-1). The thickness may be dictated by loads acting when the vessel is not under pressure. Conditions during pressure test shall be the subject of special consideration.

B.2 Notation

For the purpose of this annex the following symbols apply. All dimensions are in the corroded condition, unless otherwise indicated (see **3.1.5**):

D is the mean diameter of spherical or cylindrical section of shell;

[1)] Remote means "at distance not less than $\sqrt{Re}$".

e is the shell analysis thickness (see **1.6**);

f is the design stress;

f_m is the general primary membrane stress intensity [see Equation (B-12)]

f_1 f_2 are the principal stresses in a plane tangential to shell surface, positive if tensile [see Equations (B-1) and (B-2)];

M is the bending moment on shell acting in a plane containing shell axis, at transverse section considered;

p is the design pressure, defined in **3.2.3**. For this Annex $p > 0$ for internal pressure and $p < 0$ for external pressure. For the calculation of the meridional stress in spheres, and the longitudinal stress in vertical cylindrical or conical shells, p excludes the static head of liquid;

R_i is the inside radius of shell (for conical shell, inside radius measured normal to axis of shell at the transverse section considered);

T is the torque acting about shell axis at transverse section;

W is the axial force on shell (positive if tensile) at transverse section considered (this force excludes pressure load);

α is the semi-apex angle of conical shell;

φ is the angle included by normal to shell at transverse section considered and shell axis (spherical shell only);

θ is the angle included by plane of action of moment M and an axial plane through the point considered (spherical and conical shells only);

σ_z is the meridional stress (longitudinal in a cylindrical shell), positive if tensile;

σ_θ is the circumferential stress, positive if tensile;

τ is the shear stress.

B.3 Equations for principal stresses and components thereof

B.3.1 General

For vertical vessels the axial force W includes dead weight loads, live loads and additional forces such as wind, earthquake, wave motion, blast, transport and lifting, as specified in Table 3.2-1 for the load case under consideration. The force W includes loads acting above (or below) the point under consideration, depending on whether the vessel support is below (or above) that point.

For horizontal vessels the force W includes the effect of additional forces such as wind, earthquake, wave motion, blast, transport and lifting, as specified in Table 3.2-1 for the load case under consideration.

For vertical vessels the moment M includes the effect of forces such as wind, earthquake, wave motion, blast, transport and lifting, as specified in Table 3.2-1 for the load case under consideration.

For horizontal vessels the moment M includes the effects of dead loads, live loads and additional forces such as wind, earthquake, wave motion, blast, transport and lifting, as specified in Table 3.2-1 for the load case under consideration.

Definitions and explanations of dead loads, live loads, wind loads, earthquake loads, and additional forces are given in **3.2.1.3**.

B.3.2 Principal stresses

The principal stresses f_1 and f_2, acting tangentially to the shell surface at the point under consideration, should be calculated from the following equations:

$$f_1 = 0.5\left[\sigma_\theta + \sigma_z + \sqrt{(\sigma_\theta - \sigma_z)^2 + 4\tau^2}\right] \quad \text{(B-1)}$$

$$f_2 = 0.5\left[\sigma_\theta + \sigma_z - \sqrt{(\sigma_\theta - \sigma_z)^2 + 4\tau^2}\right] \quad \text{(B-2)}$$

NOTE In these equations σ_θ and σ_z should be substituted with correct signs.

B.3.3 Stress components

The stress components σ_θ, σ_z and τ shall be calculated from the following equations.

a) *Cylindrical shell* (see Figure B.1)

$$\sigma_\theta = \frac{pR_i}{e} \quad \text{(B-3)}$$

$$\sigma_z = \frac{pR_i^2}{(2R_i + e)e} + \frac{W}{\pi(2R_i + e)e} \pm \frac{4M}{\pi(2R_i + e)^2 e} \quad \text{(B-4)}$$

$$\tau = \frac{2T}{\pi(2R_i + e)^2 e} \quad \text{(B-5)}$$

NOTE 1 The positive directions of W and M are shown in Figure B.1.

NOTE 2 The positive and negative signs before the term containing M refer to points A and B (see Figure B.1) respectively.

NOTE 3 The direction of T and sign of shear stress τ are immaterial.

NOTE 4 All stress components should be calculated for points A and B.

b) *Spherical shell* (see Figure B.2)

$$\sigma_\theta = \frac{pR_i^2}{(2R_i + e)e} - \left(\frac{W}{\pi(2R_i + e)e} \times \frac{1}{\sin^2\varphi}\right) - \left(\frac{4M}{\pi(2R_i + e)^2 e} \times \frac{1}{\sin^3\varphi} \times \cos\theta\right) \quad \text{(B-6)}$$

$$\sigma z = \frac{pR_i^2}{(2R_i + e)e} + \left(\frac{W}{\pi(2R_i + e)e} \times \frac{1}{\sin^2\varphi}\right) + \left(\frac{4M}{\pi(2R_i + e)^2 e} \times \frac{1}{\sin^3\varphi} \times \cos\theta\right) \quad \text{(B-7)}$$

$$\tau = \left(\frac{4M}{\pi(2R_i + e)^2 e} \times \frac{\cos\varphi}{sin^3\varphi} \times \sin\theta\right) + \left(\frac{2T}{\pi(2R_i + e)^2 e} \times \frac{1}{\sin^2\varphi}\right) \quad \text{(B-8)}$$

NOTE 5 The positive directions of M and W are shown in Figure B.2.

NOTE 6 Note that θ is measured from the point where M induces maximum meridional tension.

NOTE 7 The two components of shear stress should be treated as positive and additive, irrespective of the direction of T.

NOTE 8 All stress components should be evaluated at points in the range θ = 0 to θ = 180°.

c) *Conical shell* (see Figure B.3)

$$\sigma_\theta = \frac{pR_i}{e} \times \frac{1}{\cos\alpha} \quad \text{(B-9)}$$

$$\sigma_z = \left(\frac{pR_i^2}{(2R_i + e)e} \times \frac{1}{\cos\alpha}\right) + \left(\frac{W}{\pi(2R_i + e)e} \times \frac{1}{\cos\alpha}\right) +$$

$$\left(\frac{4M}{\pi(2R_{\mathrm{i}}+e)^2e} \times \frac{1}{\cos\alpha} \times \cos\theta\right) \tag{B-10}$$

$$\tau = \left(\frac{4M}{\pi(2R_{\mathrm{i}}+e)^2e} \times \tan\alpha\ \sin\theta\right) + \frac{2T}{\pi(2R_{\mathrm{i}}+e)^2e} \tag{B-11}$$

NOTE 9 The Notes to a) and b) apply, but reference should also be made to Figure B.3.

Figure B.1 **Stresses in a cylindrical shell under combined loading**

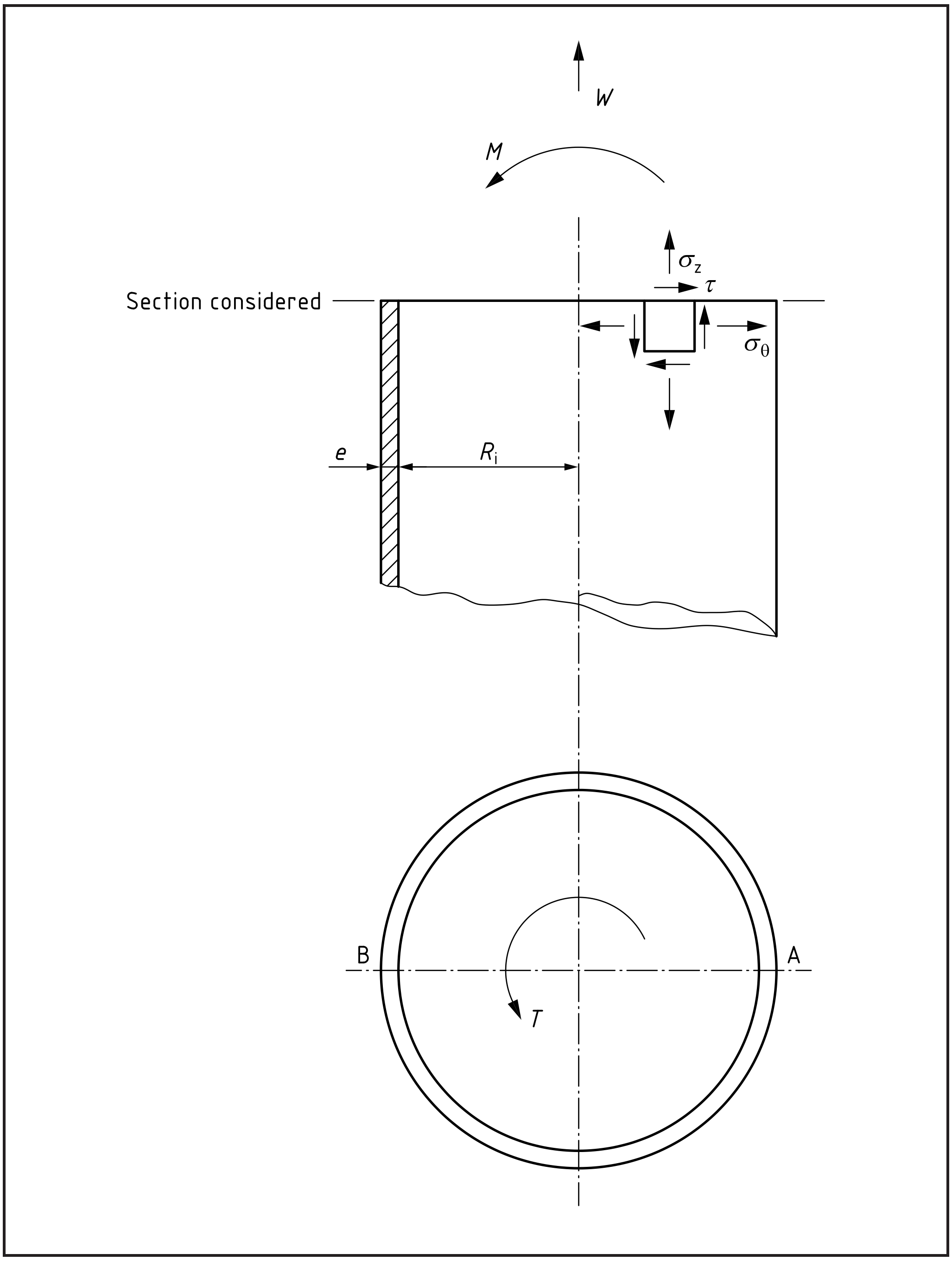


Figure B.2 **Stresses in a spherical shell under combined loading**

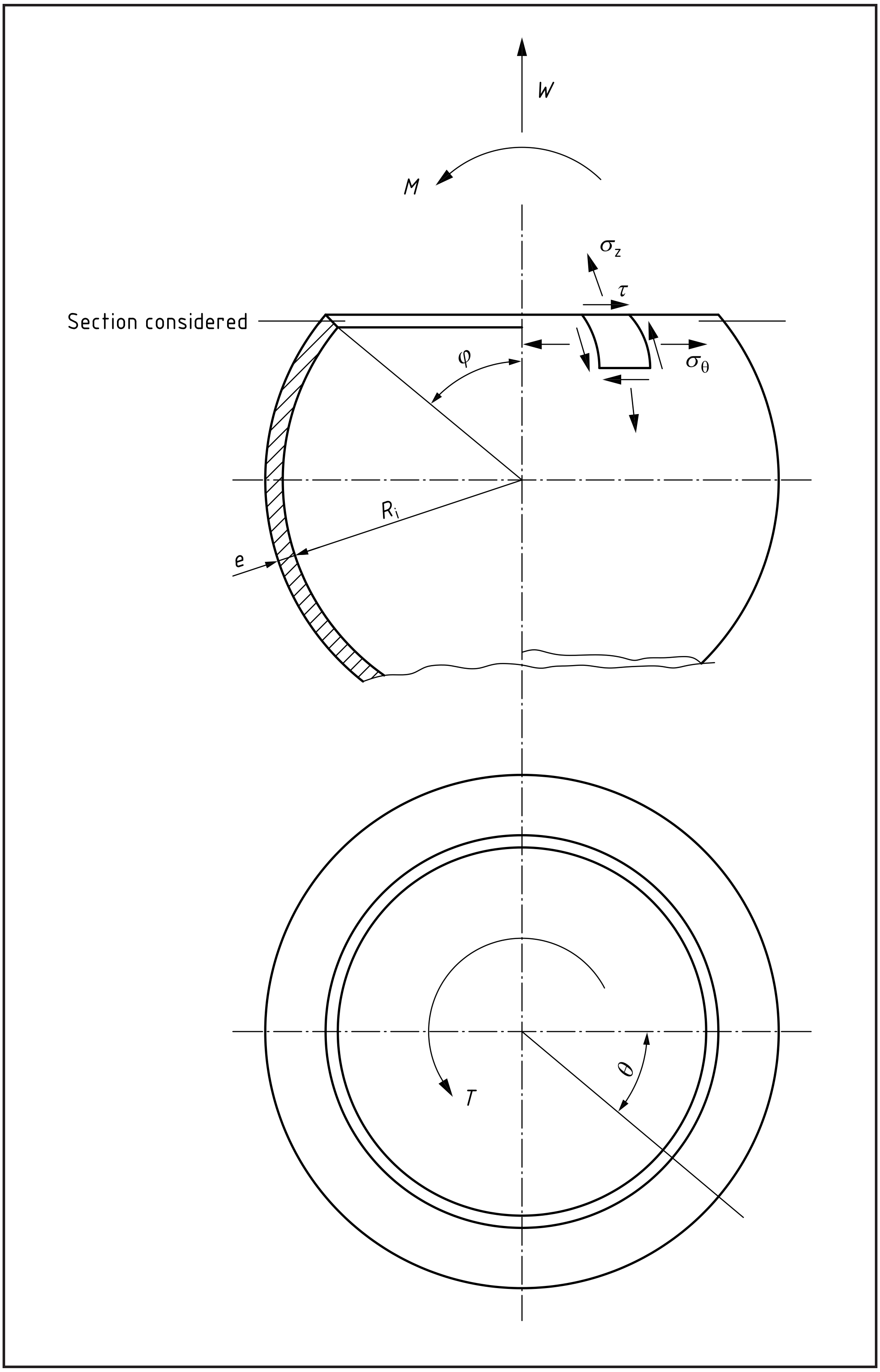
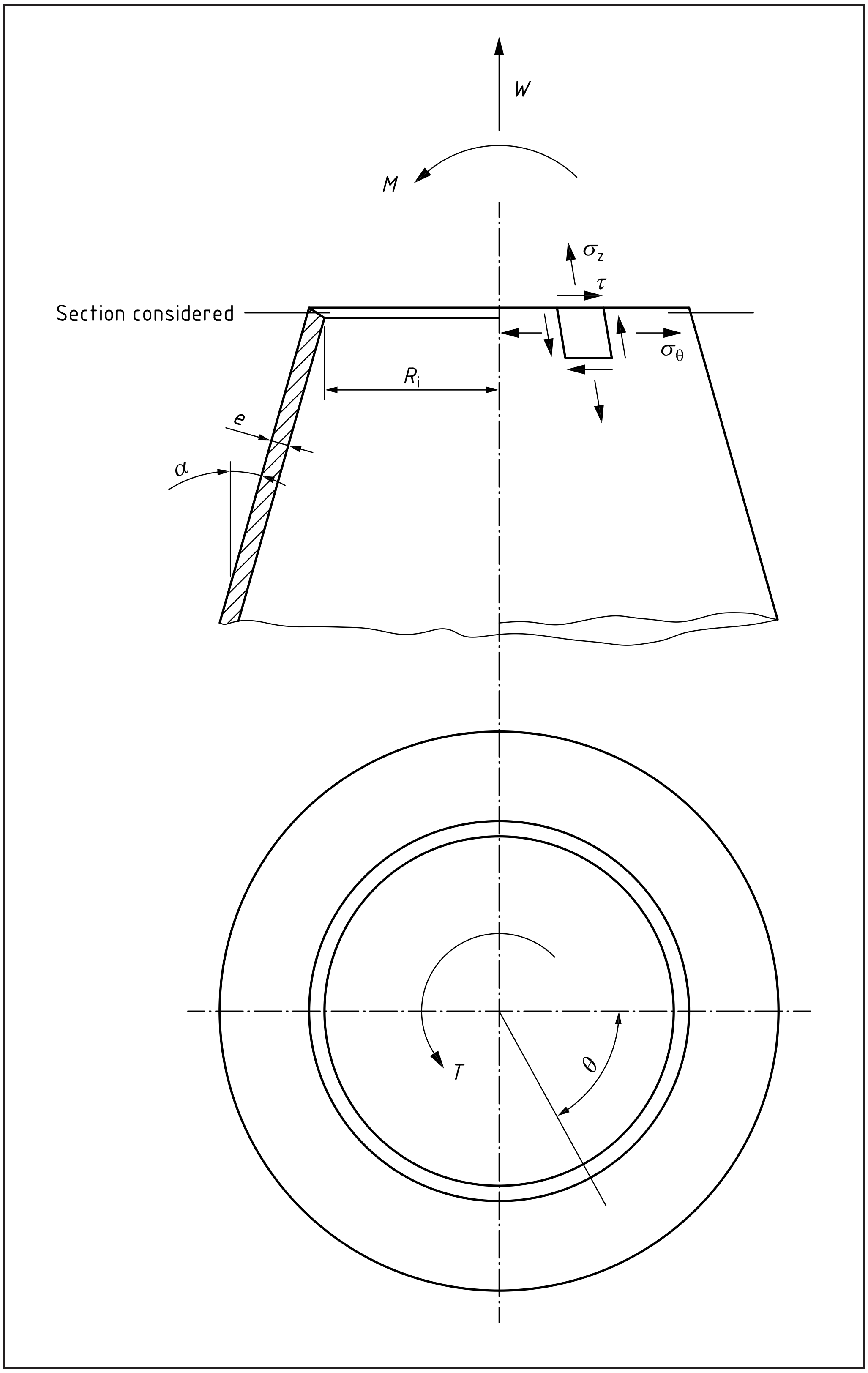


Figure B.3 **Stresses in a conical shell under combined loading**



B.4 General primary membrane stress intensity

$$f_m = \max(|f_1 - f_2|; |f_1 + 0.5p|; |f_2 + 0.5p|) \quad \text{(B-12)}$$

NOTE 1 In this equation f_1, f_2 and p should be substituted with correct signs.

NOTE 2 For a cylindrical shell subjected to internal or external pressure, a bending moment M and axial force W (without an applied torque), the stress intensity may be determined directly from the stress components as follows:

$$f_m = max(|\sigma_\theta - \sigma_z|; |\sigma_z + 0.5p|; |\sigma_\theta + 0.5p|)$$

In this equation σ_q, σ_z and p should be substituted with correct signs.

The allowable shell longitudinal stress and allowable shell compressive stress shall be as specified in Table 3.2-1, depending on the load case.

B.5 Calculation of wind loading

In order to calculate wind loadings it is necessary to determine:

a) the geographical location of the vessel and basic wind speed together with the effects of topography, height and environment;

b) factor for the design life of the vessel;

c) pressure coefficient, depending on shape and height/diameter ratio.

Information and guidance on the use of these factors and conditions is contained in BS EN 1991-1-4.

NOTE Guidance on the application of BS EN 1991-1-4 to pressure vessels is given in Enquiry Case EC 5500/127.

Special consideration shall be given to tall slender vertical vessels which might be subject to aerodynamic oscillation by wind forces.

If the frequency of shedding of eddies coincides with the natural frequency of the vessel, critical conditions can arise. These effects shall be investigated for height/diameter ratios of 10 or greater.

BRE Digest No. 119 gives information on eddy shedding frequency. Methods for finding the natural period of vibration in vessels can be found in the following publications:

FREESE, C.E. Vibrations of vertical pressure vessels. *J. Engng. Ind.* 1959, February.

DE GHETTO and LONG. Check towers for dynamic stability. *Hydrocarbon Processing.* 1966, **45**(2).

For vessels having a large *D/e* ratio there may be risk of shell instability due to high localized pressure. This aspect should be investigated in such cases, particularly for an empty vessel.

B.6 Calculation of earthquake loading

The stress limit given in **A.3.6** is applicable in cases where it is agreed that earthquake loads can be treated as equivalent static loads and where the probable incidence of the "design" earthquake is not greater than that of the wind loading given in **B.5**.

Information on design for earthquake resistance is given in BS EN 1998-1 and BS EN 1998-6, including the UK National Annexes.

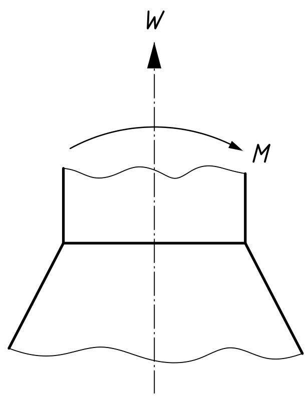
NOTE Other National Annexes apply for vessels installed in other European countries.

B.7 Global loads at cylinder to cone junctions

B.7.1 Introduction

Global loads are defined as those acting on the transverse section of the shell as shown in Figure B.4. The limitations given in **3.5.3.4**, **3.5.3.5** and **3.5.3.6** of the main text apply.

Figure B.4 **Global loads**



Key

M is the applied moment;

W is the applied axial force.

B.7.2 Nomenclature

D_c	is the mean diameter of the cylinder at the junction with the cone;
e_{aj}	is the lesser of the analysis thicknesses of the cylinder or cone, at the junction;
e_{a1}	is the analysis thickness of the cylinder at the junction
e_{a2}	is the analysis thickness of the cone at the junction;
f	is the lesser of the cylinder or cone design stresses
M	is the applied moment;
p	is the design pressure;
W	is the applied axial force, positive when putting the cylinder under tension;
α	is the semi-angle of the cone at the apex.

B.7.3 Junction of cylinder and large end of cone, without torispherical knuckle

Calculate the following:

$$\beta = \frac{1}{3}\sqrt{\frac{D_c}{e_{aj}}}\left(\frac{\tan\alpha}{1 + 1/\sqrt{\cos\alpha}}\right) - 0.15 \quad \text{(B-13)}$$

$$W_{max} = \frac{\pi D_c e_{aj} f \cos\alpha}{(2.4\beta + 1.06)} \quad \text{(B-14)}$$

$$M_{max} = \frac{D_c W_{max}}{4} \quad \text{(B-15)}$$

$$p_{max} = \frac{2e_{aj} f}{\beta D_c} \quad \text{(B-16)}$$

Check that:

$$\max\left\{\frac{|p|}{p_{max}}; \frac{|W|}{W_{max}}; \left|\frac{p}{p_{max}} + \frac{W}{W_{max}}\right|\right\} + \frac{|M|}{M_{max}} \leq 1.0 \quad \text{(B-17)}$$

If the condition in Equation (B-17) is not met, thicknesses at the junction will need to be increased.

B.7.4 Junction of cylinder and large end of cone, with torispherical knuckle

Calculate γ and ρ from Equations (3.5.3-7) and (3.5.3-8) of the main text, β from Equation (B-13), then calculate the following:

$$W_{max} = \frac{\pi D_c e_{aj} f \cos\alpha}{(2.4\beta + 1.06)}(\rho + 1) \quad \text{(B-18)}$$

$$M_{max} = \frac{D_c W_{max}}{4} \quad \text{(B-19)}$$

$$p_{max} = \frac{2e_{aj} f \gamma}{\beta D_c} \quad \text{(B-20)}$$

Check that:

$$\max\left\{\frac{|p|}{p_{max}}; \frac{|W|}{W_{max}}; \left|\frac{p}{p_{max}} + \frac{W}{W_{max}}\right|\right\} + \frac{|M|}{M_{max}} \leq 1.0 \quad \text{(B-21)}$$

If the condition in Equation (B-21) is not met, thicknesses at the junction will need to be increased.

B.7.5 Junction of cylinder and small end of cone

Derive β_H from Equation (3.5.3-10) of the main text, using e_{a1} and e_{a2} in place of e_1 and e_2, then calculate the following:

$$W_{max} = \frac{\pi D_c e_{a1} f}{2\beta_H - 1.0} \quad \text{(B-22)}$$

$$M_{max} = \frac{D_c W_{max}}{4} \quad \text{(B-23)}$$

$$p_{max} = \frac{2e_{a1} f}{\beta_H D_c} \quad \text{(B-24)}$$

Check that:

$$\max\left\{\frac{|p|}{p_{\max}};\frac{|W|}{W_{\max}};\left|\frac{p}{p_{\max}}+\frac{W}{W_{\max}}\right|\right\}+\frac{|M|}{M_{\max}}\leq 1.0 \qquad \text{(B-25)}$$

If the condition in Equation (B-25) is not met, thicknesses at the junction will need to be increased.

Annex C Assessment of vessels subject to fatigue

C.1 Introduction

C.1.1 General methodology

This annex covers the assessment of vessels subject to fatigue and contains requirements to ensure that the vessel is designed to have a fatigue life which is at least as high as the required service life.

An introduction to fatigue and the factors that can influence fatigue life is given in **C.1.2**. A detailed fatigue analysis of a vessel, or a component of a vessel or bolting, need not be carried out if criteria given in **C.2** are satisfied. If these criteria are not satisfied then a full fatigue assessment shall be carried out in accordance with **C.3**. The basic principles of this assessment and the basis of the associated S-N curves are described in **C.3.1**. The application of the S-N curves, together with the modifying effects of various materials and plate thicknesses, are described in **C.3.2**. **C.3.3** identifies the stresses to be used with the S-N curves. Each curve refers to a group of details, identified as class C, D, E, F, F2, G, G2 or W1. The class of any given detail is chosen using **C.3.4.1**, **C.3.4.2**, **C.3.4.3** and **C.3.4.4**. Stresses for the detail are then estimated using **C.3.4.5** and **C.3.4.6** and the design life found from Figure C.3.

The detailed fatigue assessment of bolts is carried out in accordance with **C.3.5**. Recommendations for reducing the risk of fatigue at a weld toe by dressing, are given in **C.4**.

The assessment of weld defects is described in **C.3.4.2**.

The rules given in this annex are intended to provide the designer with the means to establish the service fatigue life of vessels built to normal standards of workmanship and containing no significant pre-existing and/or manufacturing defects beyond the quality levels defined by this document.

Calculations carried out using the methods presented herein are based on test data that can be considered to account for the time taken to incubate and initiate small defects, with factors of conservatism applied to take account of the statistical nature of fatigue performance. The presence of shallow surface breaking fatigue defects at the end-of-life condition according to these rules is therefore very unlikely but would not be inconsistent with the application of these rules.

NOTE Methods are available in BS 7910 to enable the designer to establish whether longer periods of safe operation can be demonstrated by calculation of subsequent fatigue crack growth and determination of critical defect size.

C.1.2 Factors which influence fatigue life

C.1.2.1 Cyclic service loadings

During service, pressure vessels may be subjected to cyclic or repeated stresses (see Figure C.1). Examples of sources of such stresses include the following:

a) application or fluctuations of pressure (including testing);

b) temperature transients;

c) restrictions of expansion or contraction during normal temperature variations;

d) forced vibrations;

e) variations in external loads.

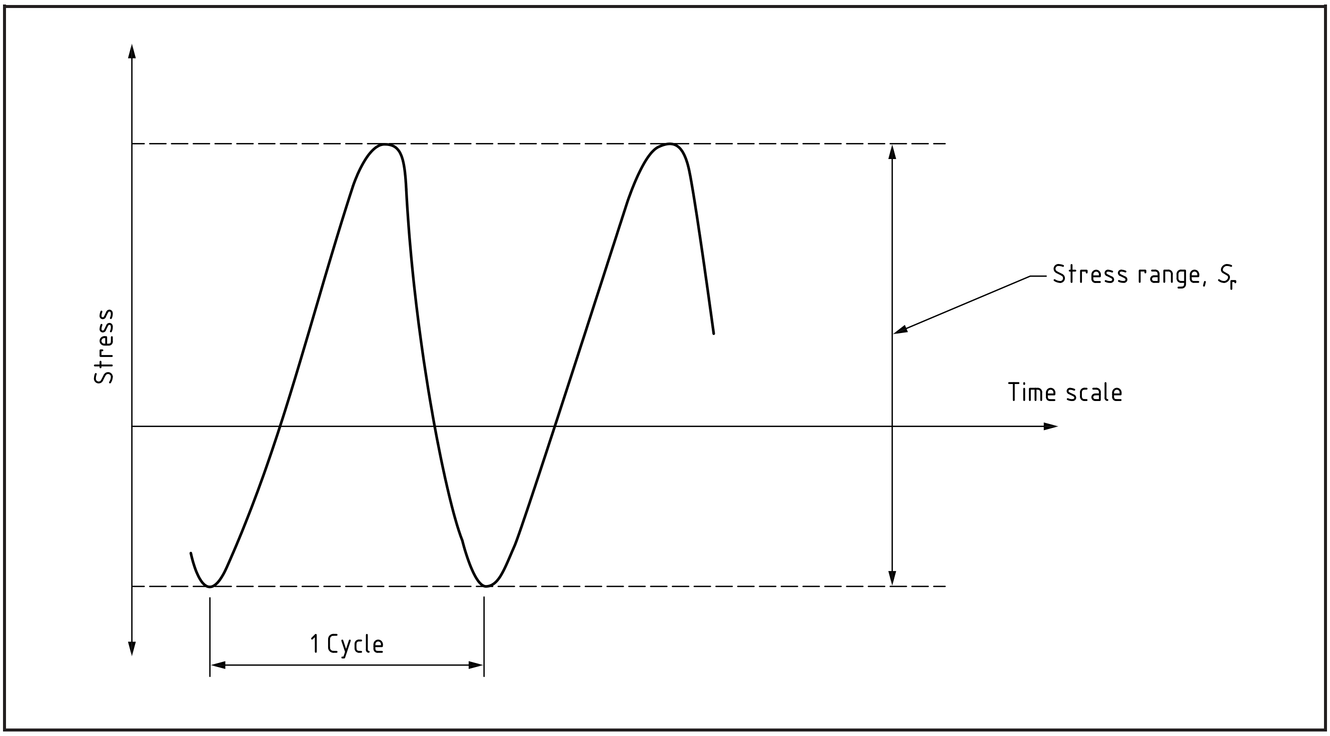
Fatigue failure can occur during service if the fatigue life of the material or any structural detail is exceeded.

C.1.2.2 Corrosion

Corrosive conditions are detrimental to the fatigue strength of steels, aluminium alloys, nickel alloys and titanium alloys. Fatigue cracks can occur under such conditions at lower levels of fluctuating stress than in air and the rate at which they propagate can be higher. The provisions of this annex do not include any allowances for corrosive conditions.

Therefore, where corrosion fatigue is anticipated and effective protection from the corrosive medium cannot be guaranteed, a factor is chosen on the basis of experience or testing provisions by which the stresses specified in this annex are reduced to compensate for the corrosion. If because of lack of experience it is not certain that the chosen stresses are low enough, it is advisable that the frequency of inspection is increased until there is sufficient experience to justify the factor used.

Figure C.1 **Illustration of fluctuating stress**



C.1.2.3 Temperature

There are no restrictions on the use of the fatigue design curves for vessels which operate at sub-zero temperatures, provided that the material through which a fatigue crack might propagate is shown to be sufficiently tough to ensure that fracture will not initiate from a fatigue crack (see Annex D).

There is a lack of data on the influence of creep on the elevated temperature fatigue strength of steel and aluminium, and this annex is therefore only applicable to vessels which operate at temperatures below the creep range of the material. Thus, the design curves are applicable for temperatures up to and including:

a) 350 °C for ferritic steels in material groups 1 to 6, 9 and 11;

b) 430 °C for austenitic stainless steels in material group 8;

c) 100 °C for aluminium alloys in material groups 21 to 23;

d) 300 °C for duplex steels in material sub-group 10.1;

e) 250 °C for super duplex steels in material sub-group 10.2;

f) 450 °C for nickel alloys in material groups 41 to 48; and

g) 150 °C for titanium alloys in material groups 51 to 54.

NOTE For titanium alloys at temperatures above 150 °C specialist advice should be sought.

Where a pressure vessel is intended for cyclic operation within the creep range, the design conditions shall be given by the purchaser in the purchase specification, having regard to the available service experience and experimental information.

C.1.2.4 Vibration

Pulsations of pressure, wind-excited vibrations or vibrations transmitted from plant (e.g. rotating or reciprocating machinery) may cause vibrations of piping or local resonance of the shell of a pressure vessel. Due to the high number of stress fluctuations induced by vibration, fatigue cracking can occur at welded joints, even at very low stress ranges.

Since in most cases vibrations cannot be anticipated at the design stage, it is advisable to make an examination of plant following initial start-up. If vibration occurs which is considered to be excessive, the source of the vibration shall be isolated, or stiffening[1)], additional support or damping introduced at the location of the local vibration. If vibration remains and it was not taken into consideration as a source of fatigue loading at the design stage, a re-evaluation using the detailed assessment (see **C.3**) shall be performed.

C.1.3 Symbols

For the purposes of this annex the following general symbols apply. Particular symbols are defined throughout the text.

A	is the constant in equation of fatigue design curve (see Table C.1);
b	is the exponent in correction term for thickness;
d_b	is the bolt outside diameter;
D_{max}	is the maximum inside diameter of cylindrical vessel, including corrosion allowance (in mm);
D_{min}	is the minimum inside diameter of cylindrical vessel, including corrosion allowance (in mm);
e	is the nominal thickness of section being considered, including corrosion allowance (in mm) (see **1.6**);
E	is the modulus of elasticity at the maximum operating temperature from Table 3.6-3 (in N/mm^2);
f	is the design stress (in N/mm^2);
f_f	is the maximum stress used in **C.2.2** (in N/mm^2);
f_1, f_2, f_3	are the principal stresses at point being considered (in N/mm^2);
k_e	is the mechanical loading, plastic correction factor [see Equations (C-17) and (C-18)];
k_{tb}	is the correction factor for plate thickness and bending [see **C.3.2.3**];
k_ν	is the thermal loading, plastic correction factor [see Equation (C-19)];
K_m	is the stress magnification factor due to misalignment [see Equations (C-23) and (C-24)];
K_t	is the elastic stress concentration factor;
m	is the index in equation of fatigue design curve (see Table C.1);
N	is the design curve fatigue life (cycles);
n_i	is the number of cycles experienced under stress range S_{ri};
p	is the design pressure (in N/mm^2);

[1)] Note that if stiffening introduces additional welds, they may need to be assessed using this annex.

p_r is the pressure fluctuation range;
R is the mean radius of vessel at point considered (in mm);
S_b is the design stress value for bolting material at maximum operating temperature, from Table 3.8-1 (in N/mm^2);
S_{max} is the maximum stress at periphery of bolt due to tension plus bending (in N/mm^2);
S_{nom} is the maximum nominal stress in bolt due to direct tension (in N/mm^2);
S_r is the stress range used in conjunction with fatigue design curves (in N/mm^2);
α is the coefficient of thermal expansion (per °C);
δ is the total deviation from mean circle at seam weld (see Figure C.7) (in mm);
$\triangle T$ is the change of temperature difference (in °C);
$\triangle\sigma_b$ is the bending stress range;
$\triangle\sigma_m$ is the membrane stress range;
ν is Poisson's ratio;
σ is the direct stress (in N/mm^2);
τ is the shear stress (in N/mm^2);
Ω is the degree of bending.

C.2 Criteria for establishing need for detailed fatigue analysis

C.2.1 General

A detailed fatigue analysis of the vessel or component or bolting shall be carried out in accordance with **C.3** unless the conditions of either **C.2.2** or **C.2.3** are met, or if the design is based on previous and satisfactory experience of strictly comparable service.

C.2.2 Limitation on number of stress fluctuations

A detailed fatigue analysis need not be carried out if the total number of stress fluctuations arising from all sources does not exceed the following:

$$\frac{3.8 \times 10^9}{f_f^{\,3}}\left(\frac{25}{e}\right)^{0.75}\left(\frac{E}{2.09 \times 10^5}\right)^3 \qquad \text{(C-1)}$$

where e is the maximum of greatest thickness or 25 mm, and where, using f_f as a design stress, all the relevant rules of Section **3** (e.g. stability criteria need not be considered) and **A.3.4.2.4** are satisfied, thermal stress being treated as secondary and not peak.

*NOTE f_f need not be the same as the nominal design strength given in **2.3.1.1**. It may be less (to reduce stresses to increase the fatigue life) or it may be greater, in order to encompass thermal stresses.*

Stress ranges that do not exceed 5 N/mm^2 can be neglected when assessing the need for a detailed fatigue analysis in accordance with **C.2.2**.

C.2.3 Simplified fatigue analysis using design curves

The following steps shall be used to carry out a simplified fatigue analysis using design curves.

Step 1

Identify the various events to be experienced by the vessel which will give rise to fluctuating stresses and the frequencies at which they occur, as follows:

n_1 is the expected number of stress cycles at the lowest frequency;
n_2 is the expected number of stress cycles at the second lowest frequency;
n_3 is the expected number of stress cycles at the third lowest frequency; etc.

Step 2

For each frequency, calculate the maximum stress range (see **C.3.3**) due to pressure, due to change of temperature difference and due to mechanical loading. Add them to obtain S_{r1}, S_{r2}, S_{r3} etc. The stresses due to all sources of fatigue loading will be included in S_{r1}; S_{r2} will include stresses due to all sources except that which determines n_1; S_{r3} will include stresses due to all sources except those which determine n_1 and n_2 etc.; (note that discrete events, such as a pressure test, which will never be combined with another load source, are considered separately). An example is given in Figure C.2.

NOTE that a conservative estimate of the stress range due to pressure change, p_r, is:

$$S_r = \left(\frac{p_r}{p}\right)3f \quad \text{(C-2)}$$

and a conservative estimate of the stress range due to change of temperature difference $\triangle T$ between adjacent points[2)] is:

$$S_r = 2E\alpha\Delta T \quad \text{(C-3)}$$

Step 3

Check that the following equation is satisfied:

$$\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+\frac{n_4}{N_4}+\ldots\text{etc.} = \sum\frac{n_i}{N_i} \leq 0.6\left(\frac{25}{e}\right)^{0.75} \quad \text{(C-4)}$$

where

i = 1, 2, 3 etc.;

e is the maximum of greatest thickness or 25 mm;

N_i values are numbers of cycles obtained from the appropriate fatigue design curve in Figure C.3 or Figure C.4, at S_{ri} values calculated in step 2, adjusted where necessary for elastic modulus by first multiplying S_r by $2.09 \times 10^5/E$ (see **C.3.2.2**).

The class D fatigue design curve or, if the vessel or part under consideration contains any welds other than flush-ground butt or flush-ground repair welds, the curve for the lowest class weld detail (see **C.3.4.1**) to be incorporated in the vessel or part under consideration or the class G2 curve shall be used. The design curve in Figure C.4 shall be used to assess bolts.

2) Adjacent points are defined as points which are spaced less than the distance $2.5\sqrt{Re}$ apart, where R and e refer to the vessel, nozzle, flange or other component considered. For temperature differences over greater distances, there is sufficient flexibility between the points to produce a significant reduction in thermal stress.

C.3 Methods for detailed assessment of fatigue life

C.3.1 Basic principles of assessment method

C.3.1.1 Introduction

The fatigue strength of a pressure vessel is usually governed by the fatigue strength of details (e.g. openings, welds, bolting, attachments). Even plain material might contain flush-ground weld repairs and the presence of such welds leads to a reduction in the fatigue strength of the material. In view of this, apart from bolting and material which is certain to be free of welding, the fatigue strength of a vessel is assessed on the basis of the fatigue behaviour of test specimens containing weld details similar to those under consideration, using *S*–*N* curves, in which the fluctuating or repeated stress range, S_r, is plotted against number of cycles to failure, *N*. *S*–*N* curves based on fatigue test data obtained from threaded bolts are used to assess bolts.

C.3.1.2 S–N curves for assessment of weld details

The design *S*–*N* curves for the assessment of weld details given in Figure C.3 have been derived from fatigue test data obtained from welded specimens, fabricated to normal standards of workmanship, tested under load-control or, for applied strains exceeding yield (low-cycle fatigue), under strain control. Continuity from the low- to high-cycle regime is achieved by expressing the low-cycle data in terms of the pseudo-elastic stress range (i.e. strain range multiplied by elastic modulus).

Such data are compatible with results obtained from pressure cycling tests on actual vessels when they are expressed in terms of the nominal stress range in the region of fatigue cracking [see [1][3)].

The curves are used in conjunction with the fluctuating stress range, S_r, regardless of applied mean stress, as illustrated in Figure C.1.

Regression analysis of the fatigue test data gave the mean *S*–*N* curve and standard deviation of log *N* [2], [3]. The curves in Figure C.3 are two standard deviations below the mean, representing approximately 97.7% probability of survival. Comparison of these *S*–*N* curves and fatigue test data obtained from cyclic pressure tests on welded vessels indicates that they are conservative, but not excessively so [1].

The design procedures given in **C.2** and **C.3.4** incorporate *S*–*N* curves three standard deviations below the mean, representing approximately 99.8% probability of survival.

The *S*–*N* curves in Figure C.3 have the form:

$$S_r{}^m N = A \qquad \text{(C-5)}$$

where *m* and *A* are constants whose values are given in Table C.1. Different values apply for lives up to 5×10^7 cycles and for above 5×10^7 cycles.

3) The numbers in square brackets used in this annex relate to the bibliographic references given in **C.5**.

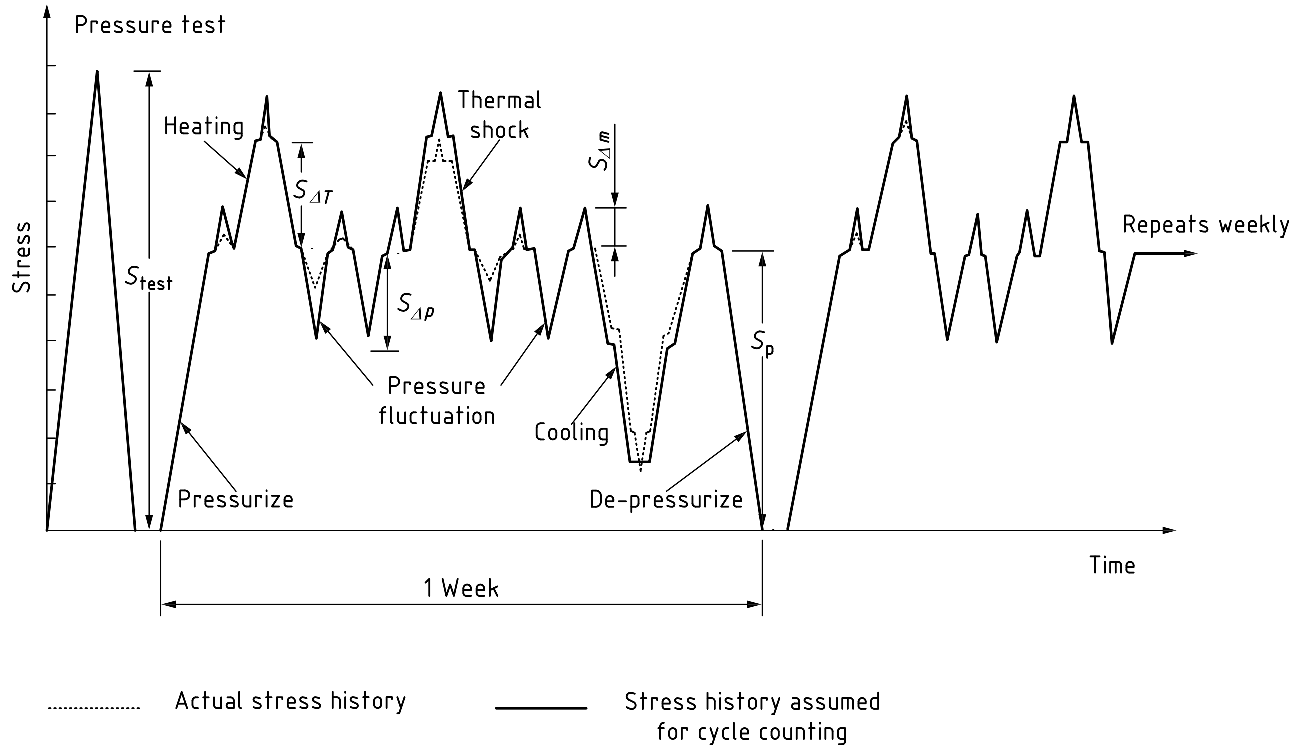
Table C.1 **Details of fatigue design curves**

Class	Constants of S–N curve				Stress range at $N = 5 \times 10^7$ cycles N/mm^2
	for $N < 5 \times 10^7$ cycles		for $N > 5 \times 10^7$ cycles		
	m	A[a]	m	A[a]	
C[b]	3.5	4.23×10^{13}	5.5	1.03×10^{17}	49
D	3	1.52×10^{12}	5	1.48×10^{15}	31
E	3	1.04×10^{12}	5	7.87×10^{14}	28
F	3	6.32×10^{11}	5	3.43×10^{14}	23
F2	3	4.31×10^{11}	5	1.81×10^{14}	21
G	3	2.48×10^{11}	5	7.21×10^{13}	17
G2	3	1.59×10^{11}	5	3.44×10^{13}	15
W1	3	9.33×10^{10}	5	1.41×10^{13}	12
S1	5	2.00×10^{16}	5	2.00×10^{16}	53
S2	5	6.55×10^{15}	5	6.55×10^{15}	42

[a] For $E = 2.09 \times 10^5$ N/mm^2.

[b] If $S_r > 774$ N/mm^2 or $N < 3\,272$ cycles, use class D curve.

Figure C.2 **Example of pressure vessel fatigue loading cycle and determination of stress ranges**



Operating conditions				Stress cycles used in simplified fatigue analysis		
Source	**Maximum stress range**	**Number of cycles**		**Stress range, S_r**	**Number of cycles, n**	
		per week	**per year**		**week**	**year**
Pressure test	S_{test}	—	1	$(S_r)_1 = S_{test}$	—	1
Full range pressure	S_p	1	50	$(S_r)_2 = S_p + S_{\Delta T} + S_{\Delta p} + S_{\Delta m}$	1	50
Temperature difference fluctuation	$S_{\Delta T}$	3	150	$(S_r)_3 = S_{\Delta T} + S_{\Delta p} + S_{\Delta m}$	3	150
Pressure fluctuation	$S_{\Delta p}$	5	250	$(S_r)_4 = S_{\Delta p} + S_{\Delta m}$	5	250
Mechanical loading	$S_{\Delta m}$	9	450	$(S_r)_5 = S_{\Delta m}$	9	450

Figure C.3 **Fatigue design *S–N* curves for weld details applicable for temperatures up to and including those specified in C.1.2.3**

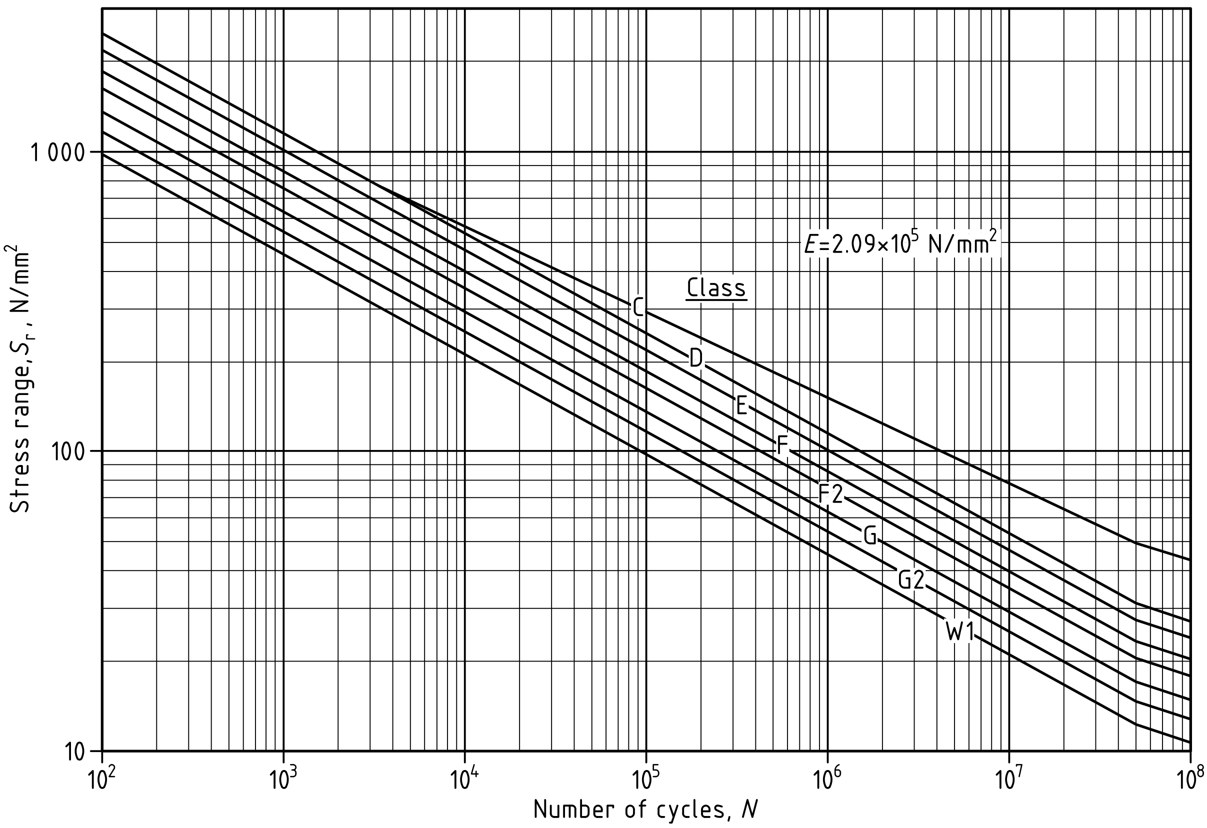
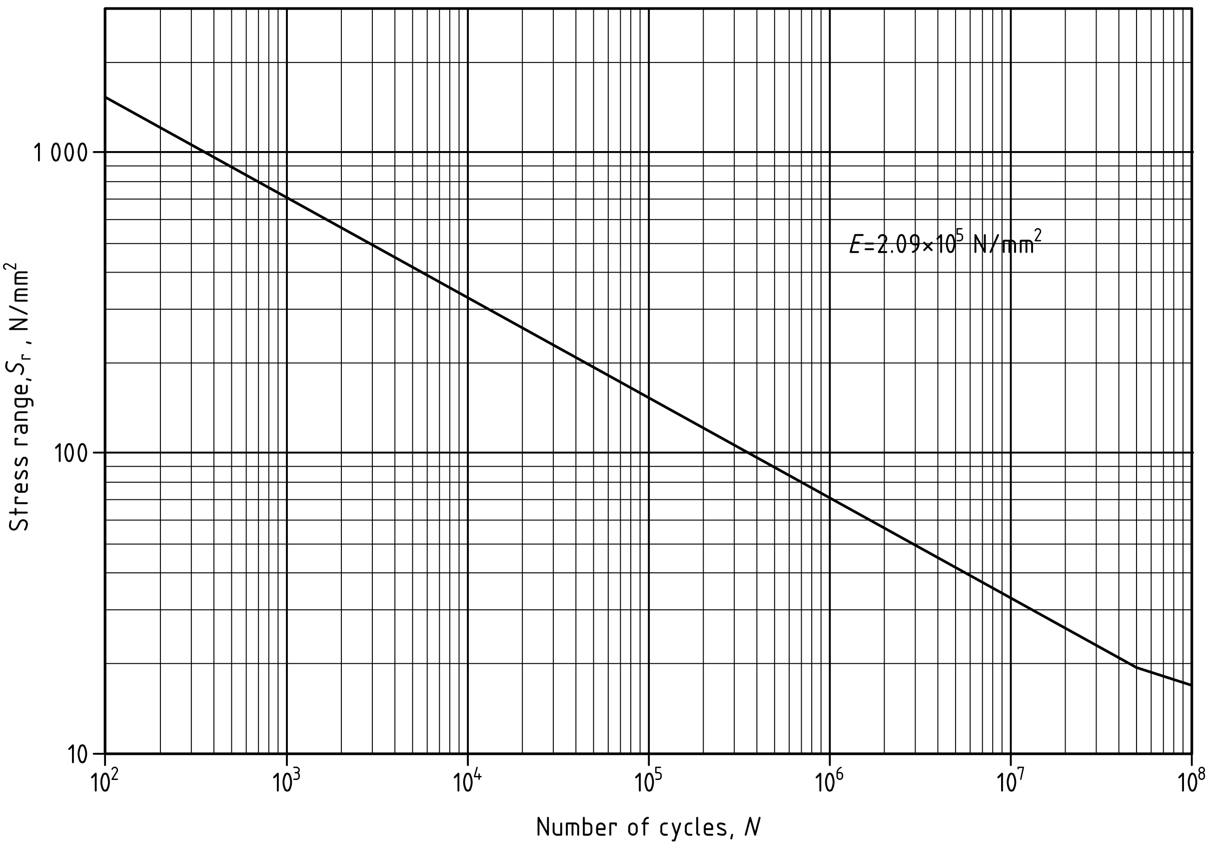


Figure C.4 **Fatigue design *S–N* curves for bolting applicable for temperatures up to and including those specified in C.1.2.3**



C.3.1.3 S–N curves for steel bolting

The *S–N* curve for steel bolting given in Figure C.4 has been derived from fatigue test data from threaded bolts (see BS 7608:2014+A1, Table 18, Class X), and is two standard deviations below the mean. The *S–N* curve in Figure C.4 has the form given by Equation (C-5), where the constants m and A are as follows:

For $N < 5 \times 10^7$ cycles, $m = 3$ and $A = 3.48 \times 10^{11}$

For $N > 5 \times 10^7$ cycles, $m = 5$ and $A = 1.27 \times 10^{14}$

The *S-N* curve is used in conjunction with the fluctuating stress range, S_r, in the bolts regardless of the applied mean stress.

In addition the following conditions apply:

a) The *S–N* curve in Figure C.4 is directly applicable to bolts with cut or ground threads and to rolled threads that have been heat treated.

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

c) If the bolt is electroplated the fatigue strength should be reduced by 20%.

C.3.2 Application of S–N curves

C.3.2.1 Types of operational cycle

The fatigue design curves are directly applicable (after any necessary adjustment for elastic modulus and thickness, see **C.3.2.2** and **C.3.2.3**) in circumstances in which the operational cycle being considered is the only one which produces significant fatigue loading. Thus the fatigue life corresponding to S_r is the allowable number of cycles at that stress range.

If there are two or more types of stress cycle, their cumulative effect shall be evaluated and the following condition met:

$$\frac{n_1}{N_1}+\frac{n_2}{N_2}+\frac{n_3}{N_3}+\ldots\text{etc.}=\sum\frac{n_i}{N_i}\leq 1 \qquad \text{(C-6)}$$

where n_i are the numbers of times that each type of stress cycle, S_{ri}, will occur during the life of the vessel and N_i are the fatigue lives corresponding to S_{ri} obtained from the appropriate fatigue design curve.

A cycle counting method is required to take account of either of the following.

a) The superposition of cycles from various sources of loading which produce a total stress range greater than the stress ranges resulting from individual sources.

b) When a stress variation does not start and finish at the same level.

This cycle counting method shall be used to determine effective stress cycles and hence the values of S_{ri} and n_i. The reservoir method as described in BS 7608:2014+A1, Annex H is an acceptable method.

The rule given in Equation (C-6) can be affected by the type of loading spectrum and BS 7608 provides some preliminary guidance on the link between spectrum type and the validity of this rule. In cases where there is uncertainty about the validity of the rule in Equation (C-6) for the type of load spectrum being assessed, the rule $\Sigma(n_i/N_i) \leq 0.5$ should be adopted.

C.3.2.2 Effect of material

These provisions are applicable to all the materials described in Section **2**. However, since the fatigue lives of weld details are independent of material yield strength, for a given detail, the same set of *S*–*N* curves (see Figure C.3) is applicable for all steels (ferritic and austenitic) and for all aluminium, nickel and titanium alloys. The *S*–*N* curves in Figure C.3 and Figure C.4 are actually related to material with a modulus of elasticity of 2.09×10^5 N/mm^2, which is the typical value for ferritic steel at ambient temperature. When other materials and/or temperatures are being considered, the modulus of elasticity *E* (in N/mm^2), the allowable stress range S_r for a particular life and the stress range obtained from the appropriate design curve at the same life, *S*, are related as follows:

$$\frac{S_r}{S}=\frac{E}{2.09\times 10^5} \qquad \text{(C-7)}$$

C.3.2.3 Effect of plate thickness and bending

The fatigue strengths of members containing surface welds can decrease with increase in plate thickness. The *S*–*N* curves apply for section thicknesses, *e*, up to 25 mm, but for *e* > 25 mm, stress ranges obtained from the design curves for the details indicated in Table C.2 should be multiplied by the factor $(25/e)^b$. In all cases, fatigue cracking from the weld toe into a stressed member is being considered and *e* is the thickness of that member. The value of the exponent *b* is obtained from the relevant detail in Table C.2.

In the case of potential fatigue failure from either the toe or end of a weld through the thickness of a 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.

If the membrane and bending components of the total stress range are known then the degree of bending is given by:

$$\Omega = \frac{\Delta\sigma_b}{\Delta\sigma_m + \Delta\sigma_b} \qquad \text{(C-8)}$$

If the membrane and bending components are not known then it is conservative to assume that the stress range is predominantly membrane and $\Omega = 0$. For cases of increasing stress range gradient through the thickness, or if the bending stress component is due to deviations from the design shape (see **C.3.4.6.4**) then Ω shall be assumed to be zero.

NOTE An increasing stress range gradient is one where the stress range increases with distance from the point being analysed towards the mid-thickness of the component. An example of this is a plate subject to a cyclic stress which varies from zero to a maximum value where the tensile membrane component exceeds the bending component, so the stress is tensile all the way through the thickness. At the surface where the stress range is minimum the stress range will increase with distance through the plate thickness. This is referred to as an increasing stress range gradient and $\Omega = 0$. At the other plate surface the stress range gradient is decreasing and Ω is obtained from Equation (C-8).

The correction factor for thickness and bending is calculated as follows.

For $e > 25$ mm:

$$k_{tb} = \left(\frac{25}{e}\right)^{b}\left(1 + 0.18\Omega^{1.4}\right) \qquad \text{(C-9)}$$

For 4 mm $\leq e \leq$ 25 mm and $\Omega > 0$:

$$k_{tb} = \left\{1 + \Omega\left[\left(\frac{25}{e}\right)^{b} - 1\right]\right\} \times \left(1 + 0.18\Omega^{1.4}\right) \qquad \text{(C-10)}$$

For $e < 25$ mm and $\Omega = 0$ or for $e < 4$ mm:

$$k_{tb} = 1 \qquad \text{(C-11)}$$

Taking different materials and plate thicknesses into account, Equation (C-5) can therefore be modified to the following:

$$N = A\left(\frac{k_{tb}E}{S_r \times 2.09 \times 10^5}\right)^{m} \qquad \text{(C-12)}$$

C.3.2.4 Effect of bolt diameter

The *S–N* curve in Figure C.4 is applicable for bolts up to 25 mm diameter. For bolts with an outside diameter d_b greater than 25 mm the stress range obtained from the *S–N* curve should be multiplied by the factor $(25/d_b)^b$, where the value of the exponent b is 0.25.

Taking different materials and bolt diameter into account, Equation (C-12) can therefore be used with k_{tb} calculated as follows.

For $d_b > 25$ mm:

$$k_{tb} = \left(\frac{25}{d_b}\right)^b \tag{C-13}$$

For $d_b \le 25$ mm:

$$k_{tb} = 1 \tag{C-14}$$

Table C.2 **Classification of weld details**

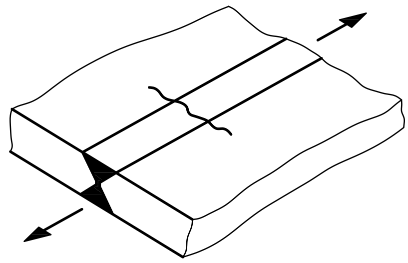
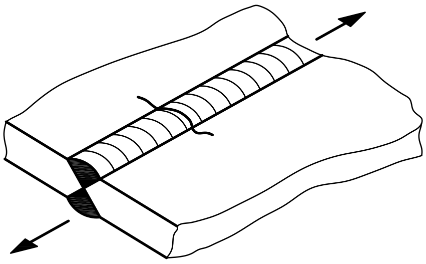
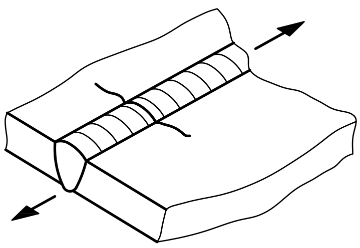
a) Seam welds			
*NOTE The highest fatigue strength transversely loaded seams are full penetration butt welds made from both sides or from one side using consumable inserts or a temporary non-fusible backing medium. For welds where the loading is transverse to the line of the weld allowance should be made for any misalignment of plates that is known or assumed to be present at the welded joint (see **C.3.4.6.4**).* *There is a reduction in the fatigue strength of transverse butt welds if they are made from one side only, unless a joint resembling one made from both sides can be achieved. This is possible using special consumable inserts or a temporary non-fusible backing medium. However, in all cases the weld should be inspected to ensure that full penetration and a satisfactory overfill shape have been achieved on the inside of the joint.* *As far as seam welds under longitudinal loading are concerned, there is an incentive to avoid the introduction of any discontinuous welds. In the absence of significant defects, their fatigue strengths are only reduced if they contain discontinuous welds.*			
Joint type	**For stresses acting essentially along the weld**		
	Sketch of detail	**Class**	**Comments**
Full penetration butt weld flush ground	 Fatigue cracks usually initiate at weld flaws	D	Weld shall be proved free from surface-breaking defects and significant sub-surface defects (see **C.3.4.2**) by non-destructive testing
Full penetration butt weld made from both sides or from one side on to consumable insert or temporary non-fusible backing		D	Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing
Full penetration butt welds made from one side without backing		D	Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing

Table C.2 **Classification of weld details *(continued)***

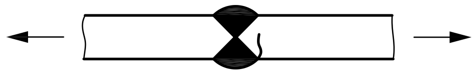
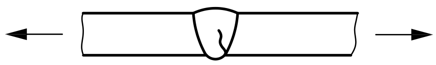
a) Seam welds *(continued)*			
Joint type	**For stresses acting essentially normal to the weld**		
	Sketch of detail	**Class**	**Comments**
Full penetration butt weld flush ground	 Fatigue cracks usually initiate at weld flaws	D	Weld shall be proved free from surface-breaking defects and significant sub-surface defects (see **C.3.4.2**) by non-destructive testing
Full penetration butt weld made from both sides or from one side on to consumable insert or temporary non-fusible backing		D E	Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing and, for welds made from one side, full penetration For welds made onto temporary backing, provided that the surfaces in contact with the backing are aligned to within ± 1 mm Thickness correction applicable (see **C.3.2.3**) assuming *b* = 0.2
Full penetration butt welds made from one side without backing			Not recommended for fatigue loaded joints since fatigue life critically dependent on root condition. If full penetration can be assured, then class E. Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing Thickness correction applicable (see **C.3.2.3**) assuming *b* = 0.2

Table C.2 **Classification of weld details *(continued)***

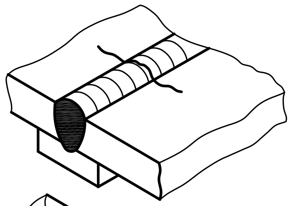
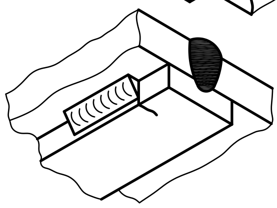
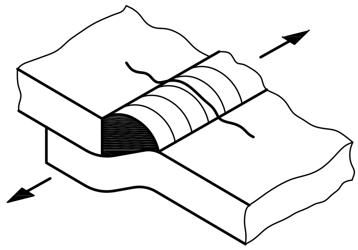
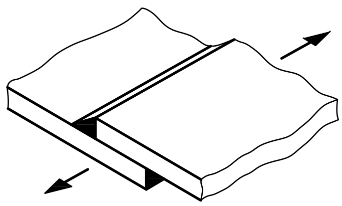
a) Seam welds *(continued)*			
Joint type	**For stresses acting essentially along the weld**		
	Sketch of detail	**Class**	**Comments**
Full penetration butt welds made from one side on to permanent backing	 	D E	Backing strip shall be continuous and, if attached by welding, tack welds shall be ground out or buried in main butt weld, or continuous fillet welds shall be used Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing Backing strip attached with discontinuous fillet weld
		D	Joggle joint Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing
Fillet welded lap joint		D	Welds shall be continuous Based on stress range on cross-section of weld

Table C.2 **Classification of weld details *(continued)***

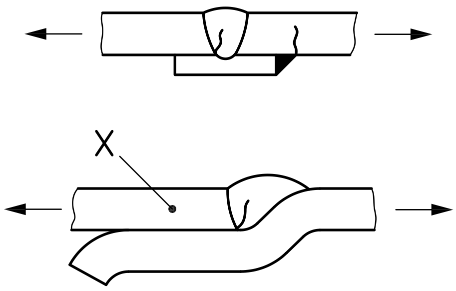
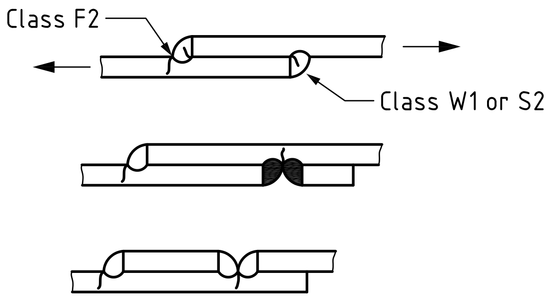
a) Seam welds *(continued)*			
Joint type	**For stresses acting essentially normal to the weld**		
	Sketch of detail	**Class**	**Comments**
Full penetration butt welds made from one side on to permanent backing		F F	Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing For hot spot stress use class F Weld shall be proved free from significant defects (see **C.3.4.2**) by non-destructive testing For hot spot stress use class F Thickness correction applicable (see **C.3.2.3**) assuming b = 0.2 and e is the thickness of plate X in the joggle joint
Fillet welded lap joint		F2 W1 or S2	Refers to fatigue failure in shell from weld toe Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)

Table C.2 **Classification of weld details *(continued)***

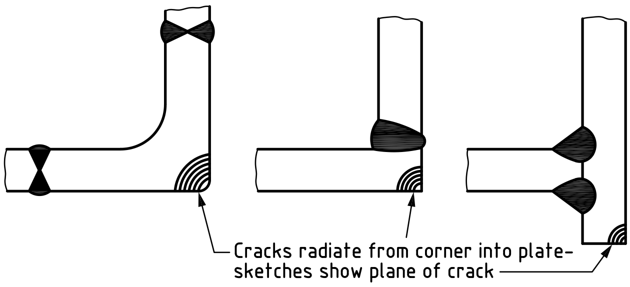
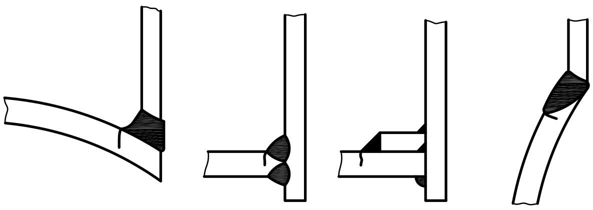
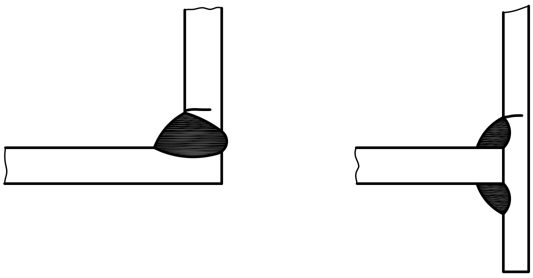
<table>
<tr><td colspan="4">b) Branch connections</td></tr>
<tr><td colspan="4">NOTE The most likely sites for fatigue cracking in branch connections are at the crotch corner and at the weld toes in the shell and the branch. Other branch-like connections shown in this table should also be considered. See C.3.4.6 for guidance on the stresses to be used in the assessment of branches.</td></tr>
<tr><th>Joint type</th><th>Sketch of detail</th><th>Class</th><th>Comments</th></tr>
<tr><td>Crotch corner</td><td></td><td>D</td><td>Can be treated as class C provided region is free from welds (including flush-ground repairs)</td></tr>
<tr><td>Weld toe in shell</td><td></td><td>F</td><td>Constants of S–N curves from Table C.5 can be used if weld toe dressed according to procedure in C.4. Thickness correction applicable (see C.3.2.3) assuming b = 0.25 and e is the shell thickness</td></tr>
<tr><td>Weld toe in branch</td><td></td><td>F</td><td>Constants of S–N curves from Table C.5 can be used if weld toe dressed according to procedure in C.4. Thickness correction applicable (see C.3.2.3) assuming b = 0.25 and e is the branch thickness</td></tr>
</table>

Table C.2 **Classification of weld details *(continued)***

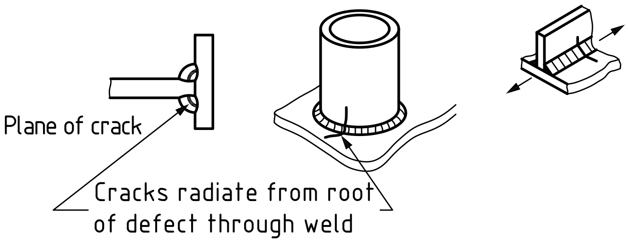
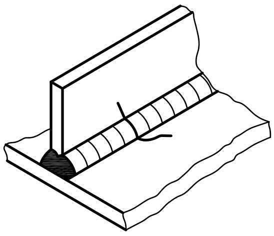
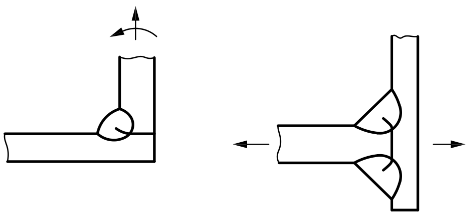
b) Branch connections *(continued)*			
Joint type	**Sketch of detail**	**Class**	**Comments**
Weld metal stressed along its length	 Fillet or partial penetration welds  Full penetration butt weld	F D	Based on stress range on cross-section of weld
Weld metal stressed normal to its length		W1 or S2	Based on stress range on weld throat (see **C.3.3.3**)

Table C.2 **Classification of weld details *(continued)***

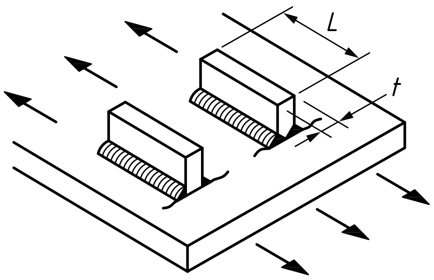
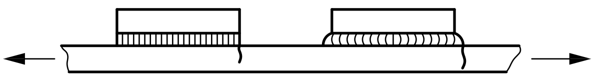
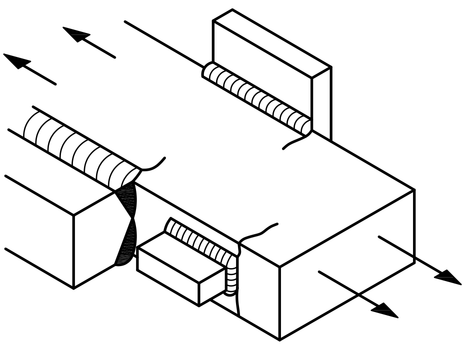
c) Attachments			
NOTE The most likely potential mode of fatigue failure at a welded attachment is from the weld toe, or the weld end in the case of welds lying essentially parallel to the direction of applied stress, into the stressed member. Transverse attachments welded only on one side may fail by fatigue crack propagation from the weld root, also into the stressed member. Such cracks are virtually undetectable and therefore this practice is not recommended. The fatigue strength of members with edge attachments is lower than that of members with only surface attachments; to allow for the accidental occurrence of edge welds, surface attachments less than 10 mm from an edge are assumed to be on the edge. *Fatigue design is based on the normal strength in the stressed member in the vicinity of the attachment. The thickness correction (see* ***C.3.2.3****) is applicable to all surface attachment details, e being the thickness of the stressed member.*			
Joint type	**For stresses acting essentially along the weld**		
	Sketch of detail	**Class**	**Comments**
Attachment of any shape with an edge fillet or bevel – butt welded to the surface of a stressed member, with welds continuous around the ends or not	 	F F2	$L \leq 150$ mm and $t \leq 50$ mm $L > 150$ mm and $t \leq 50$ mm Thickness correction applicable (see **C.3.2.3**) assuming $b = 0.25$ and e is the shell thickness Care is needed to avoid undercutting plate edge if weld ends at or close to plate edge. Any such undercutting should be ground out.
Attachment of any shape on or within 10 mm of the edge of a stressed member		G	Thickness correction does not apply Care is needed to avoid undercutting plate edge if weld ends at or close to plate edge. Any such undercutting should be ground out.

Table C.2 **Classification of weld details *(continued)***

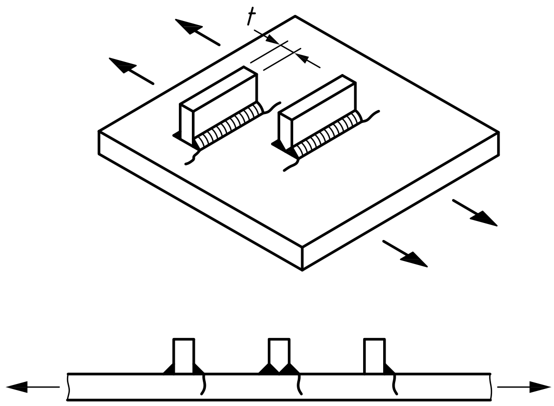
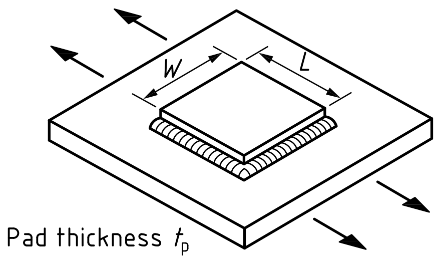
c) Attachments *(continued)*			
Joint type	**For stresses acting essentially normal to the weld**		
	Sketch of detail	**Class**	**Comments**
Attachment of any shape with an edge fillet or bevel, butt welded to the surface of a stressed member, with welds continuous around the ends or not		F F2	$t \leq 50$ mm $t > 50$ mm Thickness correction applicable (see **C.3.2.3**) assuming $b = 0.25$ and e is the shell thickness Care is needed to avoid undercutting plate edge if weld ends at or close to plate edge. Any such undercutting should be ground out.
Attachment of any shape with surface in contact with stressed member, with welds continuous around ends or not		F F2 G G2	$L \leq 150$ mm and $W \leq 50$ mm $L > 150$ mm and $W \leq 50$ mm L > 150 mm, $W > 50$ mm and $t_p \leq 32$ mm $L > 150$ mm, $W > 50$ mm and $t_p > 32$ mm Thickness correction applicable (see **C.3.2.3**) assuming $b = 0.25$ and e is the shell thickness Care is needed to avoid undercutting plate edge if weld ends at or close to plate edge. Any such undercutting should be ground out.

Table C.2 **Classification of weld details *(continued)***

d) Supports

*NOTE For classifications which refer to potential fatigue failure from the weld toe the constants of S–N curves from Table C.5 can be used, if the weld toes are dressed (see **C.4**). However, the class for potential fatigue failure through the weld throat is not affected. Therefore, for toe dressing to be effective, full penetration welds should be used for directly loaded welds.*

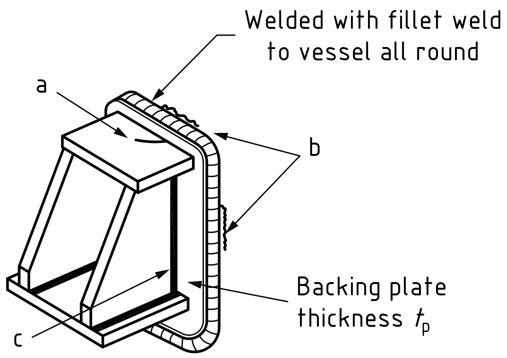
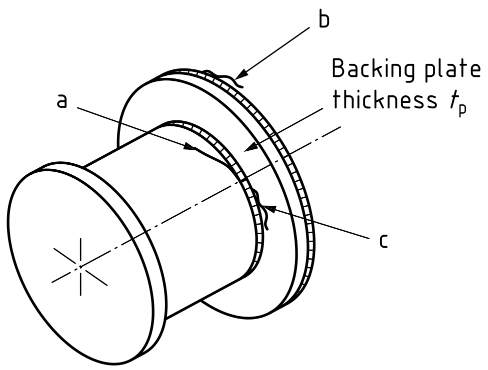
Joint type	Sketch of detail		Class	Comments
Support on either horizontal or vertical vessel		a:	F2	Refers to fatigue failure from weld toe
		b:	G	Refers to fatigue failure from weld toe, $t_p \leq 32$ mm
			G2	Refers to fatigue failure from weld toe, $t_p > 32$ mm
				Thickness correction applicable (see **C.3.2.3**) assuming *b* = 0.25 and *e* is the shell thickness
		c:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)
Trunnion support		a:	F2	Refers to fatigue failure from weld toe
		b:	G	Refers to fatigue failure from weld toe, $t_p \leq 32$ mm
			G2	Refers to fatigue failure from weld toe, $t_p > 32$ mm
				Thickness correction applicable (see **C.3.2.3**) assuming *b* = 0.25 and *e* is the shell thickness
		c:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)

Table C.2 **Classification of weld details *(continued)***

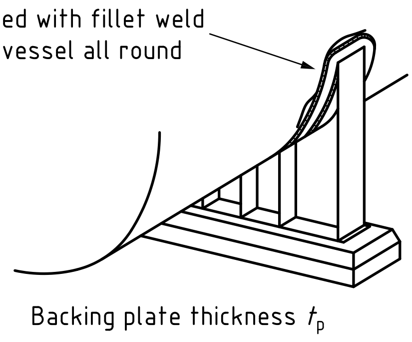
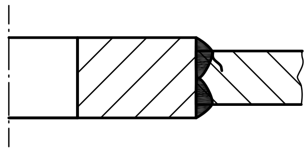
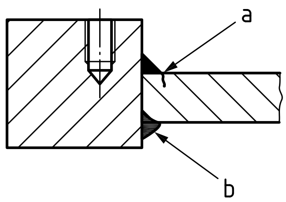
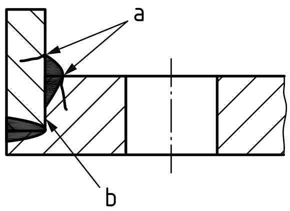
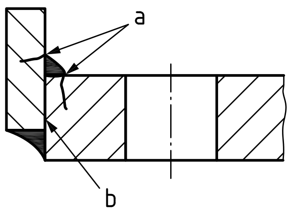
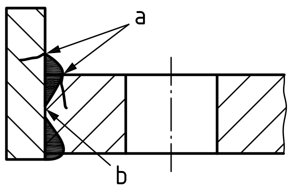
d) Supports *(continued)*			
Joint type	**Sketch of detail**	**Class**	**Comments**
Saddle support		G G2	Refers to fatigue failure from weld toe, $t_p \leq 32$ mm Refers to fatigue failure from weld toe $t_p > 32$ mm Thickness correction applicable (see **C.3.2.3**) assuming b = 0.25 and e is the shell thickness

Table C.2 **Classification of weld details *(continued)***

e) Flanges			
*NOTE For classifications which refer to potential fatigue failure from the weld toe the constants of S–N curves from Table C.5 can be used, if the weld toes are dressed (see **C.4**). However, the class for potential fatigue failure through the weld throat is not affected. Therefore, for toe dressing to be effective, full penetration welds should be used.*			
Joint type	**Sketch of detail**	**Class**	**Comments**
Full penetration butt weld made from both sides		E	Refers to fatigue failure from weld toe
Fillet welded from both sides	 a:	F2	Refers to fatigue failure from weld toe. Thickness correction applicable (see **C.3.2.3**) assuming b = 0.25
	b:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld at throat (see **C.3.3.3**)
Welded from both sides	 a:	F2	Refers to fatigue failure from weld toe. Thickness correction applicable (see **C.3.2.3**) assuming b = 0.25
	b:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)
Fillet welded from both sides	 a:	F2	Refers to fatigue failure from weld toe. Thickness correction applicable (see **C.3.2.3**) assuming b = 0.25
	b:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)
Welded from both sides	 a:	F2	Refers to fatigue failure from weld toe. Thickness correction applicable (see **C.3.2.3**) assuming b = 0.25
	b:	W1 or S2	Refers to fatigue failure in weld; based on stress range in weld throat (see **C.3.3.3**)

C.3.3 Stresses to be used with fatigue design S–N curves

C.3.3.1 General

The fatigue assessment shall be based on the primary plus secondary stress category, as defined in **A.3.4.2.4**. Direct stress (see **A.3.4.1.4**) is used rather than the stress intensity (see **A.3.4.1.1**) used elsewhere in this specification.

The full stress range is used, regardless of applied or effective mean stress. The design *S–N* curves in Figure C.3 take account of localized stress concentration factors due to weld shape and type. Thus, the calculations for primary and secondary stress need only take account of stress concentrations due to gross structural discontinuities and deviations from design shape. The full stress range is used with the design curves, regardless of applied or effective mean stress. These design curves already take account of welding residual stresses. Post weld heat treatment does not have an influence on these design *S–N* curves.

See **C.3.4.6** for guidance on the calculation of stress at gross structural discontinuities and due to deviations from design shape.

The fatigue design curves for bolting take account of stress concentrations for the threads and therefore it is not necessary to apply a stress concentration factor or fatigue strength reduction factor to the calculated bolt stress (see **C.3.3.4**).

C.3.3.2 Stress in parent plate

In the case of parent plate stresses, S_r is the maximum range of direct or normal stress. S_r shall be determined at all points where there is a risk of fatigue cracking (see **C.3.5.1** for individual weld details). In some circumstances, not all stress directions need be considered (see **C.3.4.5.1**).

Where stress cycling is due to the application and removal of a single load, S_r is the same as the maximum principal stress caused by the load acting alone.

Where stress cycling is due to more than one load source but the directions of principal stresses remain fixed, S_r is the maximum range through which any of the principal stresses changes. That is the greatest of:

$f_{1max} - f_{1min}$;

$f_{2max} - f_{2min}$;

$f_{3max} - f_{3min}$;

where f_1, f_2 and f_3 are the three principal stresses. Tensile stresses are considered positive and compressive stresses are considered negative. In practice, the through-thickness component, f_3, is rarely relevant and it can usually be ignored. This is certainly true for assessments at welded joints.

When the principal stress directions change during cycling between two load conditions, S_r is calculated as follows. Determine the six stress components (three direct and three shear) at each load condition with reference to some fixed axes. For each stress component, calculate the algebraic difference between the stresses. Calculate principal stresses from the resulting stress differences in the usual way. S_r is the numerically greatest of these principal stresses.

Where cycling is of such a complex nature that it is not clear which two load conditions will result in the greatest value of S_r, they shall be established by carrying out the above procedure for all pairs of load conditions. Alternatively, it will always be safe to assume that S_r is the difference between the algebraically greatest and smallest principal stresses occurring during the whole cycle regardless of their directions.

If the directions of the principal stresses do not remain fixed then the fatigue life N shall be taken as half the value calculated from Equation (C-12) using the values of A and m for class S1 from Table C.1.

In the case of parent plate subject to pure shear, S_r shall be taken as the nominal shear stress range and the fatigue life N shall be calculated from Equation (C-12) using the values of A and m for class S1.

C.3.3.3 Stress in weld metal

In the case of weld metal in fillet or partial penetration joints,[4)] S_r is the maximum range of stress across the effective weld throat, calculated as the load carried by the weld divided by the weld throat area, with the assumption that none of the load is carried by bearing between the components joined. Since this can be expressed as a vector sum, S_r is the scalar value of the greatest vector difference between different stress conditions during the cycle.

Where stress cycling is due to the application and removal of a single load:

$$S_r = \sqrt{\sigma^2+\tau^2} \tag{C-15}$$

where

σ is the direct stress on weld throat;

τ is the shear stress on weld throat.

Where stress cycling is due to more than one load source, but the directions of the stresses remain fixed, S_r is based on the maximum range of the load on the weld. If the shear stress τ is less than 30% of the direct stress σ the fatigue life shall be calculated using the class W1 curve, otherwise the class S2 curve shall be used.

Where the direction of the stress vector on the weld throat changes during a cycle between two extreme load conditions, S_r 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 will result in the greatest values of S_r, then the vector difference shall be found for all pairs of extreme load conditions.

Alternatively, it will always be safe to assume:

$$S_r = \sqrt{(\sigma_{max} - \sigma_{min})^2+(\tau_{1max} - \tau_{1min})^2 + (\tau_{2max} - \tau_{2min})^2} \tag{C-16}$$

where τ_1 and τ_2 are the two components of shear stress.

If the directions of the stresses do not remain fixed then the fatigue life N shall be taken as half the value calculated from Equation (C-12) using the values of A and m for class S2 from Table C.1.

In the case of weld metal subject to pure shear, S_r shall be taken as the nominal shear stress range and the fatigue life N shall be calculated from Equation (C-12) using the values of A and m for class S2.

C.3.3.4 Stress in bolts

In the case of bolts, S_r is the maximum stress range calculated on the root area of the bolt (see Table 3.8-2), and should include the effects of direct tensile and bending loads. It should take into account the pre-load on the bolt and the compressibility and specified fit-up of the connected parts of the bolted joint. Where the fatigue design of a bolted 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.

[4)] Not applicable to butt joints (e.g. seams).

The compressibility effect in a bolted joint, and the portion of the applied load that is carried by the bolts can be determined from the relative stiffness of the bolt and clamped components.

NOTE BS 7608:2014+A1, 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 the load.

The *S–N* curve in Figure C.4 takes account of the stress concentration factor for the threads, so it is not necessary to apply a stress concentration factor or fatigue strength reduction factor to the calculated bolt stress.

In the case of bolts subject to pure shear, S_r shall be taken as the nominal shear stress range and the fatigue life N shall be calculated from Equation (C-12) using the values of A and m for class S1 from Table C.1.

C.3.3.5 Elastic-plastic conditions

If the calculated pseudo-elastic stress range exceeds twice the yield strength of the material under consideration (i.e. $\Delta\sigma > 2R_e$), it shall be increased by applying a plasticity correction factor, as follows (these correction factors are discussed in [9]):

C.3.3.5.1 Mechanical loading

For mechanical loading, the corrected stress range is $k_e\Delta\sigma$, where:

$$\text{for } 2 \leq \Delta\sigma/R_e \leq 3,\ k_e = M_1[(\Delta\sigma/\{2R_e\}) - 1]^{0.5} + 1 \quad \text{(C-17)}$$

$$\text{or for } \Delta\sigma/R_e \leq 3,\ k_e = M_2 + M_3\Delta\sigma/R_e \quad \text{(C-18)}$$

where M_1, M_2 and M_3 are given in Table C.3.

Table C.3 **Values of M_1, M_2 and M_3**

Steel	M_1	M_2	M_3
Ferritic and austenitic, $R_m \leq 500$ N/mm^2	0.443	0.823	0.164
Ferritic, R_m = 500 N/mm^2 to 800 N/mm^2	$0.318 + (2.5R_m/10^4)$	$0.998 - (3.5R_m/10^4)$	$0.077 + (1.73R_m/10^4)$
Ferritic, R_m = 800 N/mm^2 to 1 000 N/mm^2	0.518	0.718	0.216

C.3.3.5.2 Thermal loading

For thermal loading, the corrected stress range is $k_\nu\Delta\sigma$, where:

$$k_\nu = \frac{0.7}{0.5 + \dfrac{0.4}{\Delta\sigma/R_e}} \quad \text{(C-19)}$$

C.3.3.5.3 Combined loading

If stressing is due to a combination of mechanical and thermal loads, the mechanical and thermal stresses shall be separated and the correction factors k_e and k_ν calculated. The corrected stress range is then the sum of the corrected stresses due to mechanical and thermal loading.

C.3.3.5.4 Elastic-plastic analysis

If the total strain range $\Delta\varepsilon_{\tau}$ (elastic-plastic) due to any source of loading is known from theoretical or experimental stress analysis, correction for plasticity is not required and:

$$\Delta\sigma = E\Delta\varepsilon_{\tau} \qquad \text{(C-20)}$$

C.3.4 Detailed assessment of welded and unwelded components

C.3.4.1 Classification

For the purpose of fatigue assessment, each part of a constructional detail which is subject to fluctuating stress is placed into one of seven classes, designated D, E, F, F2, G, G2 and W1, corresponding to the seven fatigue design curves in Figure C.3. The classifications are described in Table C.2.

The classification of each part of a detail depends upon the following:

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

b) the location of possible crack initiation at the detail;

c) the geometrical arrangement and proportions of the detail;

d) the methods of manufacture and inspection.

Thus, more than one class may apply for a given weld detail, since the class refers to one particular mode of fatigue failure, but there are a number of ways in which a weld detail might fail. The sketches in Table C.2 indicate the potential mode of fatigue cracking considered and the position and direction of the relevant fluctuating stress.

Load-carrying fillet or partial penetration joints shall be assessed as class F2, corresponding to fatigue failure from the weld toe in the stressed plate, and class W1, corresponding to fatigue failure from the weld root in the weld. The possibility of failure from the weld root is avoided if the effective weld throat thickness is such that the stress range in the weld (see **C.3.3.3**) does not exceed 0.6 times the stress range in plate. It should be noted that conformity to the requirements in Section 3 relating to weld size does not necessarily meet this criterion.

Parts of the vessel that are unwelded shall be considered as Class D on the basis that repair welds may be required. Class C only relates to parts which are certain to be free from welding.

C.3.4.2 Assessment of weld defects

Fatigue cracks can propagate from weld defects and the fatigue life of a joint may be limited by this mode of failure. This is true even for defects which are regarded as acceptable in Table 5.7-1 and Table AI.5.7-1.

Planar defects (e.g. unwelded land in partial penetration welds, lack of fusion) are particularly severe but non-planar defects (e.g. slag inclusions, porosity) may also be significant.

The fatigue lives of defects or the tolerable defects for a given fatigue life shall be assessed using an established defect assessment method such as that in BS 7910:2019. The fatigue strengths of defects are expressed in terms of quality categories, Q1 to Q10, and a design *S–N* curve is assigned to each level. The *S–N* curves for categories Q1 to Q6 (only those described as being applicable to as-welded joints shall be used) correspond to the classes D, E, F, F2, G and G2 fatigue design curves in Figure C.3. Thus, the fatigue strengths of defects can be readily compared with those of other weld details.

NOTE 1 The S–N curves in BS 7910 differ from those in the present procedures in the high-cycle regime ($N > 5 \times 10^7$ cycles) in that they include a cut-off stress at $N = 2 \times 10^7$ cycles. They should be modified to be consistent with the present procedures by extrapolating them beyond 5×10^7 cycles at a slope of $m = 5$.

NOTE 2 Annex U gives guidance on the application of BS 7910.

Acceptance levels for embedded non-planar defects are summarized in Table C.4, and are 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). If there is any doubt that a defect is non-planar or that it is embedded, it shall be treated as being planar. Multiple slag inclusions on the same cross-section (see Figure C.5) which are closer than 1.25 times the height of the larger defect, shall also be treated as a planar defect.

Planar defects can be assessed using fracture mechanics. BS 7910 describes the general procedure and also gives a simplified method of assessment which is related to the quality categories.

Table C.4 **Weld defect acceptance levels**

Class required	Maximum length of slag inclusion (in mm)		Maximum % of area porosity on radiograph
	97.72 % survival probability	99.86 % survival probability	
Q1 (D)	2.5	2	3
Q2 (E)	4	2.5	3
Q3 (F)	10	5	5
Q4 (F2)	35	9	5
Q5 (G)	No limit	66	5
Q6 (G2) and lower	No limit	No limit	5

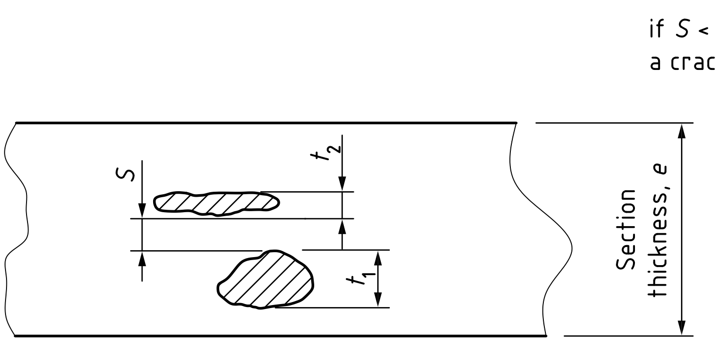
NOTE 1 Tungsten inclusions in aluminium alloy welds do not affect fatigue behaviour and need not be considered as defects from the fatigue viewpoint.

NOTE 2 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.

NOTE 3 Individual pores are limited to a diameter of e/4 or 6 mm, whichever is the lesser.

NOTE 4 The above levels can be relaxed in the case of steel welds which have been thermally stress relieved, as described in BS 7910.

Figure C.5 **Interaction criteria for assessing coplanar embedded slag inclusions**



C.3.4.3 Change of classification

C.3.4.3.1 General

By agreement between the manufacturer, purchaser and Inspecting Authority, the classification of some weld details may be raised if the conditions in **C.3.4.3.2** or **C.3.4.3.3** are met.

C.3.4.3.2 Detailed stress analysis

If, as a result of the stress analysis method used, the calculated or measured stress range adjacent to a weld in class F, F2, G or G2 incorporates the effect of the stress concentration due to the joint geometry (see **C.3.4.6**), class D may be assumed, unless otherwise indicated in the comments column of Table C.2 for the relevant joint type.

The aim of the analysis is to determine the stress at the weld toe excluding its local stress concentration. This stress at the weld toe is generally referrred to as the "structural hot spot" stress, or simply "hot spot" stress, which is equivalent to the primary plus the secondary stress of Annex A (see **A.3.4.2.4**). This stress can be determined using strain measurements or detailed stress analysis and using either extrapolation to the weld toe from the surface stress distribution approaching the weld, or directly from the through-thickness stress distribution at the weld toe.[5)]

If the stresses are determined from strain measurements on prototype or actual vessels, the structural hot spot stress is calculated from the principal stress which acts closest to the normal to the weld toe by extrapolation from stresses located at specified distances from the weld toe using the procedure detailed in Figure C.6. The maximum strain gauge size shall be 0.2*e*, where *e* is the nominal plate thickness.

If the stresses are determined by detailed stress analysis [e.g. finite element analysis (FEA)], the extrapolation procedure in Figure C.6 is still applicable. The maximum finite element mesh size shall be 0.2*e*, where *e* is the nominal plate thickness. Alternatively, the structural hot spot stress can be obtained directly from the equivalent linear membrane and bending stress distributions through the plate thickness, obtained by enforcing through-thickness equilibrium through the thickness below the weld toe. The structural hot spot stress is then the sum of the membrane and bending stresses at the weld toe.

If the stress distribution is generally linear, stresses shall be measured as indicated in Figure C.6a) and determined in accordance with Equation (C-21):

$$f_{hs} = 1.67 f_A - 0.67 f_B \qquad \text{(C-21)}$$

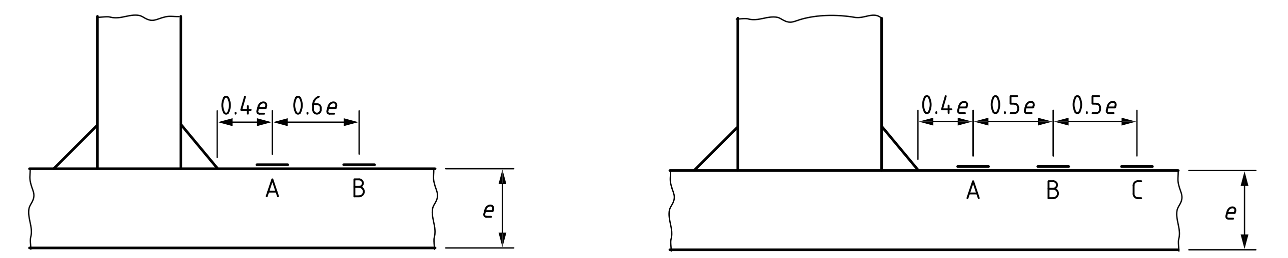
If the stress distribution is non-linear, stresses shall be measured as indicated in Figure C.6b) and determined in accordance with Equation (C-22):

$$f_{hs} = 2.52 f_A - 2.24 f_B + 0.72 f_C \qquad \text{(C-22)}$$

where f_{hs} is the hot spot stress and f_A is the stress at location A, etc.

[5)] Methods for determining the structural hot-spot stress are still being developed and are described in more detail in BS 7608:2014+A1, Annex C, and in reference [12], both of which provide practical guidance on the use of FEA. They have been validated against fatigue test results obtained from pressure vessels and components (see references [13] and [14]). However, other methods are available and better ones might become available in future. Provided they have been validated against relevant experimental data, they may be used instead.

Figure C.6 **Stress measurement points for determining structural hot spot stress at a weld toe**



a) Generally linear stress distribution

b) Non-linear stress distribution

C.3.4.3.3 Weld toe dressing

The fatigue strength of welded joints involving potential fatigue cracking from the weld toe may, where indicated in Table C.2, be raised when dressing of the toes is carried out. When joints are treated in accordance with **C.4**, the fatigue strength is increased by 50% at $N = 10^7$ cycles and the *S–N* curves have a slope of $m = 3.5$. The values of the constants *A* and *m* for the resulting fatigue curves are given in Table C.5.

NOTE These fatigue curves give no improvement in calculated fatigue life for N < 2000 cycles.

Table C.5 **Details of fatigue design curves for joints with weld toe improvement**

Class	Constants of *S-N* curve		Stress range at *N* = 2000 cycles N/mm^2
	m	*A*	
E	3.5	2.93×10^{13}	803
F	3.5	1.65×10^{13}	681
F2	3.5	1.05×10^{13}	599
G	3.5	5.53×10^{12}	498
G2	3.5	3.30×10^{12}	430

No benefit in terms of improved fatigue strength is allowed for the dressing or flush-grinding of seam welds.

Previously buried defects revealed by dressing, which could limit the fatigue strength of the joint, should be assessed (see **C.3.4.2**).

C.3.4.4 Unclassified details

Except for partial penetration butt welds, which are not classified, details not covered fully in Table C.2 shall be treated as class G, or class W1 for load-carrying weld metal.

NOTE 1 A higher classification could be used if superior resistance to fatigue is proved by special tests or reference to relevant test results.

To justify a particular design *S–N* curve, tests shall be performed at stress levels which result in lives of no more than 2×10^6 cycles, and the geometric mean fatigue life obtained from tests performed at a particular stress range shall be not less than the life from the *S–N* curve at that stress range multiplied by the factor *F* from Table C.6.

NOTE 2 Additional guidance on the statistical analysis of fatigue test data is given in BS 7608:2014+A1, Annex E.

Table C.6 **Fatigue test factor *F***

Number of tests	*F*
1	5.4
2	4.3
3	3.9
4	3.7
10	3.2

C.3.4.5 Stresses to be considered

C.3.4.5.1 Classes D to G2

The fatigue lives of weld details which fall into classes D to G2 are expressed in terms of the primary plus secondary stress range on the parent metal surface adjacent to the weld (see **C.3.3.2**), ignoring any stress concentration due to the welded joint itself but including the effect of other stress concentrations (see **C.3.4.6**).

Short or discontinuous welds, where the relevant potential failure mode is by fatigue cracking from the weld end or weld toe shall be assessed on the basis of the maximum principal stress range S_r, and classified on the basis that the weld is oriented in the least favourable direction with respect to S_r.

Continuous welds (e.g. seams, ring stiffener welds) may be treated differently if the maximum principal stress range acts in a direction which is within 30° of the direction of the weld. Then the weld can be classified as being parallel to the direction of loading with respect to the maximum principal stress range and normal to the loading direction with respect to the minimum principal stress range (see **C.3.3.2**).

C.3.4.5.2 Class W1

The fatigue lives of class W1 details are expressed in terms of the maximum stress range on the weld throat (see **C.3.3.3**).

C.3.4.6 Estimation of stress

C.3.4.6.1 General

In arriving at the primary plus secondary stresses required for use in this annex, it is necessary to take full account of structural discontinuities (e.g. nozzles) but also some sources of stress not normally considered, in particular:

a) discontinuities such as cylinder to end junctions, changes in thickness and welded-on rings;

b) deviations from the intended shape such as ovality, peaking and mismatched welds;

c) temperature gradients.

Methods in this specification (e.g. Annex G) give the primary and secondary stresses that take account of gross structural discontinuities in pressure vessels. References in published literature (e.g. [5], [7] and [8]) give guidance on stress concentration factors for fatigue assessment of a wider range of geometries.

C.3.4.6.2 Nozzles

Three possible stress concentrations due to structural discontinuities in nozzles shall be considered when calculating S_r.

a) *Crotch corner*. The class D, or exceptionally class C (see **C.3.4.1**), fatigue design curve shall be used in conjunction with the maximum circumferential (with respect to the nozzle) stress range at the crotch corner. K_t is usually referred to the nominal hoop range in the shell.

b) *Weld toe in shell*. The class F or F2 fatigue design curve, depending on the weld detail, shall be used in conjunction with the maximum stress range in the shell at the welded toe. Consideration shall be given to stresses in the shell acting in all radial directions with respect to the nozzle in order to determine the maximum stress at the weld toe. The possibility of stresses arising in the shell as a result of mechanical loading on the nozzle as well as pressure loading shall be considered.

c) *Weld toe in branch*. This region shall be treated as described in item b), except that the maximum stress range in the branch shall be used. Again, the possibility of mechanical as well as pressure loading shall be considered.

C.3.4.6.3 Supports and attachments

Local concentrations of stress can arise in the shell where it is supported (see Annex G) or loaded through an attachment. The appropriate fatigue design curve shall be used in conjunction with the maximum stress range in the shell at the weld toe determined using the same criteria as for nozzle weld toes in the shell (see **C.3.4.6.2**).

The primary membrane and bending stresses that arise due to local loads in both spherical and cylindrical vessels may be derived using **G.2**, **G.3.1** or **G.3.2**, as relevant, in order to calculate the stress range. In the case of the stresses in horizontal vessels on saddle supports, as determined in **G.3.3**, this procedure is not appropriate. An alternative procedure to provide more accurate values of the relevant stresses is given in references [10] and [11].

*NOTE A method of calculating the maximum stresses in the saddle area of a horizontal vessel loaded by contents that can be used in a fatigue assessment in accordance with Annex C, is given in **G.3.3.2.7**.*

C.3.4.6.4 Deviations from design shape

Local increases in pressure-induced stresses in shells which arise as a result of secondary bending stresses due to discontinuities and departures from the intended shape[6)] shall be taken into account when calculating pressure stresses for the fatigue assessment of the shell at seams and attachments, even if the allowable assembly tolerances in **4.2.3** are met.

In the absence of detailed stress analysis of the particular case being considered, a conservative estimate of the effect of the additional bending stresses due to departure from design shape may be obtained by multiplying the appropriate nominal stress range by the following stress magnification factor, K_m:

[6)] Departures from intended shape include misalignment of abutting plates, an angle between abutting plates, roof-topping where there is a flat at the end of each plate, weld peaking and ovality, as illustrated in Figure C.7. In most cases these features cause local increases in the hoop stress in the shell but deviations from design shape associated with circumferential seams cause increases in the longitudinal stress.

When the stresses greater than yield arise as a result of deviation from design shape, the pressure test will lead to an improvement in the shape of the vessel due to plastic deformation. It may be noted that vessels made from materials with yield strengths considerably higher than the specified minimum are less likely to benefit in this way. The beneficial effect of the pressure test on the shape of the vessel cannot be predicted and therefore if some benefit is required in order to satisfy the fatigue analysis, it is necessary to measure the actual shape after pressure test. Similarly, strain measurements to determine the actual stress concentration factor should be made after pressure test.

$$K_m = 1 + A_1 + A_2 + A_4 \text{ for cylinders} \quad \text{(C-23)}$$

$$K_m = 1 + A_1 + A_3 + A_4 \text{ for spheres} \quad \text{(C-24)}$$

where A_1 to A_4 are calculated using Equations (C-25) to (C-29), and are the maximum values of the factors for the location being considered. The value of p_i in Equation (C-26) shall be taken as the maximum cyclic internal pressure for the stress cycle being assessed, and the value of p_e in Equation (C-27) shall be taken as the maximum cyclic external pressure for the stress cycle being assessed.

A less conservative estimate of the effect of the additional bending stresses may be obtained by calculating separate values of the stress multiplication factor K_m for the maximum and minimum cyclic pressure and for the longitudinal and circumferential stresses, taking account of the orientation of the welded joint being assessed. Factor A_2 only applies to the circumferential stress in a cylinder and the value will be different for the maximum and minimum pressure cases. Each of the nominal longitudinal and circumferential direct stresses shall be multiplied by the relevant value of K_m, and these stresses used in the procedures in **C.3.3.2** to evaluate the maximum and minimum principal stresses and hence the stress range.

A_1 caters for misalignment and is given by:

$$A_1 = \left(\frac{6\delta_1}{e_1}\right)\left(\frac{{e_1}^n}{{e_1}^n + {e_2}^n}\right) \quad \text{(C-25)}$$

where

δ_1 is the offset of centrelines of abutting plates;

$e_1 \le e_2$ where e_1 and e_2 are the thicknesses of two abutting plates;

n is 1.5 for a sphere or circumferential seam in a cylinder and 0.6 for a longitudinal seam in a cylinder.

A_2 caters for ovality in cylinders and is given by:

a) for internal pressure

$$A_2 = \frac{1.5(D_{max} - D_{min})}{e\left\{1 + \frac{0.5p_i(1-\nu^2)}{E}\left(\frac{D}{e}\right)^3\right\}} \quad \text{(C-26)}$$

where

D is the mean diameter.

p_i is the internal pressure applied for the case being considered.

b) for external pressure

$$A_2 = \frac{1.5(D_{max} - D_{min})}{e\left(1 - \frac{3p_e}{p_m}\right)} \quad \text{(C-27)}$$

where

p_e is the external pressure applied for the case being considered;

p_m is the elastic instability pressure for collapse of a perfect cylindrical shell, from **3.6.2.1** equation (3.6.2-8), calculated at 20 °C;

A_3 caters for poor angular alignment of plates in spheres and is given by:

$$A_3 = \frac{\theta}{49}\sqrt{\frac{R}{e}} \quad \text{(C-28)}$$

where

θ is the angle between tangents to the plates, at the seam (in degrees);

A_4 caters for local peaking and is given by:

$$A_4 = \frac{6\delta}{e} \tag{C-29}$$

where

δ is the deviation from true form, other than above;

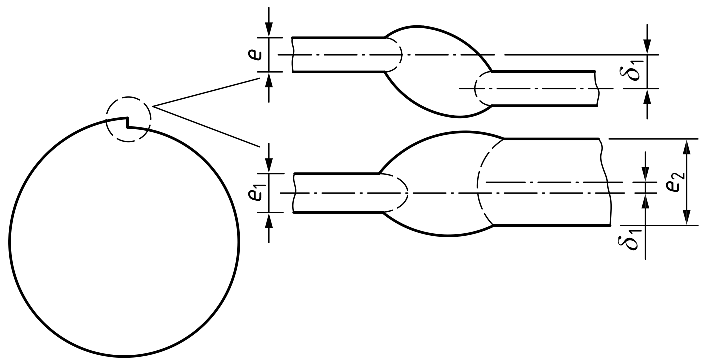
and other terms are defined in Figure C.7.

In the case of seam welds, the incorporation of a transition taper at thickness changes to conform to **4.2.3** does not affect the value of A_1, and the offset δ_1 shall be measured from the centre lines of the untapered section.

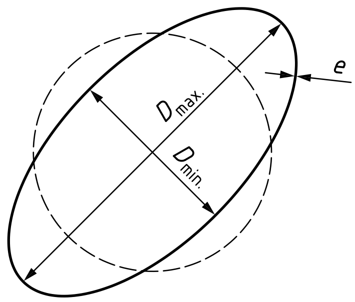
Equations (C-23) and (C-24) will overestimate K_m if local bending is restricted, for example in the case of short shape defects, when there will be a stress redistribution around the defect, or for defects in short cylindrical vessels, which can get support from the ends, or when adjacent attachments stiffen the shell. Also, ovality in long cylinders may not cause the estimated stresses because of the shape improvement due to elastic deformation under pressure.

By agreement between the manufacturer, purchaser and Inspecting Authority, the effect of departures from design shape for which $K_m \leq 2$ may be ignored if the weld toes are burr machined using the procedure given in **C.4**.

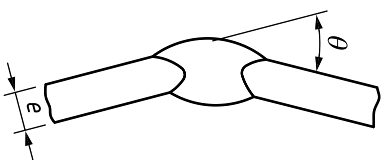
Figure C.7 **Deviations from design shape at seam welds**



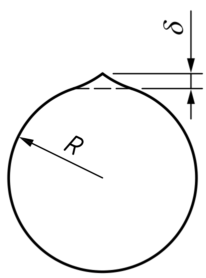
a) Axial misalignment (factor A_1)



b) Ovality (factor A_2)



c) Angular misalignment (factor A_3)



d) Local peaking (factor A_4)

C.3.5 Detailed fatigue assessment of bolts

C.3.5.1 Maximum stresses in bolts

Service stresses in bolts arising from the combination of such factors as pre-load, pressure and differential thermal expansion may be higher than S_b values in Table 3.8-1. However, in bolts subjected to fluctuating stress they shall be limited as follows.

a) The maximum nominal stress S_{nom} due to direct tension, averaged across the bolt cross-section and neglecting stress concentration, shall not exceed $2S_b$.

b) The maximum stress S_{max} at the periphery of the bolt cross-section resulting from direct tension plus bending and neglecting stress concentrations shall not exceed $3S_b$. A lower value may be applicable for high strength steel bolts.

C.3.5.2 Welding of bolts

These rules are not applicable if any bolts which will be subjected to fluctuating stress are welded.

C.3.5.3 Use of fatigue design curves

The method of analysis is as described in **C.3.2**.

C.4 Recommendations for reducing risk of fatigue at weld toe

Fatigue cracks readily initiate at weld toes on stressed members, partly because of the stress concentration resulting from the weld shape but chiefly because of the presence of inherent flaws. For members at least 6 mm thick, the fatigue lives of welds which might fail from the toe may be increased (see Table C.2 for details) by locally machining and grinding the toe to reduce the stress concentration and remove the inherent flaws, as follows.

The weld toe is machined using a rotating conical tungsten-carbide machining burr. In order to ensure that weld toe flaws are removed, the required depth of machining is 0.5 mm below any undercut (see Figure C.8). In addition, the root radius of the resulting weld toe groove, r, shall meet the following:

$$r \geq 0.25e \geq 4d$$

The area should be inspected using dye penetrant or magnetic particles. Such inspection is facilitated if the machined toe is ground using emery bands, a measure which is also beneficial from the fatigue viewpoint. The resulting profile should produce a smooth transition from the plate surface to the weld, as shown in Figure C.8, with all machine marks lying transverse to the weld toe.

The above technique is particularly suitable for treating weld toes. The ends of short or discontinuous welds can only be treated effectively if the weld can be carried around the ends of the attachment member to provide a distinct weld toe.

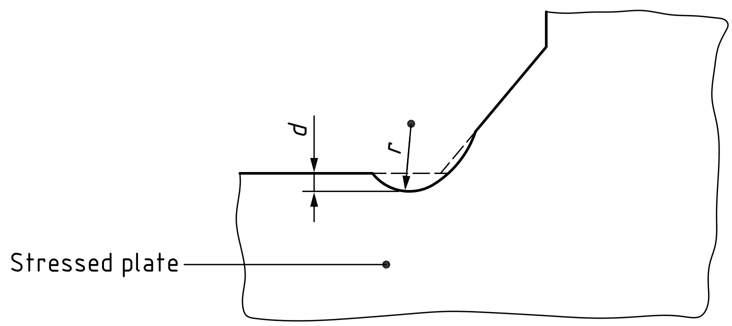
Weld toe dressing inevitably results in a region of reduced plate thickness. However, the stress concentration associated with this feature is less severe than the original weld and so, from the fatigue viewpoint, the loss of section in the weld toe region is acceptable. However, it should be considered when calculating the minimum pressure thickness.

As an alternative to burr grinding, TIG or plasma dressing may be used 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, as described in BS 7608:2014+A1, Annex F.

Toe dressing only affects the fatigue strength of a welded joint from the point of view of failure from the weld toe. The possibility of fatigue crack initiation from other features of the weld (e.g. weld root in fillet welds) shall not be overlooked.

Weld toe dressing cannot be assumed to be effective in the presence of a corrosive environment which can cause pitting in the dressed region.

Figure C.8 **Weld toe dressing**



d = depth of grinding below undercut = 0.5 mm

C.5 Bibliography

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Annex D Requirements for vessels designed to operate below 0°C

D.1 Introduction

This annex provides, when applied along with the appropriate requirements of the main text, measures for the avoidance of brittle fracture. They take into account good engineering practices, which have developed in the pressure vessel and other industries to ensure that materials and design details are adequate to resist brittle fracture under the design conditions specified. The requirements also take into account a broad spectrum of results from experimental test data.

Where it is found difficult to meet the requirements of this specification using the criteria specified, alternative methods of assessment (e.g. fracture mechanics as outlined in Annex U) are permitted to be used by agreement between the purchaser, the manufacturer and the Inspecting Authority.

NOTE Whilst this specification covers the requirements for the design and construction of new pressure vessels, the principles of this annex may, with the agreement of the inspecting/insuring authority, be used for the assessment of vessels in service.

D.2 Scope

The requirements of this annex apply to the design, materials and manufacturing processes of vessels which have a minimum design temperature, θ_D, less than 0 °C.

These requirements apply to all pressure parts and attachments welded thereto but not to non-pressure parts such as internal baffles, etc. that are not welded to a pressure part and are not otherwise an integral part of a pressure part.

The application of this annex is limited to ferritic steels in groups 1, 2 and 4 inclusive except that rimming steels shall not be used at minimum design temperatures below 0 °C.

General requirements for materials for use in low temperature applications are given in **2.2**. Annex U provides recommendations for the application of fracture mechanics for cases outside the scope of this annex.

Items 18 and 19 of **K.1.4.2** restrict or qualify the use of some ferritic steel in groups 1 and 4 for vessels designed to operate below 0 °C.

D.3 Definitions

D.3.1 Design reference temperature, θ_R

A characteristic temperature, based upon the minimum design temperature, the membrane stress level, the construction category and the extent of post-weld heat treatment.

D.3.2 Design reference thickness, *e*

A characteristic dimension of the material, component or weld.

D.3.3 Required impact energy

The value of Charpy V energy required by this annex to be demonstrated for the material, component or welds of a vessel to this specification.

D.3.4 Required impact test temperature, *RITT*

The temperature at which the required impact energy is to be achieved.

D.3.5 Material impact energy

The value of Charpy V energy demonstrated for a material, component or weld.

D.3.6 Material impact test temperature

The temperature at which the material's impact energy has been determined.

NOTE The above required impact energy and required impact test temperature are minimum and maximum values, respectively, that can be satisfied by more onerous values, i.e. by demonstrating a higher material impact energy value or from testing at a lower material impact test temperature level.

D.4 Methodology

The following procedure is aimed at defining the necessary impact requirements (i.e. the minimum Charpy V impact energy and the temperature at which it is to be demonstrated) to provide protection against brittle fracture.

For each material, component and weld in a vessel:

a) define the:

  1) minimum design temperature;

  2) maximum calculated tensile membrane stress;

  3) construction category;

  4) sequence of assembly and the post-weld heat treatment (PWHT) arrangements;

b) determine the design reference temperature in accordance with **D.5.1**;

c) determine the design reference thickness in accordance with **D.5.2**;

d) determine the required impact test temperature in accordance with **D.6.1** using the above design reference temperature and design reference thickness along with either Figure D.1, Figure D.2 or the associated equation;

e) determine the required impact energy from **D.6.2**;

f) establish actual material impact properties, conforming to **D.7**, from appropriate existing test data (i.e. from material test certificates for parent materials) or through tests (i.e. to **D.7.2** for welds);

g) compare actual data from f) with requirement defined in d) and e).

The requirements above shall be satisfied in conjunction with conformance to the design, manufacturing and workmanship requirements of **D.8**.

Figure D.1 **Permissible design reference temperature/design reference thickness/required impact test temperature relationships for as-welded components (see also Table D.2, Note 2)**

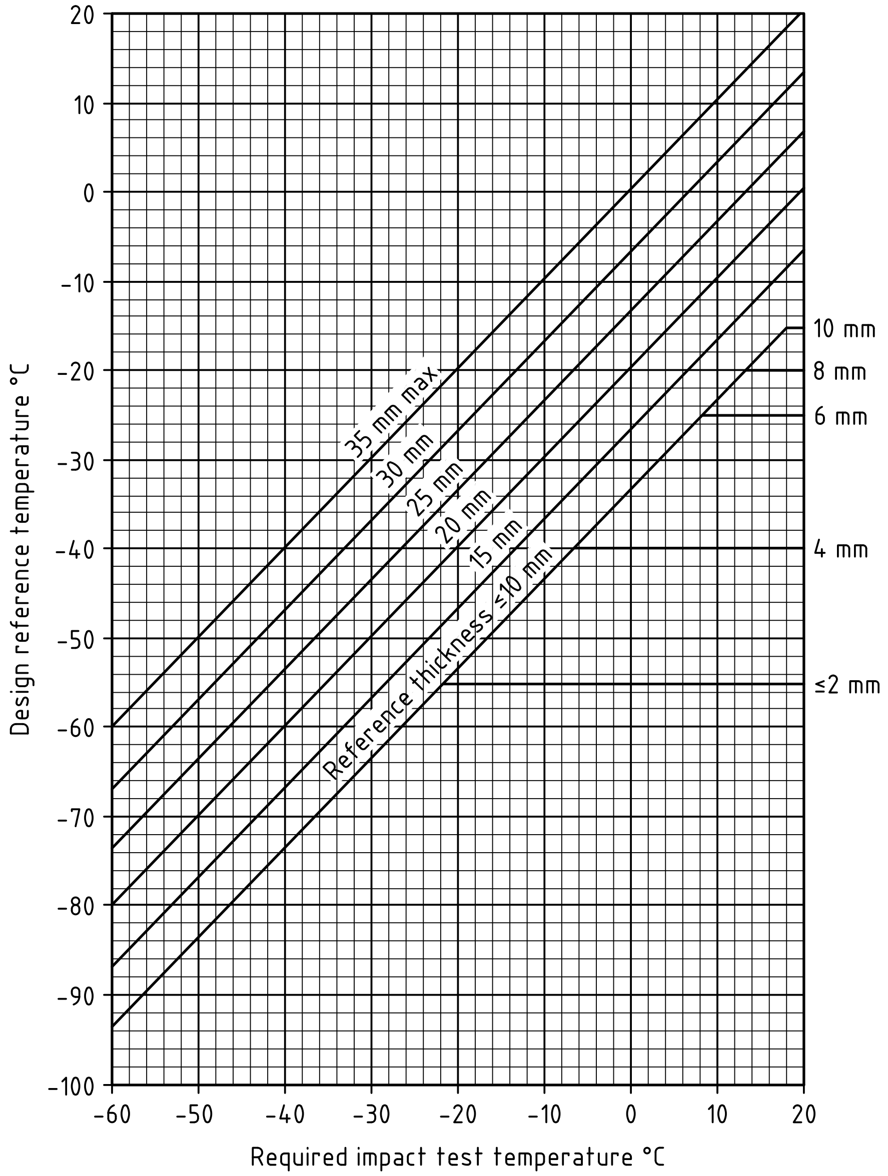
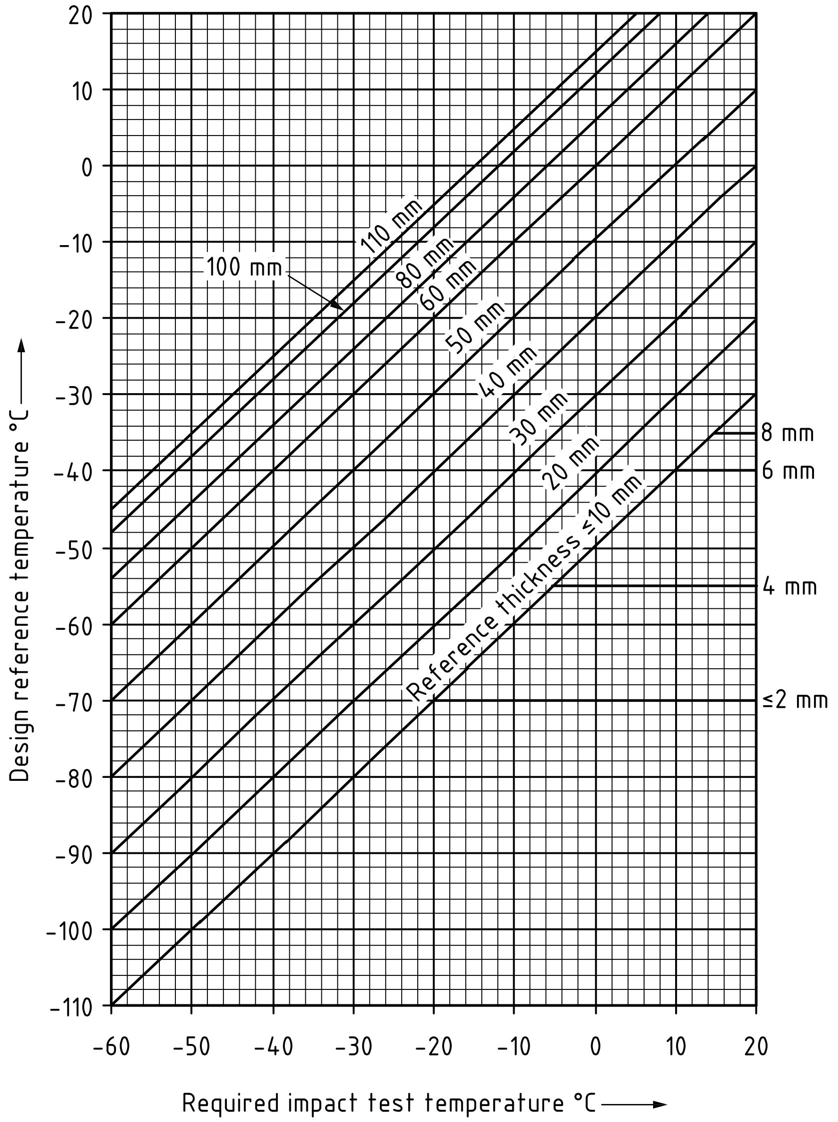


Figure D.2 **Permissible design reference temperature/design reference thickness/required impact test temperature relationships for post-weld heat-treated components (see also Table D.2, Note 2)**



D.5 Determination of reference parameters

D.5.1 Calculation of design reference temperature

D.5.1.1 General rules for calculation of design reference temperature

The design reference temperature, θ_R, shall not be greater than the minimum design temperature adjusted, as appropriate, as follows:

$$\theta_R \leq \theta_D + \theta_S + \theta_C + \theta_H$$

where

θ_D is the minimum design temperature as defined in **3.2.5**;

θ_S is an adjustment depending on the calculated membrane stress, as follows:

θ_S is 0°C when the calculated tensile membrane stress is equal to or exceeds $2f_N/3$ (where f_N is as defined in **2.3.1.1**);

θ_S is +10°C when the calculated tensile membrane stress is equal to or exceeds 50 N/mm^2 but does not exceed $2f_N/3$;

θ_S is +50°C when the calculated tensile membrane stress does not exceed 50 N/mm^2. In this case the membrane stress should take account of internal and external pressure, static head and self-weight;

θ_C is an adjustment depending upon the construction category, as follows:

θ_C is 0°C for category 1 components;

θ_C is –10°C for category 2 components;

θ_H is an adjustment in applications where all plates incorporating sub-assemblies are post-weld heat treated before they are butt-welded together, but the main seams are not subsequently post-weld heat treated. In these applications θ_H is +15°C.

NOTE 1 The calculated tensile membrane stress is the maximum calculated value of tensile stress that falls within the general primary membrane stress category, as defined in ***A.3.4.2.1****. It excludes local membrane stresses, bending stresses and secondary stresses.*

NOTE 2 In cases where the calculated tensile membrane stress can vary with the minimum design temperature, e.g. auto-refrigeration during depressurization, the coincident values of θ_D and θ_S should be evaluated, allowing, where appropriate, for the possibility of repressurization while still cold (e.g. hydraulic overfill). The condition that results in the lowest value of θ_R should be used for the purpose of selection of materials.

D.5.1.2 Rules for heat exchanger tubes

The design reference temperature for heat exchanger tubes shall not be lower than the values given in Table D.1.

Table D.1 **Design reference temperature for heat exchanger tubes (°C)**

Design reference thickness	**BS EN 10216-1: P195TR2, P235TR2, P265TR2 BS EN 10216-2: P195GH, P235GH, P265GH BS EN 10217-1: P195TR2, P235TR2, P265TR2 BS EN 10217-2: P195GH, P235GH, P265GH ASTM A179, ASTM A192, ASTM A214, ASTM A334-6**		
mm	**As welded**	**Welded and PWHT**	**Unwelded**
10	−15	−30	−70
8	−20	−35	−75
6	−25	−40	−80
4	−40	−55	−95
2	−55	−70	−110

D.5.2 Calculation of design reference thickness

D.5.2.1 General

The design reference thickness of the vessel component materials and welds shall be determined from Table D.4 and by following the requirements given in **D.5.2.2** to **D.5.2.8**.

NOTE 1 In this clause, thickness refers to the nominal thickness (see ***1.6****). When the actual thickness used exceeds this value by more than the normal manufacturing tolerance, or where thicker material has been substituted for that ordered, special consideration should be taken, e.g. fracture mechanics analysis, as discussed in Annex U.*

NOTE 2 In this clause, where reference is made to Figure D.1 and Figure D.2, the related equations given in ***D.6.1*** *may also be used.*

D.5.2.2 Butt welded components

The design reference thickness of each component shall be taken as the thickness of the component under consideration at the edge of the weld preparation, as shown in item 1 of Table D.4. For the thicker component in the as-welded condition, this thickness shall be used in Figure D.1, or the thickness remote from the weld shall be used in Figure D.2, whichever gives the lower impact test temperature.

D.5.2.3 Weld neck flanges, plate and slip-on (or hubbed) flanges, tubeplates and flat ends

The design reference thickness of each component and the welds shall be determined separately by considering the thickness of the components e_1, e_2 and e_3 as indicated for items 8 to 13 and 16 to 21 of Table D.4.

If the distance from a flange, tubeplate or flat end to a butt weld is more than four times the thickness of the butt weld (see items 10 and 20 of Table D.4), and for the construction details shown in items 11, 13, 18 and 19 of Table D.4, the design reference thickness of the flange, tubeplate or flat end for the as-welded condition shall be the thickness at the edge of the weld preparation. This thickness shall be used in Figure D.1, or one quarter of the thickness of the flange, tubeplate or flat end shall be used in Figure D.2, whichever gives the lower impact test temperature.

For the constuction details shown in items 14 and 15 of Table D.4, the design reference thickness of the blind flange, flat end or tubeplate shall be taken as one quarter of the thickness of the component for both the as-welded and post-weld heat treated conditions. For unwelded components, Figure D.2 shall be used to determine the impact test temperature. Where a blind flange or flat end is fitted with a welded branch or nozzle, the requirements of **D.5.2.4** are also applicable.

For the as-welded condition, the design reference thickness of tubeplates having tubes attached by welding (see item 21 of Table D.4) shall be the tube thickness. This thickness shall be used in Figure D.1, or one quarter of the thickness of the tubeplate shall be used in Figure D.2, whichever gives the lower impact test temperature.

NOTE Where the shell to tubeplate joint is stress relieved but the tube/tubeplate joint is as-welded, this may affect the selection of materials for the tubeplate.

D.5.2.4 Branches, nozzles and compensating plates

For items 2 to 6 of Table D.4, the design reference thickness of each component shall be determined separately by considering only the thickness of that component, as given by e_1, e_2 and e_3 in Table D.4.

Where butt-welded inserts are used (see item 7 of Table D.4), the design reference thickness of the branch or nozzle shall correspond to the greater of the thickness e_3 at the edge of the weld preparation or one quarter of the thickness e_1 of the reinforced part of the branch or nozzle. For the as-welded condition this thickness shall be used in Figure D.1, or the thickness e_1 of the reinforced part of the branch shall be used in Figure D.2, whichever gives the lower impact test temperature.

NOTE For welded vessels where there is a direct welded connection between a compensation plate and shell plate (which may be due to the use of a full penetration weld as shown in item 3 of Table D.4) and where the sum of e_2 and e_3 exceeds 40 mm, consideration should be given to the need for post weld heat treatment.

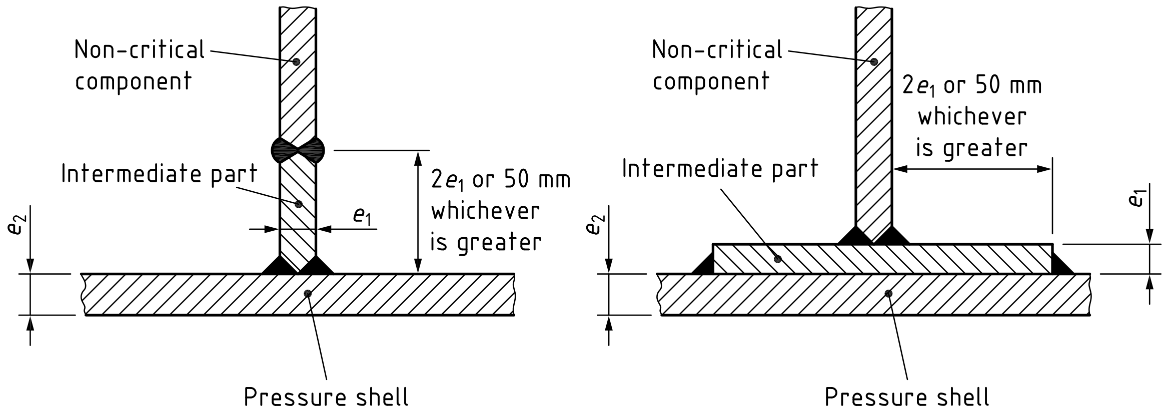
D.5.2.5 Tubes

The design reference thickness shall be that of the nominal thickness of the tube including corrosion allowance.

D.5.2.6 Attachments

Attachments welded directly to a pressure component shall be regarded as part of the pressure component, and the design reference thickness shall be that of the shell or of the attachment at the point of attachment whichever is thicker. Intermediate attachments (see Figure D.3) shall be employed where it is required to attach non-critical components to the shell.

Figure D.3 **Examples of details for attaching non-critical components to pressure shell**



D.5.2.7 Unwelded items

Unwelded items shall be taken as stress relieved and the design reference thickness shall be taken as one quarter of the thickness of the item.

D.5.2.8 Welds

The reference thickness for a range of weld details is given in Table D.4.

D.6 Required impact parameters

D.6.1 Required Charpy impact test temperature

The impact test temperature shall be determined either graphically or by calculation from Figure D.1 or Figure D.2 or their related equations. The figures and equations relate the design reference temperature, the design reference thickness and the required impact test temperature for the as-welded and the post-weld heat treated conditions respectively.

If the design reference temperature and the design reference thickness are known Figure D.1 and Figure D.2 or their related equations can be used to determine the required material impact test temperature.

Alternatively, if the actual impact test temperature for the material is known the figures can be used to determine the minimum acceptable design reference temperature for any given design reference thickness, or the maximum acceptable design reference thickness for any given design reference temperature.

Figure D.1 and Figure D.2 are based upon the following equations when the reference thickness is not less than 10 mm:

for the "as welded" case, covered by Figure D.1:

$$\theta_R = RITT + \frac{4(e-35)}{3}$$

for the "PWHT" case, covered by Figure D.2:

$$\theta_R = \min[RITT - 60 + e\ ;\ RITT - 18 + 0.3e]$$

where

e is the component reference thickness (mm);

$RITT$ is defined at **D.3.4** (°C).

When the reference thickness exceeds 110 mm an alternative method of assessment shall be used (e.g. fracture mechanics as outlined in Annex U).

These equations may be used instead of the figures and are considered valid over the range of parameters covered by Figure D.1 and Figure D.2.

For all reference thicknesses below 10 mm, the horizontal line corresponding to the reference thickness shall be used except when the 10 mm line gives a lower value of design reference temperature (the horizontal lines in Figure D.1 and Figure D.2 are the values in Table D.3). When the 10 mm lines are used, the energy requirements at 10 mm shall be scaled down in proportion to specimen thickness.

D.6.2 Required impact energy values

Table D.2 defines the required minimum impact energy to be achieved at the required impact test temperature determined in accordance with **D.6.1**. However, impact testing is not required for materials, except heat exchanger tubes, with a design reference thickness of 10 mm or less provided that the design reference temperature is not lower than the appropriate value in Table D.3. Similarly, impact testing is not required for heat exchanger tubes, with a design reference thickness of 10 mm or less, provided that the design reference temperature is not lower than the appropriate value in **D.5.1.2** and Table D.1.

Table D.2 **Required impact energy**

Specified minimum strength	Required impact energy value at the required impact test temperature			
	10 mm × 10 mm	10 mm × 7.5 mm	10 mm × 5 mm	10 mm × 2.5 mm
N/mm^2	J	J	J	J
$R_m < 450$ and $R_e < 355$	27	22	19	10
$R_m \geq 450$ or $R_e \geq 355$	40	32	28	15

*NOTE 1 Where a material specification quotes, or a test in accordance with **D.7** results in, material impact energy at a test temperature different from the required impact test temperature in accordance with this annex, the impact energy data may be converted to a common temperature base on the basis of 1.5 J per °C when converting to a temperature lower than the actual impact test temperature, or 1.0 J per °C when converting to a temperature higher than the actual impact test temperature. Such conversion shall be permitted only in the range 18 J to 47 J of Charpy V impact energy. For example, 20 J at 0°C may be regarded as equivalent to 30.0 J at +10°C, and 47 J at −20°C may be regarded as equivalent to 32 J at −30°C. The 1.5 J per °C and 1.0 J per °C relationships may also be utilized to determine the temperature adjustment when seeking to compare data at common impact energy levels.*

NOTE 2 For non-impact tested grades of standard steels listed in Table K.1-1 it may be assumed that a satisfactory impact value has been achieved at +20°C. However, see the Notes to Table K.1-2 to Table K.1-12: Note 18 prohibits certain steels for applications below 0°C; Note 19 requires certain steels to be impact tested, to the requirements of Table D.1 if they are to be used below 0°C, whether or not impact testing is normally required.

NOTE 3 This annex was originally based upon historical data from tests with the specimen in the longitudinal direction with respect to the rolling direction. BS EN and other European Standards specify that specimens should be transverse, which is more conservative. The requirements of this annex are now based on the use of transverse test specimens, notched parallel to the primary rolling direction. Many pressure vessel steels are now cross rolled, however if data from longitudinal specimens are used, the required impact temperature should be reduced by 10 °C.

Table D.3 **Minimum design reference temperature for omission of impact test (as allowed by D.6.2)**

Design reference thickness mm	As welded	PWHT
10	−15°C	−30°C
8	−20°C	−35°C
6	−25°C	−40°C
4	−40°C	−55°C
≤2	−55°C	−70°C

Table D.4 **Reference thickness of weld joint components *e***

Dimensions in mm

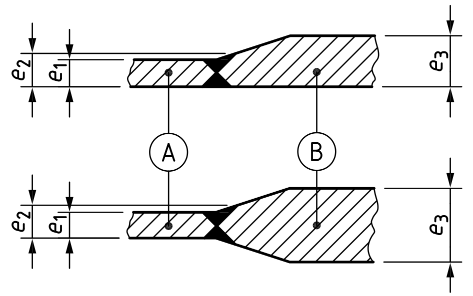
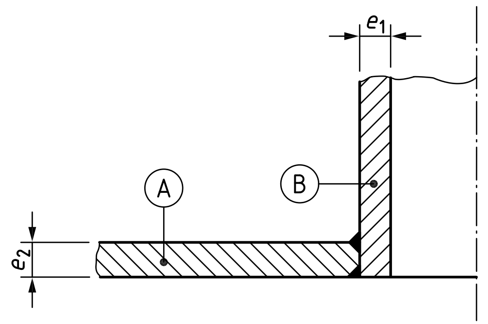
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness e		
			Part A	**Weld**	**Part B**
1		A-W	e_1	e_2	e_2 check e_3 in Figure D.2[a]
		PWHT	e_1	e_2	e_3
2	Branches and nozzles 	A-W	e_2	e_2	e_1
		PWHT	e_2	e_2	e_1

Table D.4 **Reference thickness of weld joint components *e* (continued)**

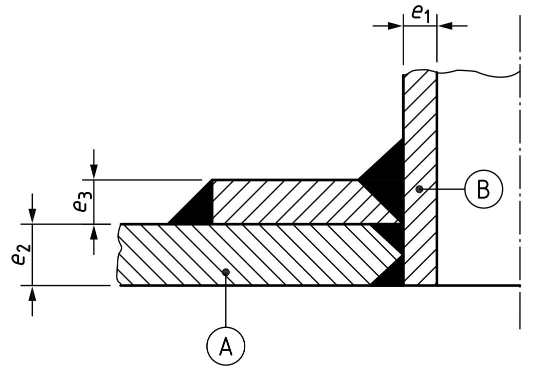
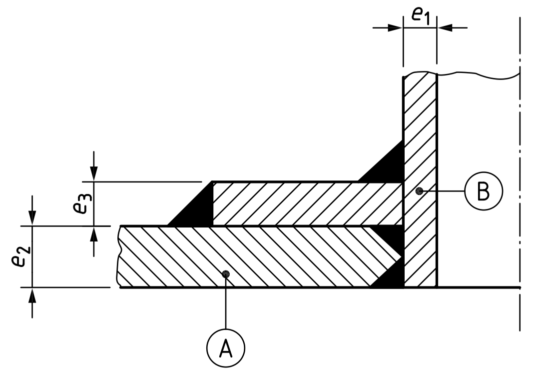
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
3		A-W	e_2	max(e_2 ; e_3)	e_1
		PWHT	e_2	max(e_2 ; e_3)	e_1
4		A-W	e_2	max(e_2 ; e_3)	e_1
		PWHT	e_2	max(e_2 ; e_3)	e_1

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

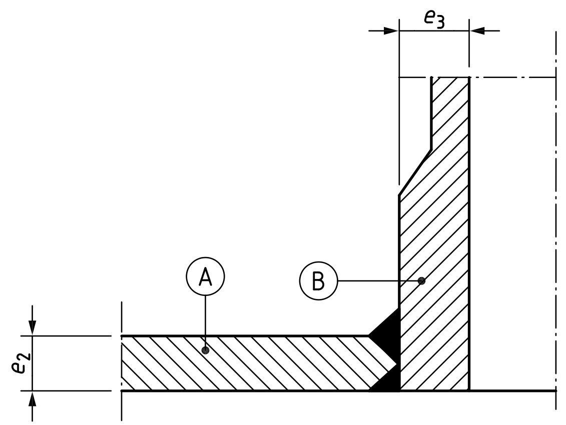
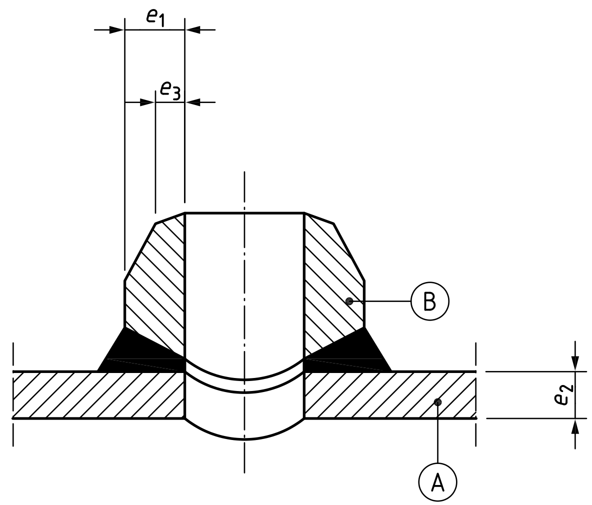
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
5		A-W	e_2	max(e_2 ; e_3)	e_3
		PWHT	e_2	max(e_2 ; e_3)	e_3
6		A-W	e_2	e_1	e_1
		PWHT	e_2	e_1	e_1

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

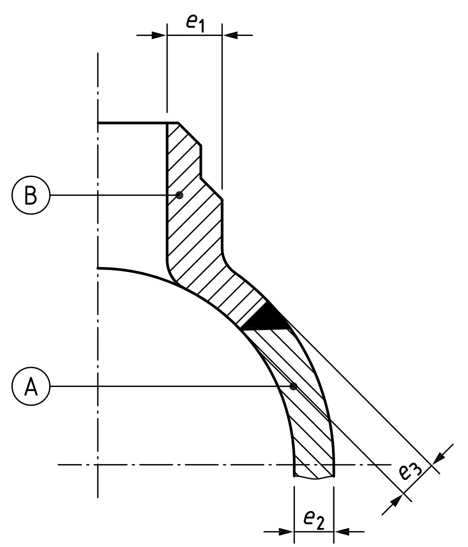
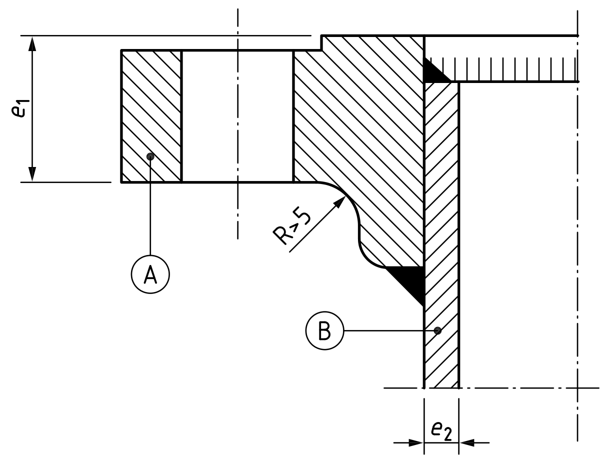
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
7		A-W	e_2	e_3	max(e_3 ; e_1/4) check e_1 in Figure D.2[b]
		PWHT	e_2	e_3	max(e_3 ; e_1/4)
8	Slip-on and plate flange 	A-W	max(e_2 ; e_1/4)	max(e_2 ; e_1/4)	e_2
		PWHT	max(e_2 ; e_1/4)	max(e_2 ; e_1/4)	e_2

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

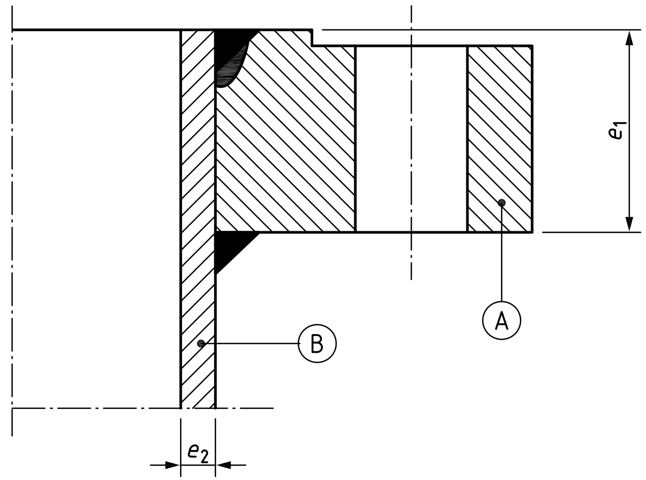
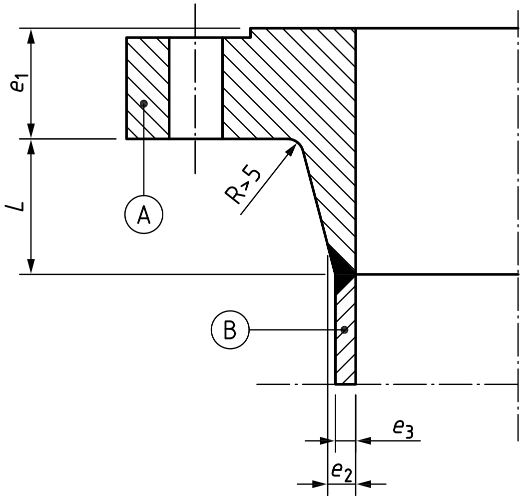
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e* Part A	Weld	Part B
9		A-W	max(e_2 ; e_1/4)	max(e_2 ; e_1/4)	e_2
		PWHT	max(e_2 ; e_1/4)	max(e_2 ; e_1/4)	e_2
10	Forged or cast welding neck flanges 	A-W with $L < 4e_2$	max(e_2 ; e_1/4)	e_2	e_3
		A-W with $L \geq 4e_2$	e_2 check e_1/4 in Figure D.2[c]	e_2	e_3
		PWHT	max(e_2 ; e_1/4)	e_2	e_3

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

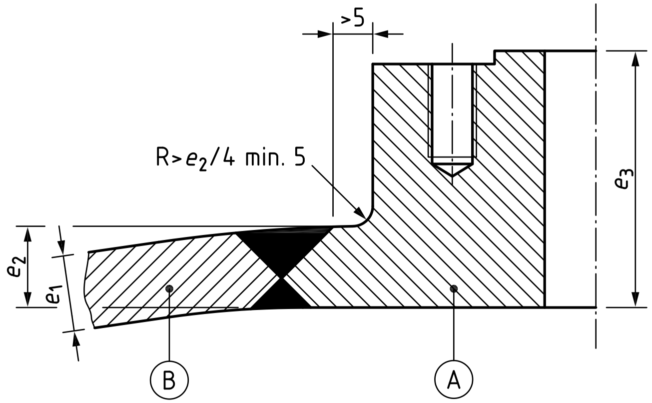
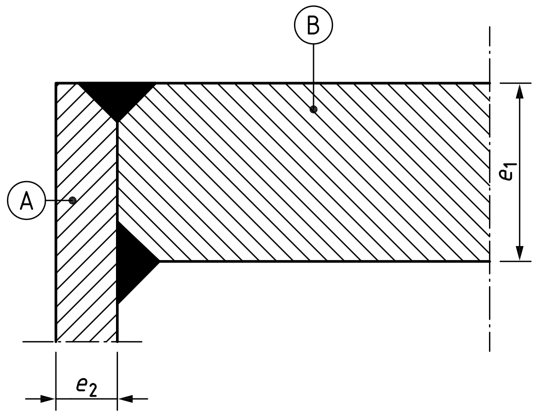
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
11	Pad-type flanges 	A-W	e_2 check $e_3/4$ in Figure D.2[d]	e_2	e_1
		PWHT	max(e_2 ; e_3/4)	e_2	e_1
12	Flat ends 	A-W	e_2	e_2	max(e_2 ; e_1/4)
		PWHT	e_2	e_2	max(e_2 ; e_1/4)

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

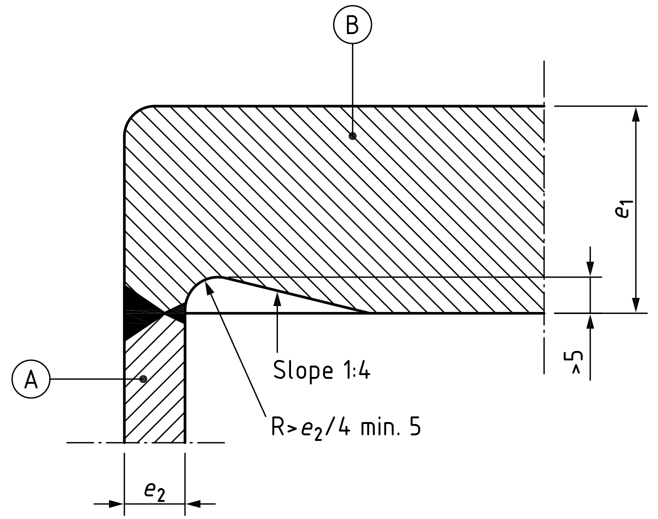
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
13		A-W	e_2	e_2	e_2 check e_1/4 in Figure D.2 c
		PWHT	e_2	e_2	max(e_2 ; e_1/4)

Table D.4 **Reference thickness of weld joint components *e* (continued)**

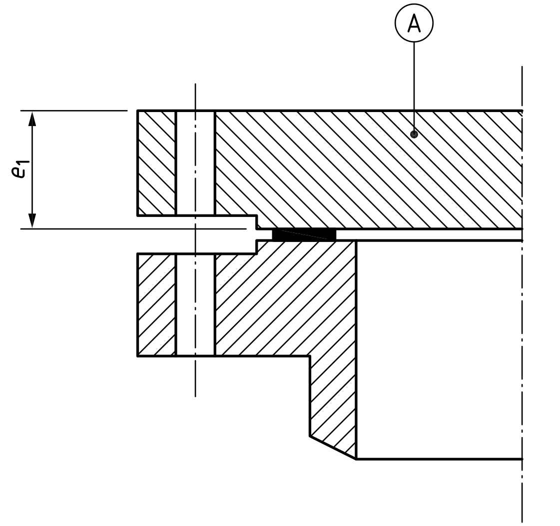
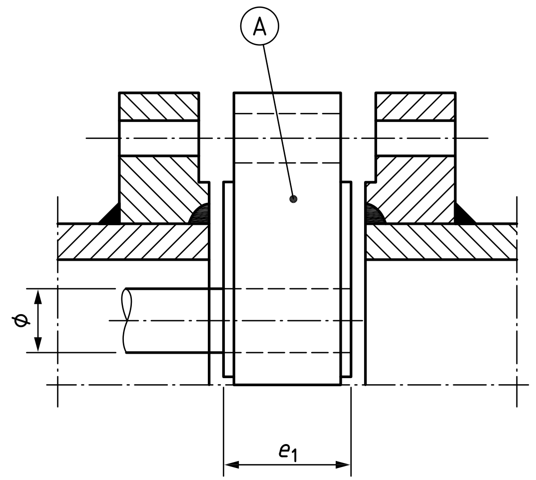
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
14	Covers and blind flanges 	A-W	$e_1/4$[e]	—	—
		PWHT	$e_1/4$	—	—
15	Tube plates 	A-W	$e_1/4$[e]	n.a.	n.a.
		PWHT	$e_1/4$	n.a.	n.a.

Table D.4 **Reference thickness of weld joint components *e* (continued)**

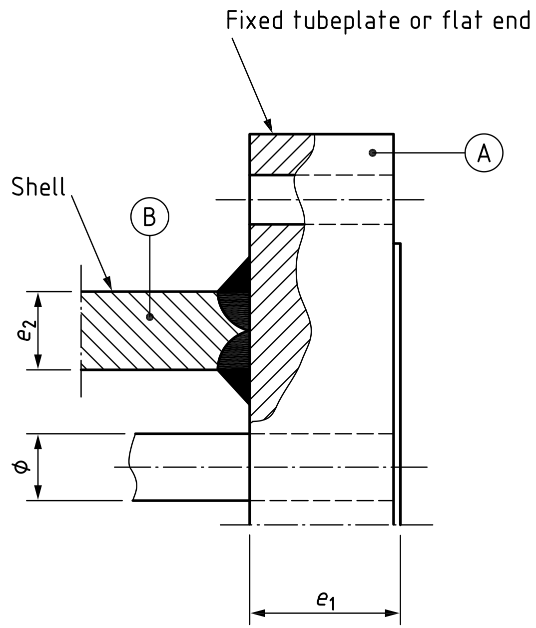
Item No.	**Construction detail**	**As-Welded (A-W) or PWHT**	**Reference thickness *e***		
			Part A	**Weld**	**Part B**
16		A-W	max(e_1/4 ; e_2)	e_2	e_2
		PWHT	max(e_1/4 ; e_2)	e_2	e_2

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

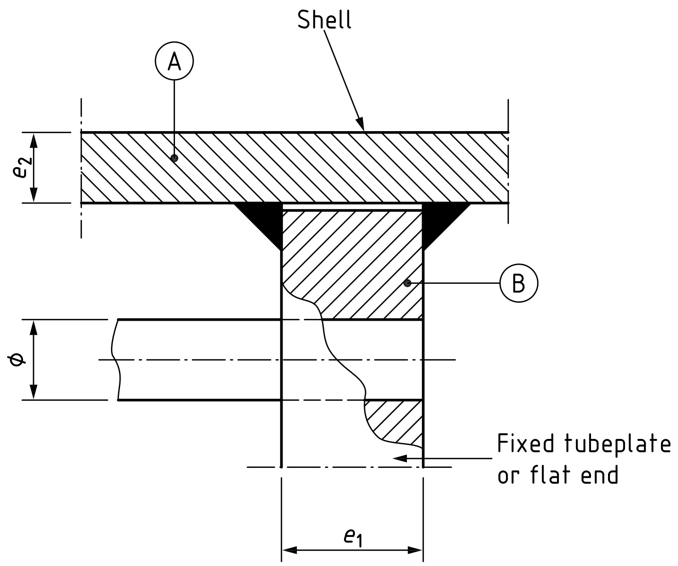
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e* Part A	Weld	Part B
17	Welded into shell/channel (preferably not to be used) 	A-W	e_2	e_2	max(e_1/4 ; e_2)
		PWHT	e_2	e_2	max(e_1/4 ; e_2)

Table D.4 **Reference thickness of weld joint components *e* (continued)**

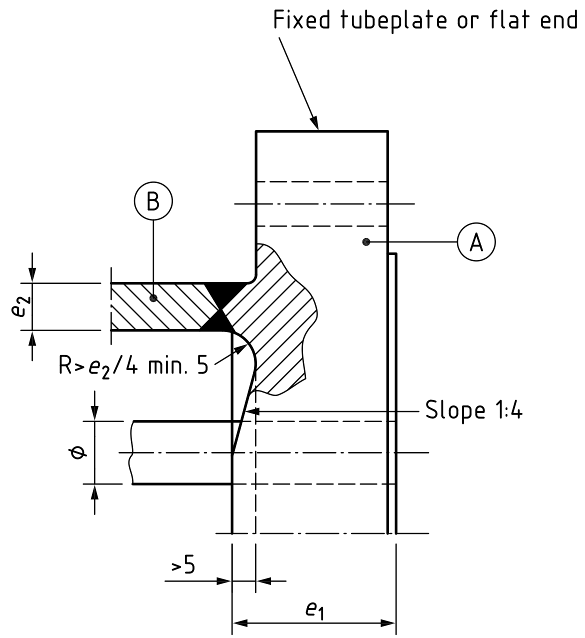
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	**Weld**	**Part B**
18	Forged tube plate with stubs 	A-W	e_2 check $e_1/4$ in Figure D.2 c	e_2	e_2
		PWHT	max($e_1/4$; e_2)	e_2	e_2

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

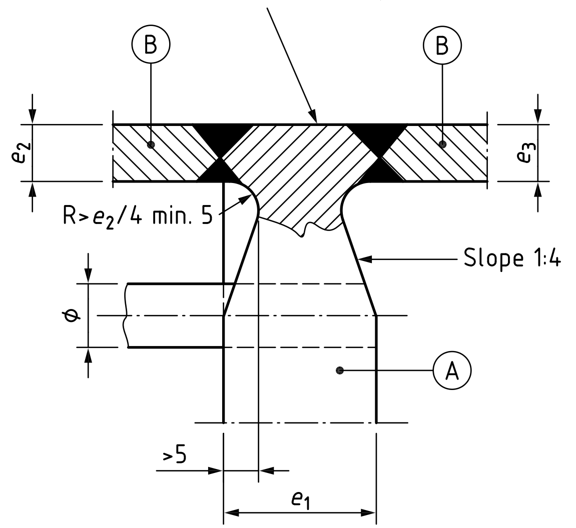
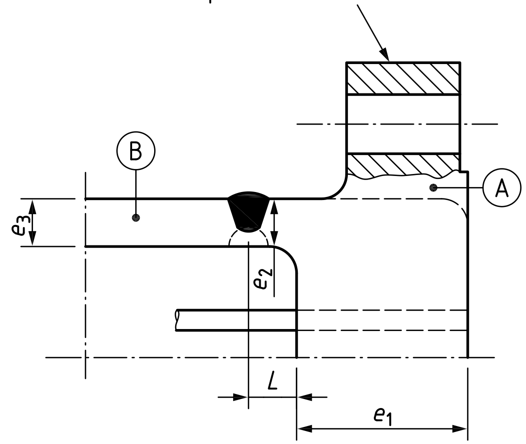
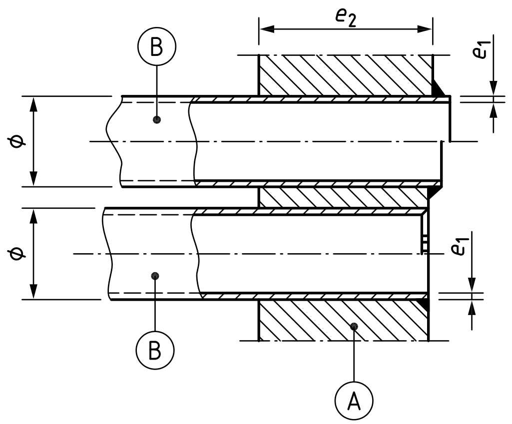
Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
19		A-W	max(e_2 ; e_3) check e_1/4 in Figure D.2 [f]	max(e_2 ; e_3)	max(e_2 ; e_3)
		PWHT	max(e_1/4 ; e_2 ; e_3)	max(e_2 ; e_3)	max(e_2 ; e_3)
20		A-W with $L < 4e_2$	max(e_2 ; e_1/4)	e_2	e_3
		A-W with $L \geq 4e_2$	e_2 check e_1/4 in Figure D.2 [c]	e_2	e_3
		PWHT	max(e_2 ; e_1/4)	e_2	e_3

Table D.4 **Reference thickness of weld joint components *e*** *(continued)*

Item No.	Construction detail	As-Welded (A-W) or PWHT	Reference thickness *e*		
			Part A	Weld	Part B
21	Tube-to-tube plate connection  	A-W	e_1 Check $e_2/4$ in Figure D.2[g]	e_1	e_1
		PWHT	max(e_1 ; $e_2/4$)	e_1	e_1

[a] The minimum test temperature of the conditions: e_2 (A-W), e_3 (PWHT) shall be taken.
[b] The minimum test temperature of the conditions: max (e_3 ; $e_1/4$) (A-W), e_1 (PWHT) shall be taken.
[c] The minimum test temperature of the conditions: e_2 (A-W), $e_1/4$ (PWHT) shall be taken.
[d] The minimum test temperature of the conditions: e_2 (A-W), $e_3/4$ (PWHT) shall be taken.
[e] For unwelded components, the test temperature of the condition $e_1/4$ (PWHT) shall be taken.
[f] The minimum test temperature of the conditions: max(e_2 ; e_3) (A-W), $e_1/4$ (PWHT) shall be taken.
[g] The minimum test temperature of the conditions: e_1 (A-W), $e_2/4$ (PWHT) shall be taken.

D.7 Material impact testing requirements

D.7.1 Testing of plates, forgings, castings and tubes

Material or component properties shall result from Charpy tests on V-notched test pieces of 10 mm, 7.5 mm, 5 mm or 2.5 mm width, tested in accordance with the requirements of the relevant material specification for parent metal. For rolled plate material the test specimens shall be transverse to the primary rolling direction, and notched parallel to the primary rolling direction.

For the application of this annex it is permissible to adopt impact test temperatures other than those specified in the relevant material specification (see Note 1 to Table D.2). Unless stated otherwise material impact energy is the average of the results of tests made on three test pieces. Unless otherwise specified in the relevant material specification, no individual value is permitted to be less than 70 % of the material impact energy.

NOTE Alternative toughness requirements may be established by reference to Annex U when so agreed between purchaser and manufacturer.

D.7.2 Testing of welds and heat affected zones

D.7.2.1 General

When materials to be joined by welding are not required by this specification to be impact tested, impact tests are not required to be undertaken on the welding procedure test plates and production weld test plates are not required. Where impact tested materials are to be joined by welding the weld properties shall result from Charpy tests on V-notched test pieces of 10 mm, 7.5 mm, 5 mm or 2.5 mm width, tested in accordance with BS EN ISO 148-1 and as follows.

D.7.2.2 Weld test plates

As detailed in this clause, additional Charpy V-notch impact tests shall be made on procedure and production weld test plates produced in accordance with Section 5 and Annex Q.

All test specimens shall be prepared after the test plates have been given a heat treatment that is the same as that which will be applied to the vessel. If the production test plates are required to be heat treated with the vessel, this shall be specified by the purchaser in the purchase specification.

a) *Procedure test plates.* Impact tests are required on procedure test plates. Alternatively the purchaser may state in the purchase specification that authenticated results of previous tests are acceptable.

b) *Production weld test plates*. Test plates are required when:

    1) specified by the purchaser in the purchase specification; or

    2) $(\theta_D - \theta_P) < 20°C$;

where

θ_P (the permissible minimum temperature) is the minimum temperature for which the vessel will be suitable;

θ_D is the minimum design temperature as defined in **3.2.5**.

θ_P can be determined by a calculation, reversing the sequence given in **D.4**, using actual material data and **D.6.1** as appropriate, to establish the lowest temperature for which each component of the vessel will be suitable. The highest of these temperatures is the permissible minimum temperature of the vessel.

NOTE Substituting (θ_D − 20) °C for θ_D, when initially determining θ_R, in accordance with ***D.5.1.1****, for the purposes of material selection, will ensure a minimum margin of 20 °C between required and permissible temperatures.*

Unless the purchaser specifies additional tests in the purchase specification the test plates shall only be subject to impact testing.

Impact testing of production weld test plates may be waived, by agreement between the manufacturer, purchaser and Inspecting Authority, in the case of welded seams made by a manual or multi-run automatic welding process in vessels made from steels for which impact testing has been waived in accordance with **D.6.2**.

Impact testing of production test plates is not required in the case of welds in materials less than 10 mm thick.

D.7.2.3 Positions of impact test specimens

All specimens shall be cut transverse to the weld with the axis of the notch perpendicular to the surface of the plate. The tests shall be done on sets of three specimens as follows.

a) *As-welded vessels*

Weld metal test pieces shall be cut so that one face of the specimen is substantially parallel to and within 3 mm of, the top surface of the weld (see Figure D.4).

NOTE Test pieces may also be taken from the root of the weld, at the purchaser's or Inspecting Authority's request, but these should be for information purposes only.

b) *Stress relieved vessels*

The number of sets of tests on the weld metal shall be related to the thickness of the test plates as follows:

Plate thickness	Number of sets
Up to 30 mm	1
30 mm to 62 mm	2
Over 62 mm	3

At least one set of test pieces shall be taken with the notch at the root of the weld (two if the root is ill-defined). The other sets shall be distributed so as to give a measure of the properties at different positions through the thickness (see Figure D.5).

c) *Heat affected zones*

No impact tests are specified for the heat affected zone when multi-run processes are used with heat inputs between 1 kJ/mm and 5 kJ/mm.

If a heat input outside this range is used and the weld has not been normalized, the heat affected zone shall be impact tested. Where impact tests are specified on the heat affected zone, the specimens detailed in a) or b) shall be duplicated but with their notches located in the heat affected zone and 1 mm to 2 mm from the fusion boundary. Individual specimens shall be etched to show the fusion boundary and heat affected zone so as to ensure accurate location of the notch (see Figure D.6).

Figure D.4 **Location of Charpy V-notch specimens in weld metal (as-welded vessels)**

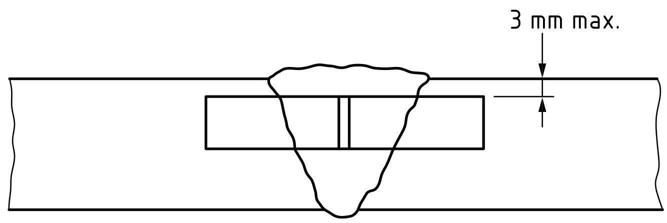


Figure D.5 **Location of Charpy V-notch specimens in weld metal (stress relieved vessels)**

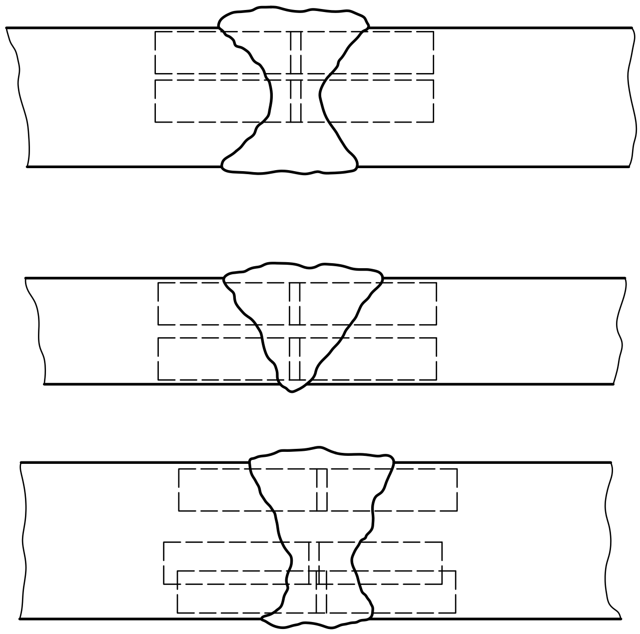
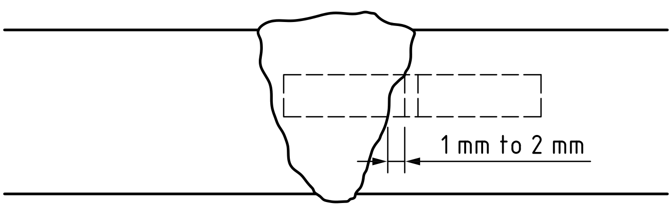


Figure D.6 **Location of Charpy V-notch specimens in heat affected zone**



D.7.2.4 Retests

If the required impact energy is not attained or if one specimen only shows a value less than 70% of the specified impact energy, then three additional specimens shall be selected from a position similar to that from which the set of specimens under consideration was taken. The average value of the six specimens shall be not less than required impact energy and not more than two specimens shall show values below the required impact energy, only one of which is permitted to be below the specified individual value.

D.8 Design, manufacture and workmanship

D.8.1 Design

Each design shall provide sufficient flexibility and be as simple as possible. The occurrence of rapid changes in temperature likely to give rise to severe temperature gradients shall be avoided but where this is not possible, consideration shall be given to special design details.

NOTE 1 A typical desirable design detail is given in Figure D.7 as an illustration.

Details that will produce local areas of high stress, e.g. lugs, gussets producing discontinuous stiffening and abrupt structural changes, shall not be permitted. Discontinuous stiffeners or continuous stiffeners attached by tack or intermittent welding shall not be used. Saddle supports for vessels shall not be welded directly to vessels; doubling plates shall always be used (see **D.5.2.6**).

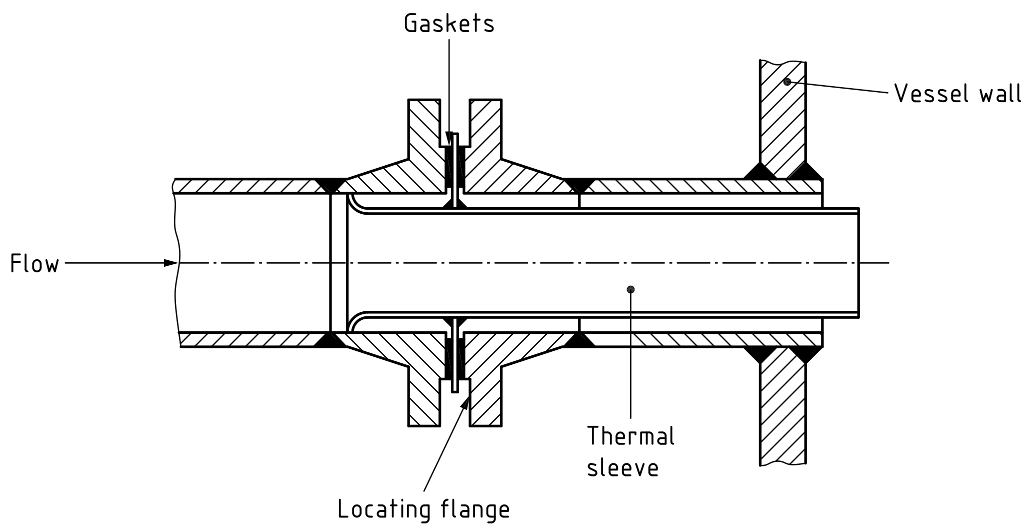
Pipe supports and anchors shall be attached to an encircling mechanically separate sleeve.

NOTE 2 Screwed connections and socket-welded valves and fittings should preferably not be used.

Nozzles and complicated structural attachments shall be welded to shell plates in the workshop and be considered as a separate sub-assembly, which may also be evaluated individually with regard to the desirability of a separate heat treatment.

Welded tubesheet to shell, and flat end plate to shell attachments shall be generally in accordance with Figure E.39 to Figure E.44 inclusive. Such attachments shall conform to Figure E.41a) or Figure E.41b), the prolongation of the tubeplate to provide a bolting flange being optional.

Figure D.7 **Example of detail for avoidance of severe thermal gradients**



D.8.2 Manufacture

All materials used shall be as specified. Pieces of plate, etc. of uncertain origin shall not be used.

Hard stamping is only permitted for the purposes of plate identification and shall be kept to a minimum. Only round nosed stamps shall be used. Marking for vessel identification is specified in **5.8.9**.

D.8.3 Heat treatment of components after forming

All plates that have been cold formed to an internal radius less than 10 times the plate thickness (more than 5% deformation) shall be given a normalizing treatment afterwards.

Cold formed dished ends with flanges shall be normalized; plates that are cold pressed to form the segments of a sphere or a hemispherical end shall be normalized if the radius is less than 10 times the thickness. Normalizing is required in all other cases, except where the manufacturer produces evidence that the forming technique used does not significantly change the impact properties.

Pipe that has been locally bent (with or without local heating) to an internal radius less than 10 times the outside diameter of the pipe shall be normalized. Unless it can be demonstrated that the temperature control during the forming operation is equivalent to the normalizing procedure, ferritic steel parts that have been hot formed shall always be normalized afterwards.

D.8.4 Welding

Because the notch ductility of weld deposit depends upon the technique used, the procedure used in making the production joints shall be the same as that used for the weld procedure test subject to the variables permitted by BS EN ISO 15614-1.

Annex E Recommendations for welded connections of pressure vessels

E.1 Typical details for principal seams

The details indicated in this clause have given satisfactory results under specific manufacturing conditions and are included for general guidance. Modification may be required to suit particular manufacturing techniques and all details adopted have to be shown by the manufacturer to produce satisfactory results by the procedure specified in Section 4.

Where no root gap is shown it is intended that the joints be close butted. For requirements governing the use of backing strips see **4.3.6.2**.

The following details are given:

a) butt welds using the manual metal-arc process (see Figure E.1);

b) circumferential butt welds where the second side is inaccessible for welding (see Figure E.2);

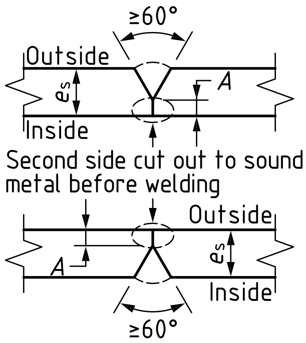
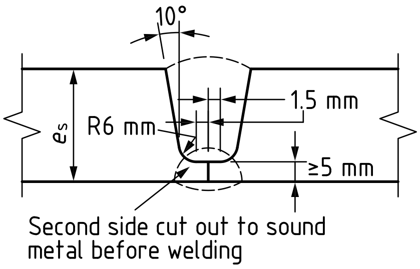
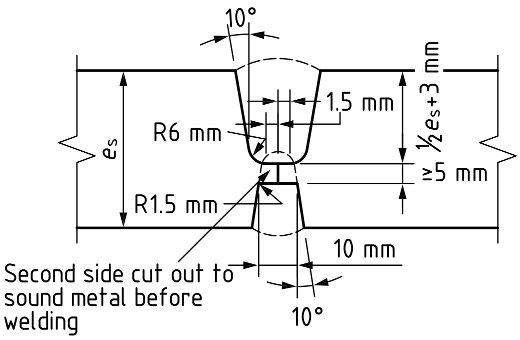
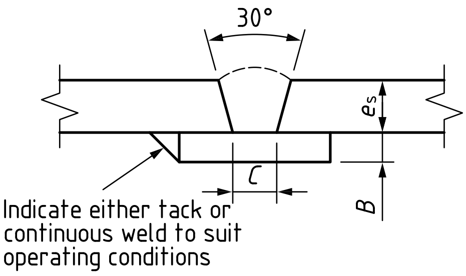
c) butt welds using the submerged arc welding process (see Figure E.3);

d) butt welds for manual inert gas welding (see Figure E.4);

e) circumferential lap welds (for category 3 vessels only) (see Figure E.5);

NOTE The thicknesses quoted in Figure E.1 to Figure E.5 are nominal thicknesses (see ***1.6****).*

Figure E.1 **Typical weld preparations for butt welds using the manual metal-arc process**

<table>
<tr><th>Figure</th><th>Joint</th><th>Name</th><th>Application</th></tr>
<tr><td>a)</td><td>1)
2) For smaller diameter vessels

</td><td>Double-welded butt joint with single “V”</td><td>Longitudinal and circumferential butt welds in plates not more than 20 mm thick.
The “V” should be on the inside of small diameter vessels as shown in (2) opposite.
A = 1.5 mm where e_s is less than 10 mm
A = 3 mm where e_s is 10 mm or over</td></tr>
<tr><td>b)</td><td>
</td><td>Double-welded butt joint with single “U”</td><td>Longitudinal and circumferential butt welds in plates where the thickness is greater than 20 mm</td></tr>
<tr><td>c)</td><td>
</td><td>Double-welded butt joint with double “U”</td><td>Longitudinal and circumferential butt welds where the thickness is greater than 20 mm</td></tr>
<tr><td>d)</td><td>

Weld dimensions are minima
<table>
<tr><th>Plate thickness e_s
mm</th><th>B
mm</th><th>C
mm</th></tr>
<tr><td>Up to 7.5</td><td>4.5</td><td>7.5</td></tr>
<tr><td>Over 7.5 to 12</td><td>6</td><td>9</td></tr>
<tr><td>Over 12</td><td>9</td><td>9</td></tr>
</table></td><td>Single-welded butt joint with backing strip (see **4.3.6.2**)</td><td>Longitudinal and circumferential butt welds. Backing strip to be removed after welding except where otherwise permitted in accordance with **4.3.6.2**</td></tr>
</table>

Figure E.2 **Typical weld preparations for circumferential welds where the second side is inaccessible for welding**

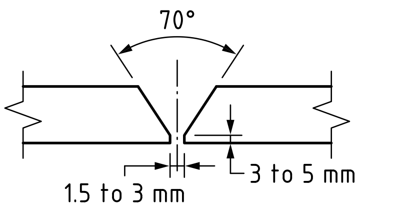
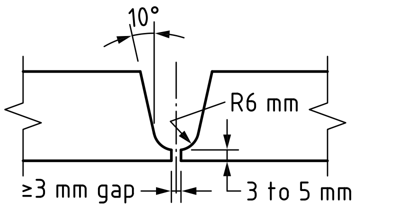
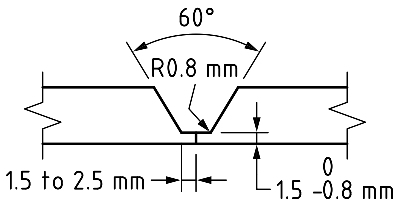
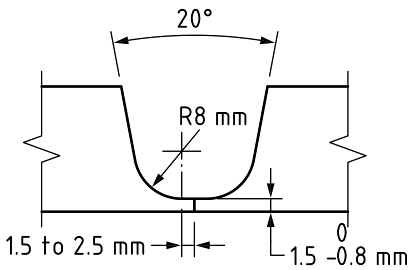
Figure	Joint	Name	Application
a)		Single-welded butt joint with "V" groove without backing strip	Butt welds in plates having a thickness not greater than 16 mm
b)		Single-welded butt joint with "U" groove without backing strip	Butt welds in plates having a thickness greater than 16 mm
c)		Single-welded butt joint with "U" groove without backing strip	Butt welds in plates up to 20 mm thick where the second side is inaccessible for welding. Initial pass to be made by the TIG process with inert gas backing
d)		Single-welded butt joint with "U" groove without backing strip	Butt welds in plates over 20 mm thick where the second side is inaccessible for welding. Initial pass to be made by the TIG process with inert gas backing

Figure E.2 **Typical weld preparations for circumferential welds where the second side is inaccessible for welding *(continued)***

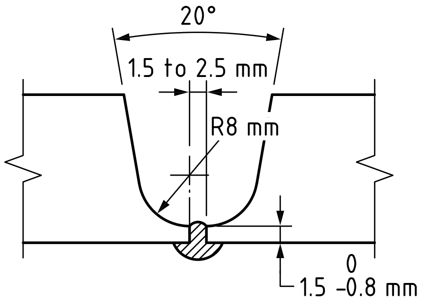
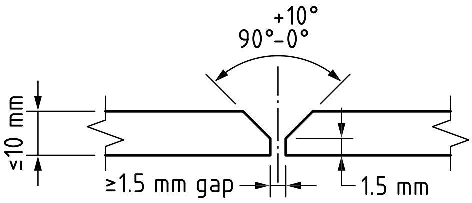
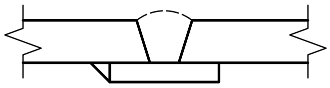
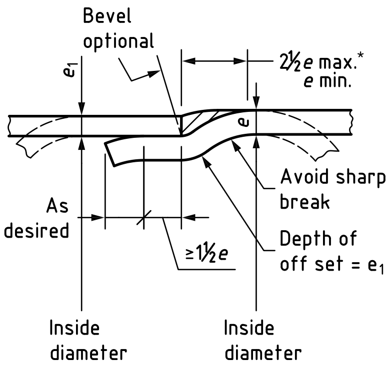
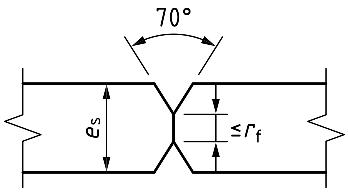
Figure	Joint	Name	Application
e)		Single-welded butt joint with “U” groove with consumable root insert	Butt welds in plates over 20 mm thick where the second side is inaccessible for welding. Initial pass to be made by the TIG process with inert gas backing
f)		Single-welded butt joint with “V” groove without backing strip	Butt welds in plates not exceeding 10 mm thickness
g)	 See Figure E.1d) for dimensions	Single-welded butt joint with with backing strip (see **4.3.6.2**)	Butt welds in all thicknesses of plate

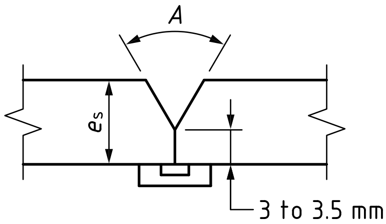
Figure E.2 **Typical weld preparations for circumferential welds where the second side is inaccessible for welding *(continued)***

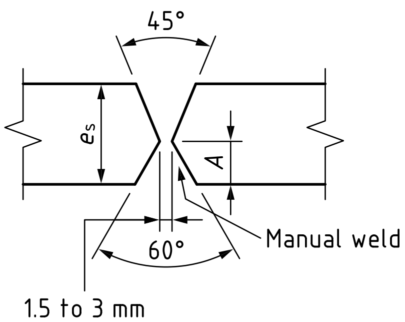
Figure	Joint	Name	Application
h)		Joggle joint	May be used for shell to shell and shell to end (excluding cone to shell) connections provided that: a) the contents are not corrosive; b) the material is restricted to group 1 with a specified minimum tensile strength not exceeding 460 N/mm^2, or group 8.1 with no limit on tensile strength; c) the greater of the thicknesses being joined does not exceed 16 mm; d) that when the flanged section of a dished end is joggled, the joggle shall be sufficiently clear of the knuckle radius to ensure that the edge of the circumferential seam is at least 12 mm clear of the knuckle; e) that when the shell with a longitudinal seam is joggled: 1) the welds are ground flush internally and externally for a distance of approximately 50 mm prior to joggling with no reduction of plate thickness; and 2) on completion of joggling, the area of the weld is subjected to magnetic crack detection or dye penetrant examination and is proven to be free of cracks; f) the offset section which forms the weld backing is a close fit within its mating section round the entire circumference (machining of the mating spigot of the offset section is permissable provided the offset thickness remaining as backing material is nowhere less than 75% of the original thickness); g) the profile of the offset is maintained and is not allowed to deteriorate through continuous production; the form of the offset is a smooth radius without sharp corners; h) that on completion of welding, the weld has a smooth profile and fills the groove to the full thickness of the plate edges being joined; i) that the junction of the longitudinal and circumferential seams are radiographed and found to be free of significant defects; j) that heat treatment as necessary is carried out on the basis of design considerations and in accordance with Figure D.1 and Figure D.2.

* This limit applies to weld preparation only; weld should be dimensioned to comply with h) in the application column.

Figure E.3 **Typical weld preparations for butt welds using the submerged arc welding process**

<table>
<tr><th>Figure</th><th>Joint</th><th>Name</th><th>Application</th></tr>
<tr><td>a)</td><td>


<table>
<tr><td>e_s (in mm)</td><td>10</td><td>15</td><td>20</td><td>25</td><td>40</td><td>50</td><td>65</td></tr>
<tr><td>r_f (in mm)</td><td>6</td><td>6</td><td>8</td><td>8</td><td>10</td><td>12</td><td>16</td></tr>
</table>
</td><td>Double-welded butt joint with double “V”</td><td>Butt welds in plates 10 mm and thicker.
Second side need not be cut back to sound metal if both root passes penetrate</td></tr>
<tr><td>b)</td><td>


<table>
<tr><td>e_s (in mm)</td><td>4.5</td><td>10</td><td>15</td><td>20</td><td>25</td><td>40</td></tr>
<tr><td>A (in degrees)</td><td>60</td><td>60</td><td>60</td><td>45</td><td colspan="2">45 (min.)</td></tr>
</table>
</td><td>Single-welded butt joint with a single “V” and a temporary backing bar</td><td>Butt welds in plate 4.5 mm to 40 mm thick.
Joint welded using temporary copper backing</td></tr>
<tr><td>c)</td><td>


<table>
<tr><td>e_s (in mm)</td><td>10</td><td>Up to 65</td></tr>
<tr><td>A (in degrees)</td><td>4.5</td><td>6 (min.)</td></tr>
</table>
</td><td>Single-welded butt joint with manual metal-arc backing</td><td>Butt welds between plates 10 mm to 65 mm thick
Manual metal-arc laid and cut back before submerged are welding</td></tr>
</table>

Figure E.4 **Typical weld preparations for butt welds using the manual inert gas arc welding for austenitic stainless and heat resisting steels only**

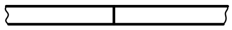
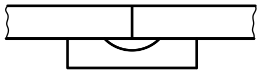
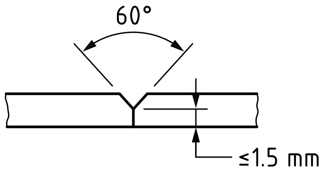
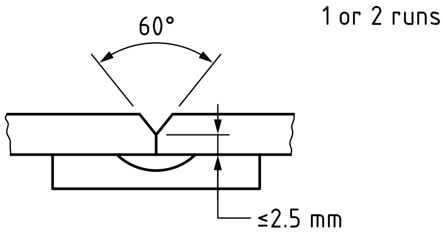
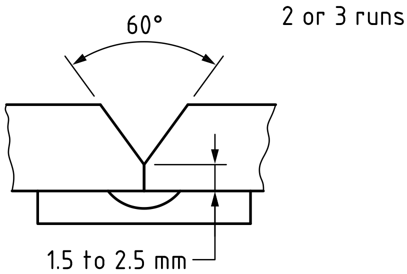
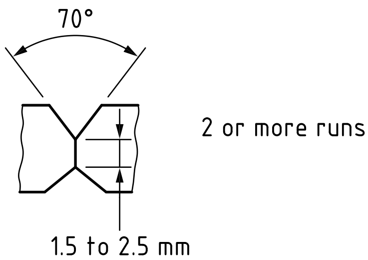
Figure	**Plate thickness**	**Edge preparation**	**Remarks**
a)	1 mm to 2 mm		Inert gas backing bar may be used (see **4.3.6.2**)
b)	3 mm		Backing bar should be used (see **4.3.6.2**)
c)	3 mm		Either a backing bar or argon backing should be used. There should be no access for air to the back of the weld (see **4.3.6.2**)
d)	4 mm		Frequently a filler rod is not used for the first run. Where the back of the joint cannot be dressed after welding, argon backing should be used, and there should be no access for air to the back of the weld (see **4.3.6.2**)
e)	6 mm		If no backing bar is used, cut back to sound metal and add sealing run (see **4.3.6.2**
f)	6 mm		Cut back after first run to sound metal before welding underside

Figure E.4 **Typical weld preparations for butt welds using the manual inert gas arc welding for austenitic stainless and heat resisting steels only *(continued)***

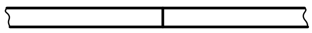
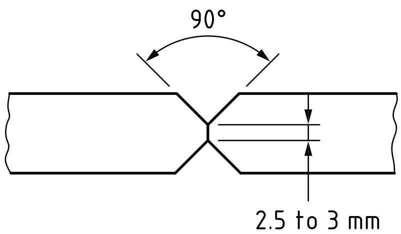
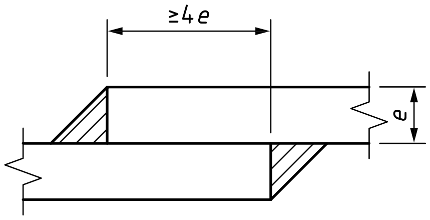
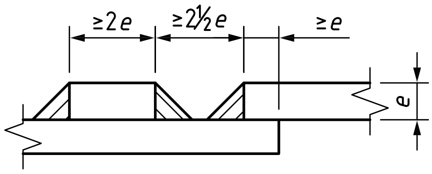
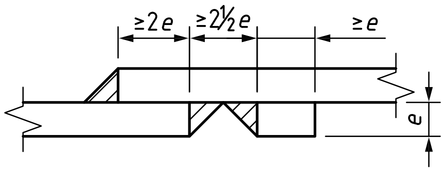
Figure	Plate thickness	Edge preparation	Remarks
g)	3 mm		Butt welds in plate not exceeding 3 mm thick Double operator single pass vertical TIG process
h)	3 mm to 6 mm		Butt welds in plate between 3 mm and 6 mm thick Double operator single pass vertical TIG process

Figure E.5 **Typical weld details for circumferential lap joints**

Figure	Joint	Name	Application Circumferential joints only
a)	 *e* is the thickness of thinner plate joined	Double full-fillet lap joint	Not exceeding 16 mm plate Category 3 vessels only Permitted for shell to end connections provided that the weld is clear of the knuckle at the end
b)	 *e* is the thickness of thinner plate joined	Double full-fillet lap joint with plug weld	Not exceeding 14 mm plate Plugs to be proportioned to take 20 % of total load Category 3 vessels only
c)	 *e* is the thickness of thinner plate joined		

E.2 Typical examples of acceptable weld details

E.2.1 General

This clause is based upon Annex G of ISO/DIS 2694[1)]. The drawings are intended to convey recommendations in regard to connections welded manually by the metal-arc process in steel pressure vessels with a shell thickness of not less than 5 mm. . The following types of connections are covered.

a) Branches without added compensation rings:
   1) set-on branches (see Figure E.9 to Figure E.15);
   2) set-in branches (see Figure E.16 to Figure E.21);
   3) forged branch connections (see Figure E.22 and Figure E.23).
b) Branches with added compensation rings:
   1) set-on branches (see Figure E.24);
   2) set-in branches (see Figure E.25, Figure E.26, Figure E.27 and Figure E.28).
c) Studded connections and couplings:
   1) butt-welded studded connections (see Figure E.29);
   2) socket welded and screwed connections (see Figure E.30).
d) Flanges (see Figure E.31, Figure E.32 and Figure E.33).
e) Jacketted vessels (see Figure E.34, Figure E.35, Figure E.36 and Figure E.37).
f) Flat ends covers (see Figure E.38).
g) Tubeplate to shell connections (see Figure E.39 to Figure E.44).
h) Flat end connections (flanges) (see Figure E.31).

NOTE 1 Typical examples of arc welded tube to tubeplate joints are given in Annex T.

*NOTE 2 The thicknesses quoted in **E.2** and Figure E.6 to Figure E.44 are nominal thicknesses (see **1.6**).*

E.2.2 Purpose

The purpose of this clause is to exemplify sound and commonly accepted practice and not to promote the standardization of connections that may be regarded as mandatory or to restrict development in any way. A number of connections have been excluded which, whilst perfectly sound, are restricted in their use to certain applications, firms or localities. Furthermore, it is appreciated that it will be desirable to introduce amendments and additions in the future to reflect improvements in welding procedures and techniques as they develop.

E.2.3 Selection of detail

The connections recommended are not considered to be equally suitable for all service conditions, nor is the order in which they are shown indicative of their relative mechanical characteristics. In selecting the appropriate detail to use from the several alternatives shown for each type of connection, consideration should be given to the manufacture and service conditions that pertain.

[1)] Failed to gain approval.

It is to be noted that for vessels subject to internal corrosion, only those connections that are suitable for applying a corrosion allowance should be used. Certain types, such as those incorporating internal attachment by fillet welds only, do not lend themselves to this and their use on internal corrosive duties should be discouraged.

E.2.4 Weld profile and size

The limitations quoted in weld profiles and sizes are based on commonly accepted sound practice, but they may be subject to modifications dictated by special welding techniques or design conditions.

E.2.4.1 Weld profiles

The weld profiles (for example bevel angles, root radii and root faces) recommended are indicated by letters and numbers in circles or squares, which refer to the profiles shown in Figure E.6. They are designed to provide correct conditions for welding and to facilitate the deposition of sound weld metal in the root of the joint. This is particularly important in the case of single-bevel and single-J welds and, where these are given as alternatives, it is recommended in general that preference be given to the latter where the depth or throat thickness of the weld exceeds about 16 mm.

E.2.4.2 Butt joints

In cases where full penetration butt joints are indicated, it is intended that they should be back chipped or gouged and back welded, or alternatively that the welding procedure should be such as to ensure sound, positive root penetration.

E.2.4.3 Weld sizes

The size of the welds, i.e. throat thicknesses, have been proportioned to develop the full strength of the parts joined.

E.2.4.4 Fillet welds

Where the leg length of a fillet weld necessary to meet the design throat thickness (or design leg length) at the edge of a plate or section is such that the parent metal does not project beyond the weld, melting of the outer corner or corners, which reduces the throat thickness, shall not be allowed. See Figure E.7a).

E.2.4.5 Modifications

Cases may well arise where sound modifications may be made with advantage:

a) to the weld profiles to suit special welding techniques;

b) to the weld sizes to suit design and service conditions.

It is recommended however, that such modifications be approved by a competent engineer.

E.2.5 Notes applicable to the various types of connections shown in Figure E.9 to Figure E.44

E.2.5.1 The dimensions and shape of the detail chosen can influence the feasibility and/or efficiency of ultrasonic examination. This may also be a function of the equipment and time available. Where ultrasonic examination is specified, these factors should be given due consideration.

E.2.5.2 When welds are made from one side only, the penetration bead is to have a smooth contour and be flat or slightly convex.

E.2.5.3 The use of ring-type compensation is not suitable for cases where there are severe temperature gradients, especially when these are of a fluctuating nature.

E.2.5.4 When ring-type reinforcement is used, the material used for the ring is to be of the same nominal strength as that of the shell.

E.2.5.5 When partial penetration joints are used, root defects may be present and these cannot always be detected or interpreted by means of non-destructive testing. The use of partial penetration joints is not suitable for cases where there are severe temperature gradients, especially when these are of a fluctuating nature.

E.2.5.6 The use of socket welded and screwed couplings, such as those shown in Figure E.30, is limited to those permitted by **3.5.4.8**.

E.2.5.7 The selection of details for parts of vessels involving jacketted construction is of a special nature, and this should be borne in mind in selecting appropriate details.

E.2.5.8 When spigots designed to permit butt welded connections between sub-components [e.g. Figure E.32a), Figure E.41 and Figure E.43] are not produced by means of forging, attention is drawn to the necessity of ensuring that the through thickness properties are adequate for the design.

This should be demonstrated by obtaining at least 25 % reduction in area from three representative test pieces from the plate in a plane perpendicular to the plate surface. In addition, the spigot and adjacent region of the plate should be subjected to appropriate non-destructive testing to confirm the absence of lamellar defects after the completion of welding and post-weld heat treatment.

E.2.5.9 When ultrasonic inspection is required, it may be necessary to examine the welded connection between the branch and shell prior to fitting the compensation ring.

E.2.5.10 These details are not suitable where crevice corrosion may occur.

E.2.5.11 Although the figure indicated is intended for another purpose, it is considered that the form of preparation illustrated is suitable for the connection between the shell and a flat end.

When cut edges are not sealed by welding and are exposed in service, they are to be inspected for laminar defects which may cause leakage.

E.2.5.12 These weld details are recommended only for shell thicknesses up to 16 mm in carbon and carbon manganese steels in material groups 1, 2 and 11 with R_m (see **2.3.2**) not exceeding 432 N/mm^2 or for austenitic steels in material group 8 without limit on shell thickness or tensile strength. These weld details are not recommended for corrosive or fatigue duty.

E.2.5.13 Acceptable only for group 1 materials. This type of weld is liable to cracking of the root runs in thick sections and should be restricted to thicknesses up to 50 mm unless subject to specially agreed welding procedures.

E.2.5.14 These details are acceptable only for group 1 materials, and either shell or pad thicknesses up to 38 mm.

E.2.6 Notes applicable to branches in Figure E.9 to Figure E.30

E.2.6.1 Sections

The drawings of the recommended connections show a transverse section [see detail A, Figure E.7b)] and a longitudinal section [see detail B, Figure E.7b)].

E.2.6.2 Weld sizes

The sizes of the welds have been proportioned to develop the full strength of the parts joined. See also **E.2.4.3**, **E.2.4.4** and **E.2.7.2.1**.

E.2.6.3 Weld profiles

While both single-bevel and single-J welds have been shown as acceptable in the smaller sizes, in general the latter are to be preferred because of the sounder root conditions obtained, and it is recommended that single-bevel welds be limited in size to about 15 mm in depth. See also **E.2.4.1** and **E.2.4.2**.

E.2.7 Notes applicable to branches without compensation rings in Figure E.9 to Figure E.23

E.2.7.1 Set-on branches

Consideration should be given to the necessity for examining the shell plate for laminations around the branch hole when set-on branches are used.

E.2.7.2 Set-in branches

E.2.7.2.1 Weld sizes

The type of branch to shell connections and the sizes of welds employed may be influenced by several factors in the operational conditions for which the vessel is designed. For general guidance in this annex weld sizes have been shown for the various connections recommended, based on the concept that the welded joints should develop the full strength in tension of the branch radial to the shell as indicated in Figure E.8a) and Figure E.8b). In general, it should therefore be unnecessary to apply larger welds than those shown.

The simple, though approximate, assumption has been accepted that the total throat thickness of the welds should equal twice the branch thickness. It has also been assumed that the welds should be reasonably symmetrical about the mid-thickness of the connection.

It is further recommended that, when the branch thickness exceeds half the thickness of the shell, full penetration joints should be used with fillet welds equal in total throat thickness to 20% of the shell thickness as shown in Figure E.8c) and Figure E.8d). This additional throat thickness is recommended to compensate for the relative practical difficulty of applying perfectly sound welds in nozzle connections and of applying non-destructive tests for their examination. These additional fillet welds are also intended to provide a reasonable geometric profile, and for practical reasons a minimum dimension of 6 mm has been applied to the fillet weld size.

There may be service conditions for which smaller welds are adequate. In such cases, when subject to study by a competent engineer, the weld sizes may be reduced.

E.2.7.2.2 Gap between branch and shell

It is recommended that the gap between the branch and shell should not exceed the following:

a) 1.5 mm for branch diameters up to 300 mm; or

b) 3 mm in other cases.

Wider gaps increase the tendency to spontaneous cracking during welding particularly as the thickness of the parts joined increases.

E.2.7.2.3 Removal of internal sharp edge in branch bore

It will be noted that the internal edges in the bores of set-in branches are shown radiused because a stress concentration occurs at this point. This precaution is recommended when the branch connection is fully stressed or subjected to fatigue, but may not be necessary where these conditions do not obtain.

E.2.7.2.4 Preparation of hole in shell

In the case of set-in branches of the types shown in Figure E.17a) to Figure E.20b) inclusive, the hole in the shell may be cut and profiled in two ways as follows.

a) The depth of the grooves B and D may be constant around the hole as shown in Figure E.8e). This, the normal case, is the concept upon which the drawings have been prepared, for example see Figure E.17b).

b) The roots of the weld grooves may be in one plane, as for example when they are machine bored, in which case the depths of the grooves will vary around the hole, as shown in Figure E.8f).

E.2.8 Notes applicable to branches with added compensation rings in Figure E.24 to Figure E.28

E.2.8.1 General

Compensation rings should be a close fit to the shell and “tell-tale” holes should be provided in them.

E.2.8.2 Set-in branches

E.2.8.2.1 Gap between branch and shell

It is recommended that the gap between the branch, shell and also the compensation ring should not exceed the following:

a) 1.5 mm for branch diameters up to 300 mm; or

b) 3 mm in other cases.

Wider gaps increase the tendency to spontaneous cracking during welding particularly as the thickness of the parts joined increases.

E.2.8.2.2 Internal compensation rings

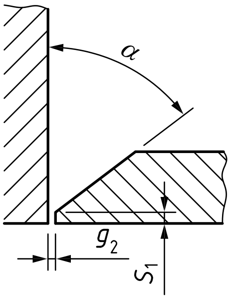
Set-in branches with single compensation rings have been shown with the rings on the outside of the shell, which is the normal case [see Figure E.25a) to Figure E.27b)]. Similar connections may be used for the attachment of internal compensation rings in the formed ends of pressure vessels and in spherical vessels.

E.2.9 Notes applicable to jacketted vessels in Figure E.34 to Figure E.37

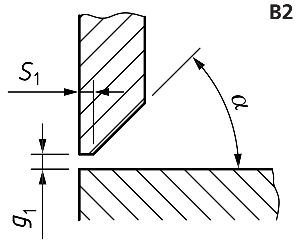
It is recommended that the gap between the shell of the vessel and the jacket or blocking ring should not exceed 3 mm. Wider gaps increase the tendency to spontaneous cracking during welding, particularly as the thickness of the parts joined increases.

Figure E.6 **Standard weld details**

B1

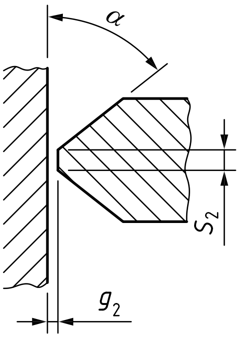


B2

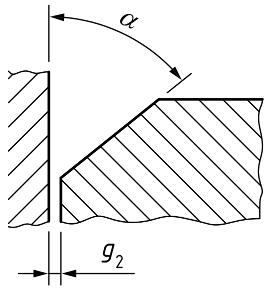


α =	50° min.
S_1 =	1.5 mm to 2.5 mm
S_2 =	0 to 3 mm
g_1 =	1.5 mm to 2.5 mm when $t < 9.5$ mm
g_1 =	2.5 mm to 4 mm when $t \geq 9.5$ mm
g_2 =	(See Note 2)

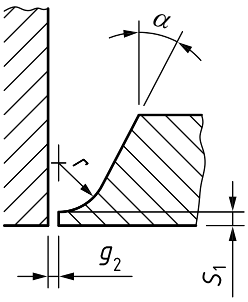
B3



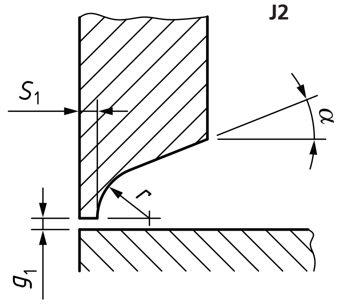
B4



J1

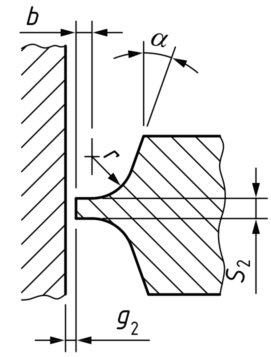


J2

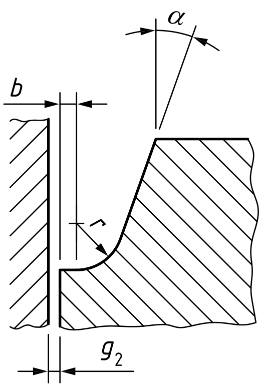


α =	15° to 35°
S_1 =	2 mm to 3 mm
S_2 =	2 mm to 3 mm
g_1 =	1.5 mm to 3 mm
g_2 =	(See Note 2)
b =	0 to 3 mm
r =	6 mm to 3 mm

J3



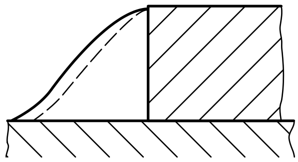
J4



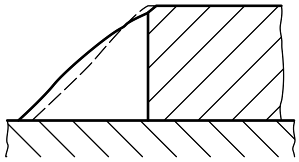
NOTE 1 These recommendations have been included for general guidance. Discretion should be used in applying the maximum and minimum dimensions quoted which are subject to variation according to the welding procedure employed (for example size and type of electrodes) and also to the position in which the welding is carried out.

NOTE 2 It is recommended that in no case should the gap between the branch and shell exceed 3 mm. Wider gaps increase the tendency to spontaneous cracking during welding, particularly as the thickness of the parts joined increases.

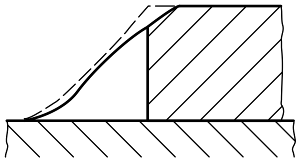
Figure E.7a) **Limitations on geometry of fillet weld applied to the edge or a part**



a) Desirable



b) Acceptable because of full throat thickness



c) Not acceptable because of reduced throat thickness

Figure E.7b) **Transverse and longitudinal sections of branch connections**

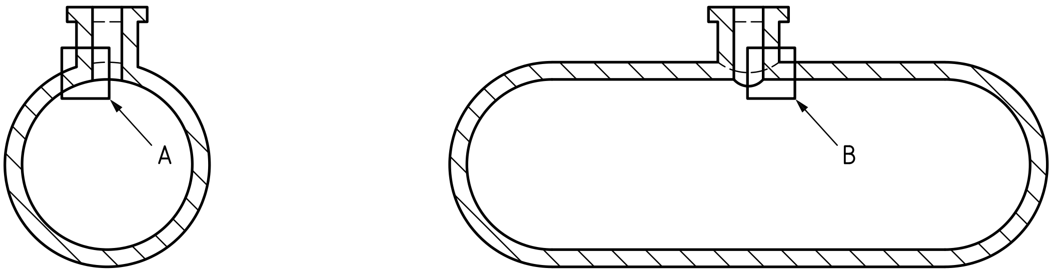
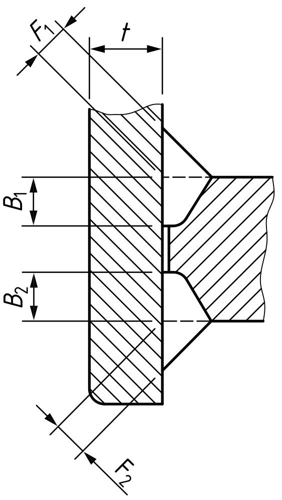
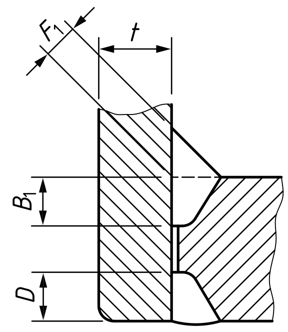


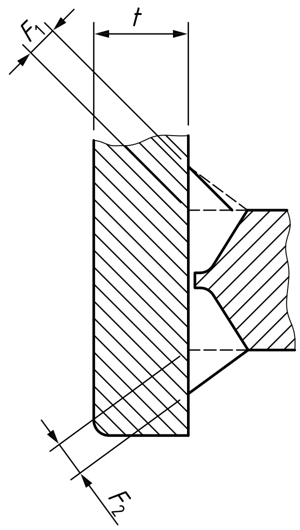
Figure E.8 **Weld details for set-in branches**



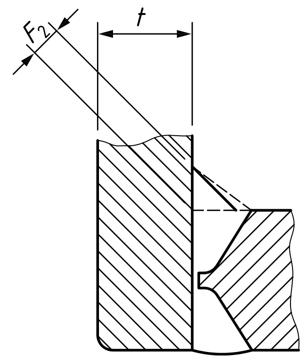
Weld sizes
$(B_1 + F_1) + (B_2 + F_2) = 2t$ approx
a) [See also Figure E.17a)]



Weld sizes
$(B_1 + F_1) + D = 2t$ approx
b) [See also Figure E.17b)]



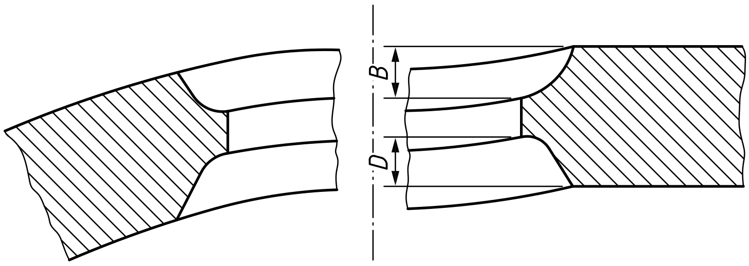
Weld sizes
$F_1 = t/10$ or 6 mm whichever is larger
c) [See also Figure E.19a)]



Weld sizes
$F_2 = t/3$ or 6 mm whichever is larger
d) [See also Figure E.19b)]

Figure E.8 **Weld details for set-in branches *(continued)***

e)



f)

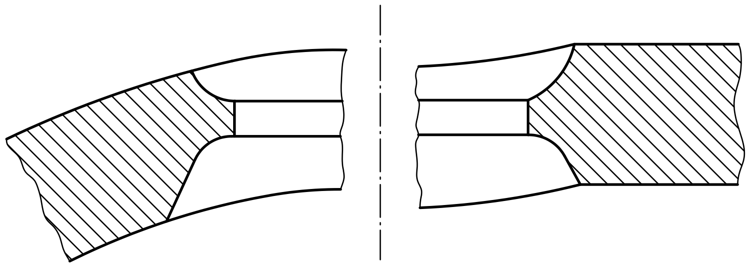
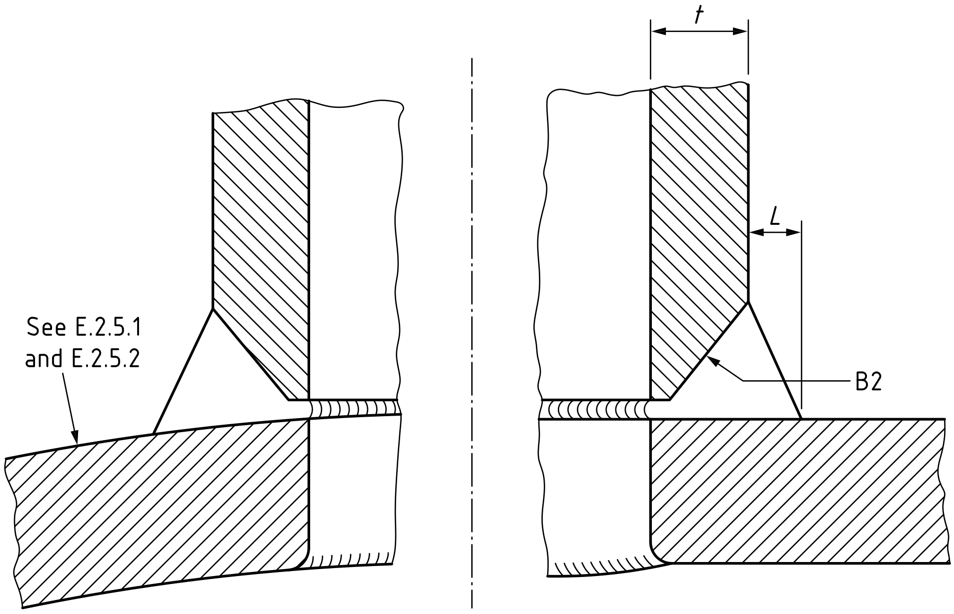


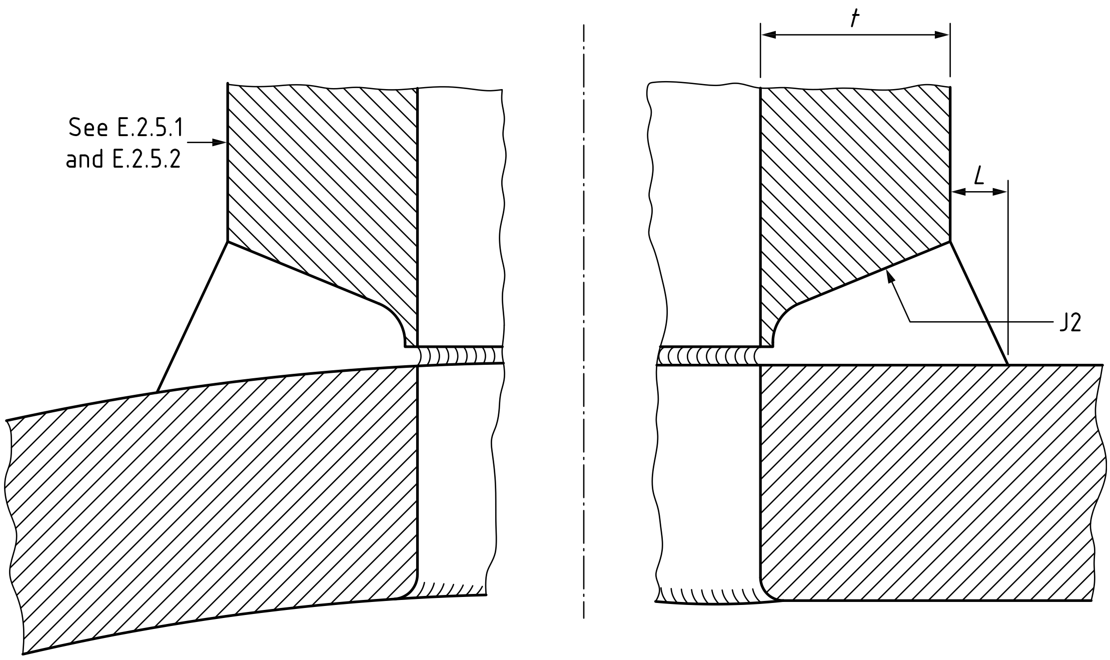
Figure E.9 **Set-on branches (see E.2.4.2)**



$L = t/3$ min. but not less than 6 mm

Preference should be given to the detail shown in b) if t exceeds about 16 mm

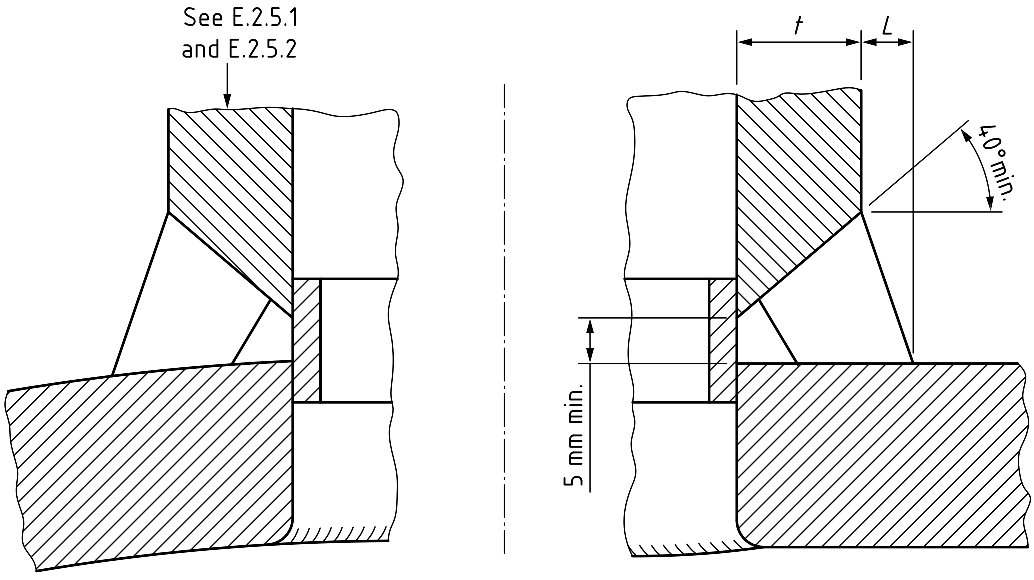
a)



$L = t/3$ min. but not less than 6 mm

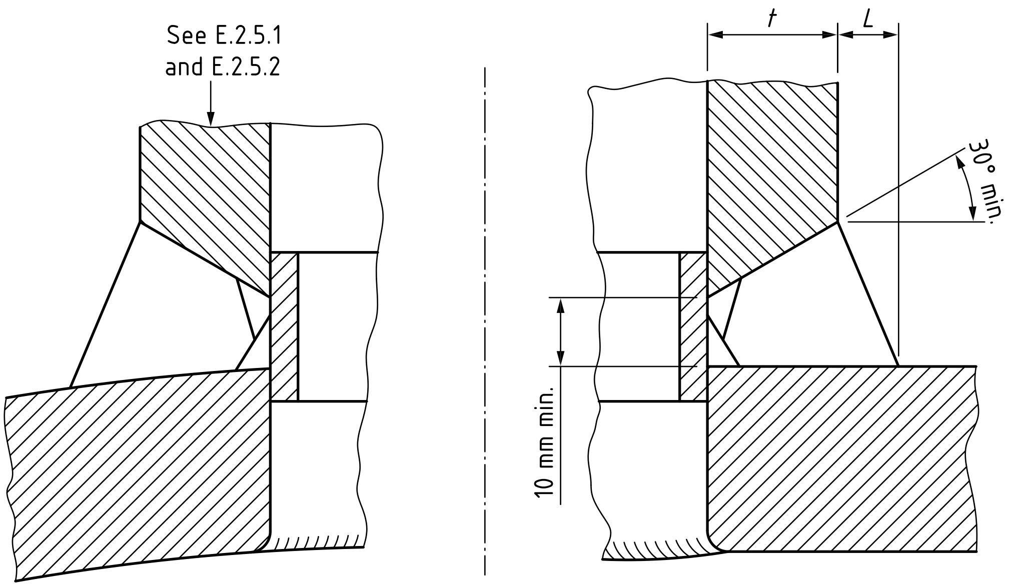
b)

Figure E.10 **Set-on branches**



$L = t/3$ min. but not less than 6 mm

a) Single root run technique



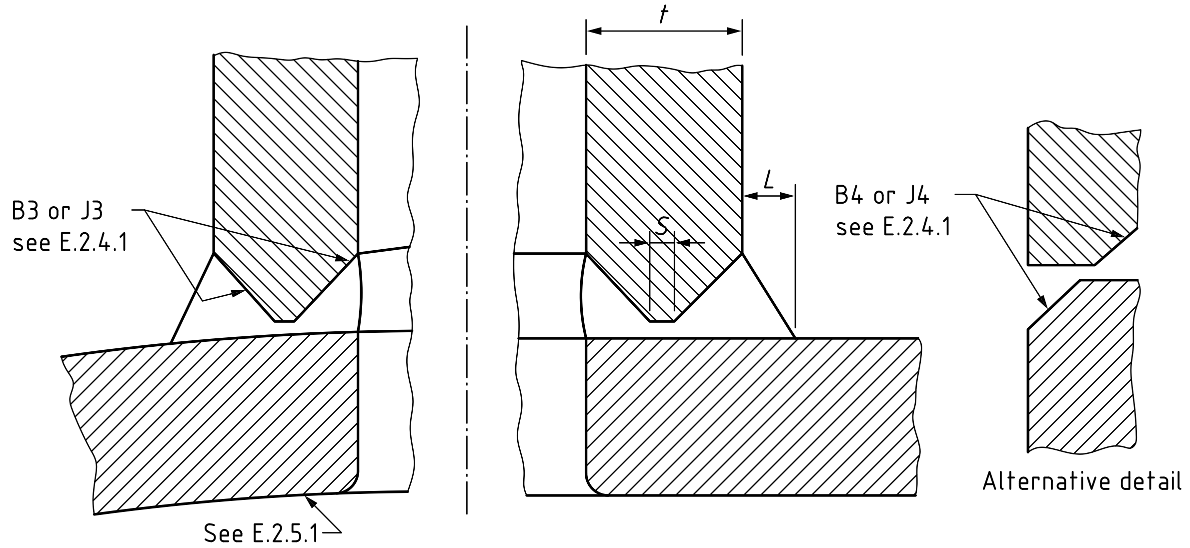
$L = t/3$ min. but not less than 6 mm

b) Double root run technique

NOTE The backing ring should be of the same nominal composition as that of the vessel shell. Care should be taken to ensure close fitting of the backing rings which should be removed after welding. After the removal of backing rings, the surface should be ground smooth and examined for cracks by dye penetrants, magnetic, or other equivalent methods.

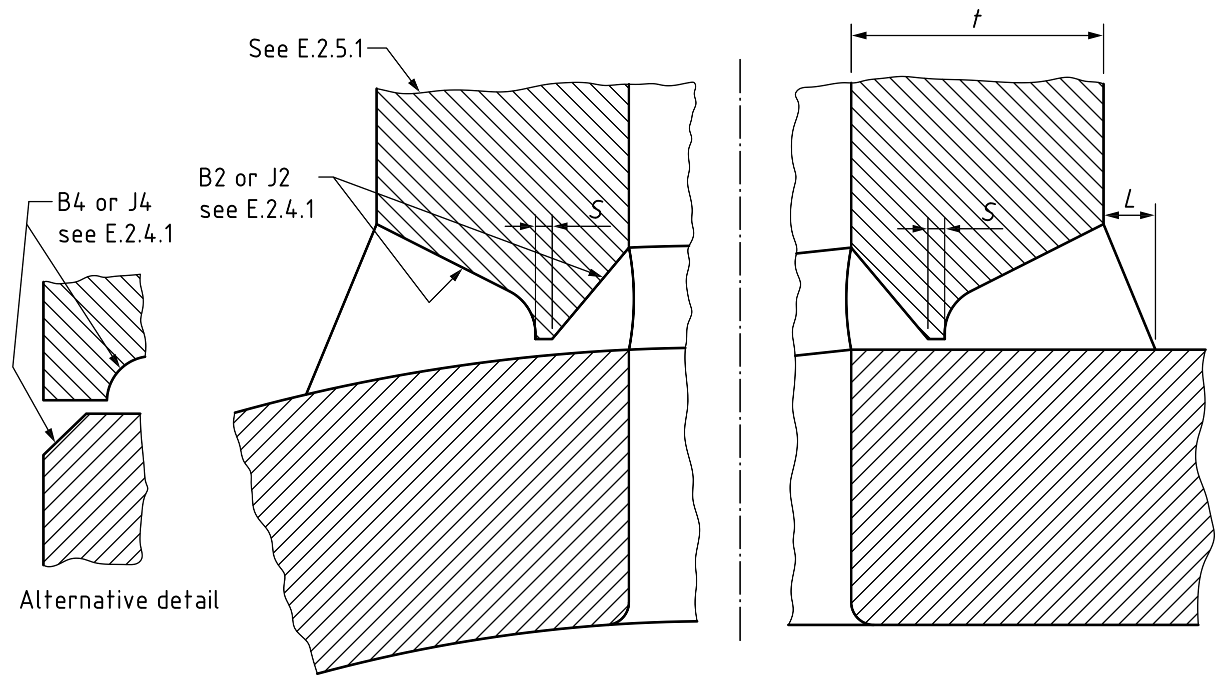
Dimensions in millimetres

Figure E.11 **Set-on branches (see E.2.4.2)**



a)

S = 1.5 mm to 2.5 mm
L = *t*/3 min. but not less than 6 mm

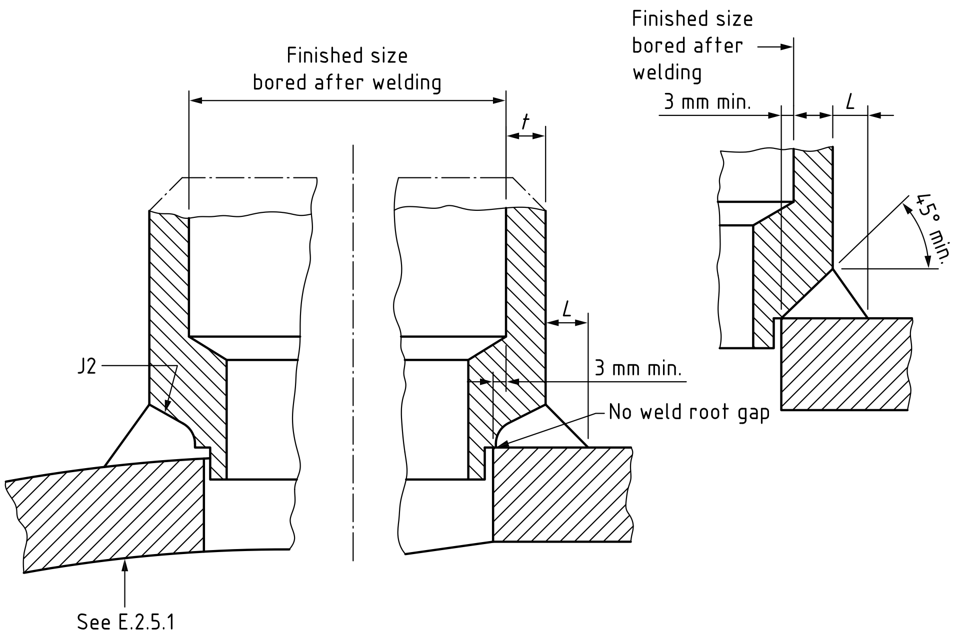


b)

S = 1.5 mm to 2.5 mm
L = *t*/3 min. but not less than 6 mm

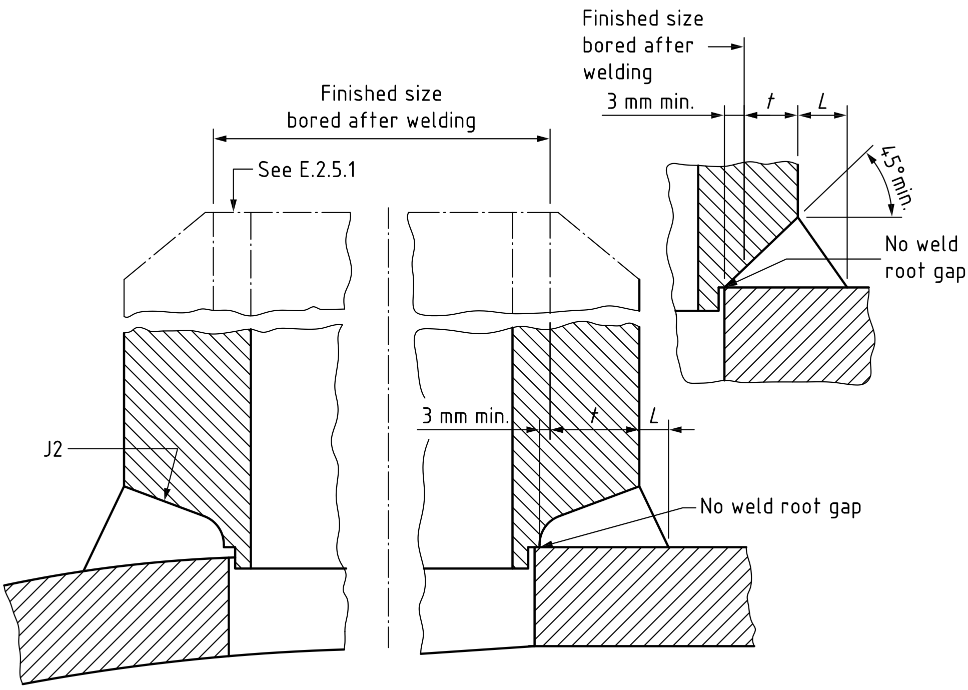
NOTE These details are recommended only where the bore of the branch is readily accessible for welding. The joint should be back-gouged from the side most accessible and suitable for this purpose, generally the outside.

Figure E.12 **Set-on branches**



L = *t*/3 min. but not less than 6 mm

a)



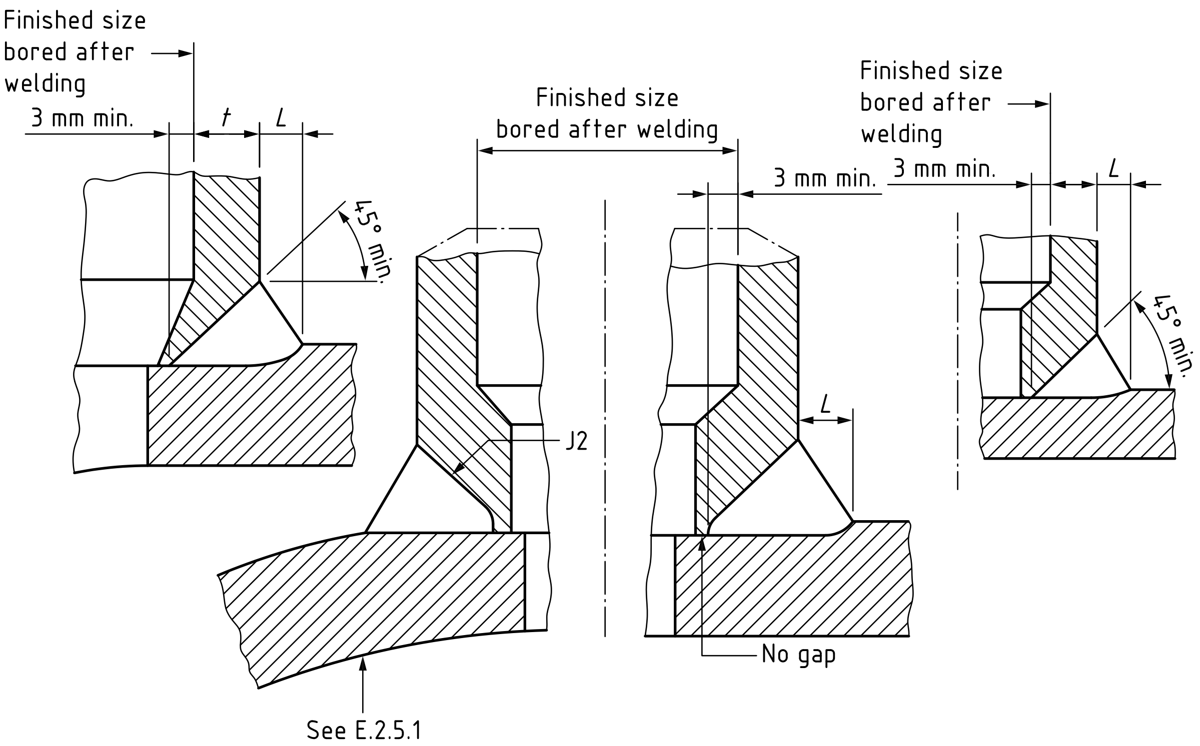
L = *t*/3 min. but not less than 6 mm

b)

NOTE Joints generally used for small branch to shell diameter ratios.

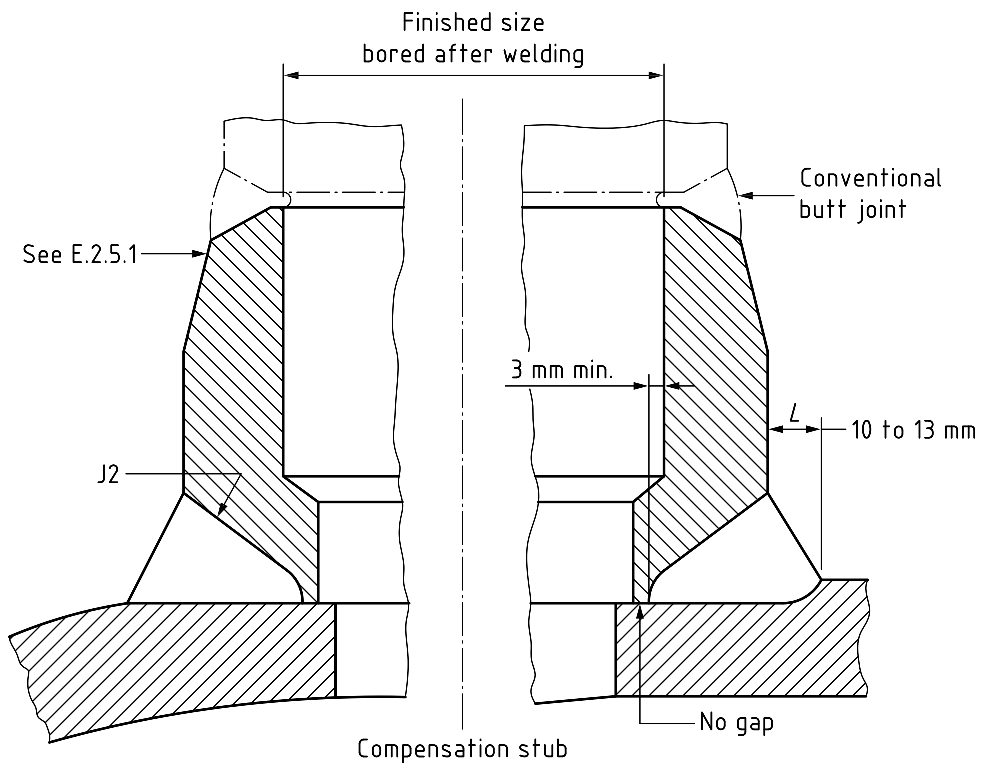
Dimensions are in millimetres

Figure E.13 **Set-on branches**



L = 10 mm to 13 mm

a)

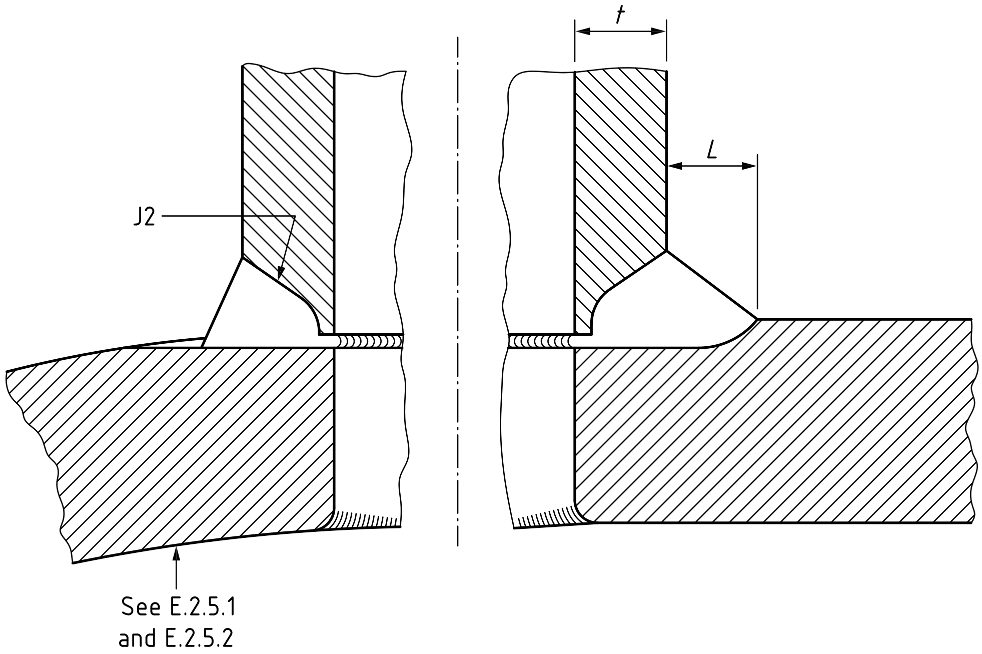


b)

NOTE Joints generally used for small branch to shell diameter ratios.

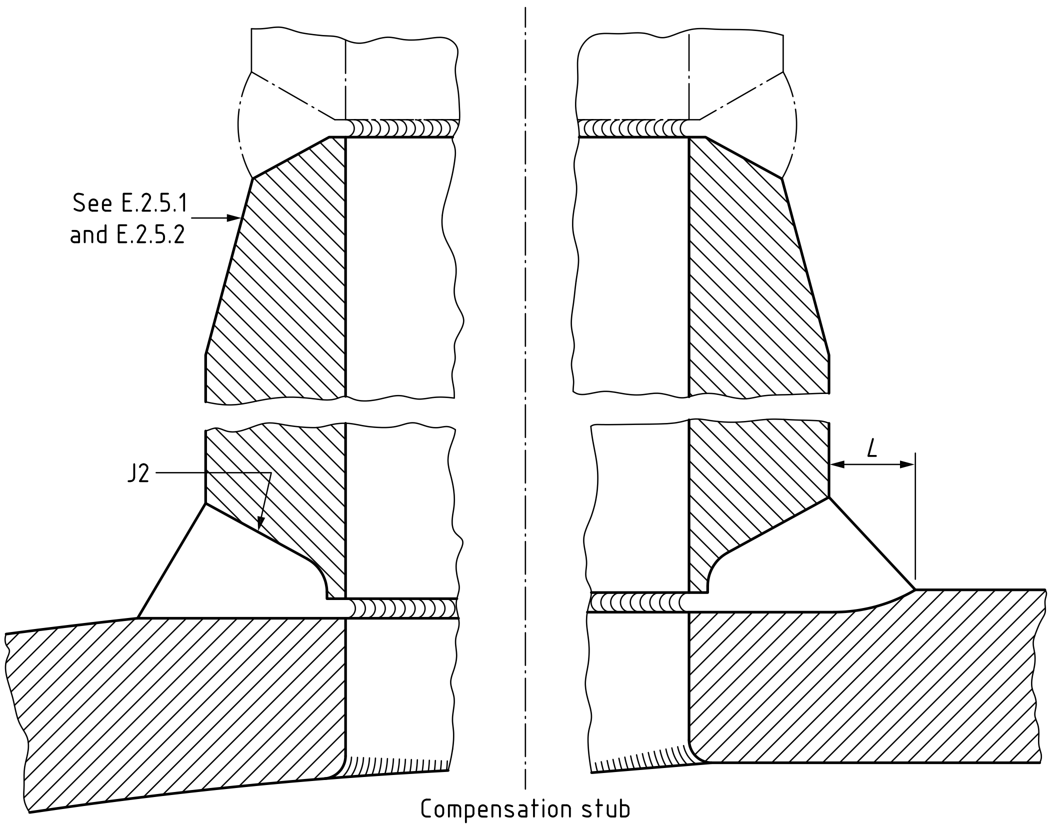
Dimensions are in millimetres

Figure E.14 **Set-on branches (see E.2.4.2)**



L = 10 mm to 13 mm

a)

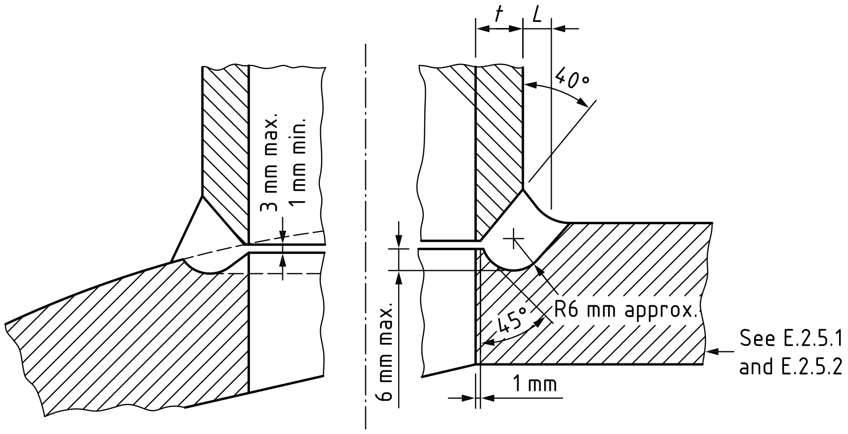


L = 10 mm to 13 mm

b)

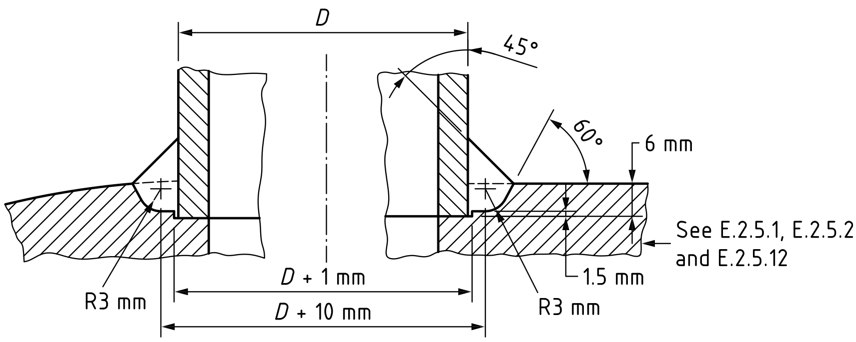
NOTE Joints generally used for small branch to shell diameter ratios.

Figure E.15 **Set-on branches (see E.2.4.2)**

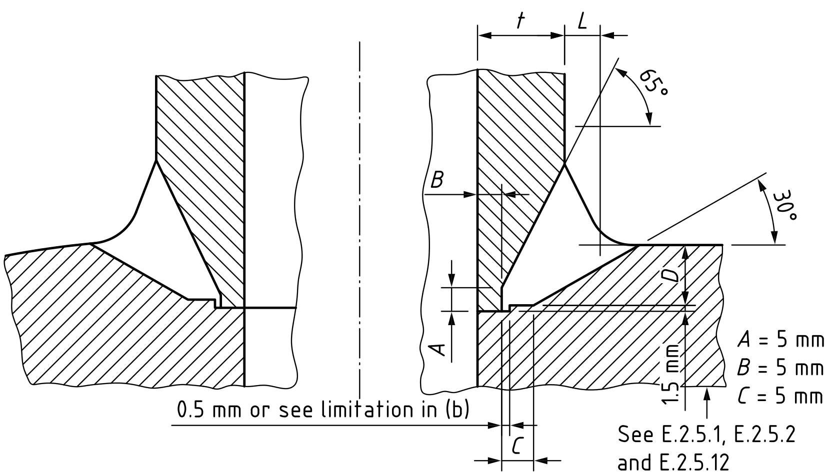


L = t/3 min. but not less than 6 mm

a) For nozzles up to approximately 100 mm bore



b) For nozzles up to approximnately 50 mm bore and 6 mm wall thickness



L = t/3 min. but not less than 6 mm

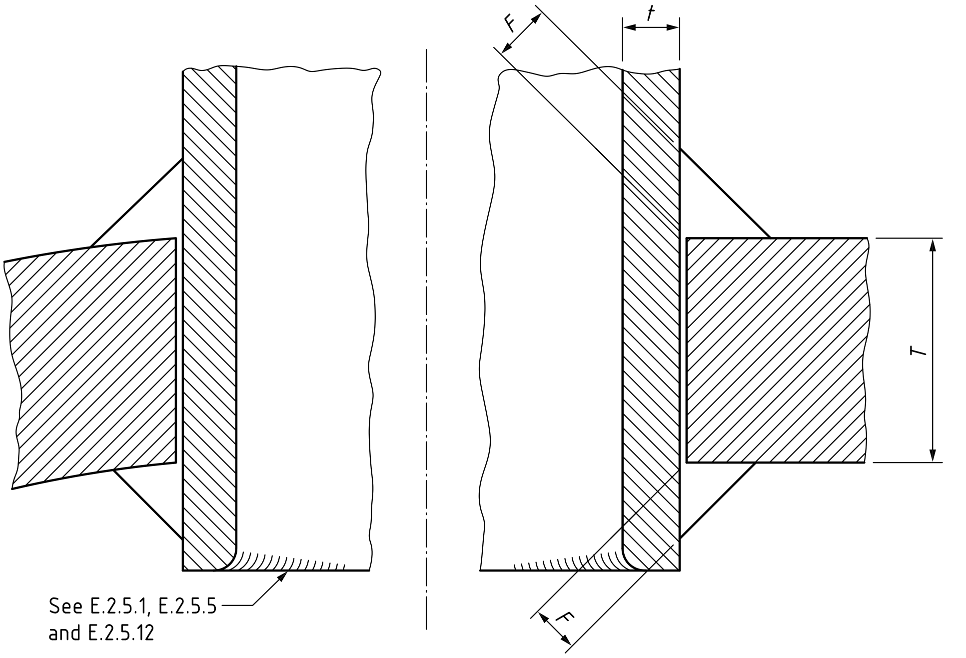
D = t with max. t = 13 mm

c) For nozzles over 50 mm bore and up to and including 150 mm bore, and with a wall thickness over 6 mm

NOTE Generally used for the attachment of nozzles to thick-walled shells.

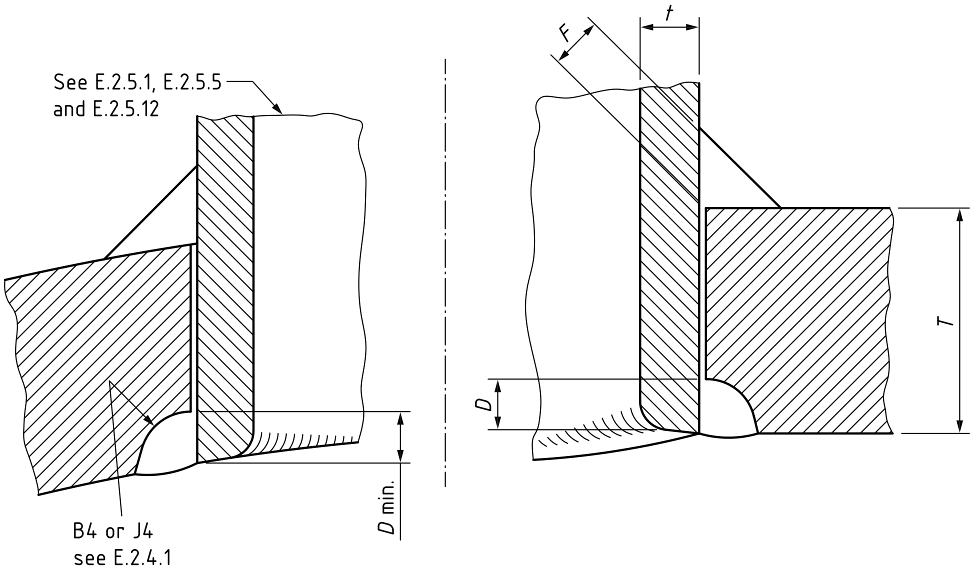
Dimensions in millimetres

Figure E.16 **Set-in branches: fillet welded connections**



F = *t*

a)



F = *t*

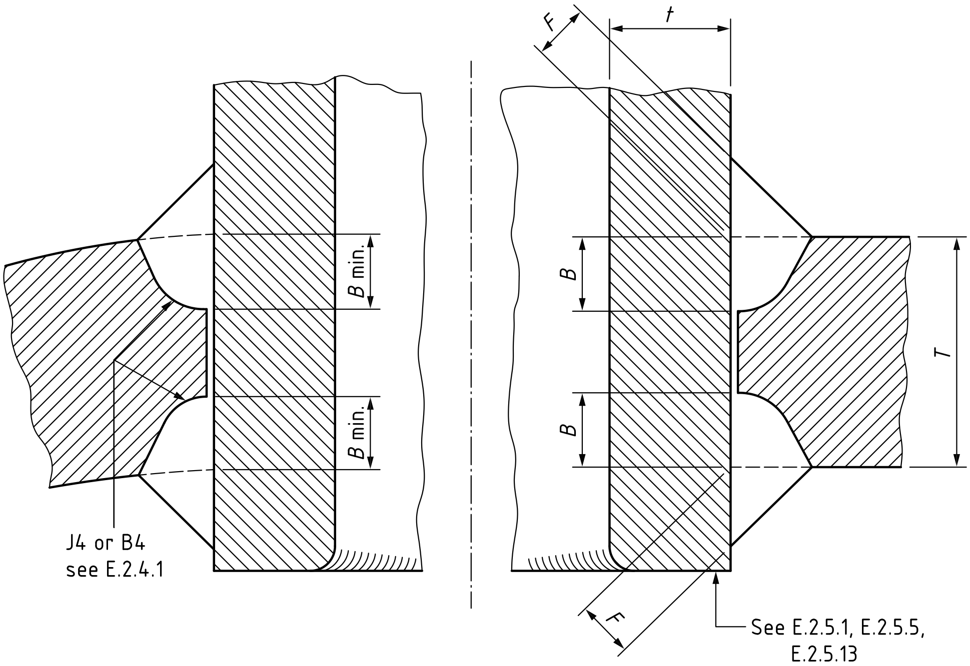
D = *t*

t = 13 mm max.

b)

NOTE Generally used when t is less than T/2. For small diameter branches, attention is drawn to the details shown in Figure E.30 which may provide a preferable solution.

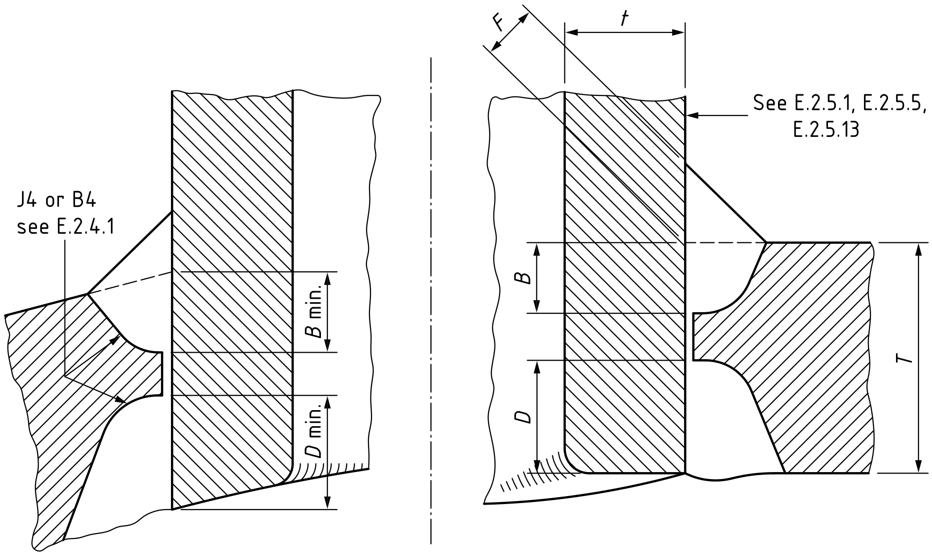
Figure E.17 **Set-in branches: partial penetration butt welded connections**



F = 6 mm min. to 13 mm max.

B + *F* = *t*

a)



F = 6 mm min. to 13 mm max.

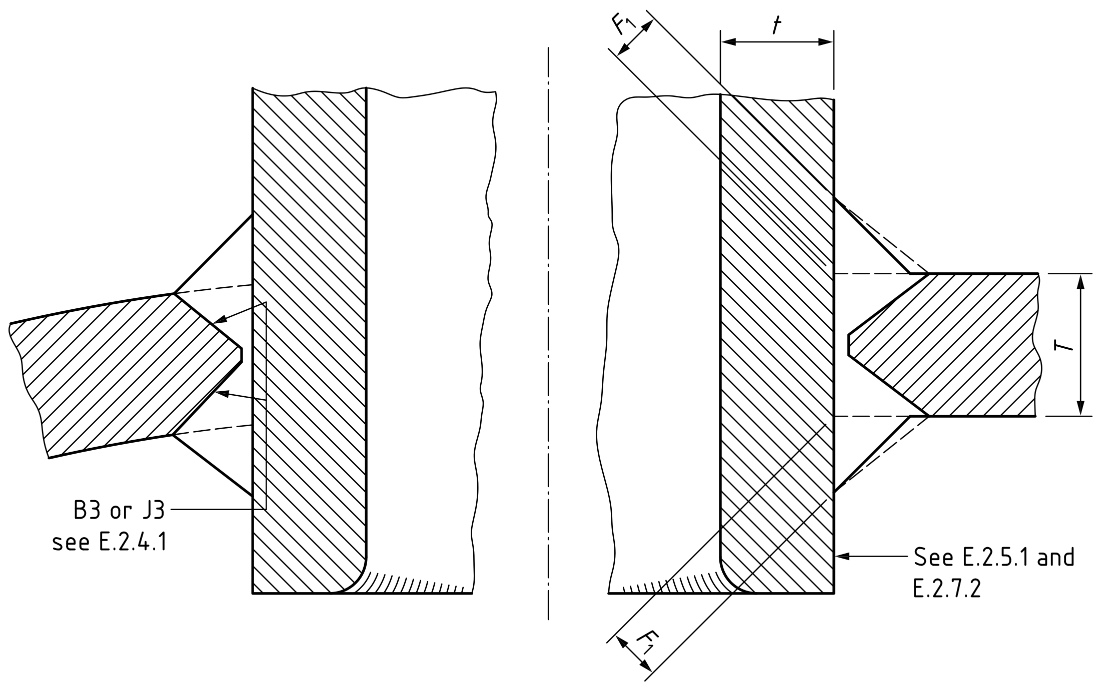
B + *F* = *t*

D = *t*

b)

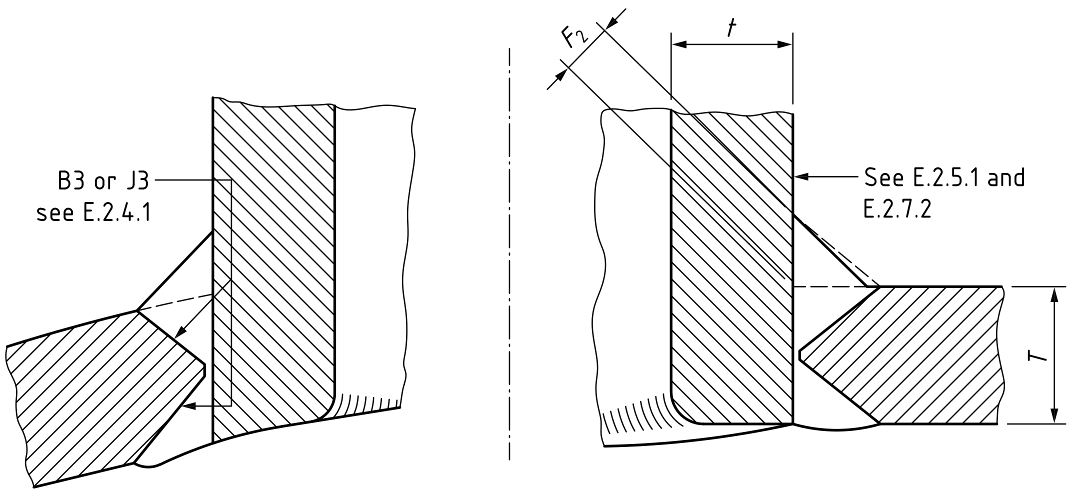
NOTE Generally used when t is approximately equal to T/2.

Figure E.18 **Set-in branches: full penetration connections (see E.2.4.2)**



F_1 = T/10 min. or 6 mm whichever is larger

a)

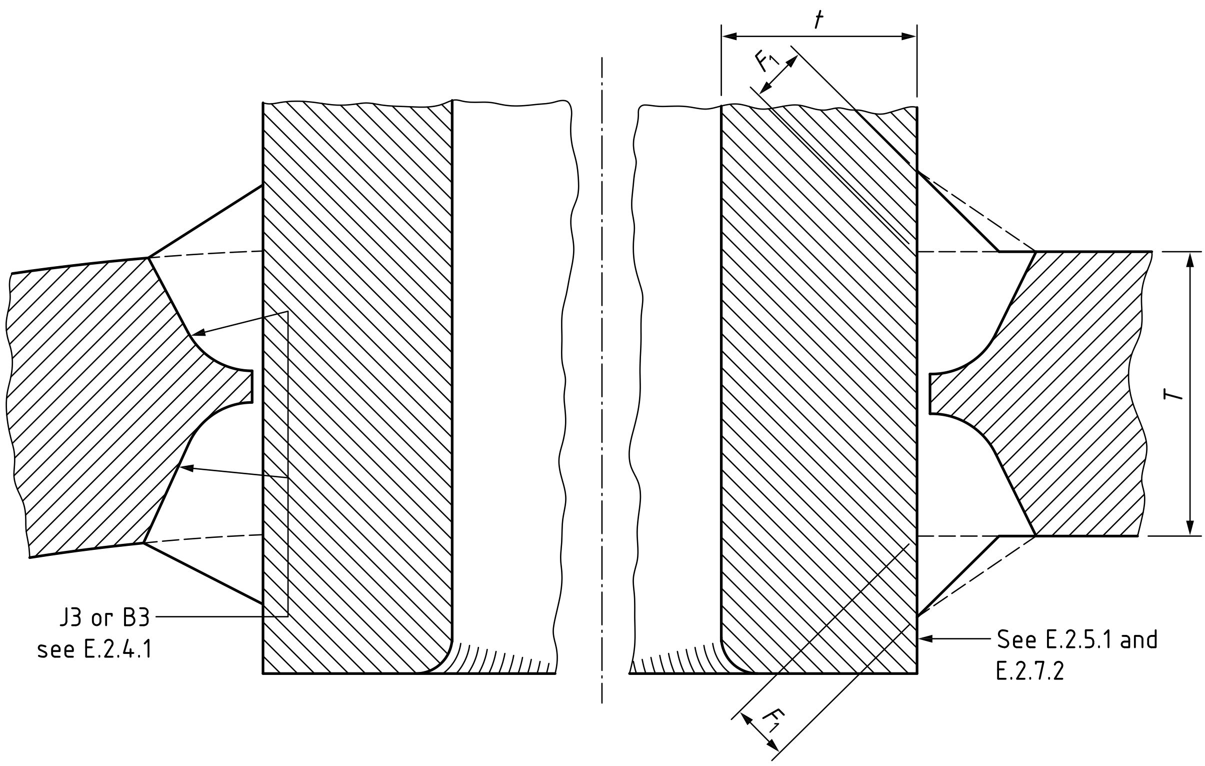


F_2 = T/5 min. or 6 mm whichever is larger

b)

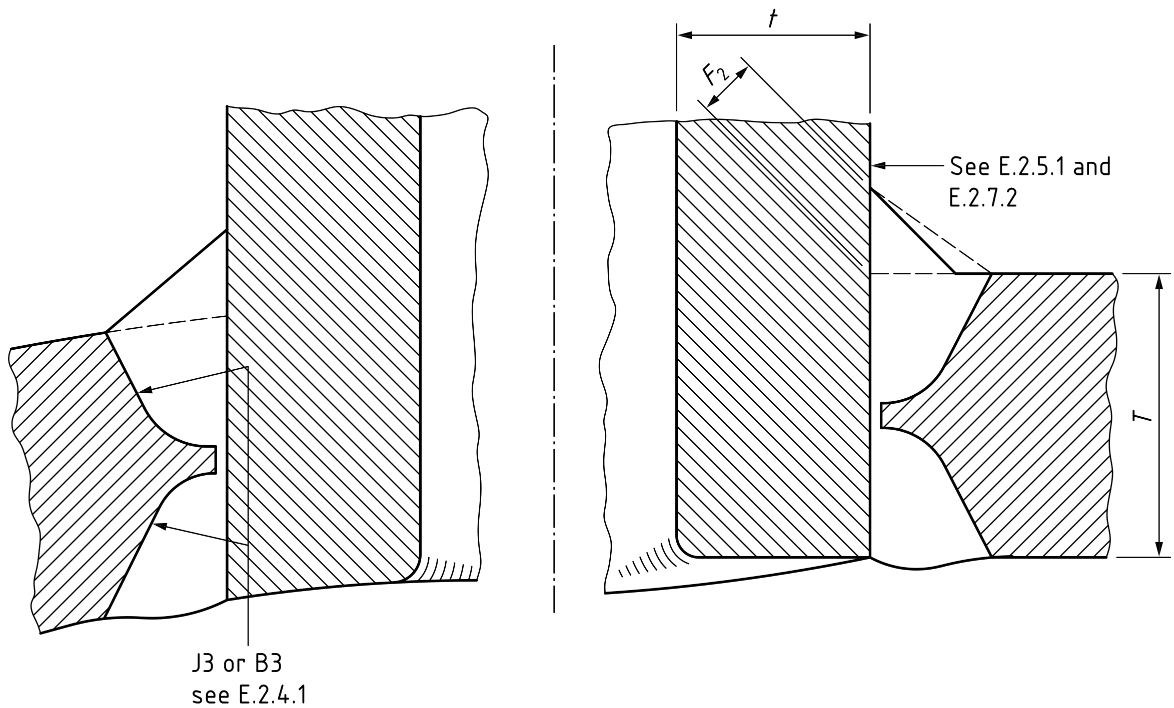
NOTE Generally used when t is greater than T/2.

Figure E.19 **Set-in branches: full penetration connections (see E.2.4.2)**



F_1 = T/10 min. or 6 mm whichever is larger

a)

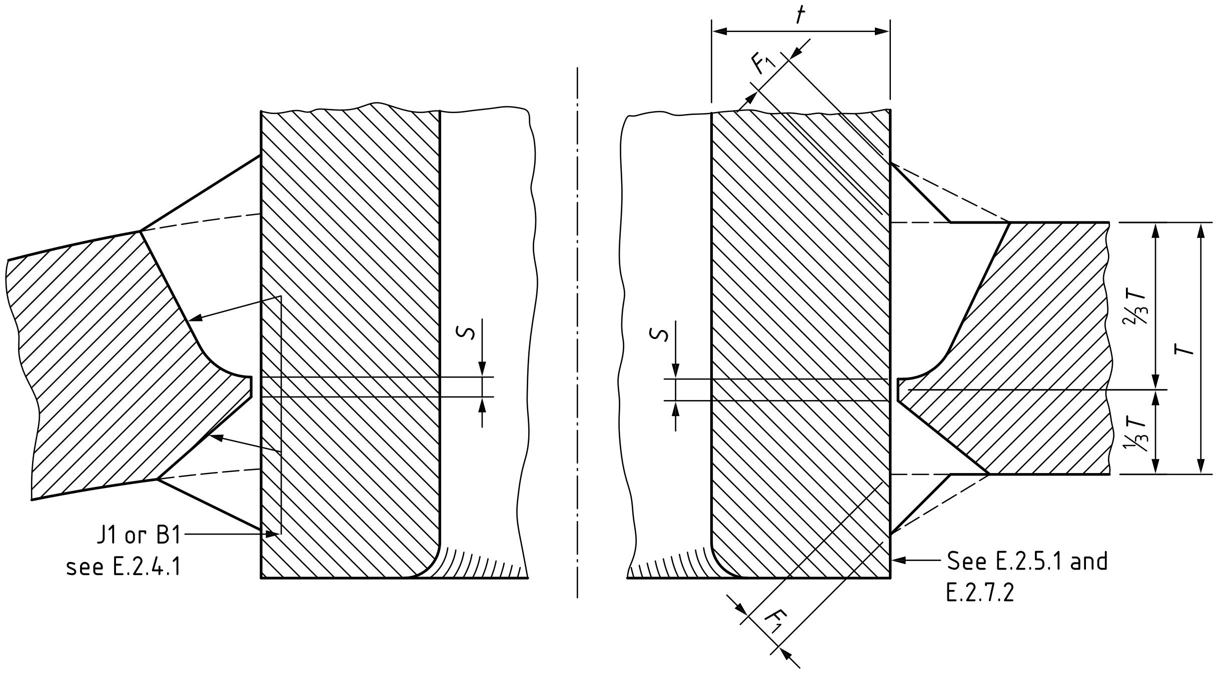


F_2 = T/5 min. or 6 mm whichever is larger

b)

NOTE Generally used when t is greater than T/2.

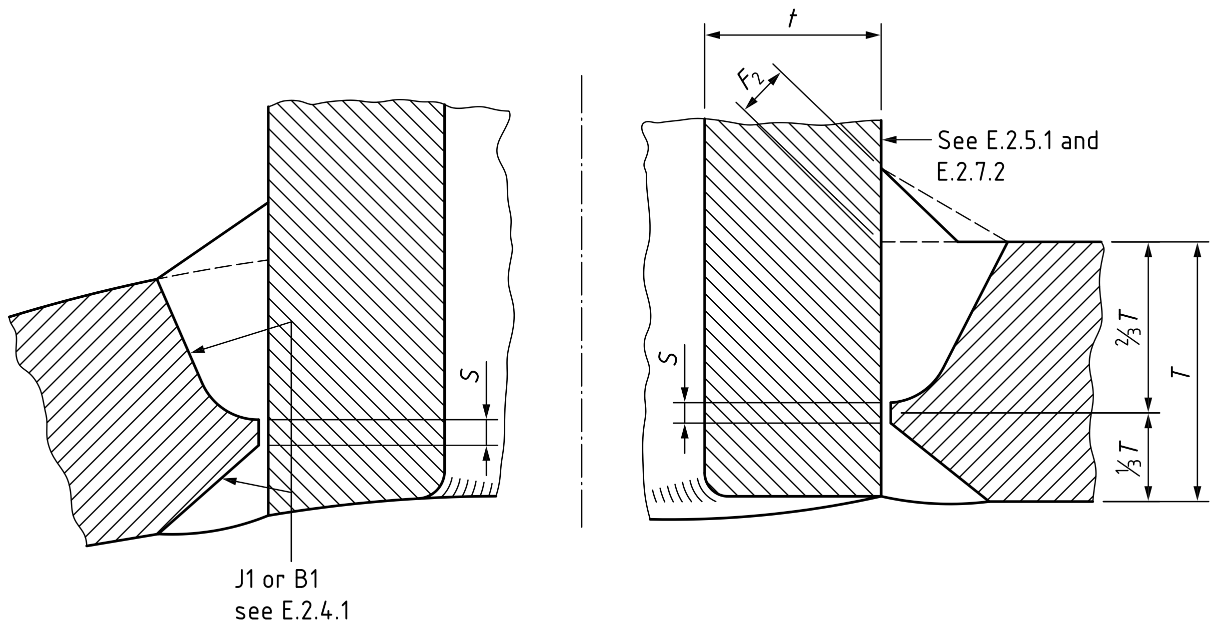
Figure E.20 **Set-in branches: full penetration connections with asymmetrical butt joints (see E.2.4.2)**



S = 1.5 mm to 2.5 mm

F_1 = T/10 min. or 6 mm whichever is larger

a)

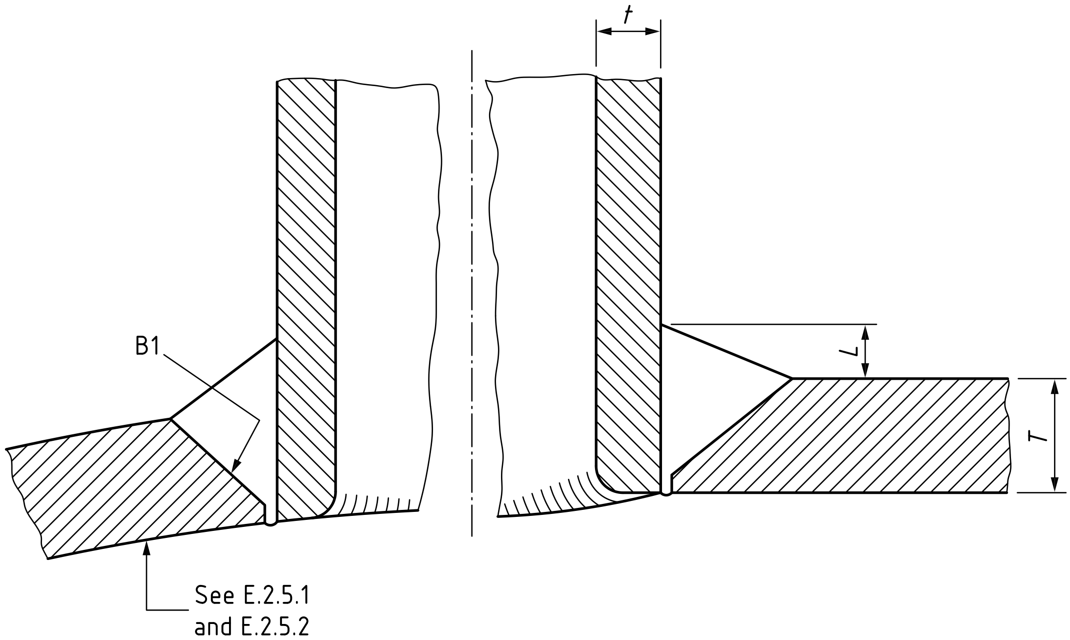


S = 1.5 mm to 2.5 mm

F_2 = T/5 min. or 6 mm whichever is larger

b)

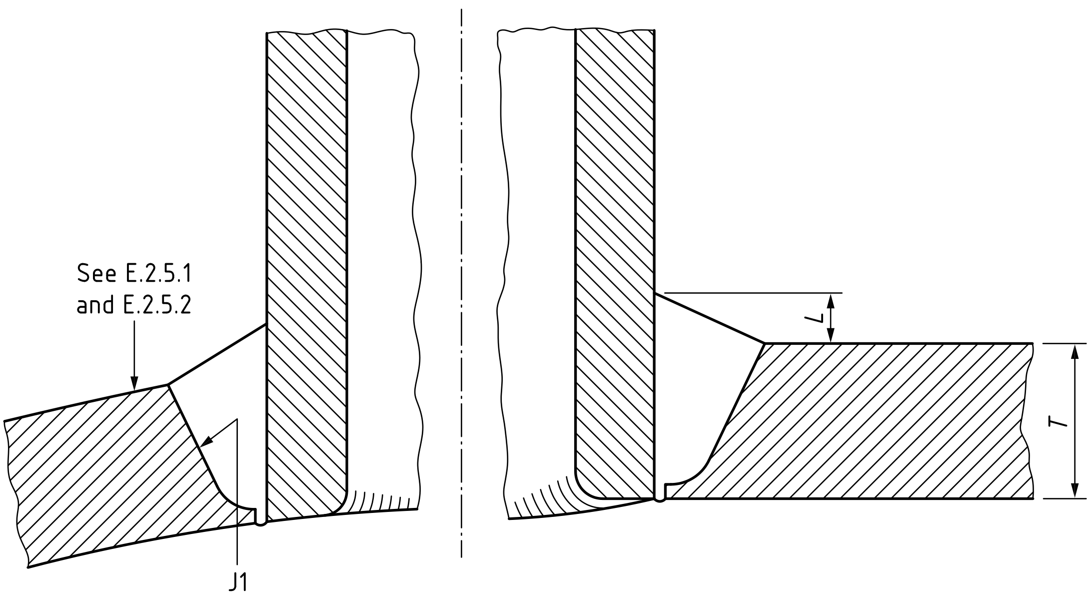
Figure E.21 **Set-in branches: full penetration connections welded from one side only**



L = *t*/3 min. but not less than 6 mm

T = 16 mm max.

a)



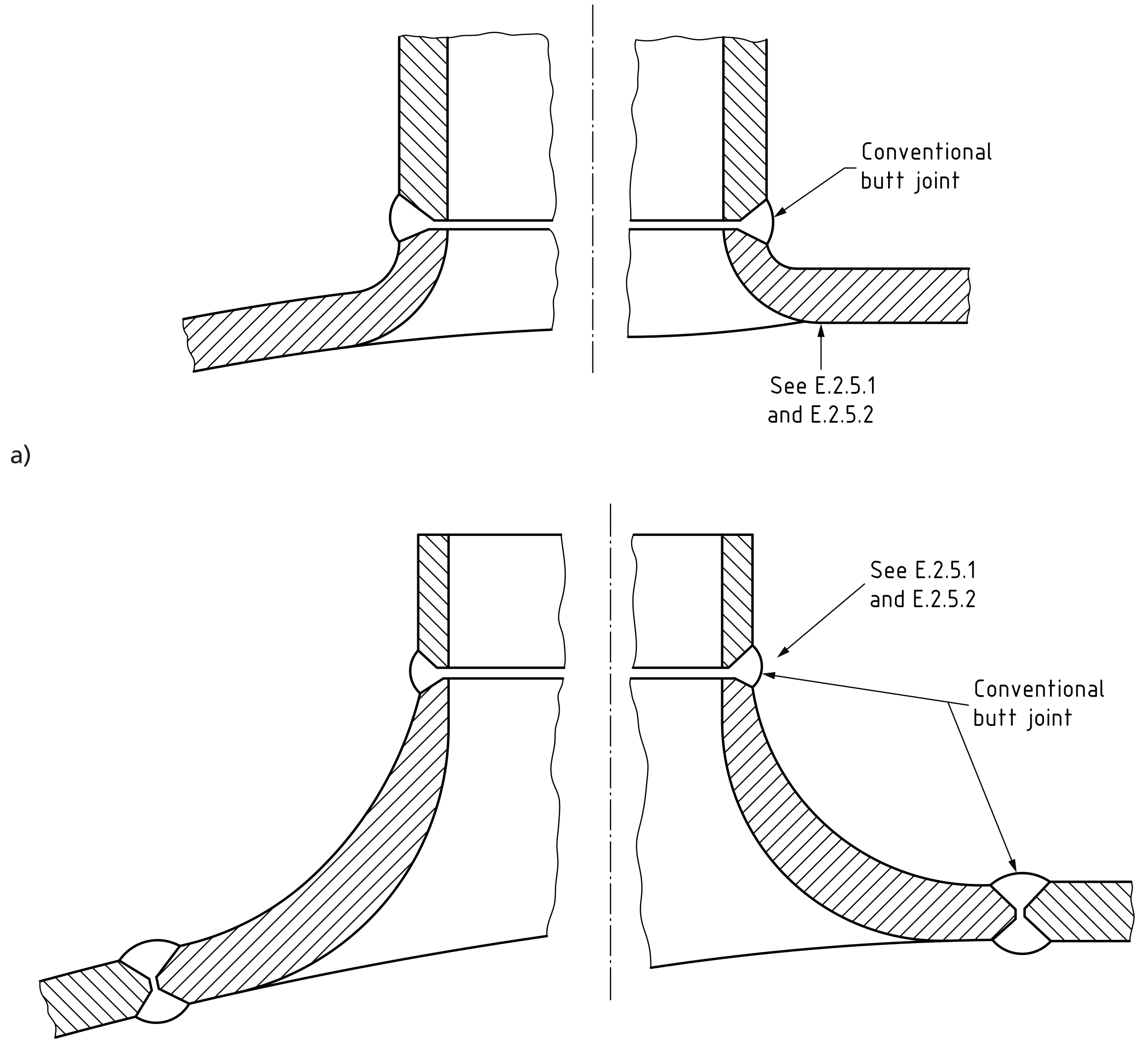
L = *t*/3 min. but not less than 6 mm

T = 25 mm max.

b)

NOTE As a general recommendation, all set-in branches should be welded on the inside of the shell as shown in Figure E.16a) to Figure E.20b) if they are accessible for the purpose, otherwise preference should be given to set-on branch connections shown in Figure E.9a) to Figure E.15c). However, the connections shown in Figure E.21a) and Figure E.21b) are considered to be acceptable but only if assurance can be provided that the welding procedure employed will ensure sound and consistent root conditions with uniform penetration.

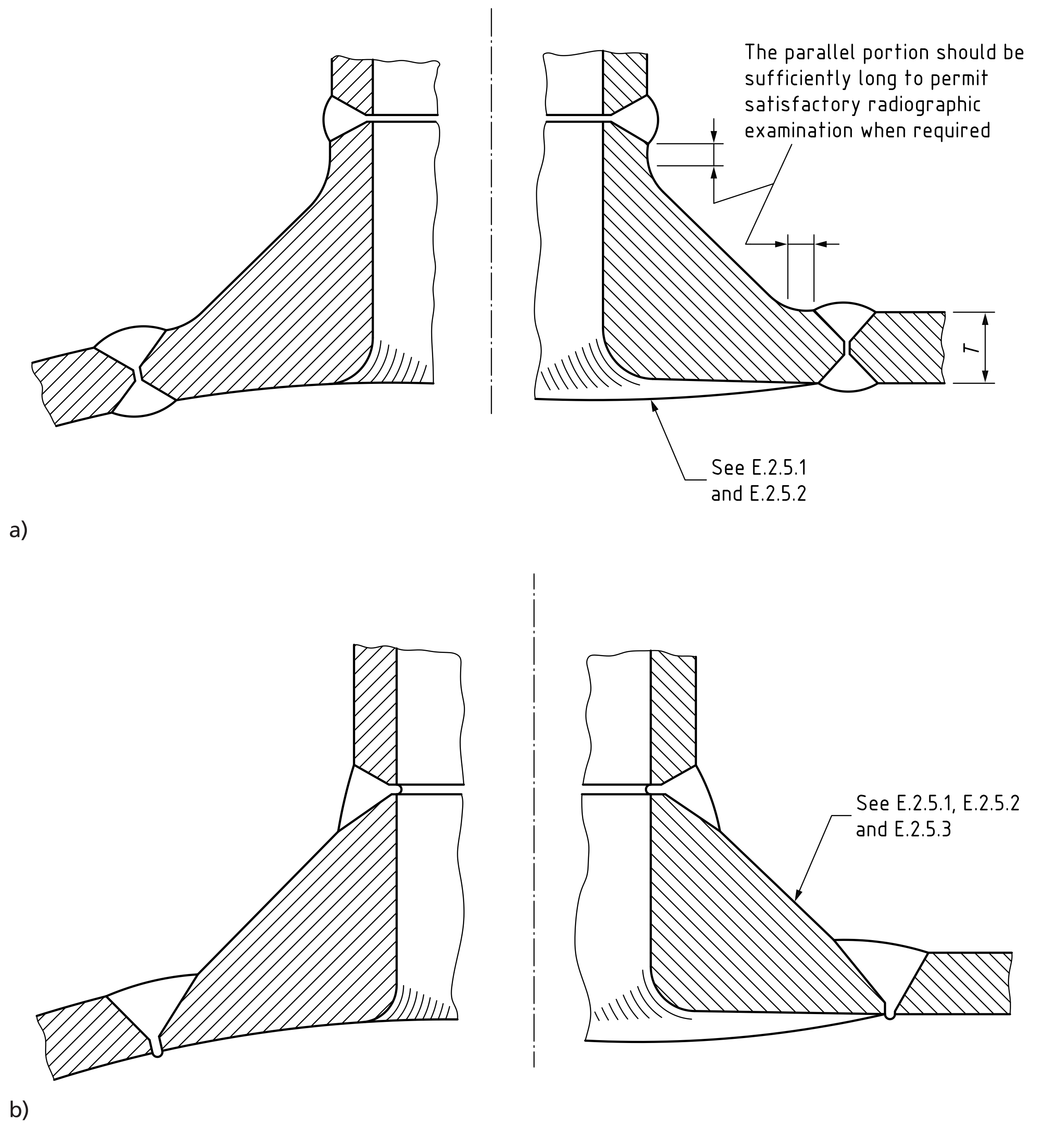
Figure E.22 **Forged branch connections (see E.2.4.1 and E.2.4.2)**



NOTE 1 Conventional butt joints are used to weld the forged branch connection to the shell and branch, and may not necessarily be of the form shown.

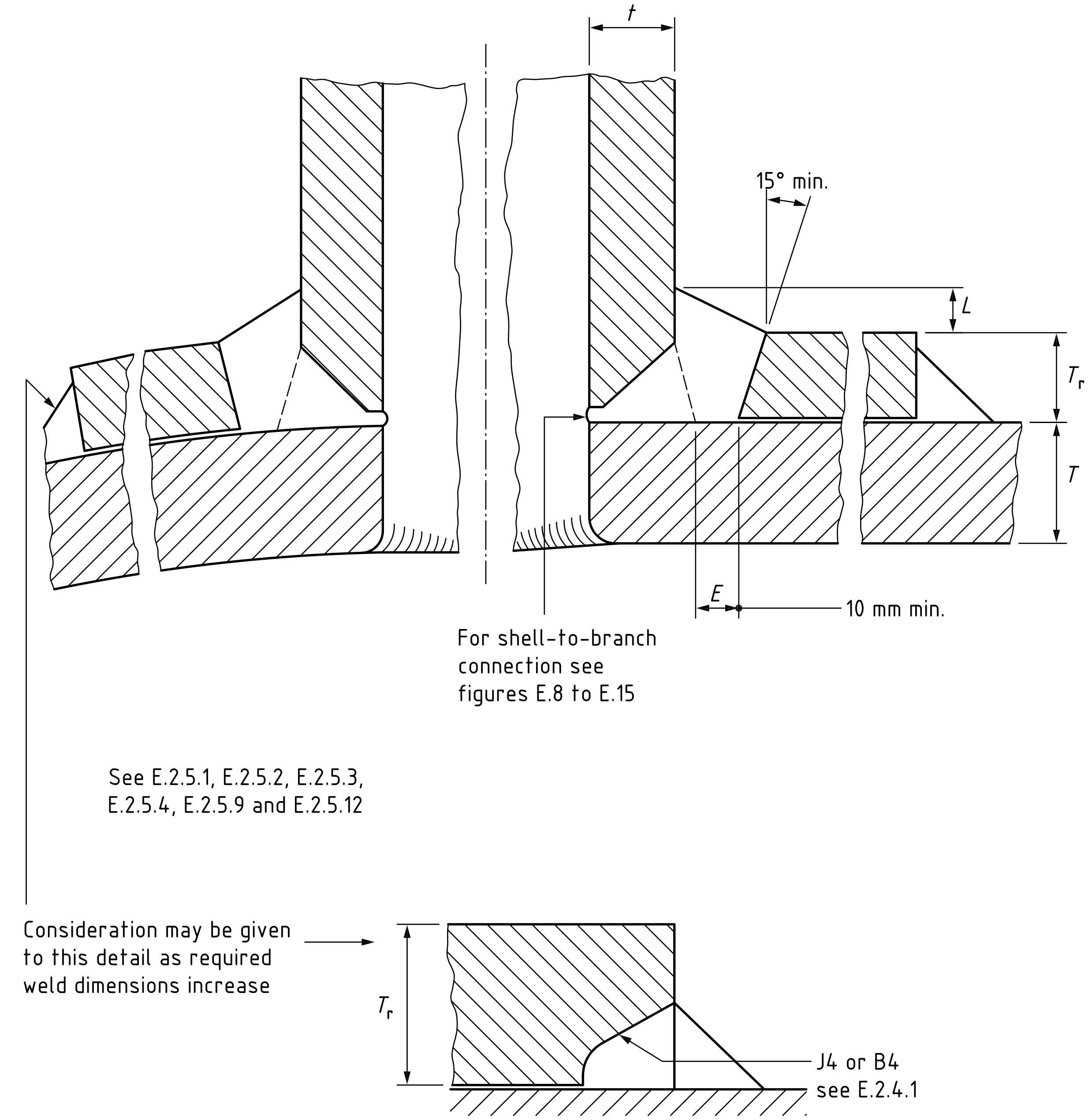
NOTE 2 Forging should be to good practice and within the correct range of temperature for the materials used.

Figure E.23 **Forged branch connections**



*NOTE Conventional butt joints are used to connect the forging to the shell and may not necessarily be of the form shown. See also **E.2.4.1** and **E.2.4.2**. These forgings connecting branches to shells are used with various forms of profile.*

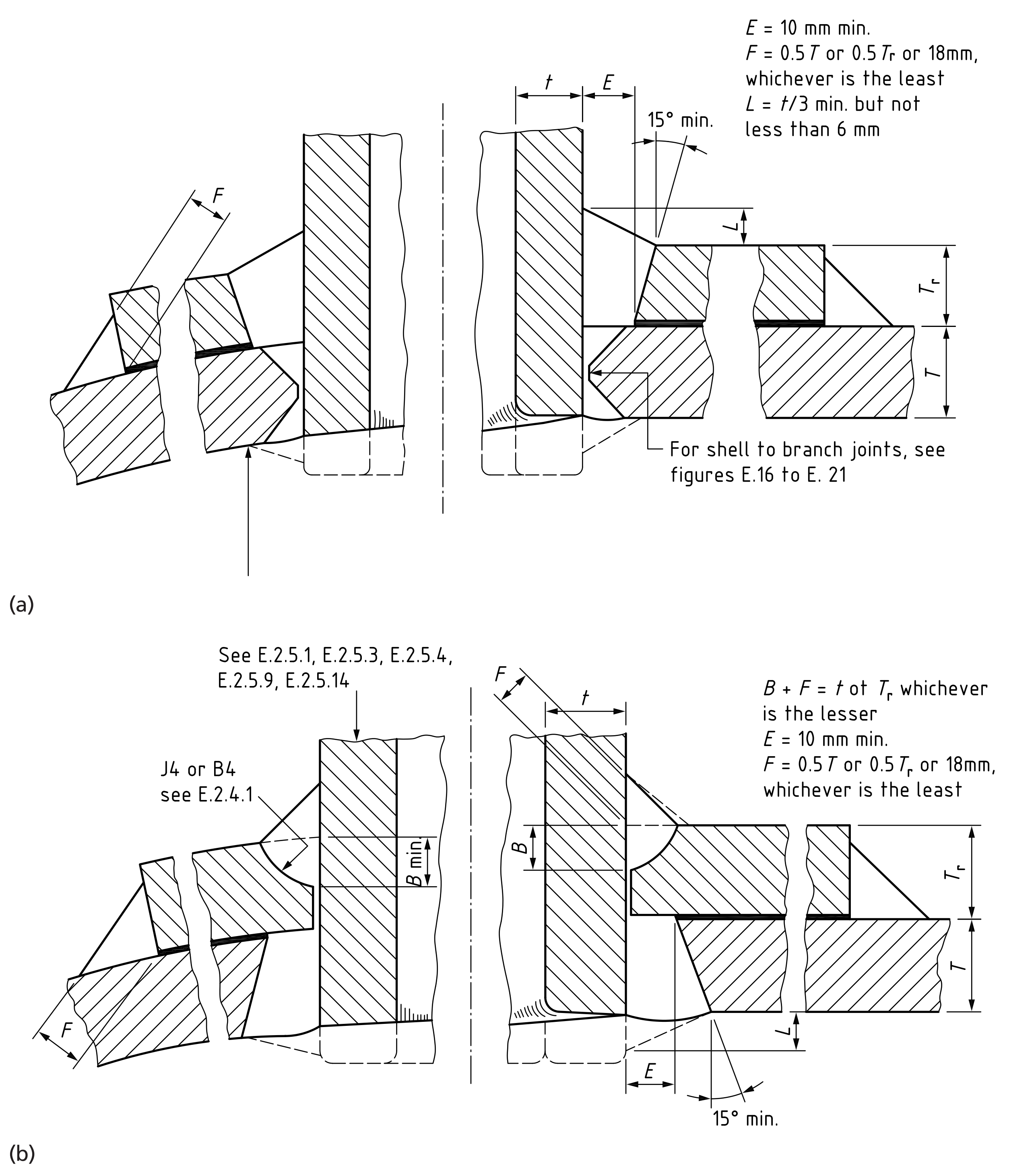
Figure E.24 **Set-on branches with added compensation rings**



$L = t/3$ min. but not less than 6 mm

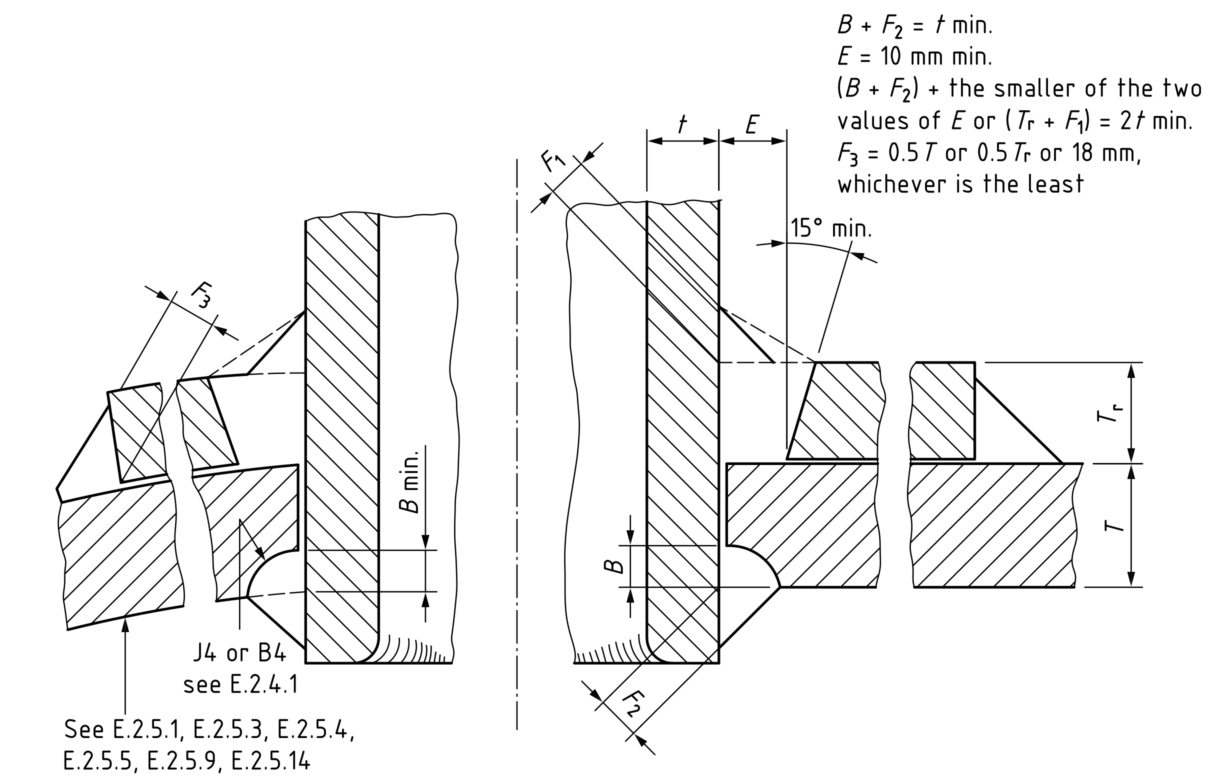
Dimensions are in millimetres

Figure E.25 **Set-in branches with added compensation rings**

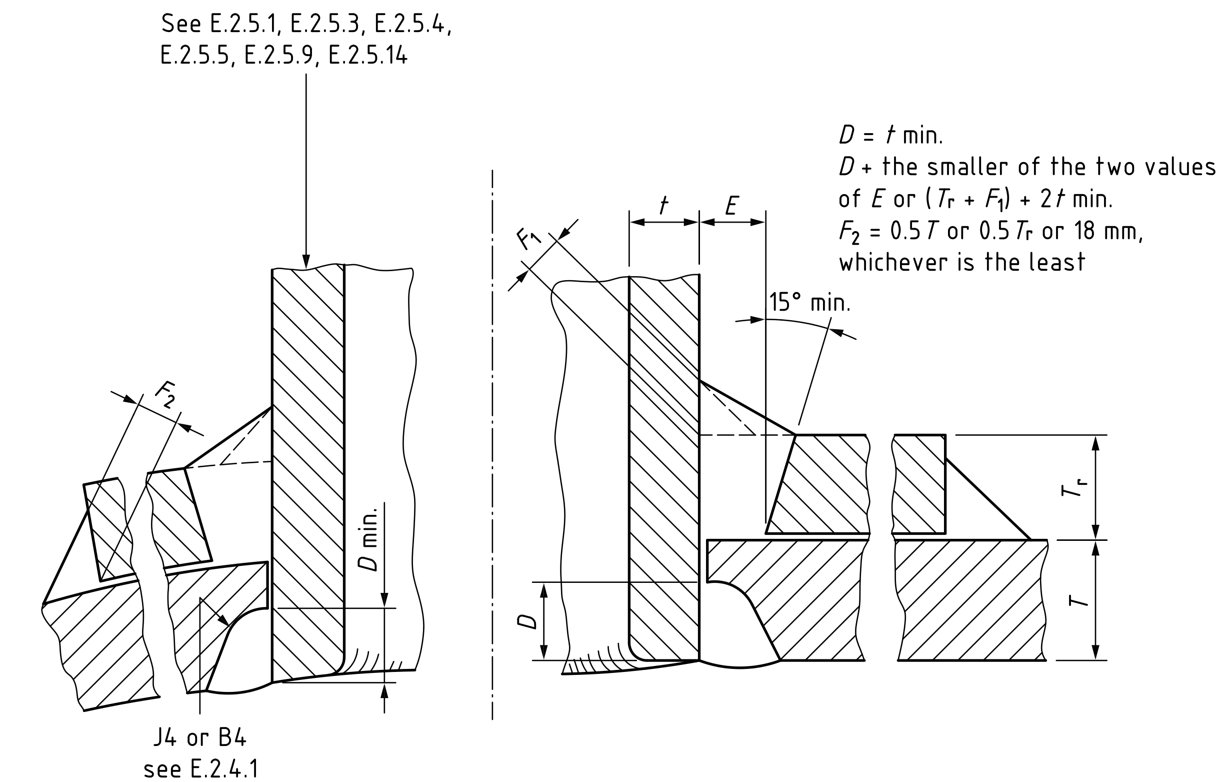


Dimensions in millimetres

Figure E.26 **Set-in branches with added compensation rings**



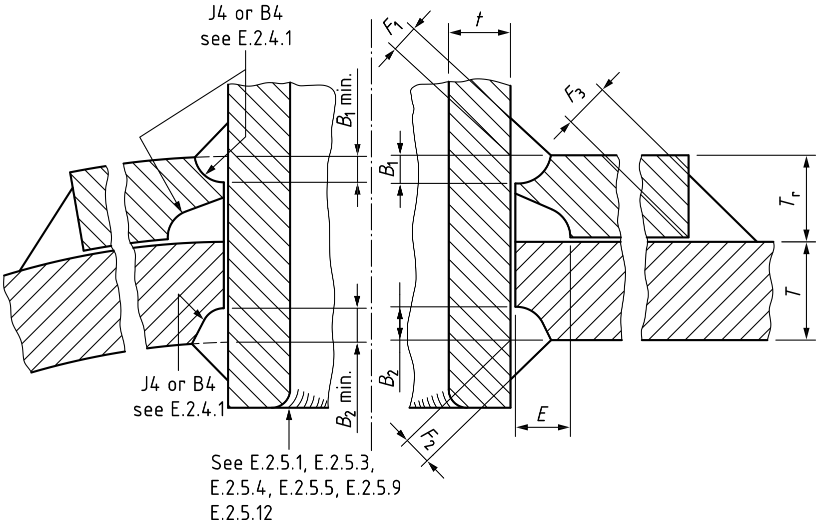
(a)



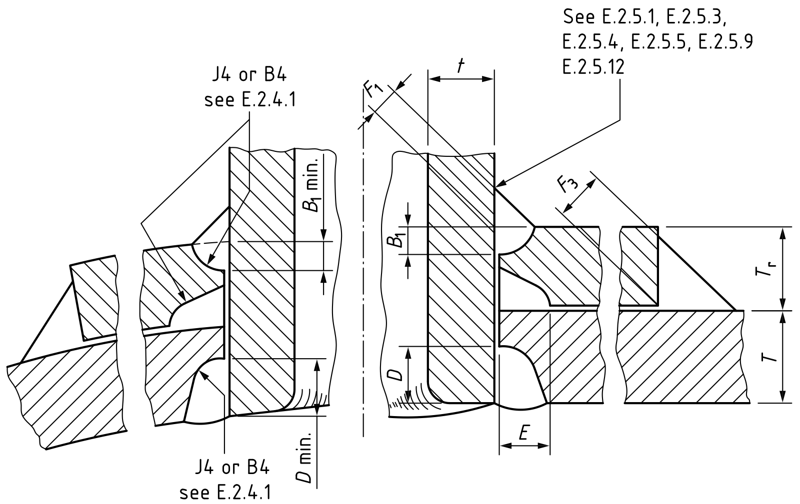
(b)

Dimensions in millimetres

Figure E.27 **Set-in branches with added compensation rings**



a)



b)

Weld sizes

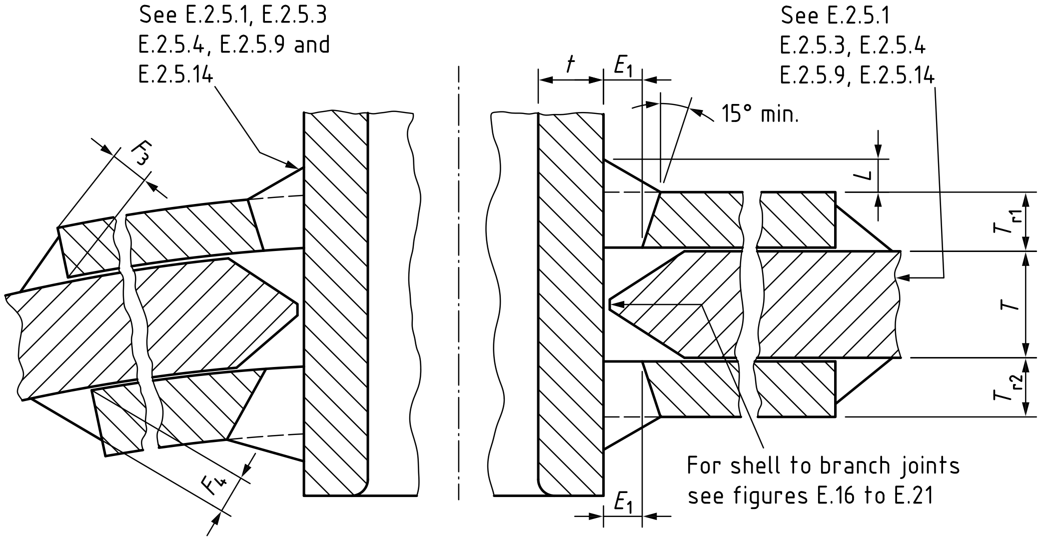
$F_3 = 0.5T$ or $0.5T_r$ or 18 mm, whichever is the least

When $T_r > t$	$(B_1 + F_1)$	$= t$
	E	$= t$
	$(B_2 + F_2)$	$= t$
When $T_r < t$	$(B_1 + F_1)$	$= T_r$
	E	$= T_r$
	$(B_2 + F_2)$	$= 2t - T_r$
When $T < t$	$(B_2 + F_2)$	$= T$

In the case of (b), for $(B_2 + F_2)$ substitute D.

Dimensions in millimetres

Figure E.28 **Set-in branches with added compensation rings**



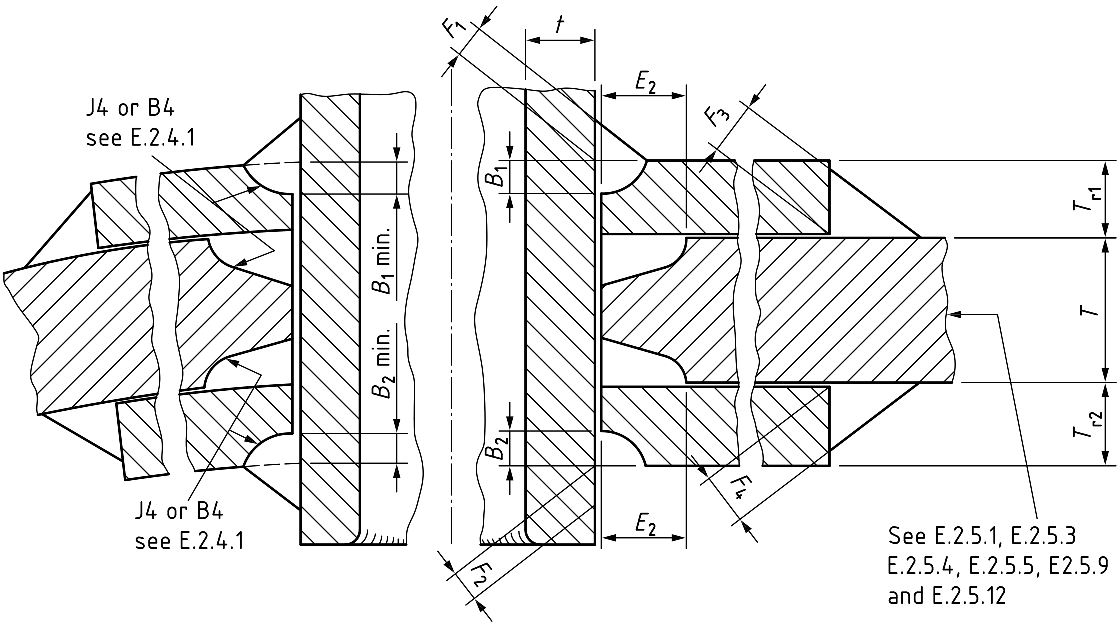
E = 10 mm min.

L = t/3 min. but not less than 6 mm

F_3 = 0.5T or 0.5T_{r1} or 18 mm, whichever is the least

F_4 = 0.5T or 0.5T_{r2} or 18 mm, whichever is the least

a) [See also Figure E.27a)]



$B_1 + F_1 = t$

$E_2 = t$ but not less than 10 mm

$B_2 + F_2 = t$

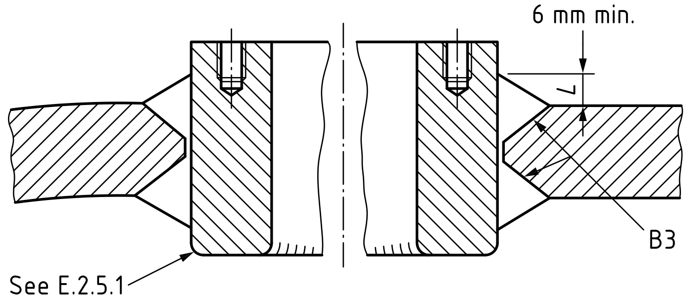
F_3 = 0.5T or 0.5T_{r1} or 18 mm, whichever is the least

F_4 = 0.5T or 0.5T_{r2} or 18 mm, whichever is the least

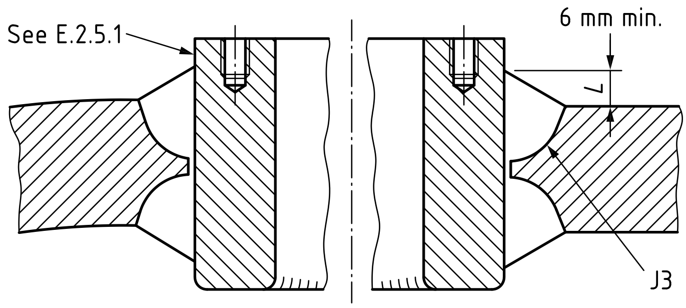
b) [See also Figure E.27a)]

Dimensions in millimetres

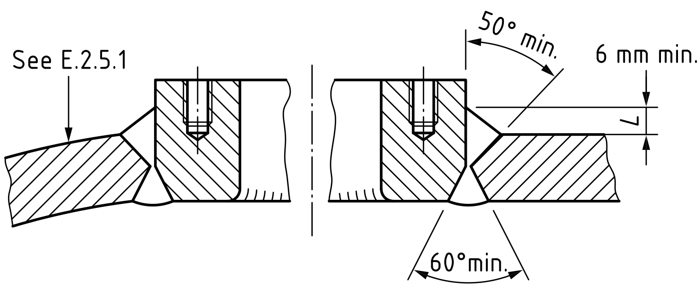
Figure E.29 **Studded connections (see also 3.5.4.8)**



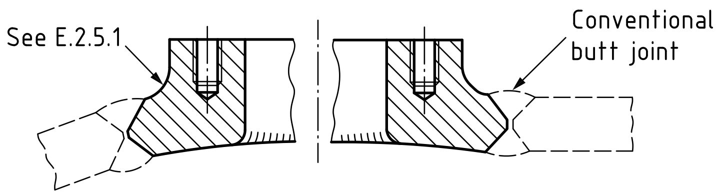
a)



b)



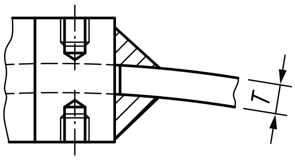
c)



d)

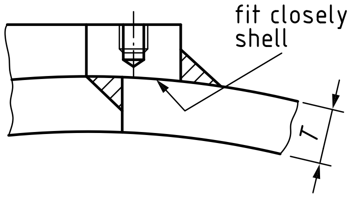
Butt welded studded connections (See **E.2.4.2**)

Figure E.29 **Studded connections (see also 3.5.4.8)** ***(continued)***

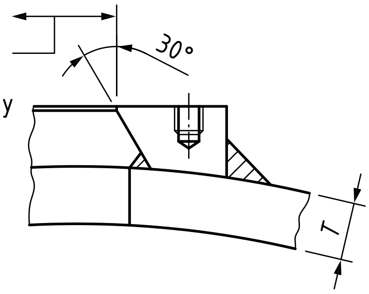


e)

The gap should not exceed 3 mm at any point



f)



g)

Fillet welded studded connections (See E.2.5.12)

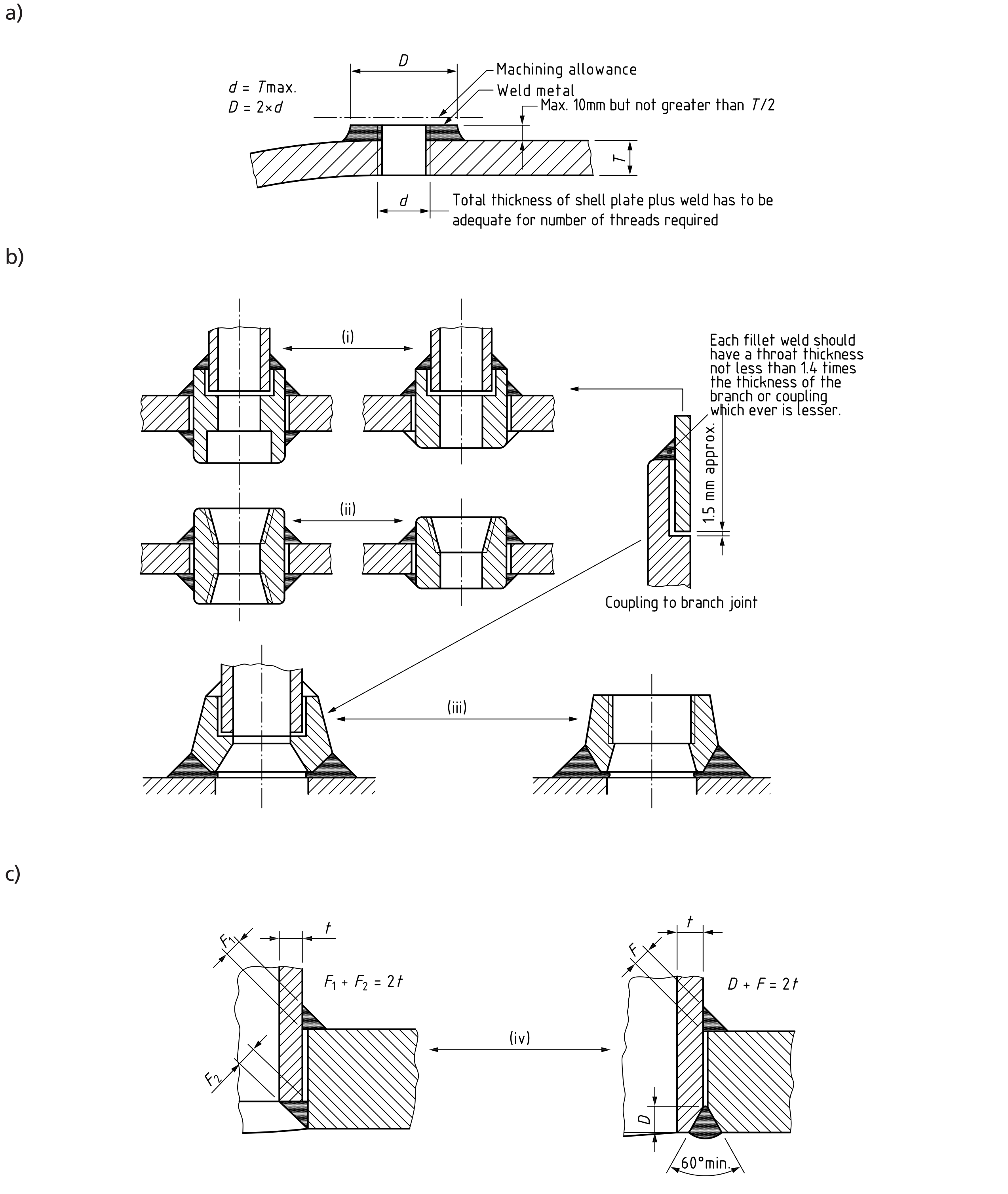
NOTE 1 Fillet welded details are not recommended if the vessel is subjected to pulsating loads when preference should be given to the details shown in a) to d).

NOTE 2 The sizes of the fillet welds should be based on the loads transmitted paying due regard to all fabrications and service requirements, but in any case should not be less than 6 mm.

NOTE 3 Each fillet weld should have a throat thickness not less than 0.7 times the thickness of the shell or pad whichever is the lesser.

Dimensions in millimetres

Figure E.30 **Socket welded and screwed connections (see also 3.5.4.8)**



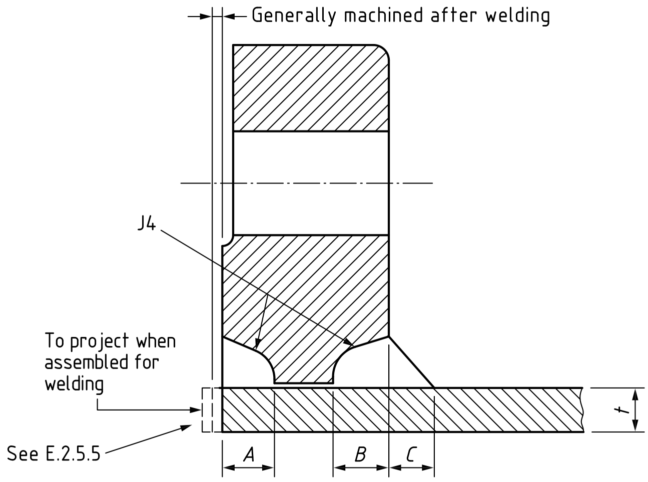
NOTE 1 Small couplings (i) to (iv) inclusive may be attached to shells by the connections shown above and by any other appropriate joint shown in Figure E.9 to Figure E.21.

*NOTE 2 For all sketches see **E.2.5.6**.*

*NOTE 3 For all sketches except for (a) and (b) (iii) see **E.2.5.12**.*

Dimensions in millimetres

Figure E.31 **Flanges**



Weld sizes

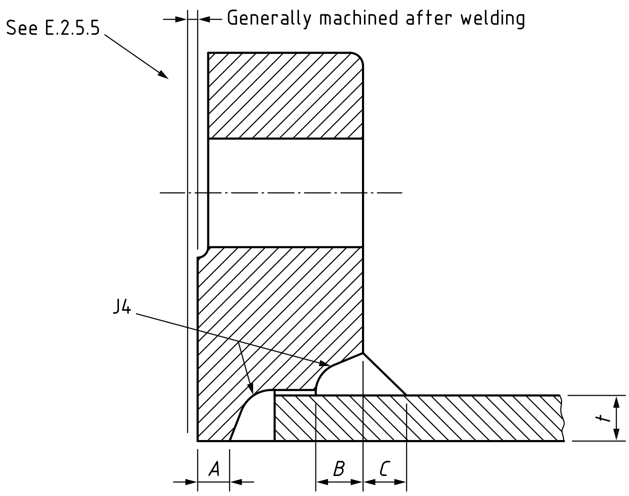
B = *t*

C = *t*

A = *t* (min.) after machining flange to final thickness

(See Note 1)

a) Face and back welded flange



Weld sizes

B = *t*

C = *t*

A = ½*t* but 5 mm min. after machining flange to final thickness

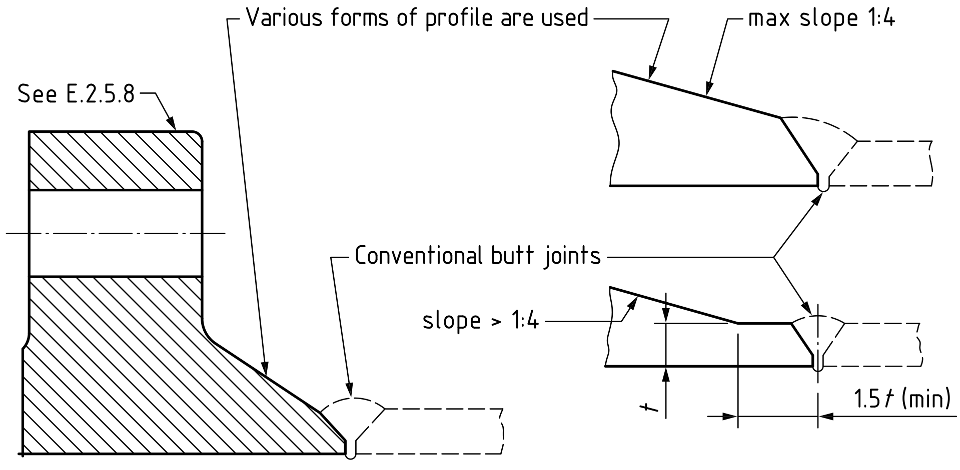
(See Note 1)

b) Bore and back welded flange

NOTE 1 The clearance between the bore of the flange and the outside diameter of the vessel should not exceed 3 mm at any point and the sum of the clearances diametrically opposite should not exceed 5 mm.

NOTE 2 The connections shown here are applicable as flat end connections, but see also ***E.2.5.11****.*

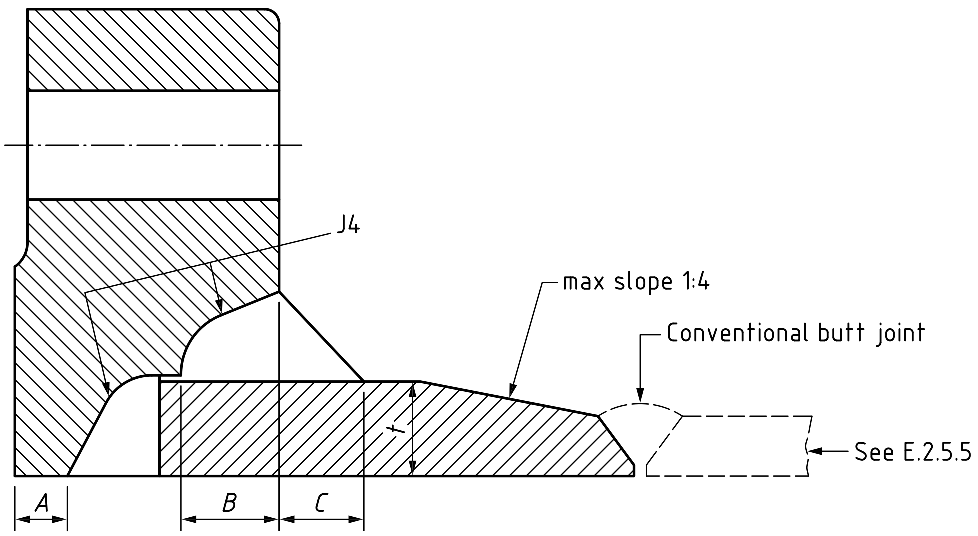
Figure E.32 **Flanges**



Alternative forms

*NOTE 1 The maximum hub slope of 1:4 is not applicable for flanges that conform to an appropriate standard for pipework flanges specified in **3.8.1** b) or c).*

a) Welding neck flange

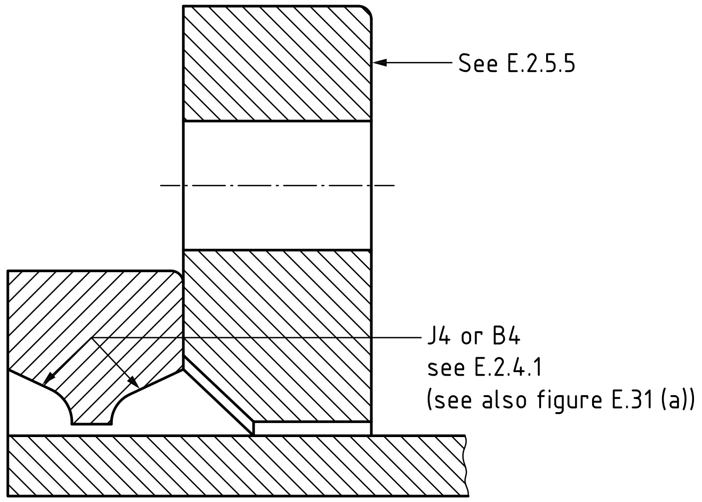


Weld sizes
B = *t*
C = *t*
A = ½*t* but 5 mm min. after machining flange to final thickness
(See Note 1 to Figure E.31)

*NOTE 2 The connection shown in b) is applicable as a flat end connection, but see also **E.2.5.11***

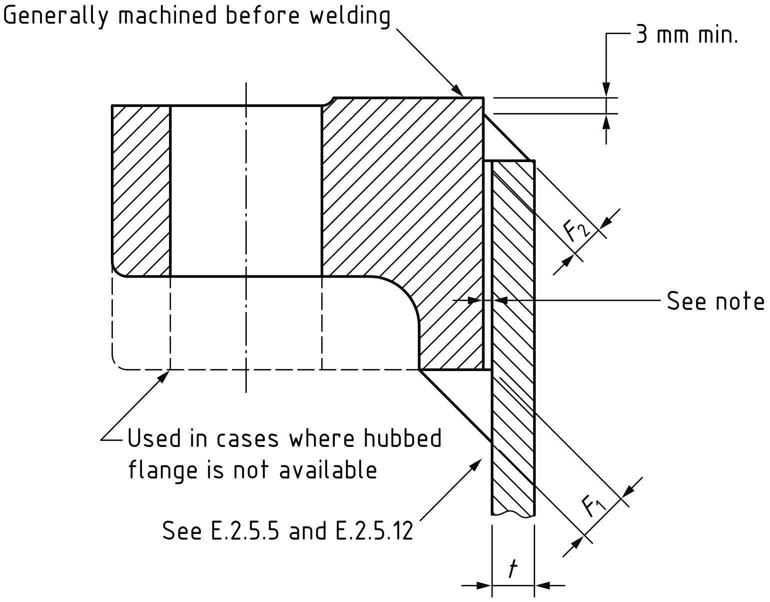
b) Welding neck flange (fabricated from plate)

Figure E.32 **Flanges *(continued)***



c) Lapped-type flange

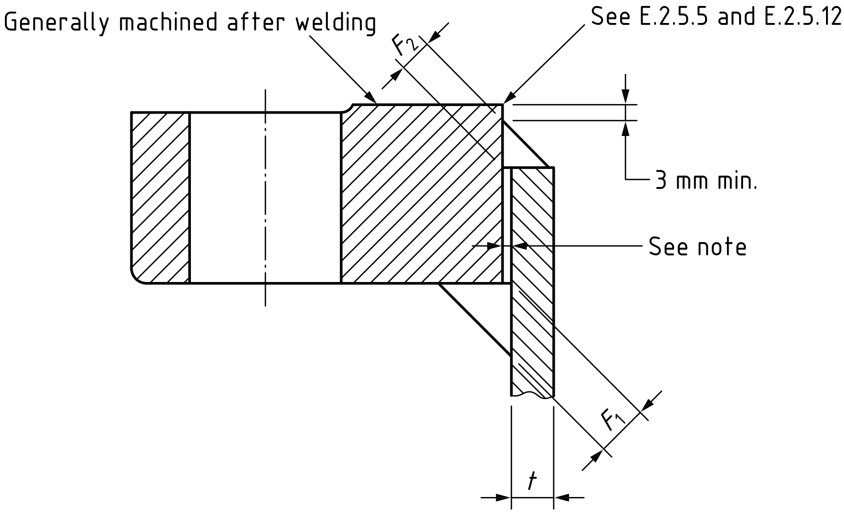
Figure E.33 **Flanges**



F_2 = 0.7t min.

F_1 = t min. but should not exceed 16 mm. See alternative details in Figure E.31a) and Figure E.31b)

a) Hubbed flange



F_2 = 0.7t min.

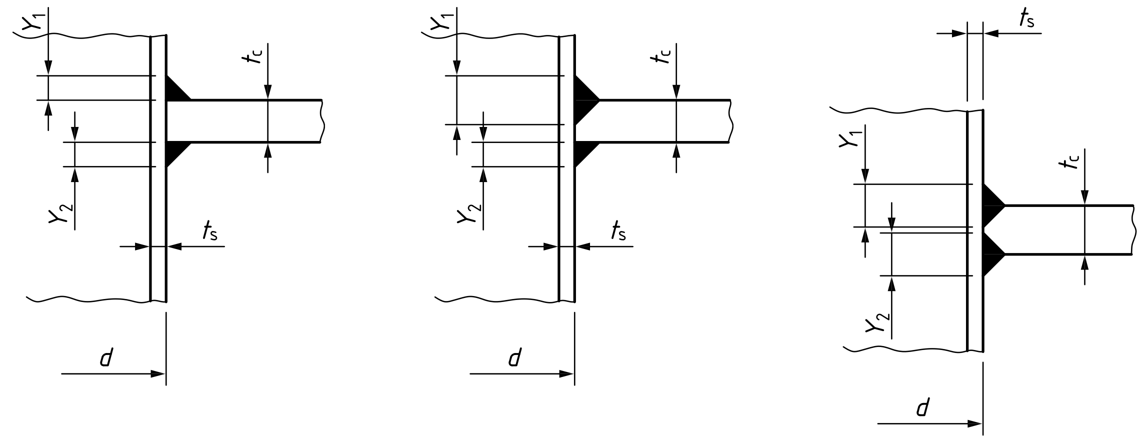
F_1 = t min. but should not exceed 16 mm. See alternative details in Figure E.31a) and Figure E.31b)

b) Fillet welded flange

NOTE The clearance between the bore of the flange and the outside diameter of the shell or branch should not exceed 3 mm at any point and the sum of the clearances diametrically opposite should not exceed 5 mm.

Dimensions in millimetres

Figure E.34 **Jacketted vessels: typical vessel/blocking ring attachments (see E.2.5.1, E.2.5.5 and E.2.5.7)**

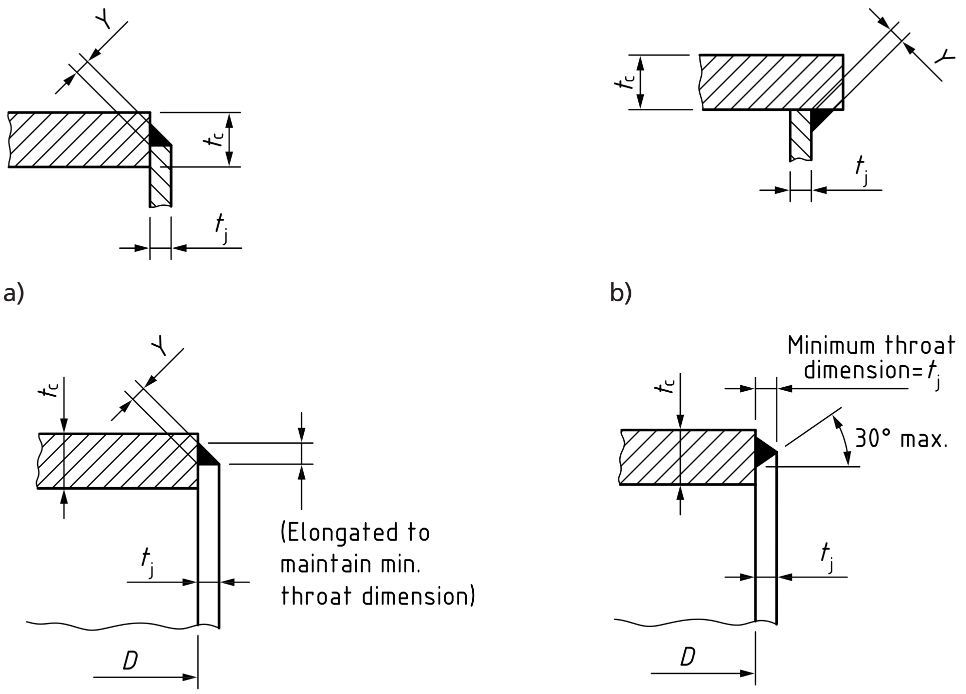


a) b) c)

$Y_1 + Y_2 \geq 1.5t_c$ or $1.5t_s$
(whichever is smaller)

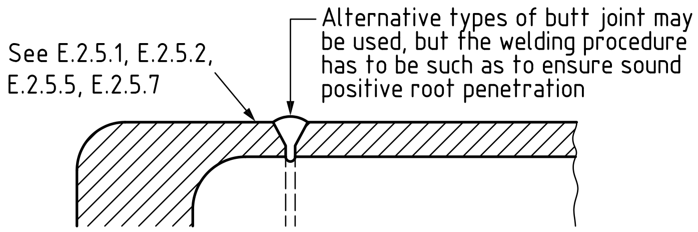
Where d is the outside diameter of the vessel, t_c is the nominal thickness of the blocking ring and t_s is the nominal thickness of the vessel.

Figure E.35 **Jacketted vessels: typical blocking ring/jacket attachments (see E.2.5.1, E.2.5.5 and E.2.5.7)**

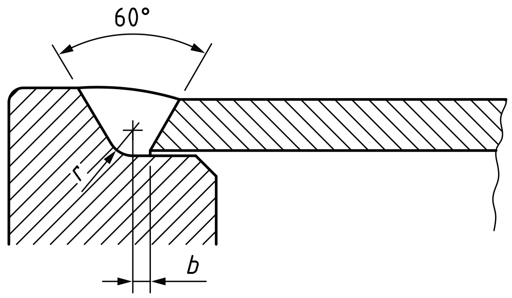


a) b) c) d)

Figure E.35 **Jacketted vessels: typical blocking ring/jacket attachments (see E.2.5.1, E.2.5.5 and E.2.5.7)** ***(continued)***



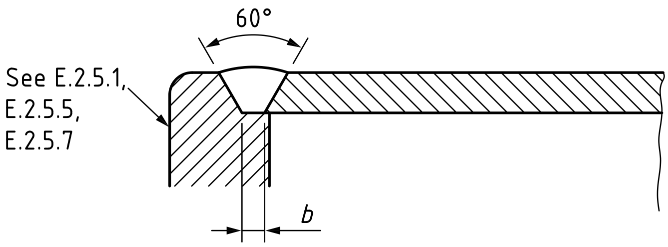
e)



r = 4 mm

b = 3 mm

f)



b = 6 mm

g)

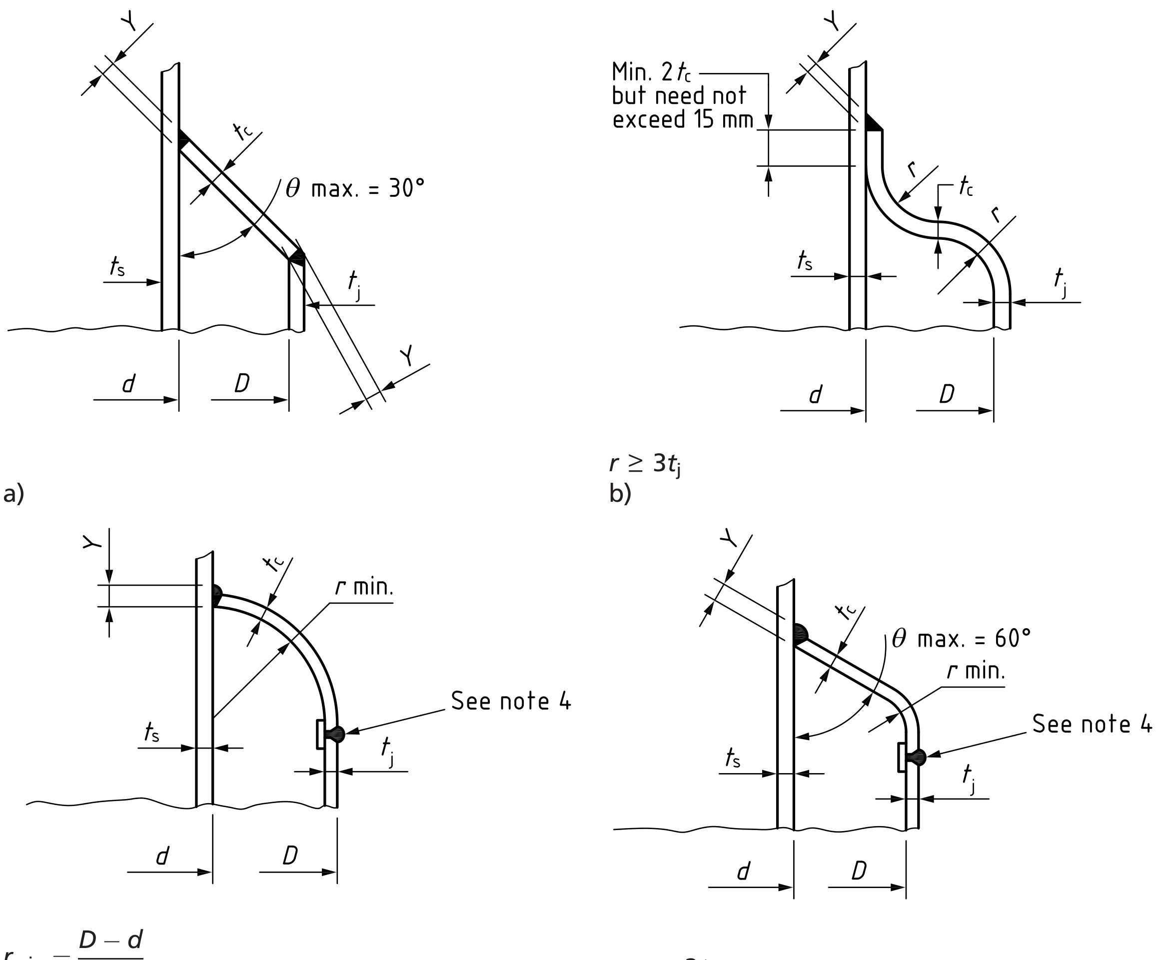
Where D is the inside diameter of the jacket, t_c is the nominal thickness of the blocking ring and t_j is the nominal thickness of the jacket.

NOTE 1 For a) and b), $Y \geq 0.7t_j$; these welds are recommended for type 1 jackets.

NOTE 2 For c), $Y \geq 0.83t_j$; these welds are recommended for type 2 jackets.

NOTE 3 For d), e), f) and g), these welds are recommended for both type 1 and 2 jackets.

Figure E.36 **Jacketted vessels: typical sealer rings (see E.2.5.1, E.2.5.5 and E.2.5.7)**



a)

$r \geq 3t_j$

b)

$r_{min} = \dfrac{D - d}{2}$

c)

$r_{min} = 2t_c$

d)

Where D is the inside diameter of the jacket, d is the outside diameter of the shell, t_c is the nominal thickness of the sealer ring, t_j is the nominal thickness of the jacket and t_s is the nominal thickness of the shell.

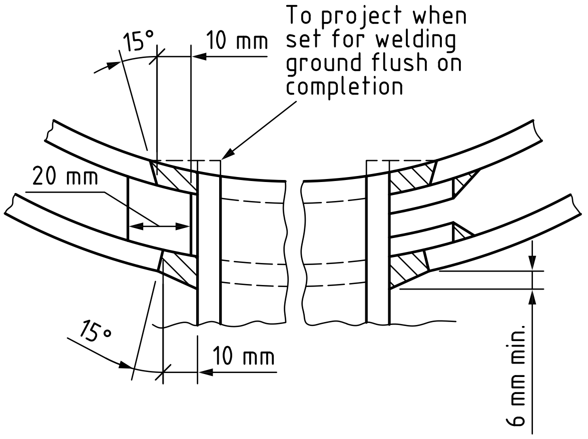
NOTE 1 For a), $Y = t_c$; this is recommended for type 1 jackets only.

NOTE 2 For b), $Y = 0.7t_c$ for type 1 jackets and $Y = 0.83t_c$ for type 2 jackets. This is recommended where $t_j \leq 16$ mm.

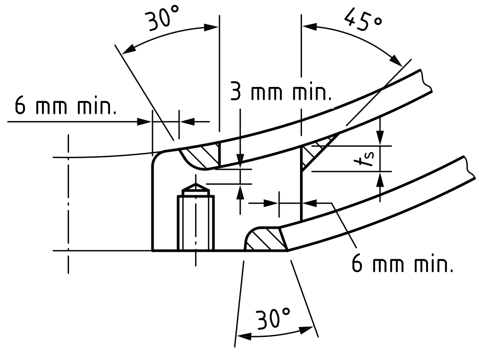
NOTE 3 For c) and d), $Y = 1.25t_c$ for type 2 jackets. For type 1 jackets a fillet weld ($Y = 0.7t_c$) may be used.

NOTE 4 For the sealer ring to shell welds and jacket to sealer ring welds (if any) the welding procedure should ensure sound root penetration.

Figure E.37 **Jacketted vessels: typical through connections**

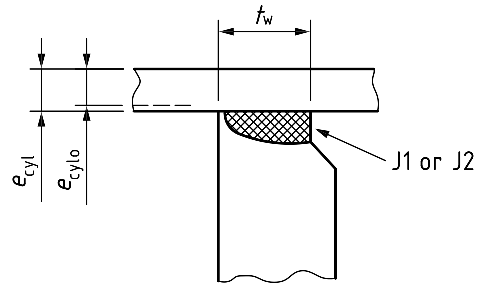


Flush type branch attatchment using a block (left-hand side) or backing rings (right-hand side)
a)



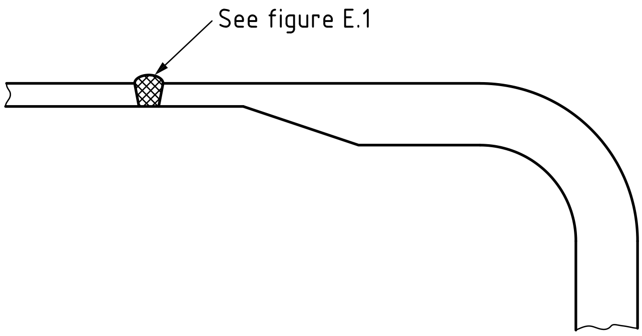
b)

Figure E.38 **Flat ends and covers (see E.2.5.1)**

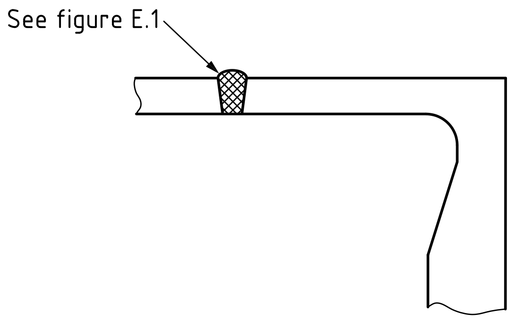


$t_w = 2e_{cylo}$ but not less than $1.25e_{cyl}$

a) Welded from one side only



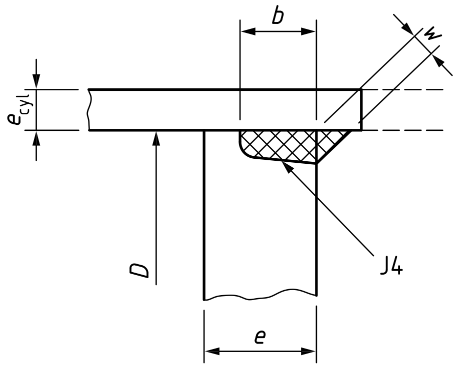
b) Butt joints (see **E.2.5.8**)



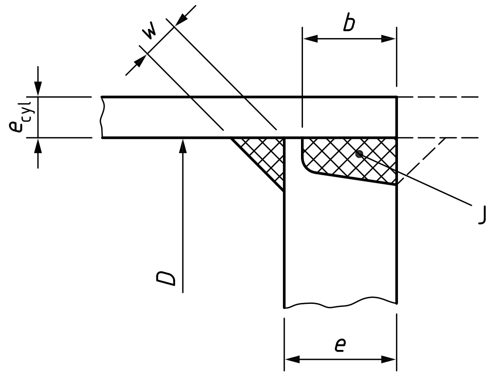
c) Butt joints (see **E.2.5.8**)

Dimensions in millimetres

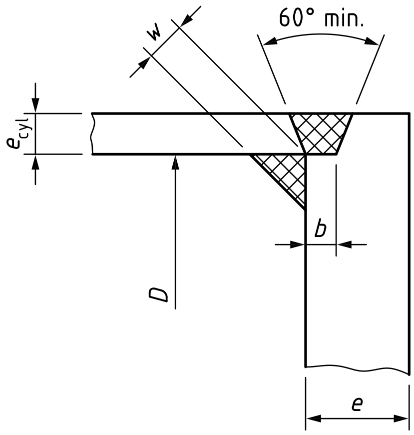
Figure E.38 **Flat ends and covers (see E.2.5.1)** ***(continued)***



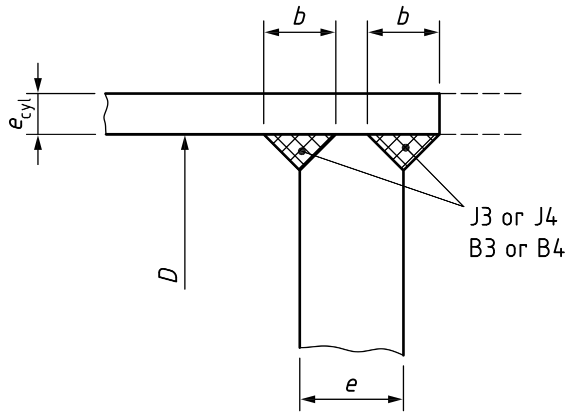
d) Welded from one side only
$b \geq 2e_{cyl}$ or e - 1.5 mm, whichever is less.
$w \geq 0.7e_{cyl}$ or 5 mm, whichever is less.
See **E.2.5.12**



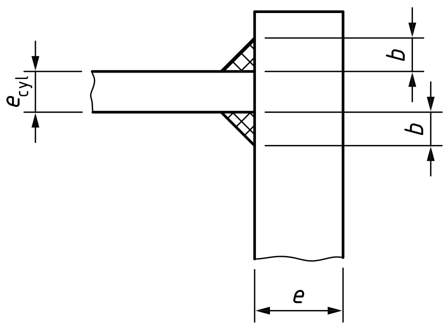
e) Welded from both sides
$b \geq 2e_{cyl}$ or e - 1.5 mm, whichever is less.
$w \geq 0.7e_{cyl}$ or 3 mm, whichever is less.



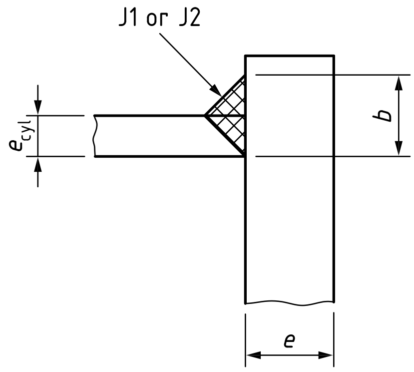
f) Welded from both sides (see **E.2.5.1**)
Penetration weld $b \geq 6$ mm
Throat of fillet weld $w \geq 0.25e_{cyl}$ or
5 mm, whichever is less



g) Welded from both sides
Penetration welds $b \geq e_{cyl}$

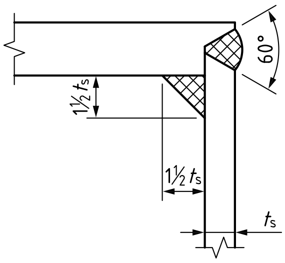


h) Welded from both sides
$b \geq e_{cyl}$
See **E.2.5.12**



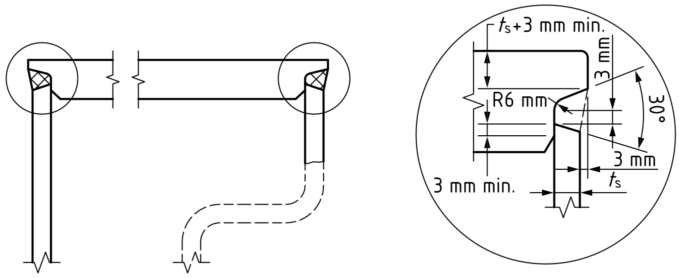
i) Welded from one side only
$b \geq 2e_{cyl}$
See **E.2.5.12**

Figure E.38 **Flat ends and covers (see E.2.5.1)** ***(continued)***



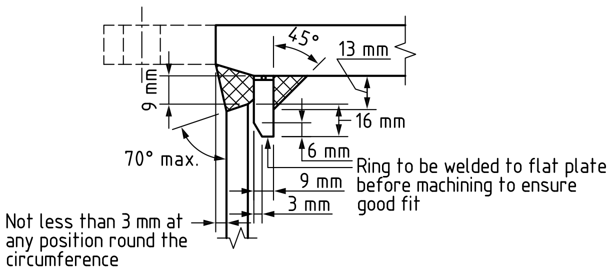
j)

NOTE This form of construction should not be used on vessels with an internal diameter exceeding 610 mm

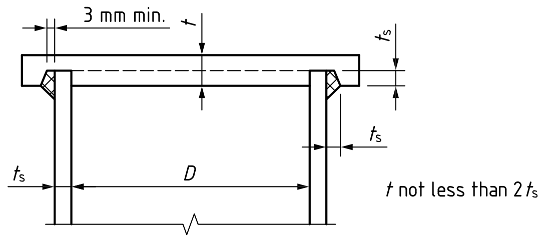


k)

See **E.2.5.12**



l)

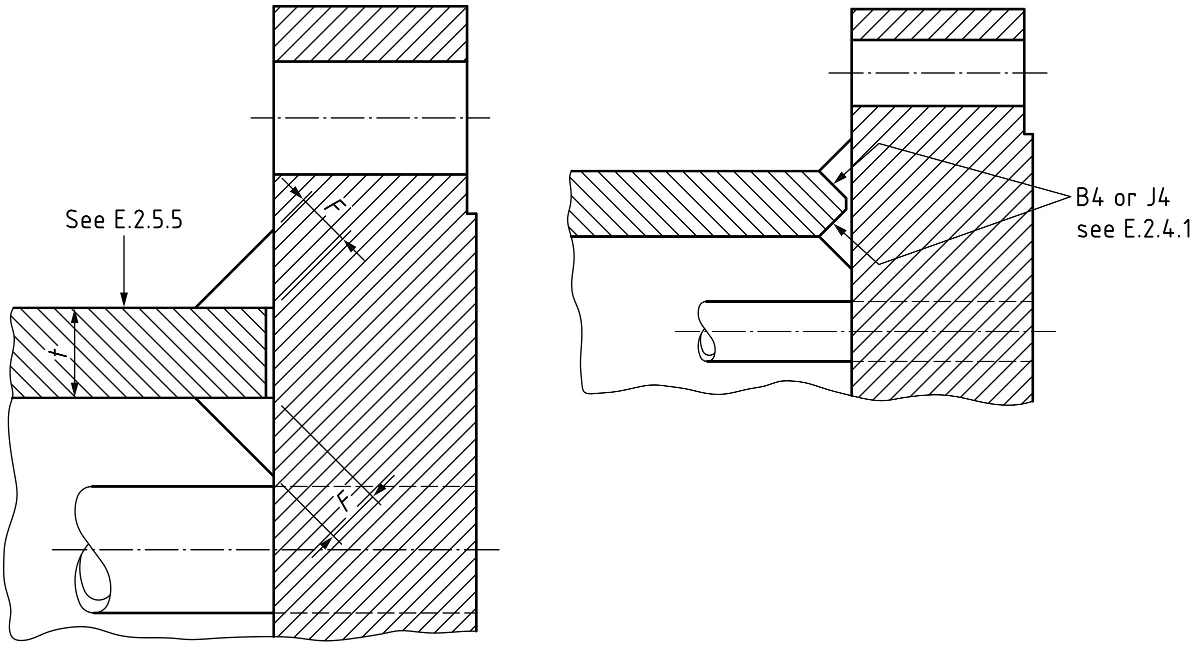


m)

See **E.2.5.12)**

NOTE For details of weld preparations (J1 etc.) see Figure E.6.

Figure E.39 **Tubeplate to shell connections: accessible for welding on *both* sides of the shell**



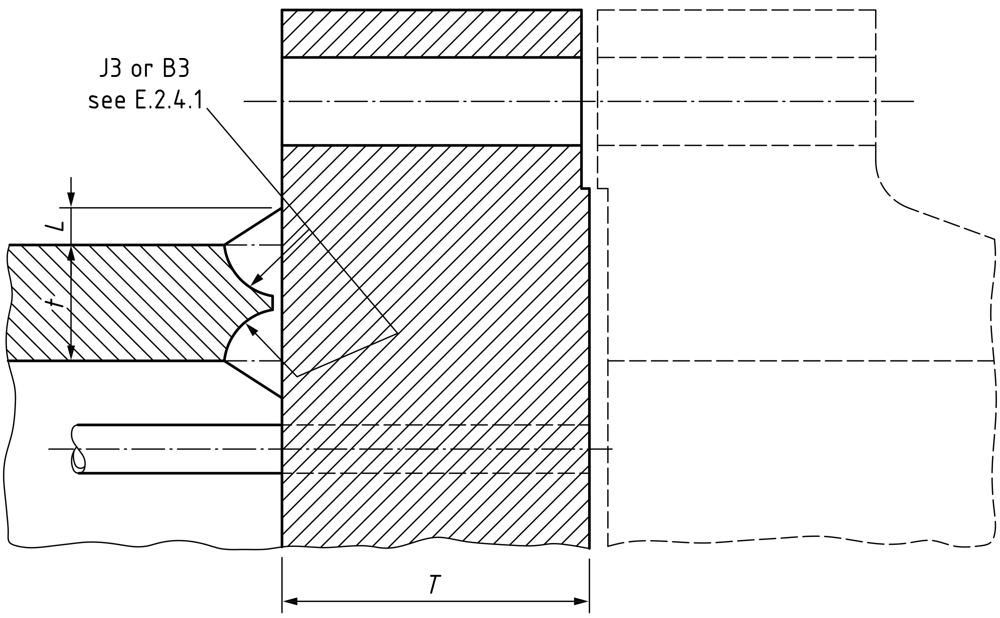
Alternative shell to tubeplate joint with partial penetration

a)

Weld size

$F = 0.7t$ min.

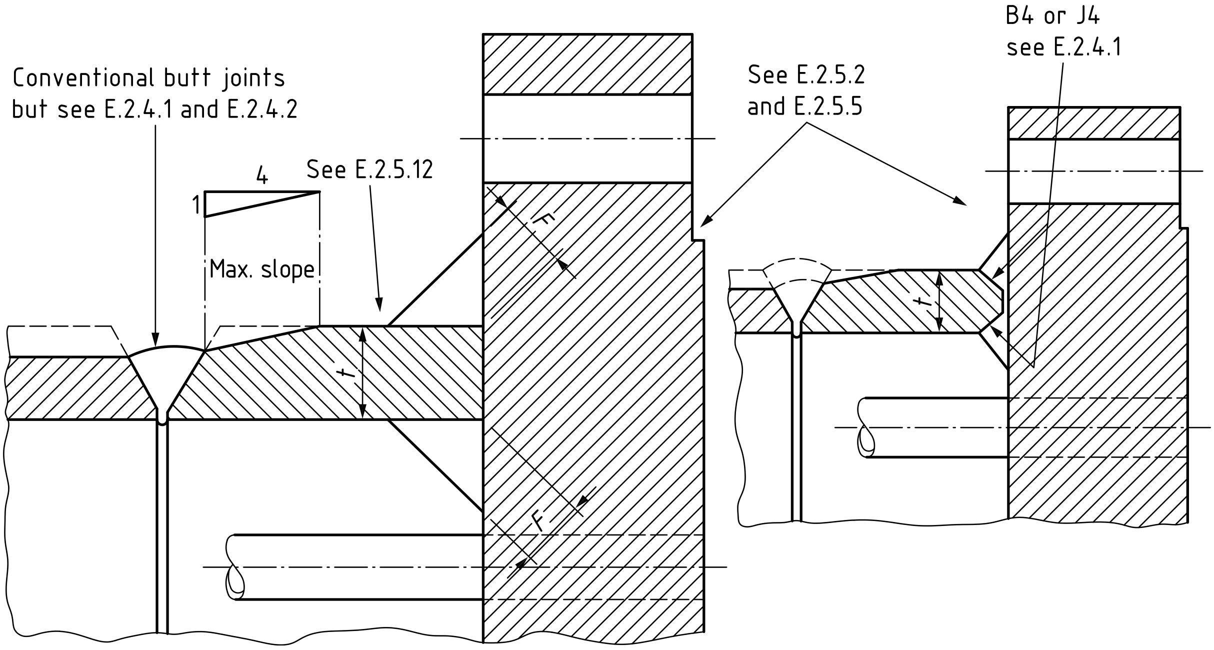
If t exceeds 16 mm, the shell should be bevelled as shown in the alternative sketch above or consideration should be given to the full penetration joint shown in b).



$L = t/3$ but not less than 6 mm.

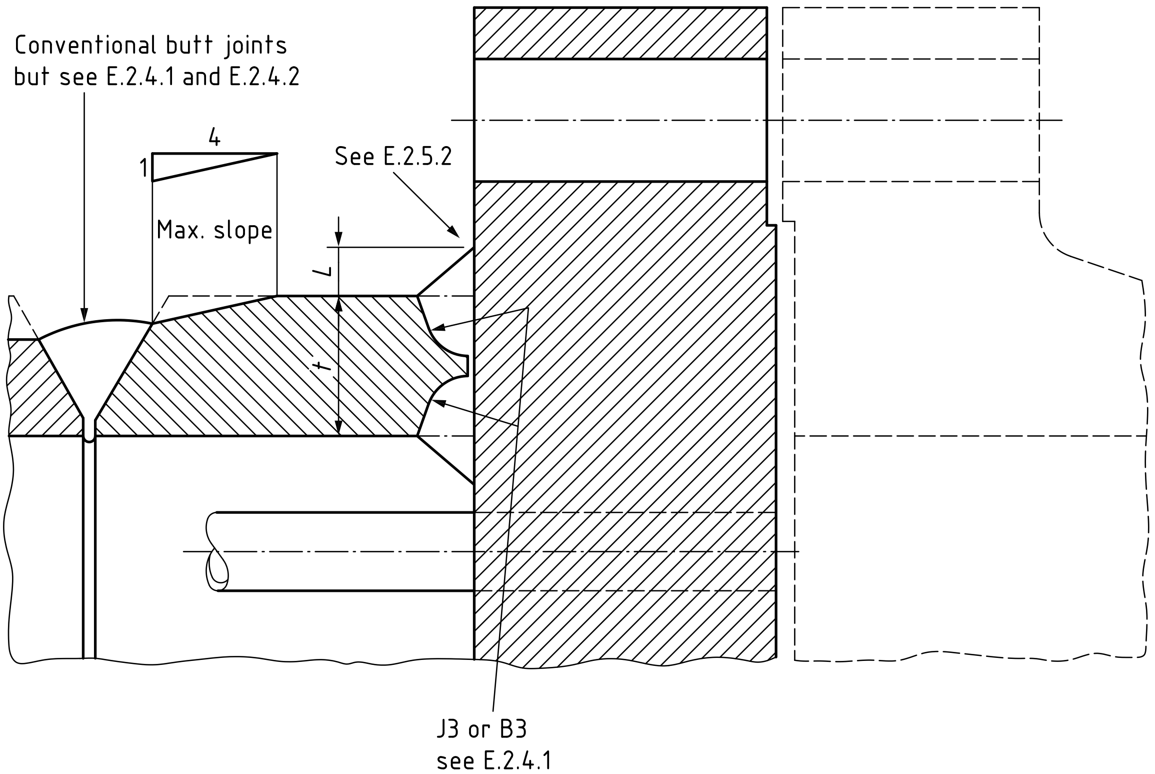
b)

Figure E.40 **Tubeplate to shell connections: accessible for welding from *outside* of shell only**



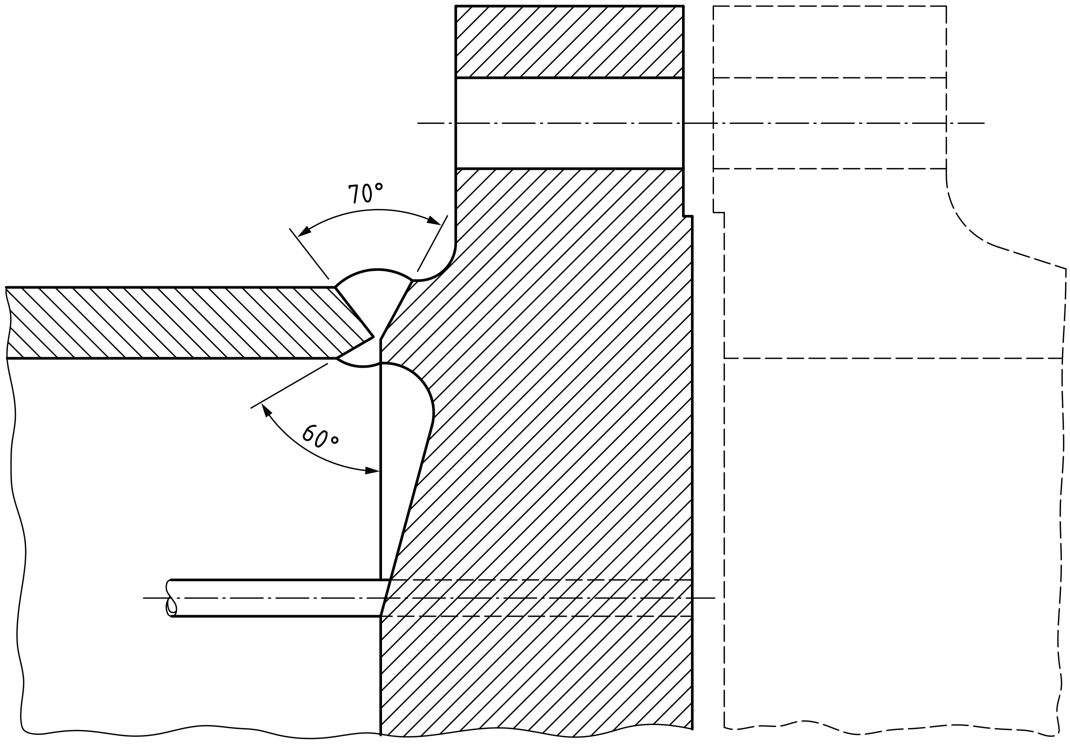
a) Alternative shell to tubeplate joint with partial penetration

Weld size
$F = 0.7t$ min.
If t exceeds 16 mm, the shell should be bevelled as shown in the alternative sketch above or consideration should be given to the full penetration joint shown in b).

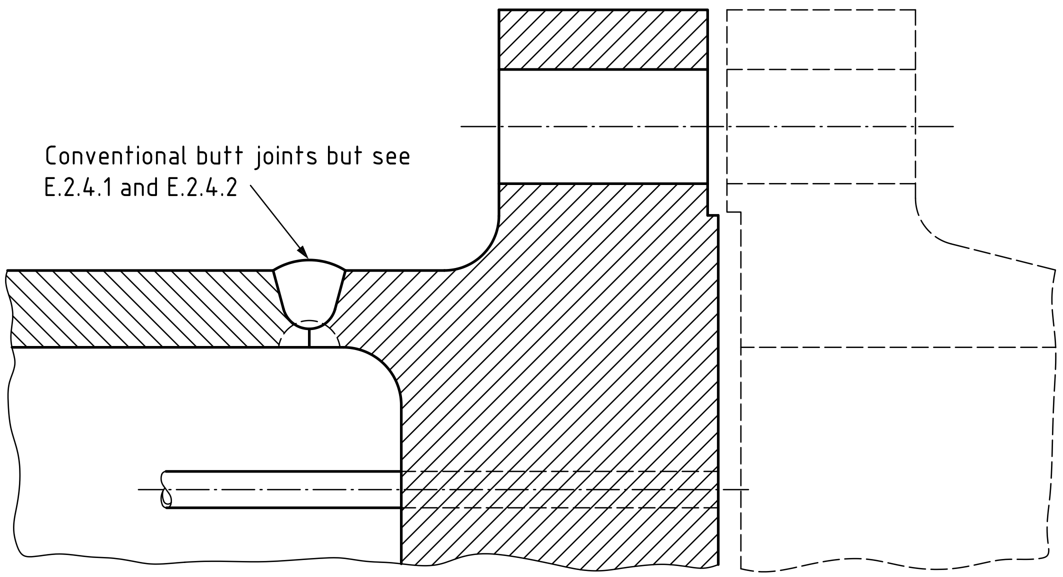


$L = t/3$ but not less than 6 mm.
b)

Figure E.41 **Tubeplate to shell connections: accessible for welding on *both* sides of shell (see E.2.5.8)**



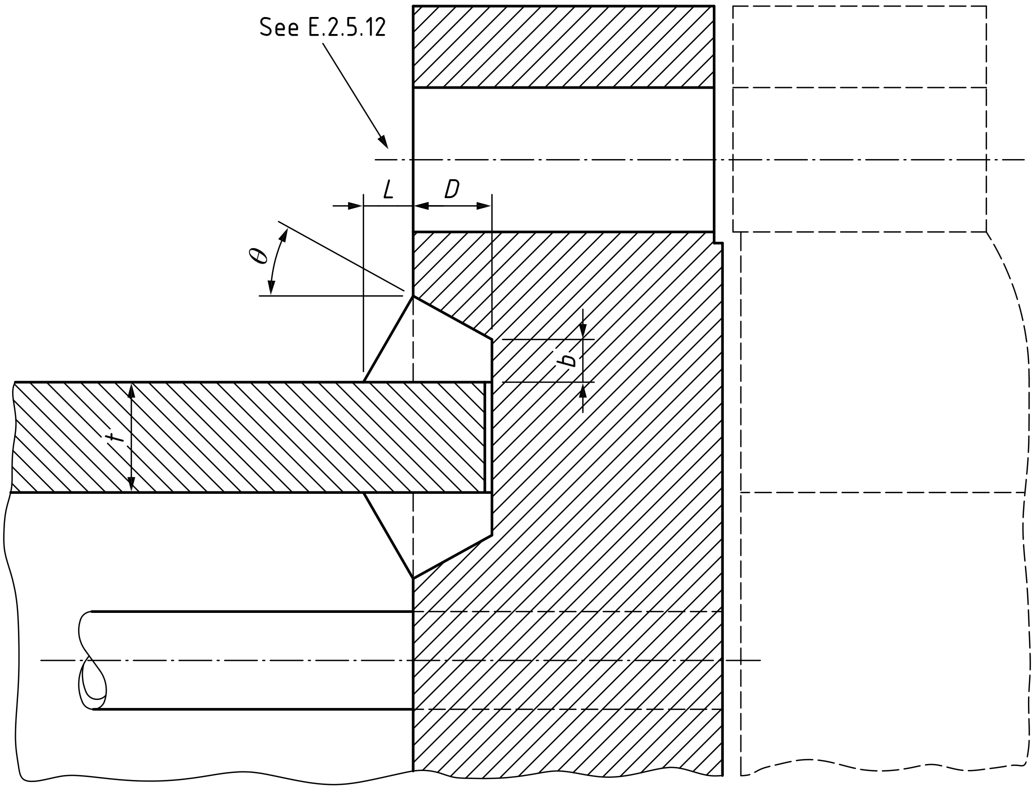
a)



b)

NOTE When using these details, special care should be taken to ensure that the tubeplate is not laminated.

Figure E.42 **Tubeplate to shell connections**



Weld sizes

D = 0.7t min.

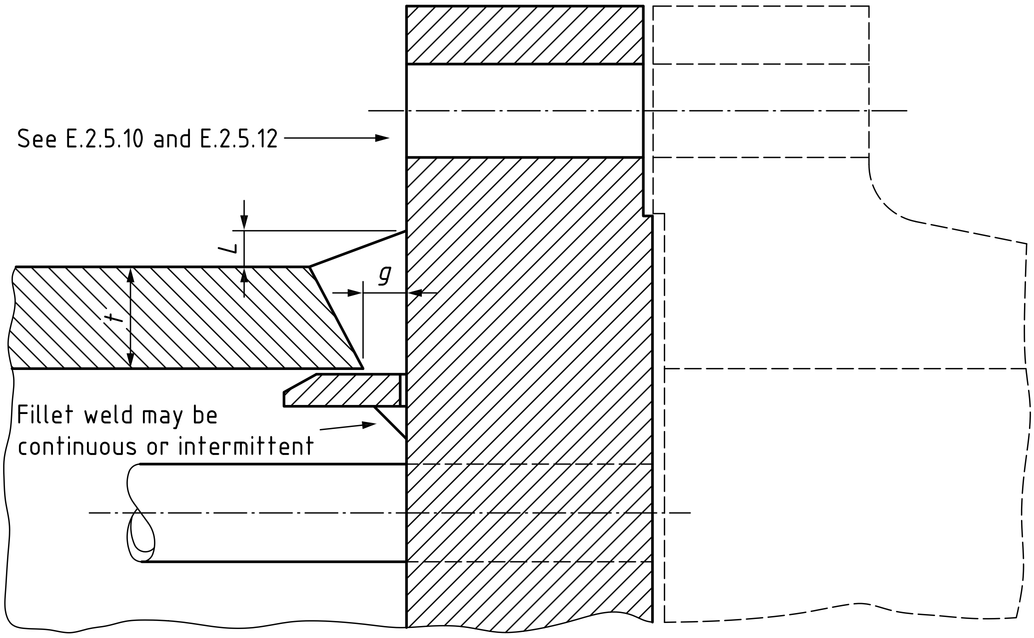
b = 6 mm min.

θ = 30° min.

L = t/3 or 6 mm whichever is larger

a) Accessible for welding on both sides of shell

Figure E.42 **Tubeplate to shell connections *(continued)***



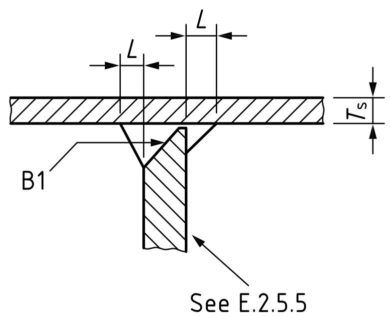
Weld sizes

L = t/3 but not less than 6 mm

g = 5 mm min.

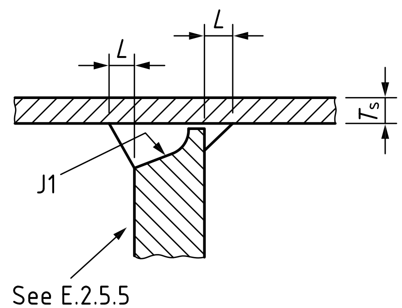
b) Accessible for welding on *outside* of shell only. (This detail is recommended for non-corrosive operating conditions only).

Figure E.43 **Tubeplate to shell connections**



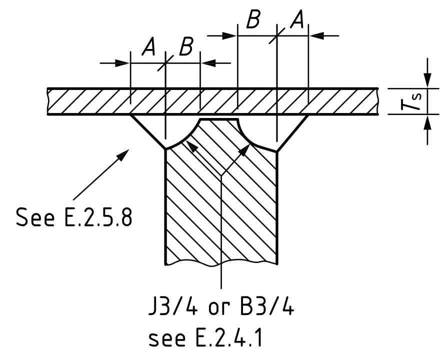
$L = T_s/3$ but not less than 6 mm

1)



$L = T_s/3$ but not less than 6 mm

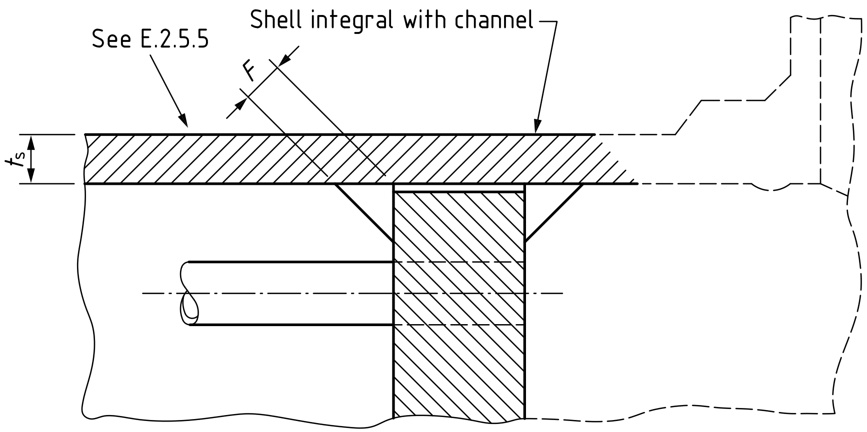
2)



$A + B = T_s$ min.

3)

Alternative joint details for a)



a)

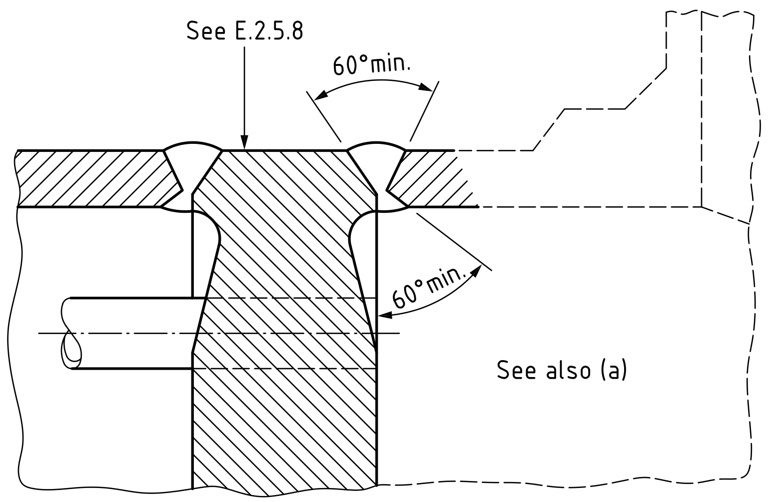
Weld size

$F = 0.7\ t$ min.

If F exceeds 13 mm preference should be given to the alternative joint details above.

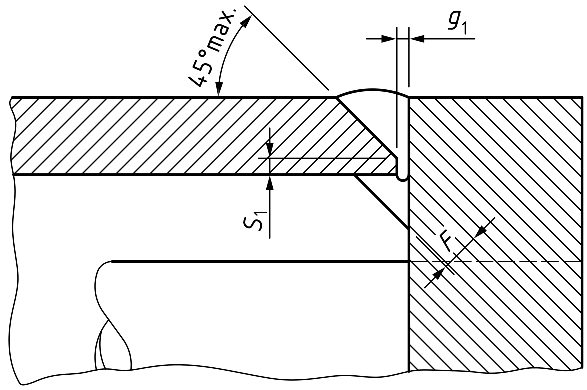
*NOTE The connections shown in the a) alternatives are applicable as flat end connections, but see also **E.2.5.11**. For alternative 3) see also **E.2.5.5**.*

Figure E.43 **Tubeplate to shell connections *(continued)***

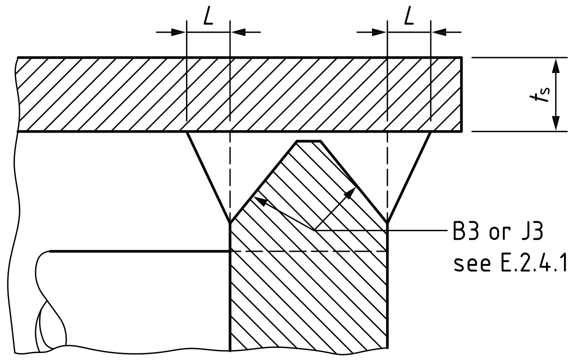


b)

Figure E.44 **Tubeplate to shell connections**



Weld sizes
S_1 as in B2
g_1 as in B2
F = 5 mm min.
a)



$L = T_s/3$ or 6 mm whichever is larger
b)

Annex G Recommendations for the design of local loads, thermal gradients, etc.

G.1 General

This annex, which has been updated in accordance with the recommendations in [22][1], deals with methods of calculating stresses due to local attachments on pressure vessels in some common cases. The determination of stress intensities from calculated stresses and stress limits is covered in Annex A and Annex B.

Although it is impracticable in view of the many variables involved to provide charts for use in the design and analysis of pad reinforced nozzles, references to some work in this field, which has been published in a form consistent with the approach in this specification, have been included [33] to [35].

Although a simplified method for estimating transient thermal stresses at a pressure vessel nozzle is included, it is not considered practicable to provide design charts for more general use in estimating thermal stresses because of the large number of variables involved (see [22]). The designer will therefore have to treat each vessel on an individual basis, and consider the thermal stresses which arise, during both transient and steady state operation, according to the duty that the vessel has to perform. Where a comprehensive stress analysis is not justified, the various components of thermal stress in the most highly stressed regions of the vessel can be considered separately. These are the stresses brought about by the following:

a) the local through thickness temperature gradient;

b) the axisymmetrical component of the mid-wall temperature distribution throughout the structure;

c) the non-symmetric component of the mid-wall temperature distribution; and

d) the variation in through thickness temperature gradient throughout the structure.

The bending and membrane components of the local thermal stress, when added to the stresses at the same position due to local loads and the stresses due to internal pressure, should satisfy the criteria of Annex A. Attention is also drawn to the recommendations given in Annex C to avoid fatigue cracking.

*NOTE 1 A fatigue assessment carried out in accordance with Annex C uses maximum stress ranges not stress intensities (see **C.3.3.1**).*

NOTE 2 If the loaded nozzle area or opening is less than 2.5 $\sqrt{R_e}$ from another stress concentrating feature, stresses as calculated in accordance with Annex G become unreliable and some other method of assessing the total stress, for example finite element stress analysis or proof test, is required.

G.2 Local loads on pressure vessel shells[2]

G.2.1 General

G.2.1.1 Introduction

This clause is concerned with the effect on the shell of a pressure vessel of local forces and moments which may come from supports, equipment supported from the vessel, or from thrusts from pipework connected to branches. Limits on vessel/attachment geometry, without which the methods given may be unreliable, are also stated.

[1] The numbers in square brackets used throughout this annex relate to the bibliographical references given in **G.5**.
[2] An abbreviated procedure has been derived [43].

Stresses due to local loads and moments applied to cylindrical shells through attachments, including nozzles[3)] are dealt with in **G.2.2** and **G.2.3**.

The methods in **G.2.2** cover the determination of stresses at the edge of the loaded areas (**G.2.2.2**), stresses away from the edge of the loaded area **G.2.2.3** and deflections in a cylindrical shell due to the application of radial load (**G.2.2.4**).

Details are given in **G.2.3** of how to treat circumferential moments (**G.2.3.2**) and longitudinal moments (**G.2.3.3**) in order to determine the maximum stresses at the outer edge of the actual loaded area (**G.2.3.4**) and the rotation of the attachment due to the application of these moments (**G.2.3.5**) to a cylindrical shell.

Stresses due to local loads and moments applied to spherical shells through attachments including nozzles[2)] are dealt with in **G.2.4**, **G.2.5** and **G.2.6**.

A method is given in **G.2.4** for calculating stresses and deflections due to radial loads (**G.2.4.3**) and stresses and deflections and slopes due to an external moment (**G.2.4.4**) when applied to a spherical shell. **G.2.5** and **G.2.6** deal with the method of calculating stresses arising at a nozzle/shell junction due to application of pressure, external load and external moment to a spherical shell. The method is based on the analysis given in [25]. Additional information based on [27] is supplied on the method of calculating shakedown conditions (**G.2.6**) and a shell/nozzle junction due to any combination of pressure, external load and external moment.

The application of the data to the treatment of thrusts due to thermal forces in pipework which may be connected to branches is discussed in **G.2.7**; its application to the design of supports is treated in **G.3**.

The data are presented in the form of charts in terms of non-dimensional functions of the variables so that any convenient system of consistent units may be used.

Alternative rules to check local loads on circular nozzles in spherical and cylindrical shells are given in **G.2.8**. These alternative rules are generally based upon those given in BS EN 13445-3:2021. The rules are also applicable to circular attachments that are welded to shells without an opening, e.g. lifting trunnions.

G.2.1.2 Notation

For the purposes of **G.2.1**, **G.2.2**, **G.2.3** and **G.2.4** the following symbols apply. All dimensions are in the corroded conditions unless otherwise indicated (see **3.1.5**).

C	is the half length of side of square loading area (in mm);
C_1	is the half side of equivalent square loading area (in mm) [see Equation (G.2.2-2) or Equation (G.2.2-3)];
C_f	is the correction factor for a cylindrical shell, longitudinal moment calculation;
C_x	is the half length of rectangular loading area in longitudinal direction (in mm);
C_ϕ	is the half length of rectangular loading area in circumferential direction (in mm);
C_z	is the axial length of loading area for an external longitudinal moment (see Figure G.2.3-2) (in mm);

[3)] Subclause **3.5.4** gives a basic design procedure for branches in both cylindrical and spherical vessels under pressure which requires reference to this annex in certain cases (see **3.5.4.3.1**). The procedure specified in **3.5.4** for vessels and cylinders is based on considerations of shakedown under pressure loading as described in PD 6550-2:1989.

C_θ	is the circumferential length of loading area for an external circumferential moment (see Figure G.2.3-1) (in mm);
d	is the distance from centre of applied load to mid-length of vessel (in mm);
E	is the modulus of elasticity (in N/mm^2) from Table 3.6-3;
f_p	is the circumferential or longitudinal pressure stress (see **G.2.3.6.2**);
f_x	is the resultant longitudinal stress (in N/mm^2);
f_ϕ	is the resultant circumferential stress (in N/mm^2);
i	is the rotation of a fitting by an external moment (in radians) [see Equations (G.2.3-1) and (G.2.3-2)];
i_b	is the slope of a branch due to external moment [see Equation (G.2.4-3)];
K_1, K_2	are constants;
L	is the length of cylindrical part of shell (in mm);
L_e	is the equivalent length of shell (in mm);
M	is the external moment applied to branch or fitting (in N·mm);
M_x	is the longitudinal or meridional bending moment per unit circumference (in N·mm/mm);
M_ϕ	is the circumferential bending moment per unit length (in N·mm/mm);
N_x	is the longitudinal membrane force per unit circumference (in N/mm);
N_ϕ	is the circumferential membrane force per unit length (in N/mm);
r	is the mean radius of cylinder or sphere (in mm);
r_o	is the mean radius of branch (in mm);
s	is the position in shell at which force, moment or deflection is required [see Equation (G.2.4-1)];
t	is the wall analysis thickness of shell (in mm) (see **1.6**);
u	defines the area over which the load is distributed [see Equation (G.2.4-2)];
W	is the external load distributed over the loading area (in N);
x	is the longitudinal distance of a point in the vessel wall from the centre of a loading area (in mm);
δ	is the deflection of cylinder at load or at any point of a sphere (in mm);
δ_1	is the deflection of cylinder or sphere at positions detailed in **G.2.3.5** and **G.2.4.4** (in mm);
θ	is the polar co-ordinate of point on a spherical vessel (in radians);
ϕ	is the cylindrical co-ordinate of a point in the vessel wall (in radians);
ϕ_1	is the angle formed by the radius through point A and the radius to the line load [see Figure G.2.2-10 (in radians)];
τ	is the shear stress due to torsion, circumferential shear force or longitudinal shear force (see **G.2.3.6.3**).

G.2.2 Radial loads on cylindrical shells

G.2.2.1 Applicability

The methods in this clause are not considered applicable in cases where the length of the cylinder L is less than its radius r (see [30]). This applies either to an open-ended cylinder or a closed-ended cylinder where the stiffness is appreciably modified from the case considered. For off-centre attachments the distance from the end of the cylinder to the edge of the attachment should be not less than r/2.

In addition the C_ϕ/r ratio should not exceed that given in Figure G.2.2-1, depending on the value of r/t for the vessel (see section **A.3.2** of [30]). This is because in thin shells the longitudinal axis is relatively flexible and free to deform in relation to the transverse axis, causing the latter to carry a disproportionate share of the load. The applicability of the methods to thick shells is also limited in specific cases by the range of r/t values against which data is given.

For values of $C_x/r > 0.25$, the data should be used with caution (see **2.3** of [22]).

These restrictions apply only in relation to the method of analysis in this annex. They are not intended for practical cases where experimental or other evidence may support the validity of the design falling outside these restrictions.

In cases where the applicability of the method given in this clause may be in doubt further data may be found in [30].

The rigidity of the attachment influences the way in which the local loading is transferred to the vessel. It has been found [45] that when the basic attachment thickness is the same as the vessel thickness, optimum vessel stresses occur. Factors to modify the stresses obtained from Annex G are available [45] when other attachment rigidities are used.

It has been found [46] that when local loads are applied to thin walled vessels, non-linear behaviour can occur. For example, if the radial loading is applied in an outward direction, as when lifting a vessel, the vessel displacements and stresses are less than those obtained from a linear analysis treatment, as presented in Annex G. However, when the loading is in the inward direction the displacements and stresses are greater than those predicted by a linear approach. In view of these facts the treatment given in Annex G is restricted to the geometries given in Figure G.2.2-1. Using the procedure given in [46] it is possible to modify the stresses obtained using Annex G, to cover the entire range of geometries up to $(C_\phi/r) = 0.25$ for all values of (r/t) up to 250.

G.2.2.2 Stresses at the edge of the loaded area

G.2.2.2.1 Introduction

The maximum stresses are at the edge of the loaded area. Figure G.2.2-2 shows a cylindrical vessel subjected to a radial load distributed over a central rectangular area $2C_x \times 2C_\phi$.

The cylindrical shell wall of the vessel is assumed to be simply supported at the ends, which means that the radial deflections, the bending moments and the membrane forces in the shell wall are assumed to be zero there. Since the stresses and deflection due to the load are local and die out rapidly away from the loaded area, this is equivalent to assuming that the loaded area is remote from the ends.

NOTE It is not necessary to add the thrust due to the internal pressure acting over the cross-section area of the nozzle bore to the radial load acting on the nozzle. This pressure thrust load is taken into account in the calculation of the stresses due to internal pressure in ***G.2.3.6.2****.*

G.2.2.2.2 Off-centre loading

If the loaded area is a distance d from the centre of the length of a vessel of length L, the deflections, bending moments and membrane forces may be assumed to be equal to those in a vessel of length L_e loaded at its mid-length. L_e is called the equivalent length and can be found from:

$$L_e = L - \frac{4d^2}{L} \tag{G.2.2-1}$$

Figure G.2.2-3 shows a cylindrical shell loaded in this way and Figure G.2.2-4 gives a graph of L_e/L against d/L which can be used to find L_e.

Figure G.2.2-1 **Restriction on vessel/attachment geometry (see G.2.2 and G.2.3)**

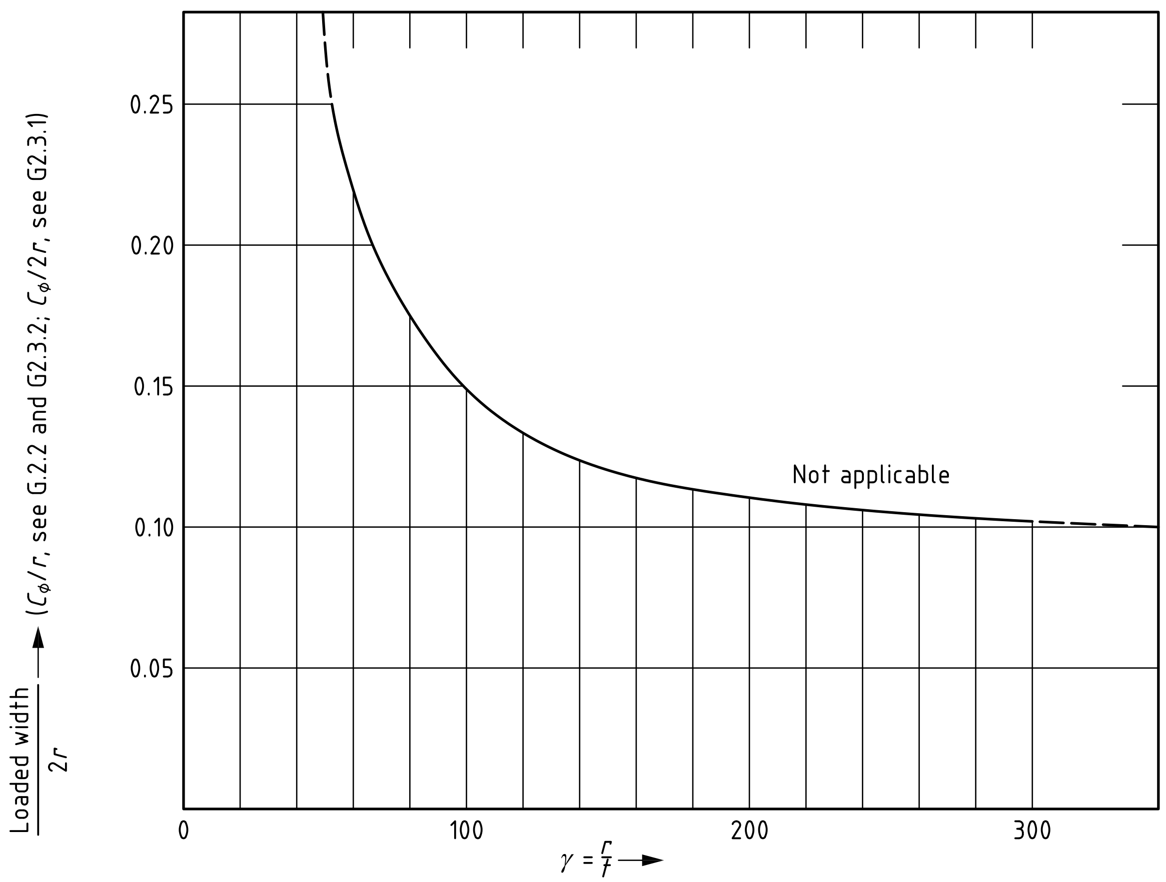


Figure G.2.2-2 **Vessel with central radial load**

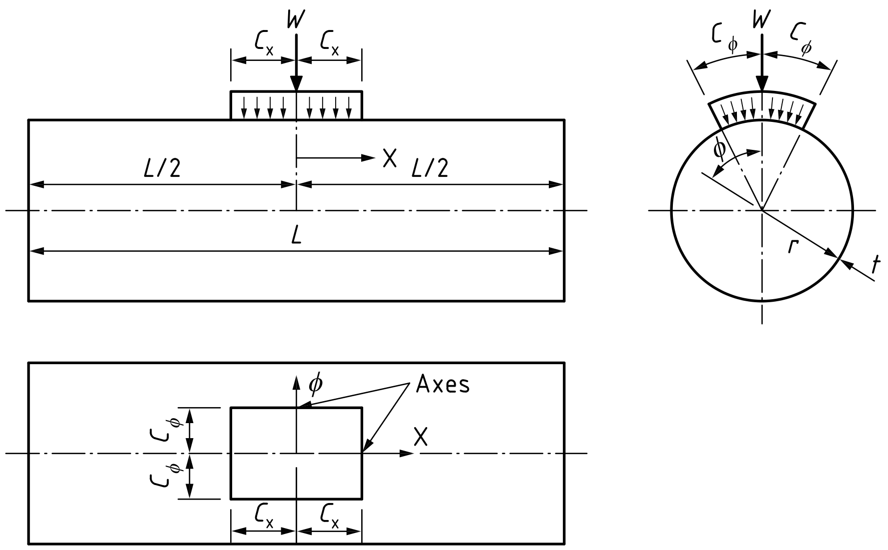


Figure G.2.2-3 **Vessel with radial load out of centre**

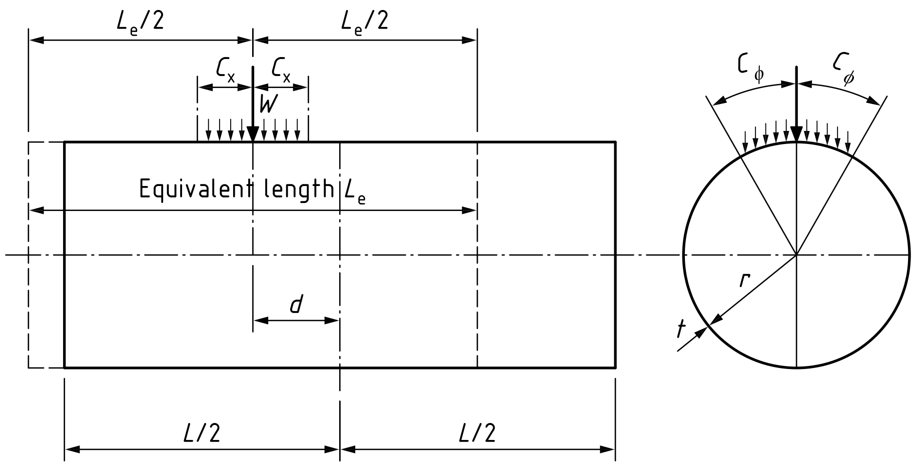
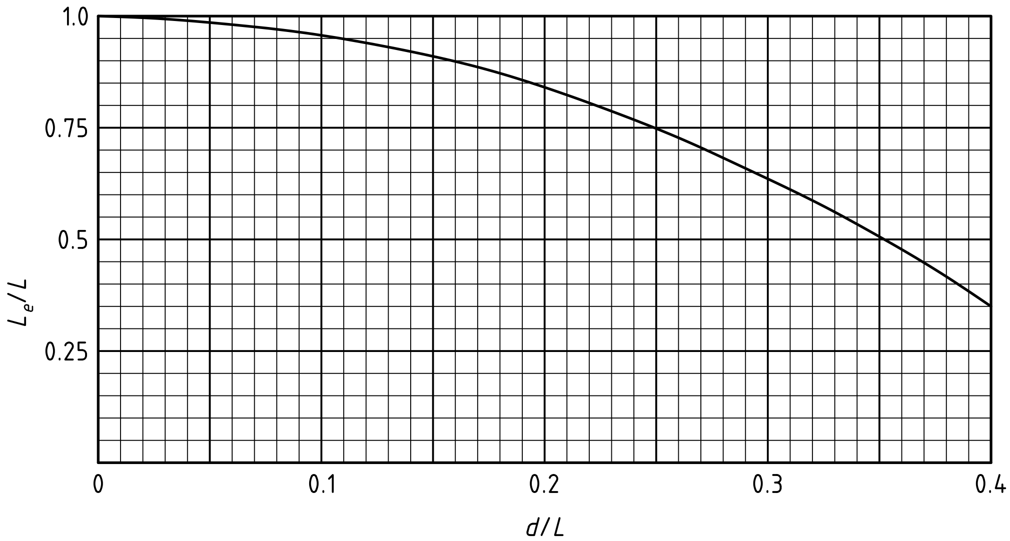


Figure G.2.2-4 **Graph for finding equivalent length L_e**



G.2.2.2.3 Determination of stresses

The resultant longitudinal stress in the shell is given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2}$$

The resultant circumferential stress is given by:

$$f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2}$$

N_x and N_ϕ are positive for tensile membrane stresses.

M_x and M_ϕ are positive when they cause compression at the outer surface of the shell.

These quantities depend on the ratios:

$$\frac{\text{axial length of load}}{\text{actual or equivalent length}} = \frac{2C_x}{L}$$

and

$$\frac{\text{circumferential length of loaded area}}{\text{axial length of loaded area}} = \frac{2C_\phi}{2C_x}$$

For a radial or a circular area of radius r_o, C_ϕ and C_x should be taken as $0.85r_o$.

For an oblique nozzle or elliptical area C_ϕ and C_x should be taken as 0.42 × the major and minor axis of the intersection of the shell or area as appropriate. Non-dimensional functions of each can be expressed in terms of the non-dimensional group:

$$64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$$

The numerical factor 64 is a scale factor without theoretical significance and the value of the expression can be found by calculation or from Figure G.2.2-5 when r, t and C_x are known. The moments and membrane forces are found by interpolation from the graphs of Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8, and Figure G.2.2-9.

Each of the four graphs in each set is for a given value of the ratio $2C_x/L$ and has curves for four values of the ratio C_ϕ/C_x.

The circumferential moment M_ϕ is found from Figure G.2.2-6. The longitudinal moment M_x is found from Figure G.2.2-7. The circumferential membrane force N_ϕ is found from Figure G.2.2-8. The longitudinal membrane force N_x is found from Figure G.2.2-9.

A moment is considered as positive if it causes compression at the outside of the vessel.

A membrane force is considered as positive if it causes tension in the vessel wall.

G.2.2.2.4 Effect of internal and external pressure

A conservative result is obtained for total stresses if the stresses due to the pressure are simply added to those due to local radial loads calculated in this clause.

This method cannot be used for vessels under external pressure because the deflection due to the radial load always increases the out-of-roundness of the shell. For the same reason it should not be applied to a cylindrical shell subject to an axial compressive load as well as a radial load. In these cases the deflection due to the radial load should be found as in **G.2.2.4** and the effect there of assessed in relation to shape requirements specified in **3.6** for such vessels. Annex M is intended for use with deflections due to shape imperfections and may not always be conservative with estimated deflections due to local loads.

Figure G.2.2-5 **Chart for finding** $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$

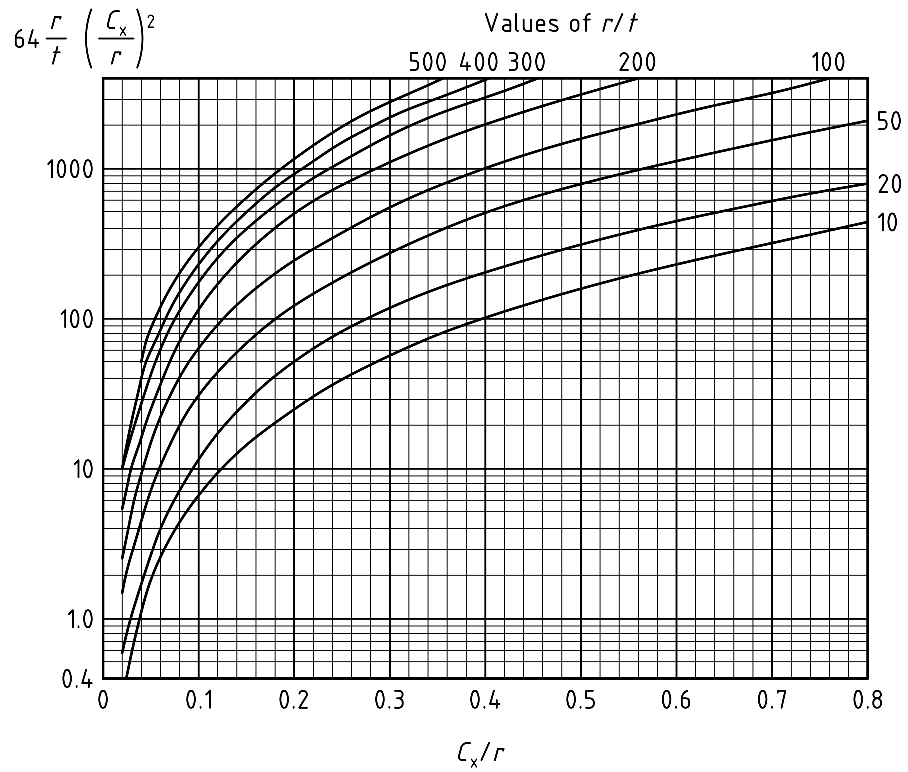
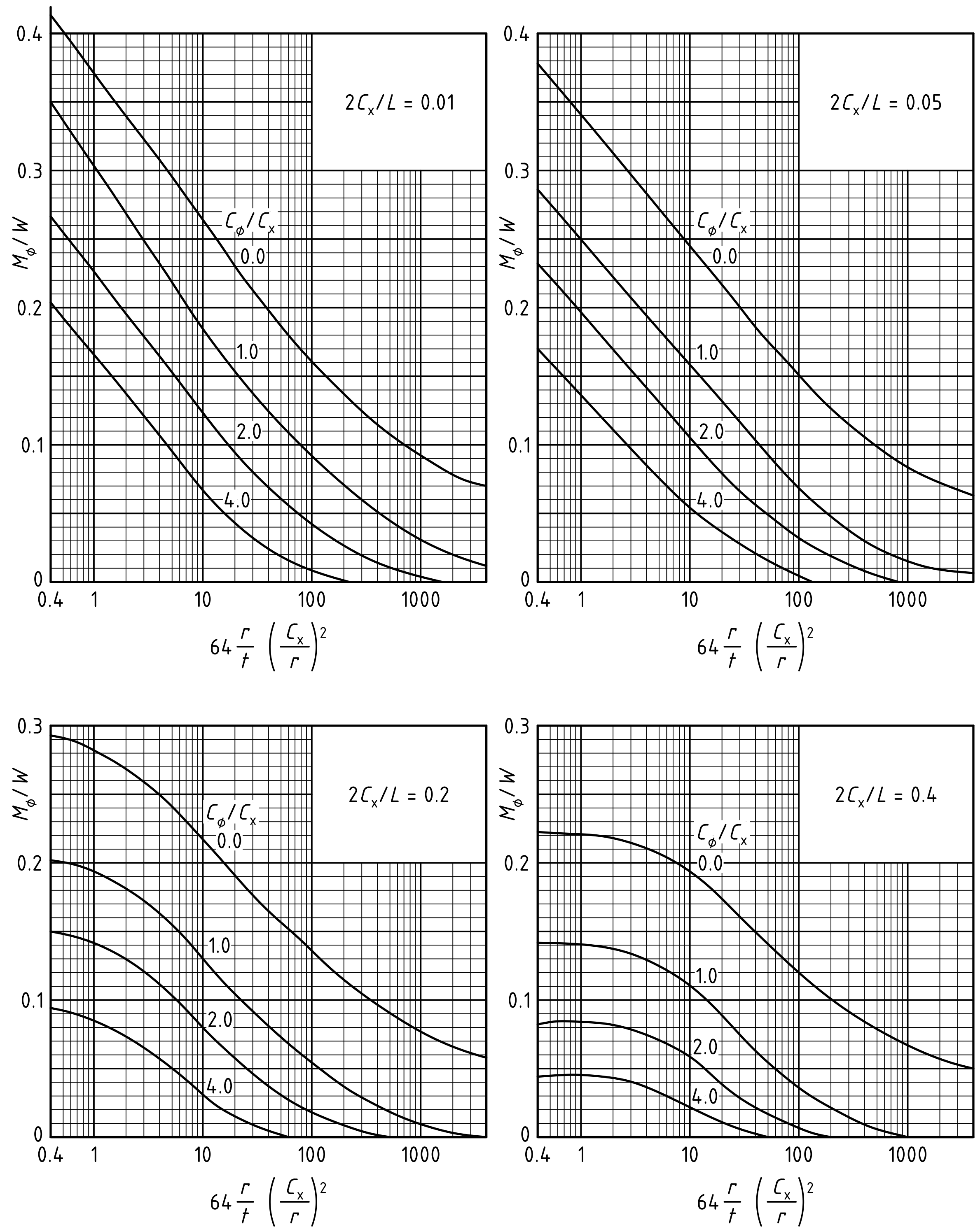
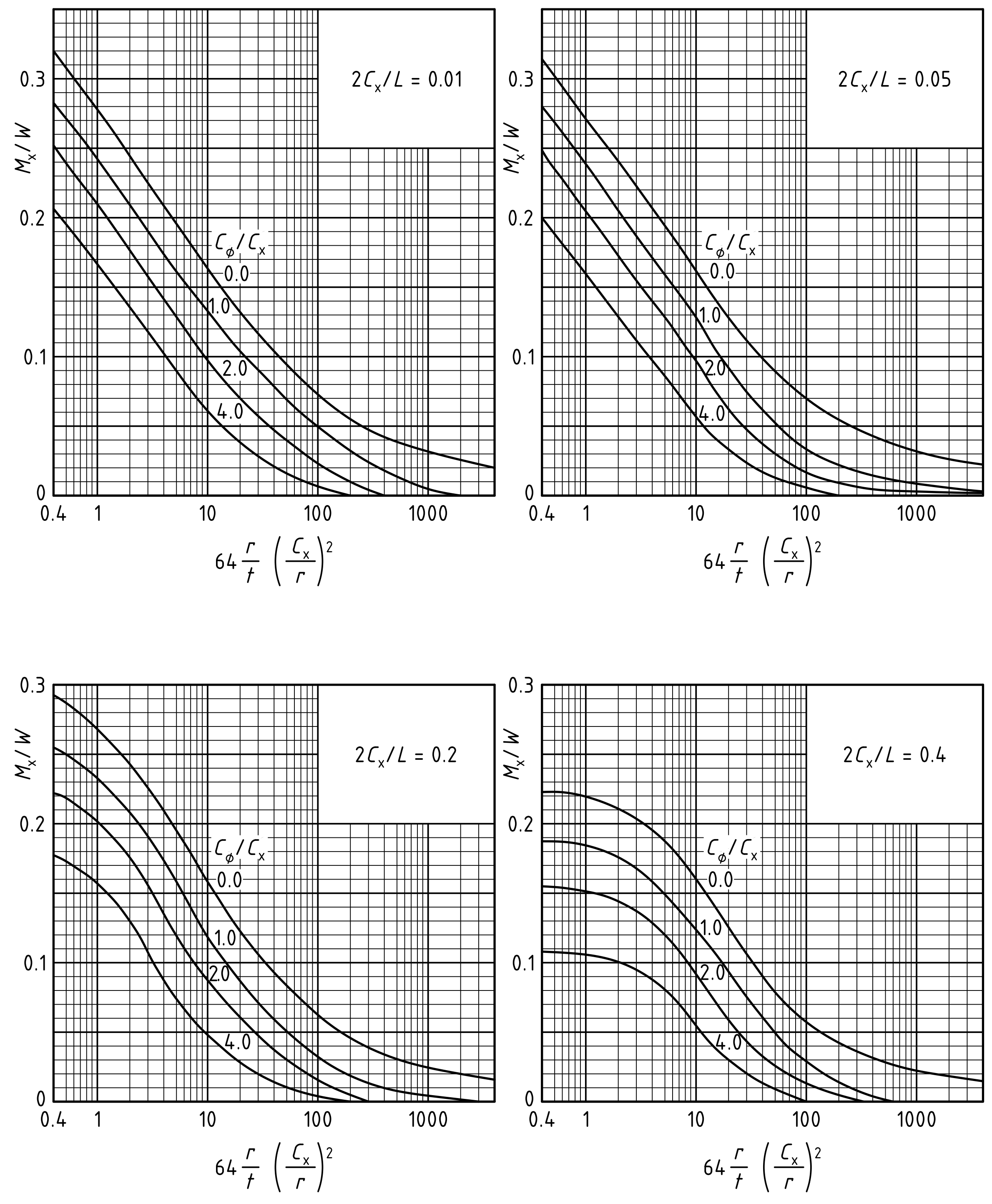


Figure G.2.2-6 **Cylindrical shells with radial load: circumferential moment per millimetre width (see G.2.2)**



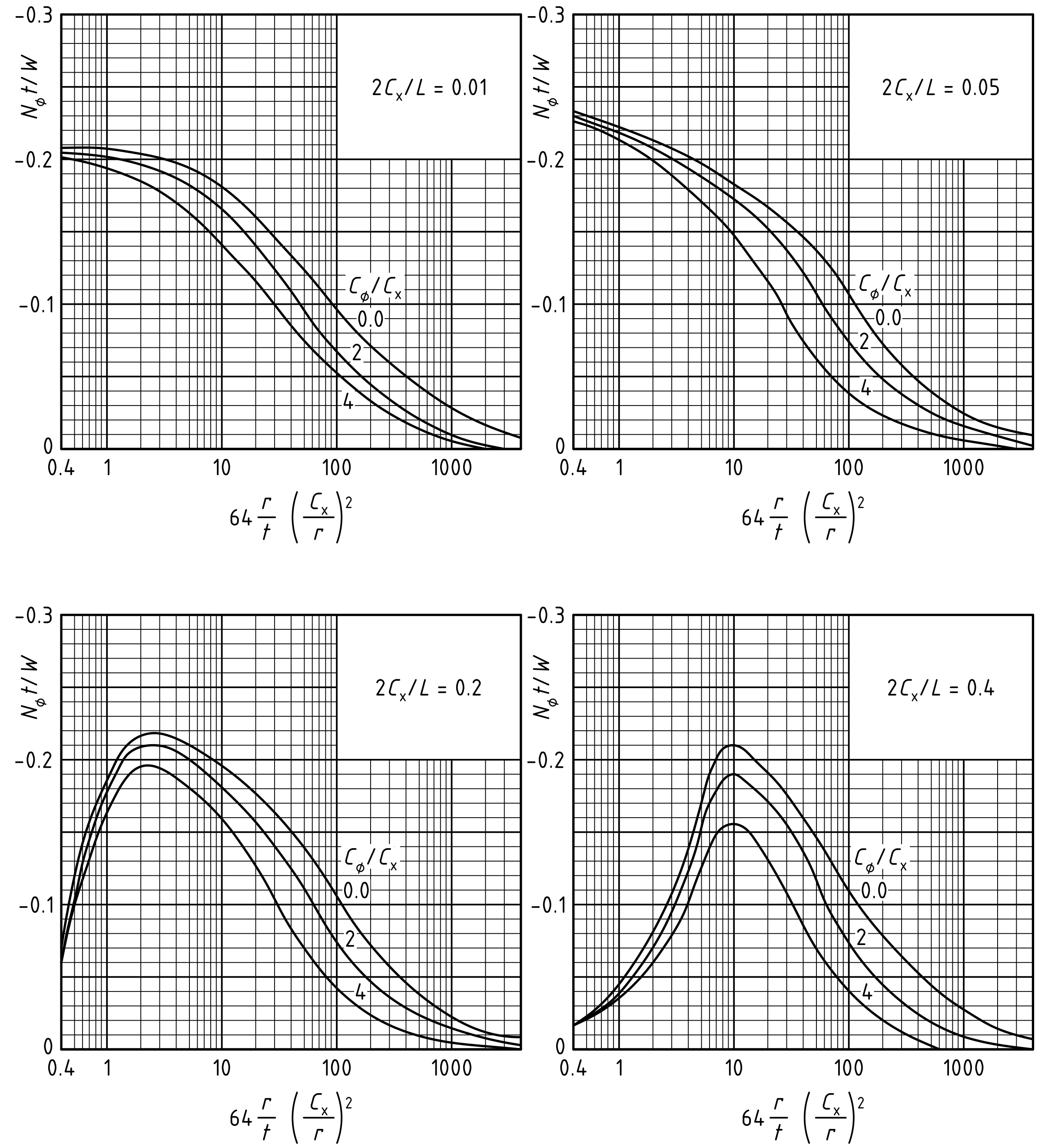
NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure G.2.2-5.

Figure G.2.2-7 **Cylindrical shells with radial load: longitudinal moment per millimetre width (see G.2.2)**



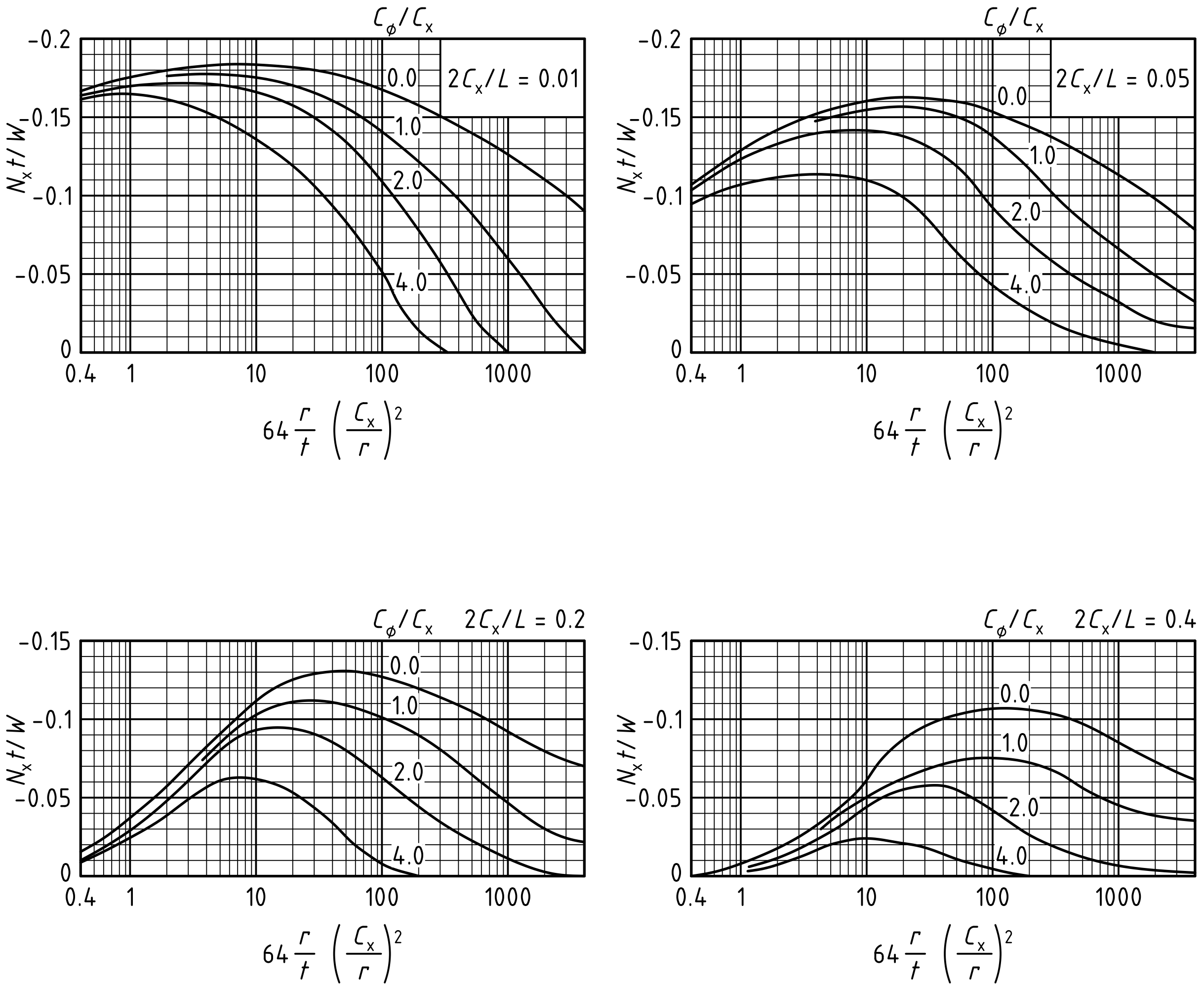
NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure G.2.2-5.

Figure G.2.2-8 **Cylindrical shells with radial load: circumferential membrane force per millimetre width (see G.2.2)**



NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure G.2.2-5.

Figure G.2.2-9 **Cylindrical shells with radial load: longitudinal membrane force per millimetre width (see G.2.2)**



NOTE $64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ is found from Figure G.2.2-5.

G.2.2.3 Stresses away from the edge of the loaded area

G.2.2.3.1 Introduction

Although the maximum stresses occur at the edge of the load, it is necessary to find those at other positions when the effect of one load at the position of another is required.

This happens:

a) when longitudinal or circumferential moments are resolved as in **G.2.3**;

b) when loads are applied close together, e.g. if a bracket is fixed close to a branch.

In general the effect of one load at the position of another can be disregarded when the distance between the centres of the loaded areas is greater than K_1C_ϕ for loads separated circumferentially or K_2C_x for loads separated axially, where K_1 and K_2 are found from Table G.2.2-1 and C_ϕ and C_x are for the greater load.

Table G.2.2-1 **Values of K_1 and K_2**

$64\,\frac{r}{t}\left(\frac{C_x}{r}\right)^2$	$\frac{2C_x}{L}$	K_1	K_2
0.4	0.01	8	8
	0.05	6	8
	0.2	3	4
	0.4	1.5	2
10	0.01	3	8
	0.05	2.5	8
	0.2	1.5	3
	0.4	1.5	2
200	0.01	Negligible	5
	0.05		4
	0.2		2.5
	0.4		1.75
3 200	All values	Negligible	2.5

NOTE The value of the non-dimensional factor $64\,\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ can be found from Figure G.2.2-5.

G.2.2.3.2 Variation of stress round the circumference

No exact analytical treatment of the variation of stress round the circumference away from the edge of the loaded area is available. The following treatment is an approximation sufficiently accurate for practical purposes. For an experimental verification of it see [17].

Consider a radial line load of length $2C_x$, applied at the mid-length of a thin cylinder as shown in Figure G.2.2-10a). The maximum stresses due to this load at points away from it are on the circumference passing through its mid-length as A in the figure. The radius through A makes an angle ϕ_1 with the line of the load.

The moments and membrane forces at A, M_ϕ, M_x, N_ϕ, N_x, can be found from the graphs of Figure G.2.2-10, Figure G.2.2-11, Figure G.2.2-12 and Figure G.2.2-13 in which the functions M_ϕ/W, M_x/W, $N_\phi t/W$ and $N_x t/W$ are plotted against the non-dimensional group $\phi_1 r/C_x$.

The diagram showing the load and its geometry, as Figure G.2.2-10a), is repeated on each chart for convenience.

Line loads are, of course, unusual in practice, and loads distributed over an area having an appreciable circumferential width $2C_\phi$ are treated as follows.

a) Find the value of the function M_ϕ/W, M_x/W, $N_\phi t/W$ or $N_x t/W$ at the edge of the load for the known values of C_ϕ/C_x and $2C_x/L$ from the graphs in Figure G.2.2-6,Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9.

b) Enter the corresponding graph in Figure G.2.2-10, Figure G.2.2-11, Figure G.2.2-12 and Figure G.2.2-13 at this value.

   The intercept on the curve for $2C_x/L$ gives a value of $\phi_1 r/C_x = Z$, e.g. if $64(r/t)(C_x/r)^2 = 10$, $2C_x/L = 0.01$ and $C_\phi/C_x = 1$. Figure G.2.2-6 gives $M_\phi/W = 0.185$. Entering Figure G.2.2-10 at $M_\phi/W = 0.185$ gives $Z = 0.55$ for $2C_x/L = 0.01$ as indicated by the dotted lines in the left-hand graph of Figure G.2.2-10.

c) The value of M_ϕ/W at A is then found by substituting ($\phi_1 r/C_x + Z — C_\phi/C_x$) for the actual value of $\phi_1 r/C_x$ in the same graph.

The other quantities M_x/W, $N_\phi t/W$, $N_x t/W$ can be found in the same way. This method is used in order to avoid the use of a separate set of four charts for each value of C_ϕ/C_x considered.

Diagrams for circumferential bending moments and forces are printed up the page to distinguish them from those for longitudinal moments and forces which are printed across the page.

When the centre of the load is away from the mid-length of the cylinder, the equivalent length L_e, found as in **G.2.2.2**, should be substituted for L in all cases.

For variation of stress along the cylinder due to radial loading see **G.2.2.3.3**.

G.2.2.3.3 Variation of stress along the cylinder

Consider a radial line load, W, distributed over a length $2C_x$ as shown in Figure G.2.2-14a).

Values of M_ϕ, M_x, N_ϕ, and N_x at A can be found from the graphs of Figure G.2.2-14, Figure G.2.2-15, Figure G.2.2-16 and Figure G.2.2-17 in which the functions M_ϕ/W, M_x/W, $N_\phi t/W$ and $N_x t/W$ are plotted against x/C_x for given values of $64\,(r/t)(C_x/r)^2$ and $2C_x/L$.

The resultant stresses in the shell at A are given by:

circumferential stress, $f_\phi = \dfrac{N_\phi}{t} \pm \dfrac{6M_\phi}{t^2}$

longitudinal stress, $f_x = \dfrac{N_x}{t} \pm \dfrac{6M_x}{t^2}$

The values for x/C_x less than 1.0, for which no curves are plotted, fall within the loaded lengths, and the curves should not be extended into this region. The values for $x/C_x = 1$ correspond to the maximum stresses found from Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9 for $C_\phi/C_x = 0$.

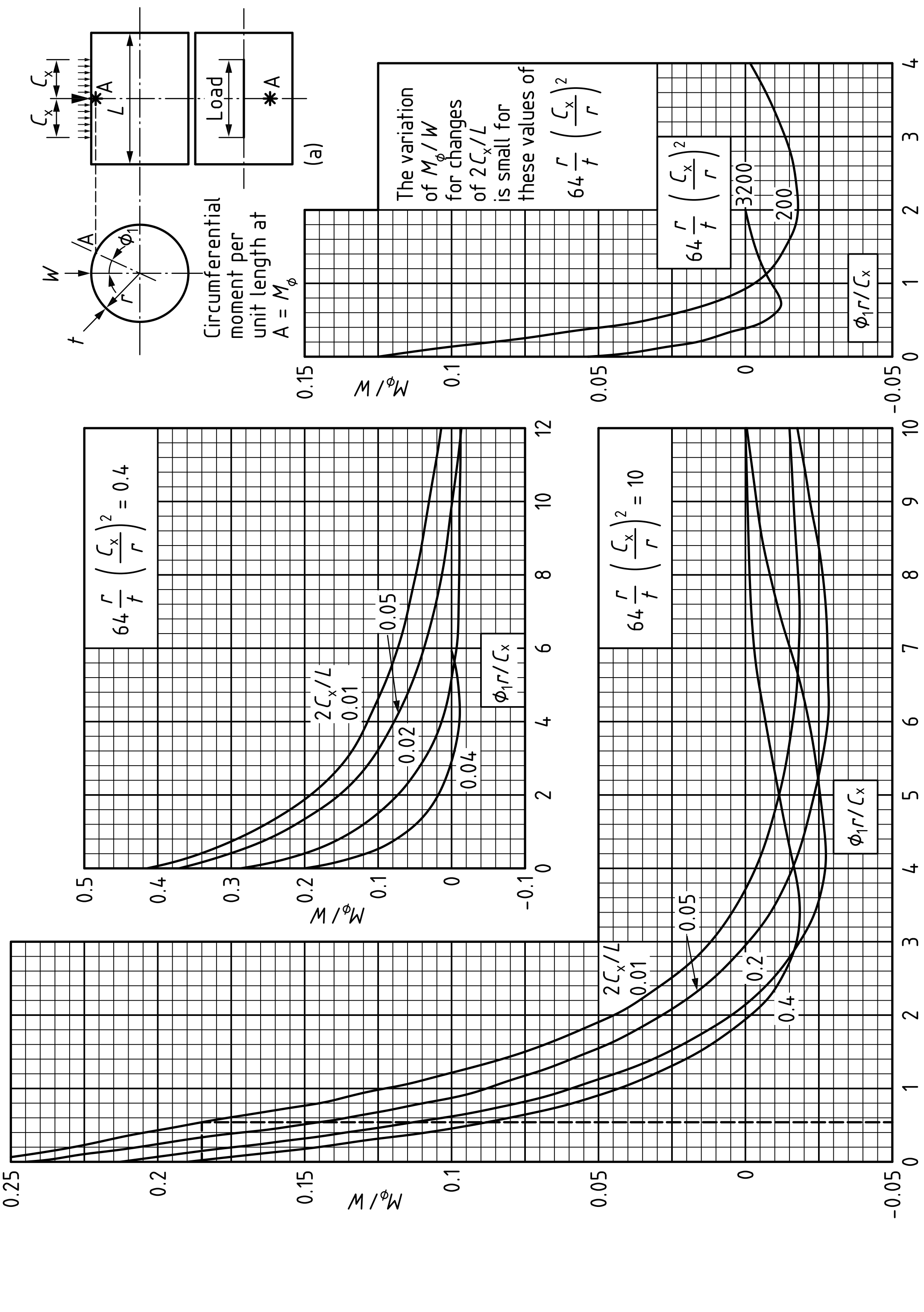
The diagram showing the load and its geometry as Figure G.2.2-14a) has been repeated on each chart for convenience.

Diagrams for circumferential bending moments and forces are printed up the page to distinguish them from those for longitudinal moments and forces which are printed across the page.

For a load distributed over an area $2C_x \times 2C_\phi$, the moments and membrane forces at any value of x/C_x are reduced in the same ratio as the corresponding values at the edge of the load found from Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9, i.e. in the ratio:

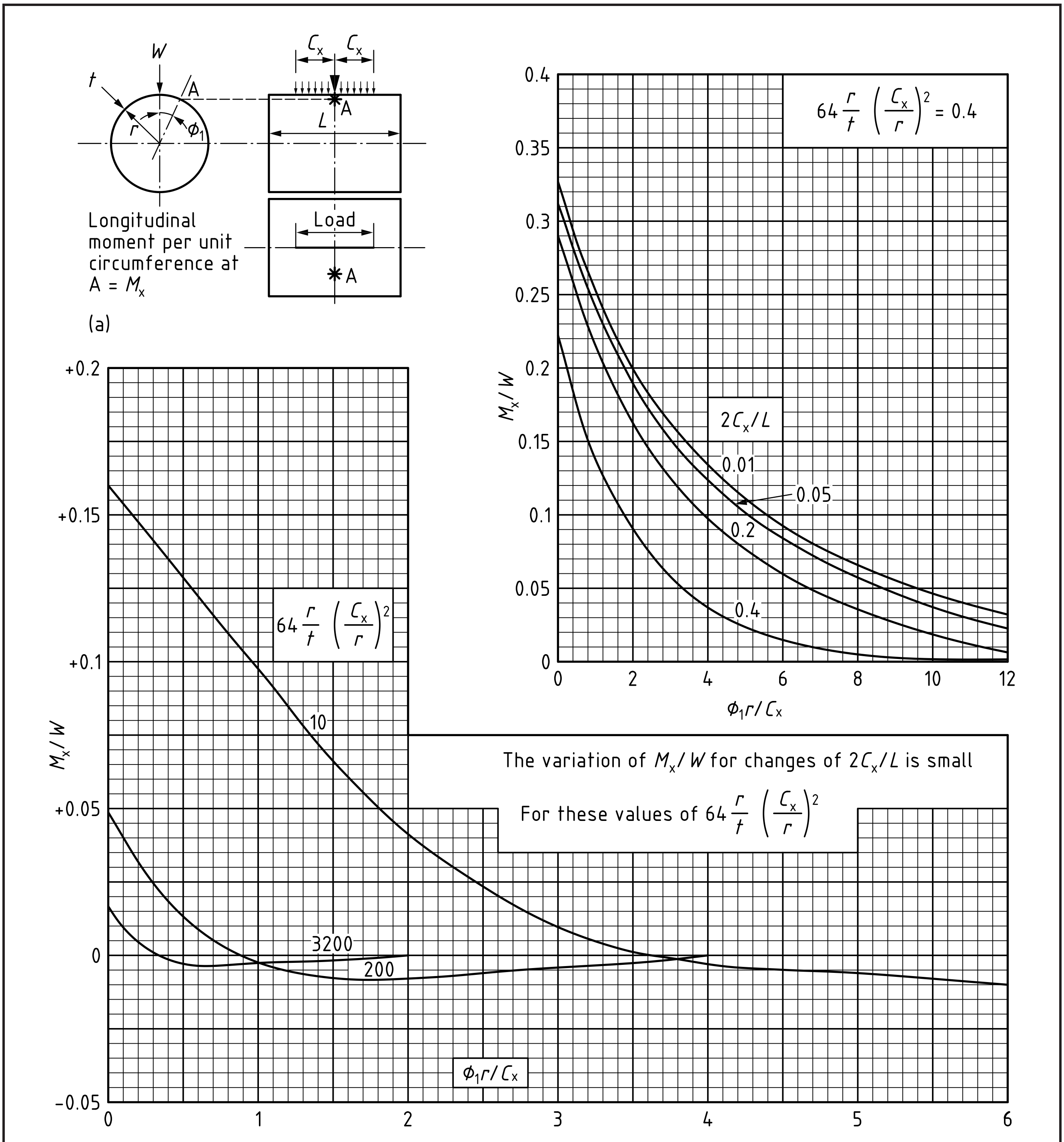
$$\frac{\text{value for actual } C_\phi/C_x}{\text{value for } C_\phi/C_x = 0}$$

Figure G.2.2-10 **Circumferential bending moment due to a radial line load variation round circumference (see G.2.2.3.2)**



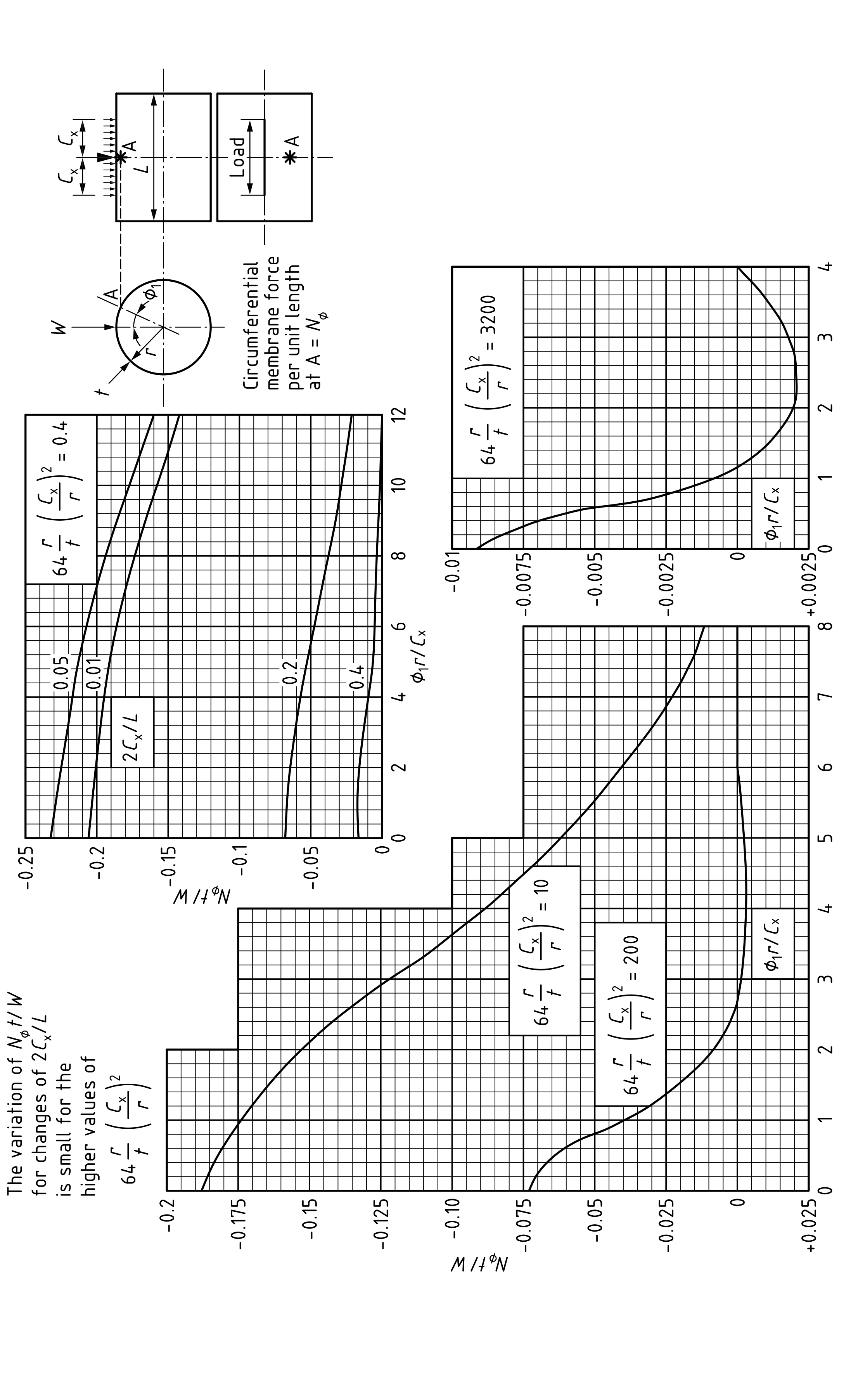
NOTE These charts apply in any consistent system of units.

Figure G.2.2-11 **Longitudinal moment from radial line load variation round circumference (see G.2.2.3.2)**



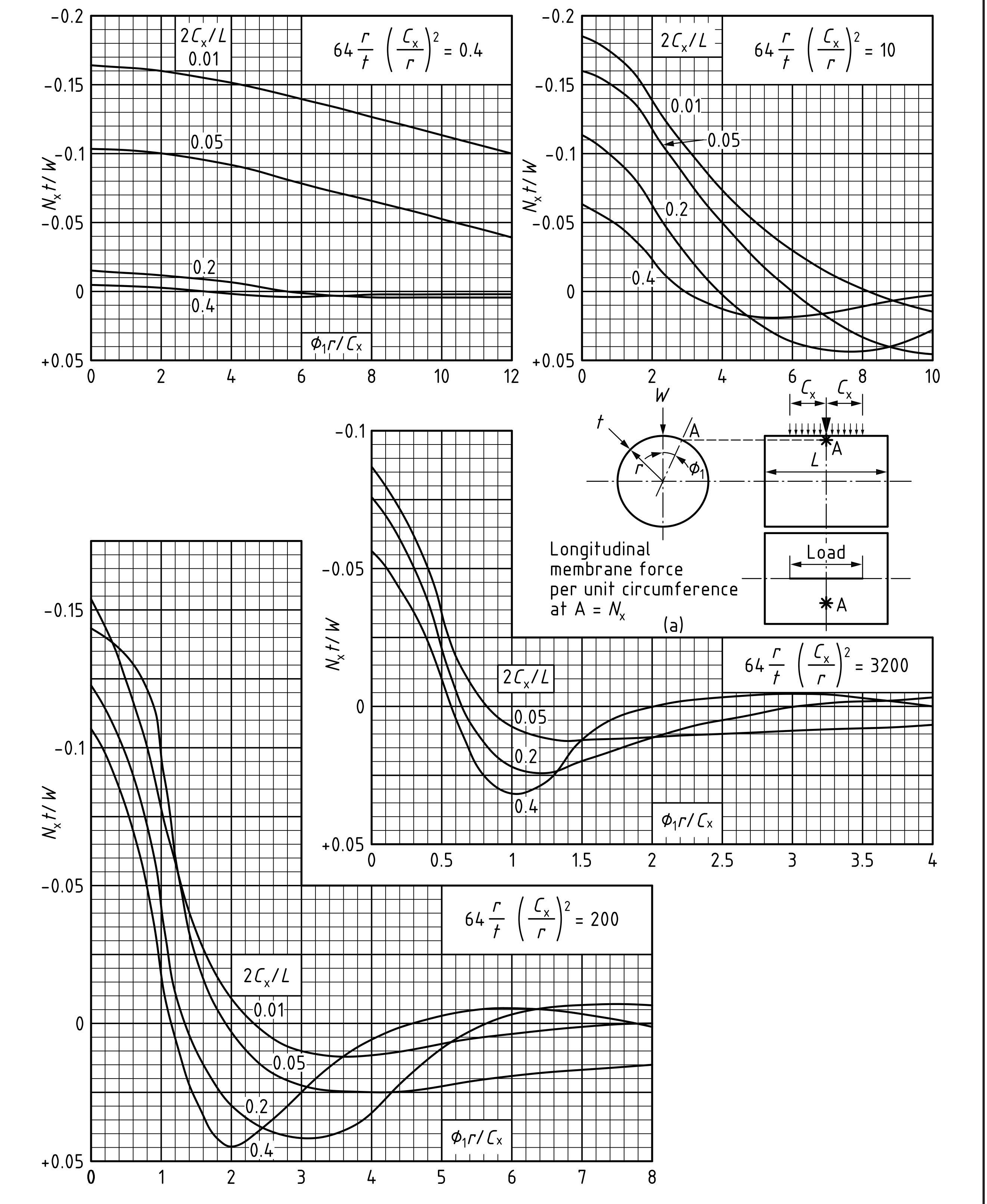
NOTE These charts apply in any consistent system of units.

Figure G.2.2-12 **Circumferential membrane force from radial line load variation round circumference (see G.2.2.3.2)**



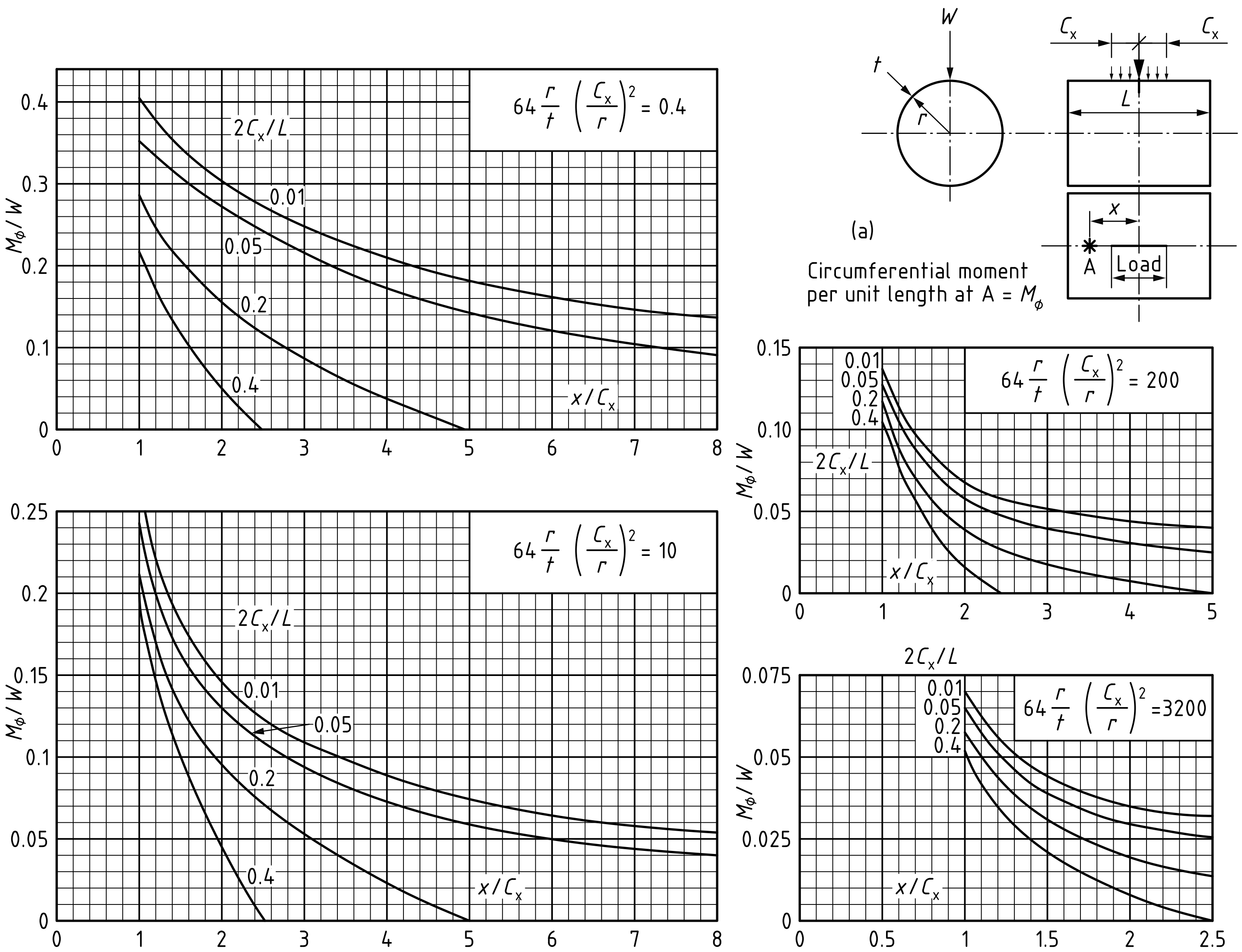
NOTE These charts apply in any consistent system of units.

Figure G.2.2-13 **Longitudinal membrane force from radial line load variation round circumference (see G.2.2.3.2)**



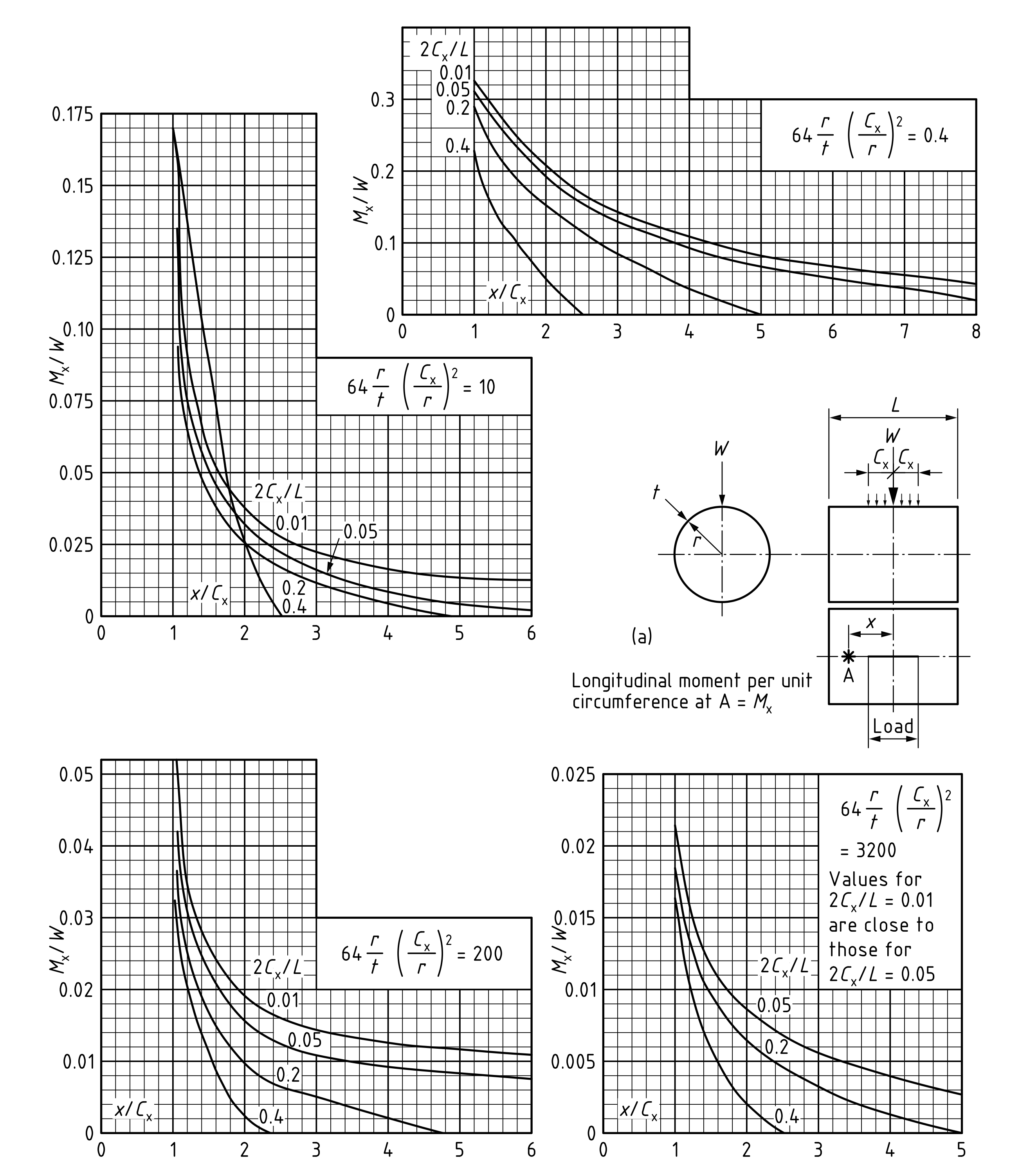
NOTE These charts apply in any consistent system of units.

Figure G.2.2-14 **Circumferential bending moment due to a radial line load variation along cylinder (see G.2.2.3.3)**



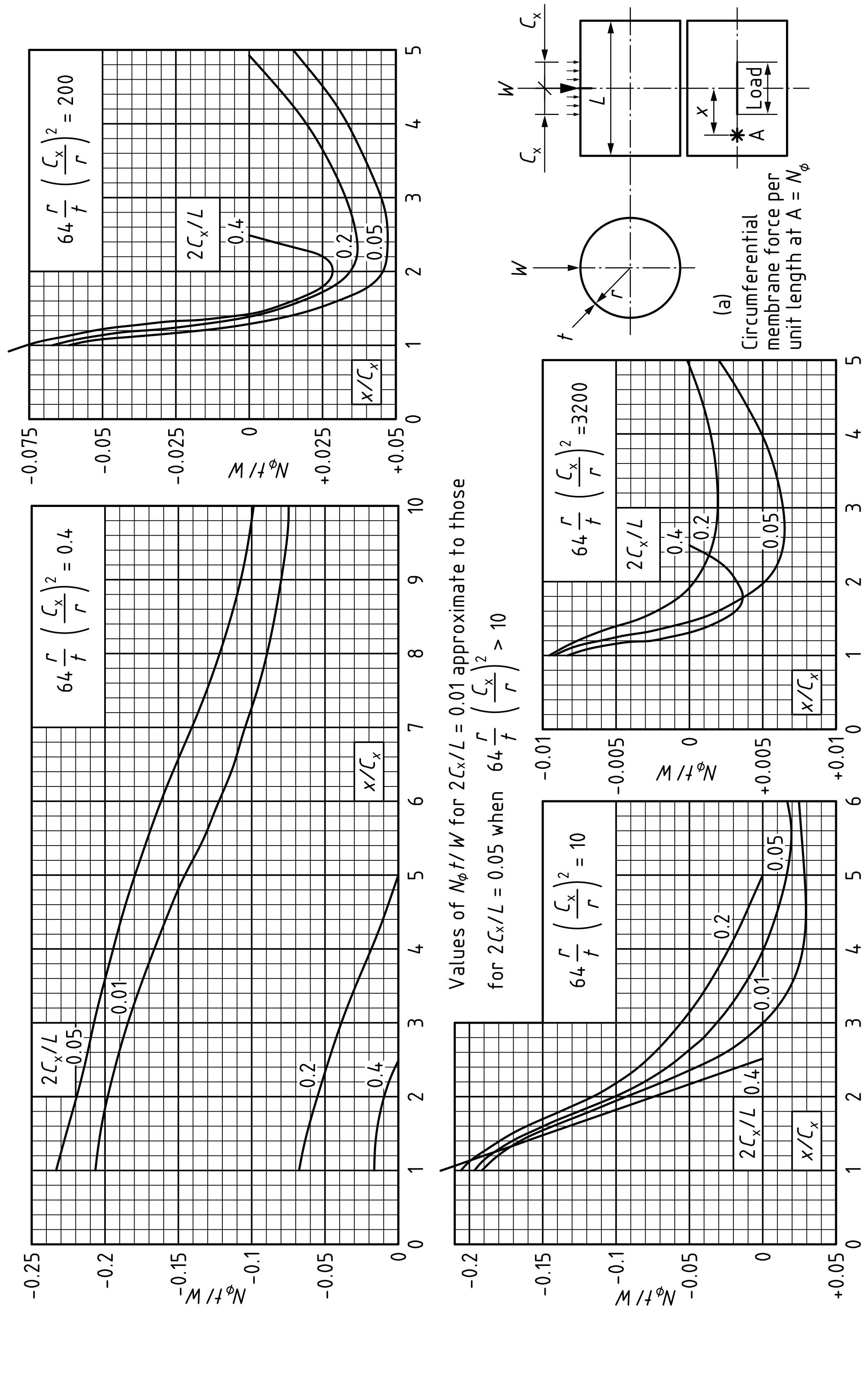
NOTE These charts apply in any consistent system of units.

Figure G.2.2-15 **Longitudinal moment due to a radial line load variation along cylinder (see G.2.2.3.3)**



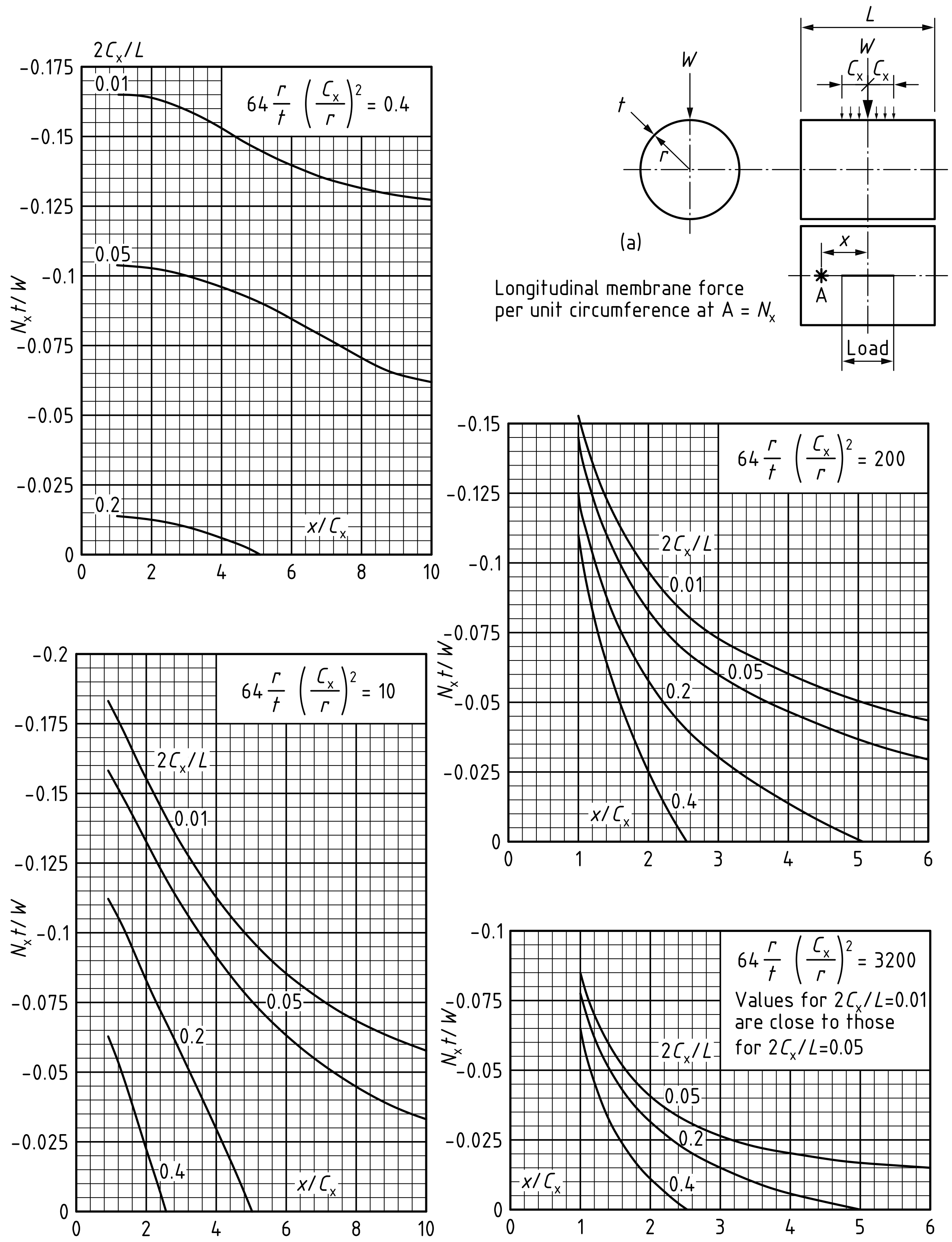
NOTE These charts apply in any consistent system of units.

Figure G.2.2-16 **Circumferential membrane force due to a radial line load variation along cylinder (see G.2.2.3.3)**



NOTE These charts apply in any consistent system of units.

Figure G.2.2-17 **Longitudinal membrane force due to a radial line load variation along cylinder (see G.2.2.3.3)**



NOTE These charts apply in any consistent system of units.

G.2.2.4 Deflections of cylindrical shells due to radial loads

The deflections of a cylindrical shell due to local load are required for:

a) finding the movement of a vessel shell due to the thrust of a pipe connected to it;

b) finding the rotation of a branch due to a moment applied by a pipe connected to it. (See **G.2.3**.)

The deflection of the shell due to radial load is a function of the non-dimensional parameters *r*/*t*, $\delta Er/W$ and *L*/*r* which is given by the full lines in the charts as follows:

Figure G.2.2-18a) for values of *r*/*t* between 15 and 40;

Figure G.2.2-18b) for values of *r*/*t* between 40 and 100;

Figure G.2.2-19 for values of *r*/*t* greater than 100.

In the case of a cylindrical shell, the deflections calculated are those at the centre of the attachment. The method does not calculate deflections at any other position.

For a central load, *L* is the actual length of the vessel.

For a load out of centre, *L* is the equivalent length L_e found as in **G.2.2.2**.

For a point load, the figures are used to determine the value of $\delta Er/W$ by entering the appropriate value of *L*/*r* in the top right hand corner extensions to the figures and then moving across the graph to the left until the $C/r = 0$ ordinate, designated "point load", is reached; thereafter follow the sloping full lines to meet the vertical line for the appropriate value of *r*/*t*, then move horizontally to read the value of $\delta Er/W$.

For a load distributed over a square of side 2*C*, the value of $\delta Er/W$ is given by a line joining the intersections of the *L*/*r* and *C*/*r* lines in the top right-hand and bottom left-hand extensions of each diagram as shown by the dotted line and example on Figure G.2.2-19.

The deflection due to a load distributed over a circular area of radius r_o is approximately the same as that for a square of side $1.7r_o$.

The deflection due to a load distributed over a rectangular area $2C_x \times 2C_\phi$ is approximately the same as that for an equivalent square of side $2C_1$ where C_1 is obtained as follows:

$$C_1 = \sqrt{C_\phi C_x} \text{ when } C_x > C_\phi \qquad \text{(G.2.2-2)}$$

$$C_1 = (C_\phi)^{0.93} \times (C_x)^{0.07} \text{ when } C_\phi > C_x \qquad \text{(G.2.2-3)}$$

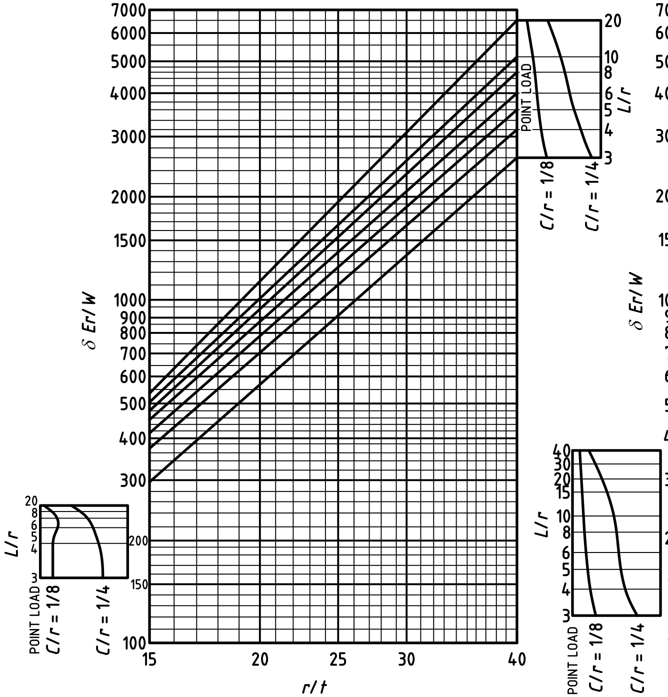
(or from Figure G.2.2-20)

Equation (G.2.2-2) applies to a rectangular area in which the long axis is parallel to the axis of the cylinder.

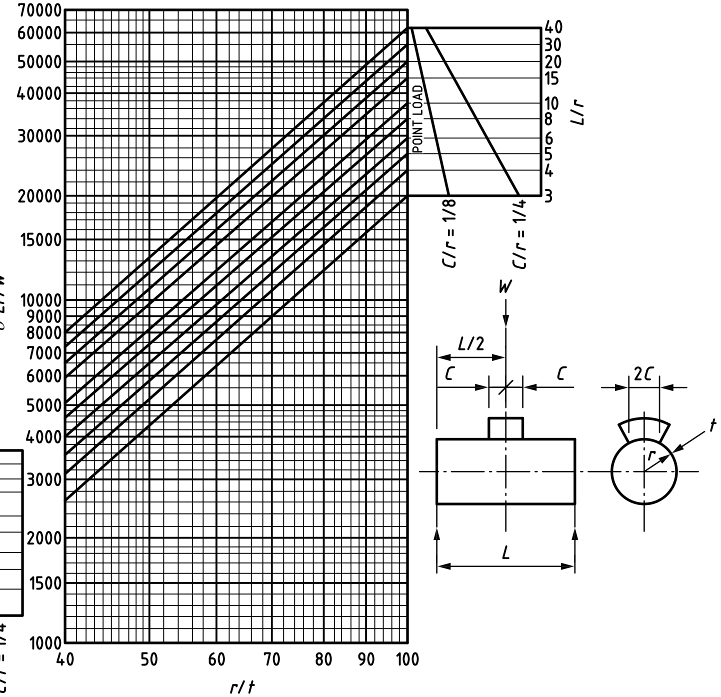
Equation (G.2.2-3) applies to a rectangular area in which the long axis is circumferential.

NOTE Enquiry Case 5500/137 gives additional guidance on the calculation of deflections and rotations for nozzles in cylindrical shells.

Figure G.2.2-18 **Maximum radial deflection of a cylindrical shell subjected to a radial load *W* for *r/t* between 15 and 100**



a) For $\frac{r}{t}$ less than 40



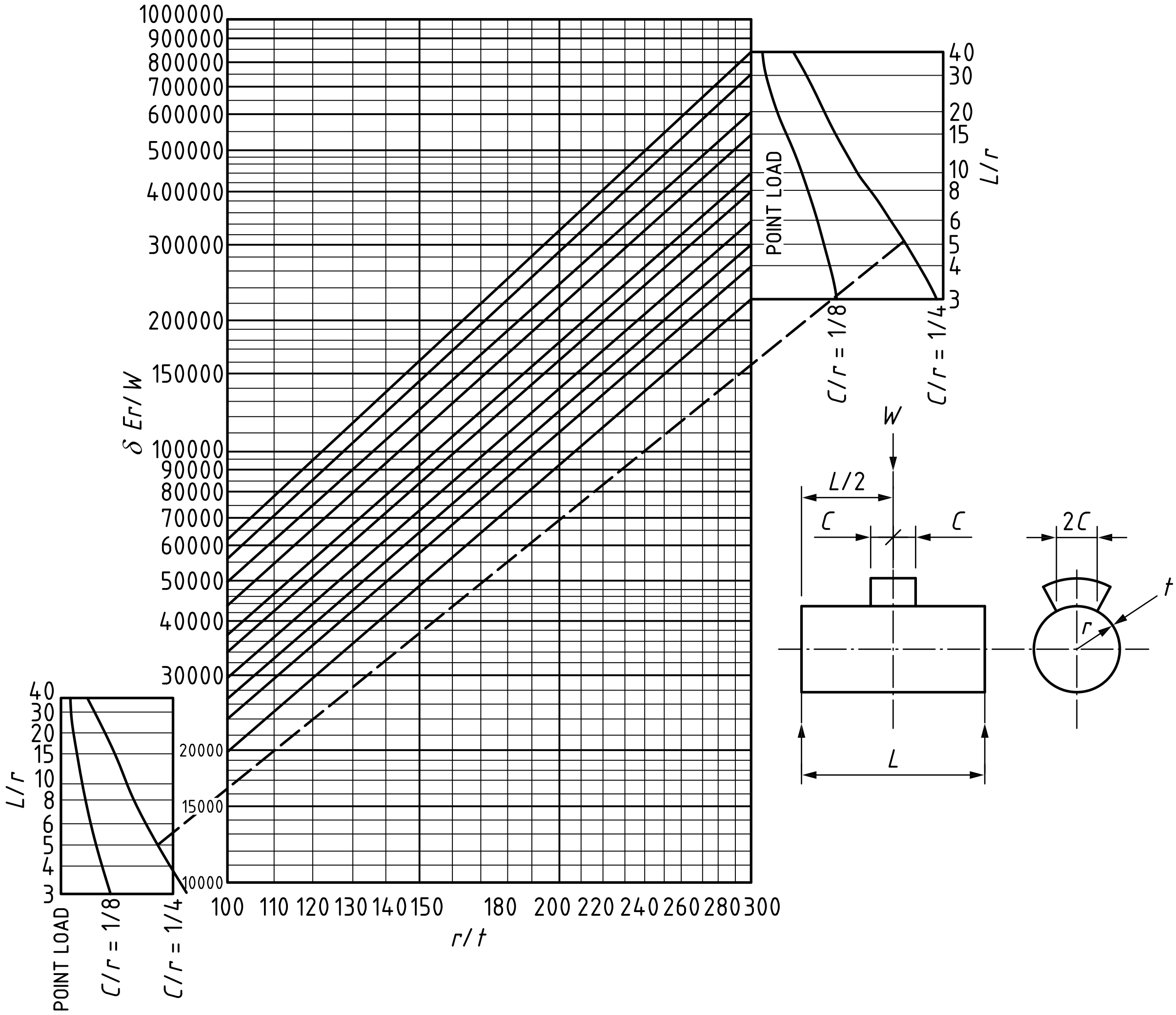
b) For $\frac{r}{t}$ between 40 and 100

Example. Find $\delta Er/W$ for L/r = 10, r/t = 70, C/r = 0 = point load. Enter L/r = 10 at right-hand side of chart, move horizontally to C/r = 0 or point load, proceed down sloping line to meet vertical line for r/t = 70, then move horizontally and read $\delta Er/W$ = 17 000.

NOTE 1 For values of r/t greater than 100 see Figure G.2.2-19.

NOTE 2 Values of δ are exclusive of the deflection of the whole shell as a beam.

Figure G.2.2-19 **Maximum radial deflection of a cylindrical shell subjected to a radial load *W* for *r*/*t* between 100 and 300**

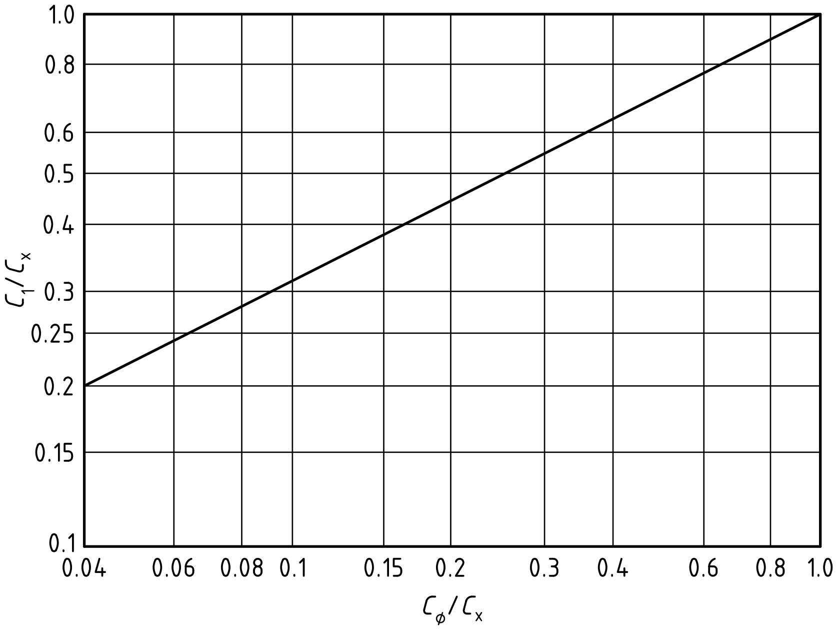


Example. Find $\delta Er/W$ for L/r = 5, C/r = ¼, r/t = 180. Enter L/r = 5 at both sides of chart, move horizontally to C/r = ¼. Join intersection points. Enter r/t at 180, move vertically to intersection line, then move horizontally and read $\delta Er/W$ = 55 000.

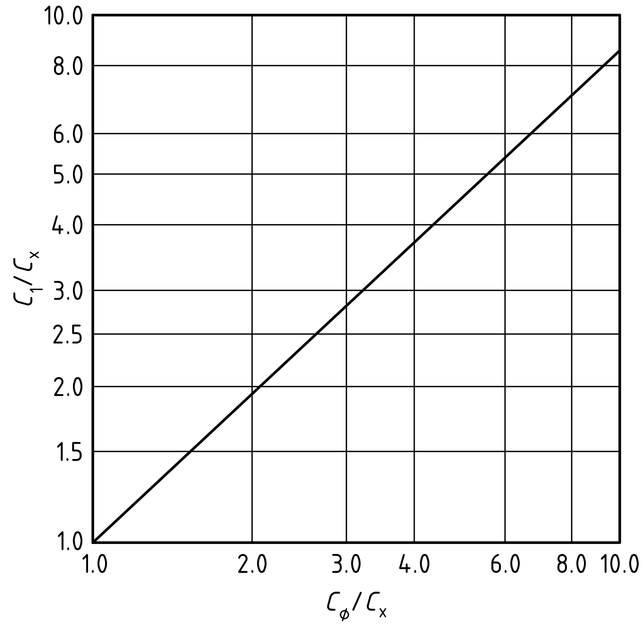
NOTE 1 For values of r/t less than 100 see Figure G.2.2-18.

NOTE 2 Values of δ are exclusive of the deflection of the whole shell as a beam.

Figure G.2.2-20 **Graphs for finding the square $2C_1 \times 2C_1$ equivalent to a rectangular loading area $2C_x \times 2C_\phi$**



a) When C_ϕ is less than C_x



b) When C_ϕ is greater than C_x

G.2.3 External moments applied to cylindrical shells

G.2.3.1 Applicability

External moments can be applied to the shell of a vessel by a load on a bracket or by the reaction at a bracket support.

For design purposes external moments are considered as described in **G.2.3.2, G.2.3.3**, **G.2.3.4** and **G.2.3.5**.

The results are not considered applicable in cases where the length of the cylinder, L, is less than its radius r (see [30]). For off-centre attachments the distance from the end of the cylinder to the edge of the attachment should be not less than $r/2$.

In addition, the ratios $C_\theta/(2r)$ (**G.2.3.2**) and C_ϕ/r (**G.2.3.3**) should not exceed those given in Figure G.2.2-1, depending on the value of r/t for the vessels.

For corresponding values of C_x/r and $C_z/(2r) > 0.25$ the data should be used with caution (see [22]).

These restrictions apply only in relation to the method of analysis in this annex. They are not intended for practical cases where experimental or other evidence may support the validity of the design falling outside these restrictions.

In cases where the applicability of the method given in this clause may be in doubt further data may be found in [30].

G.2.3.2 Circumferential moments

A circumferential moment applied to a rectangular area $C_\theta \times 2C_x$ (see Figure G.2.3-1) is resolved into two opposed loads:

$\pm W = \frac{1.5M}{C_\theta}$ acting on rectangles of sides $2C_\phi \times 2C_x$, where $C_\phi = \frac{C_\theta}{6}$, which are separated by a distance of $\frac{2C_\theta}{3}$ between centres.

For a round branch $C_\theta = 1.7r_o = 2C_x$.

G.2.3.3 Longitudinal moments

Similarly, a longitudinal moment, applied to an area $2C_\phi \times C_z$ (see Figure G.2.3-2) is resolved into two opposed loads: $\pm W = \frac{1.5M}{C_z}$ acting on rectangles of sides $2C_\phi \times 2C_x$, where $C_x = \frac{C_z}{6}$, which are separated by a distance of $\frac{2C_z}{3}$ between centres.

For a round branch $C_z = 1.7r_o = 2C_\phi$.

G.2.3.4 Maximum stresses

The maximum stresses due to the moment occur at the outer edges of the actual loaded area. The circumferential and longitudinal moments and membrane forces are given by:

$$M_\phi = M_{\phi 1} - M_{\phi 2};$$

$$M_x = M_{x1} - M_{x2};$$

$$N_\phi = N_{\phi 1} - N_{\phi 2};$$

$$N_x = N_{x1} - N_{x2}.$$

The quantities with subscript 1 are equal to those for a load W distributed over an area of $2C_\phi \times 2C_x$ and are found from Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9.

Quantities with a subscript 2 are equal to those due to a similar load at a distance $x = 5C_x$ from the centre of the loaded area for a longitudinal moment or at an angle of $\phi_1 = 5C_\phi/r$ from the radius through the centre of the loaded area for a circumferential moment. When performing manual calculations, the quantities with a subscript 2 can be neglected if the value of K_2, from Table G.2.2-1, corresponding to the value of $2C_x/L$ for a longitudinal moment, or that of K_1 corresponding to the value of $2C_x/L$ for a circumferential moment, is less than 5.0. Otherwise they are found as follows.

NOTE For computer analysis the quantities with a subscript 2 should always be included for greater accuracy.

a) *For a longitudinal moment*

1) Take $x/C_x = 5.0$ and obtain values for a radial line load from Figure G.2.2-14, Figure G.2.2-15, Figure G.2.2-16 and Figure G.2.2-17. It may be necessary to use different values of L_e (see **G.2.2.2**) for the two resolved loads if the moment is distributed over an area which is *not* small compared with its distance from the nearer end of the vessel.

2) Correct these values for a total circumferential width equal to $2C_\phi$ as in the example in **G.2.2.3.3**.

b) *For a circumferential moment*

1) Find the values at the edge of the loading area $2C_\phi \times 2C_x$ from Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9.

2) Enter the corresponding graph in Figure G.2.2-10, Figure G.2.2-11, Figure G.2.2-12 and Figure G.2.2-13 at this value. The intercept on the curve for $2C_x/L$ gives a value of $\dfrac{\phi_1 r}{C_x} = Z$.

3) The values for quantities with subscript 2 are then given by the ordinate for $\dfrac{\phi_1 r}{C_x} = \dfrac{4C_\phi}{C_x} + Z$ from the same graph.

G.2.3.5 Rotation due to external moments

It is sometimes required to find the rotation of a branch or bracket due to a moment applied to it. This is given approximately by:

$$i = \frac{3\delta_1}{C_\theta} \text{ for a circumferential moment or} \tag{G.2.3-1}$$

$$i = \frac{3\delta_1}{C_z} \text{ for a longitudinal moment} \tag{G.2.3-2}$$

where δ_1 is the deflection produced by one of the equivalent loads $W = \dfrac{1.5M}{C_\theta}$ or $= \dfrac{1.5M}{C_z}$ acting on an area of $2C_\phi \times 2C_x$ as defined in Figure G.2.3-1 or Figure G.2.3-2; δ_1 is found from Figure G.2.2-18 and Figure G.2.2-19.

NOTE Enquiry Case 5500/137 gives additional guidance on the calculation of deflections and rotations for nozzles in cylindrical shells.

Figure G.2.3-1 **Circumferential moment**

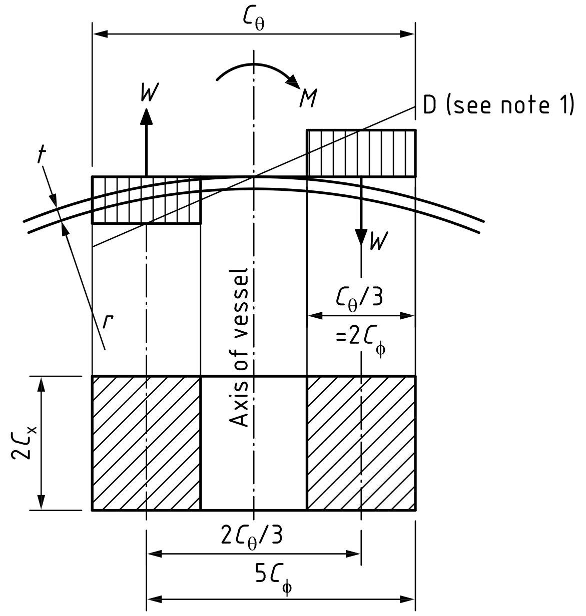
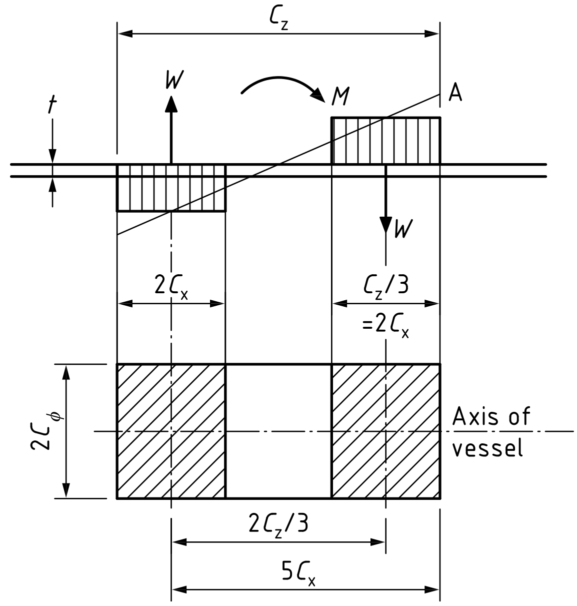
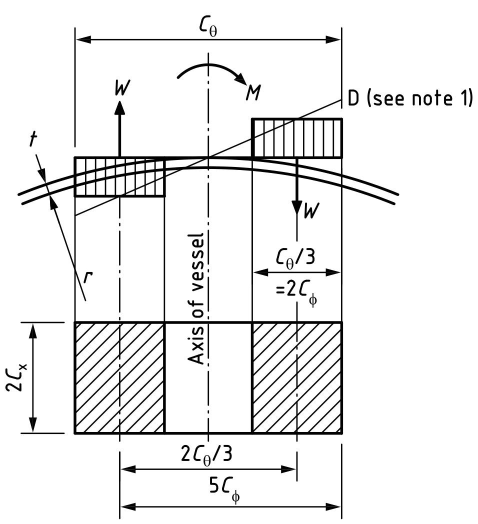


Figure G.2.3-2 **Longitudinal moment**



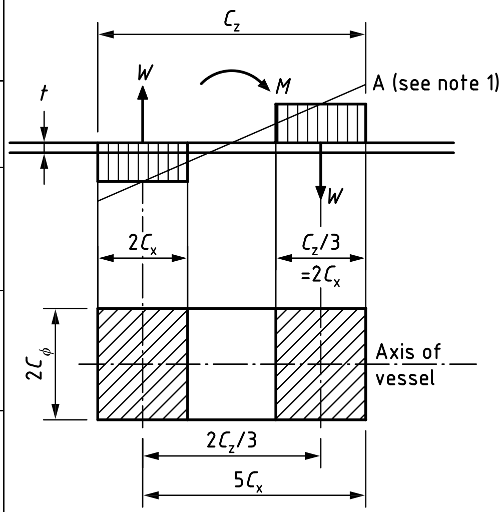
Suggested working form G1			**Load case:**	
Clause G.2.3 Circumferential moment on cylindrical shell				
Sign convention:	N_x and N_ϕ are tensile when +ve M_x and M_ϕ cause compression in the outer surface when +ve with M_C positive in the direction shown: f_x and f_ϕ are tensile when +ve			
Shell mean radius r =	Shell thickness t =	Shell length L =	Offset from centre line d =	Moment M_C =
For nozzle or circular pad	Nozzle mean radius r_o =	$C_\theta = 1.7r_o$ = $C_x = 0.85r_o$ =		
For rectangular pad	C_θ = circumferential length = C_x = (axial length) / 2 =			
$\frac{r}{t}$ =	$W = \frac{1.5M_C}{C_\theta}$ =	$\frac{W}{t^2}$ =		
$C_\phi = \frac{C_\theta}{6}$ =	$L_e = L - \frac{4d^2}{L}$ =	$\frac{C_x}{r}$ =		
$\frac{C_\phi}{C_x}$ =	$\frac{2C_x}{L_e}$ =	$64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ =		
From Table G.2.2-1 K_1 =		If $K_1 < 5.0$ assume $M_{\phi2}$, M_{x2}, $N_{\phi2}$ and N_{x2} = 0		
With $\frac{C_\phi}{C_x}$ as above	From Figure G.2.2-6 $\frac{M_{\phi1}}{W}$ =	From Figure G.2.2-7 $\frac{M_{x1}}{W}$ =	From Figure G.2.2-8 $\frac{N_{\phi1}t}{W}$ =	From Figure G.2.2-9 $\frac{N_{x1}t}{W}$ =
Z =	From Figure G.2.2-10 $\frac{\phi_1 r}{C_x}$ =	From Figure G.2.2-11 $\frac{\phi_1 r}{C_x}$ =	From Figure G.2.2-12 $\frac{\phi_1 r}{C_x}$ =	From Figure G.2.2-13 $\frac{\phi_1 r}{C_x}$ =
$\frac{4C_\phi}{C_x} + Z$ =				
With $\frac{\phi_1 r}{C_x} = \left(\frac{4C_\phi}{C_x} + Z\right)$	From Figure G.2.2-10 $\frac{M_{\phi2}}{W}$ =	From Figure G.2.2-11 $\frac{M_{x2}}{W}$ =	From Figure G.2.2-12 $\frac{N_{\phi2}t}{W}$ =	From Figure G.2.2-13 $\frac{N_{x2}t}{W}$ =
	$\frac{M_\phi}{t^2}$ = $\frac{W}{t^2} \times \left(\frac{M_{\phi1}}{W} - \frac{M_{\phi2}}{W}\right)$ =	$\frac{M_x}{t^2}$ = $\frac{W}{t^2} \times \left(\frac{M_{x1}}{W} - \frac{M_{x2}}{W}\right)$ =	$\frac{N_\phi}{t}$ = $\frac{W}{t^2} \times \left(\frac{N_{\phi1}t}{W} - \frac{N_{\phi2}t}{W}\right)$ =	$\frac{N_x}{t}$ = $\frac{W}{t^2} \times \left(\frac{N_{x1}t}{W} - \frac{N_{x2}t}{W}\right)$ =

Suggested working form G1 (*continued*)				
Clause G.2.3 Circumferential moment on cylindrical shell				
Longitudinal stresses at position D	$f_x = \frac{N_x}{t} + \frac{6M_x}{t^2} =$ G + H = (inside) $f_x = \frac{N_x}{t} - \frac{6M_x}{t^2} =$ G – H = (outside) (see Note 2)			
Circumferential stresses at position D	$f_\phi = \frac{N_\phi}{t} + \frac{6M_\phi}{t^2} =$ E + F = (inside) $f_\phi = \frac{N_\phi}{t} - \frac{6M_\phi}{t^2} =$ E – F = (outside) (see Note 2)			
For $C_x > C_\phi$	$C_1 = \sqrt{C_\phi C_x}$ or from Figure G.2.2-20a) $\frac{C_1}{C_\phi} =$	$C_1 =$ $\frac{C_1}{r} =$ $\frac{L_e}{r} =$	From Figure G.2.2-18 or Figure G.2.2-19 $\frac{\delta_1 Er}{W} =$ $E =$	$\delta_1 =$ $\theta = \frac{3\delta_1}{C_\theta} =$
For $C_\phi > C_x$	$C_1 = C_\phi^{0.93} \times C_x^{0.07}$ or from Figure G.2.2-20b) $\frac{C_1}{C_x} =$	$C_1 =$ $\frac{C_1}{r} =$ $\frac{L_e}{r} =$	From Figure G.2.2-18 or Figure G.2.2-19 $\frac{\delta_1 Er}{W} =$ $E =$	$\delta_1 =$ $\theta = \frac{3\delta_1}{C_\theta} =$

NOTE 1 Position D corresponds to quadrants Q1 and Q2 in Figure G.2.3-4.

NOTE 2 To ensure correct summation in suggested working form G3, letters have been inserted here for the stress components and their signs.

*NOTE 3 This suggested working form is provided for guidance when performing manual calculations in accordance with **G.2.3**. It does not cover additional recommendations such as those given in **G.2.2.1** and **G.2.7**.*

Suggested working form G2			**Load case:**	
Clause G.2.3 Longitudinal moment on cylindrical shell				
Sign convention:	N_x and N_ϕ are tensile when +ve M_x and M_ϕ cause compression in the outer surface when +ve with M_L positive in the direction shown: f_x and f_ϕ are tensile when +ve			
Shell mean radius $r =$	Shell thickness $t =$	Shell length $L =$	Offset from centre line $d =$	Moment $M_L =$
For nozzle or circular pad	Nozzle mean radius $r_o =$	$C_\phi = 0.85 r_o =$ $C_z = 1.7 r_o =$		
For rectangular pad	$C_\phi =$ (circumferential length) / 2 = $C_z =$ axial length =			
$\frac{r}{t} =$	$W = \frac{1.5M_L}{C_z}$ $=$	$\frac{W}{t^2} =$		
$C_x = \frac{C_z}{6} =$	$L_e = L - \frac{4d^2}{L}$ $=$	$\frac{C_x}{r} =$		
$\frac{C_\phi}{C_x} =$	$\frac{2C_x}{L_e} =$	$64\frac{r}{t}\left(\frac{C_x}{r}\right)^2$ $=$		
From Table G.2.2-1 $K_2 =$		If $K_2 < 5.0$ assume $M_{\phi 2}$, M_{x2}, $N_{\phi 2}$ and $N_{x2} = 0$		
With $\frac{C_\phi}{C_x}$ as above	From Figure G.2.2-6 $\frac{M_{\phi 1}}{W} =$	From Figure G.2.2-7 $\frac{M_{x1}}{W} =$	From Figure G.2.2-8 $\frac{N_{\phi 1}t}{W} =$	From Figure G.2.2-9 $\frac{N_{x1}t}{W} =$
With $\frac{C_\phi}{C_x} = 0$	From Figure G.2.2-6 $\frac{M_{\phi 0}}{W} =$	From Figure G.2.2-7 $\frac{M_{x0}}{W} =$	From Figure G.2.2-8 $\frac{N_{\phi 0}t}{W} =$	From Figure G.2.2-9 $\frac{N_{x0}t}{W} =$
Correction factor $C_f =$	$\frac{M_{\phi 1}}{M_{\phi 0}} =$	$\frac{M_{x1}}{M_{x0}} =$	$\frac{N_{\phi 1}}{N_{\phi 0}} =$	$\frac{N_{x1}}{N_{x0}} =$
With $\frac{x}{C_x} = 5$	From Figure G.2.2-14 $\frac{M_{\phi 3}}{W} =$	From Figure G.2.2-15 $\frac{M_{x3}}{W} =$	From Figure G.2.2-16 $\frac{N_{\phi 3}t}{W} =$	From Figure G.2.2-17 $\frac{N_{x3}t}{W} =$
	$\frac{M_{\phi 2}}{W} = \frac{M_{\phi 3}}{W} \times C_f$ $=$	$\frac{M_{x2}}{W} = \frac{M_{x3}}{W} \times C_f$ $=$	$\frac{N_{\phi 2}t}{W} = \frac{N_{\phi 3}t}{W} \times C_f$ $=$	$\frac{N_{x2}t}{W} = \frac{N_{x3}t}{W} \times C_f$ $=$
	$\frac{M_\phi}{t^2} =$ $\frac{W}{t^2} \times \left(\frac{M_{\phi 1}}{W} - \frac{M_{\phi 2}}{W}\right)$ $=$	$\frac{M_x}{t^2} =$ $\frac{W}{t^2} \times \left(\frac{M_{x1}}{W} - \frac{M_{x2}}{W}\right)$ $=$	$\frac{N_\phi}{t} =$ $\frac{W}{t^2} \times \left(\frac{N_{\phi 1}t}{W} - \frac{N_{\phi 2}t}{W}\right)$ $=$	$\frac{N_x}{t} =$ $\frac{W}{t^2} \times \left(\frac{N_{x1}t}{W} - \frac{N_{x2}t}{W}\right)$ $=$

Suggested working form G2 (*continued*)				
Clause G.2.3 Longitudinal moment on cylindrical shell				
Longitudinal stresses at position A	$f_x = \frac{N_x}{t} + \frac{6M_x}{t^2} =$ K + L = (inside) $f_x = \frac{N_x}{t} - \frac{6M_x}{t^2} =$ K – L = (outside) (see Note 2)			
Circumferential stresses at position A	$f_\phi = \frac{N_\phi}{t} + \frac{6M_\phi}{t^2} =$ I + J = (inside) $f_\phi = \frac{N_\phi}{t} - \frac{6M_\phi}{t^2} =$ I – J = (outside) (see Note 2)			
For $C_x > C_\phi$	$C_1 = \sqrt{C_\phi C_x}$ or from Figure G.2.2-20a) $\frac{C_1}{C_\phi} =$	$C_1 =$ $\frac{C_1}{r} =$ $\frac{L_e}{r} =$	From Figure G.2.2-18 or Figure G.2.2-19 $\frac{\delta_1 Er}{W} =$ $E =$	$\delta_1 =$ $\theta = \frac{3\delta_1}{C_\theta} =$
For $C_\phi > C_x$	$C_1 = C_\phi^{0.93} \times C_x^{0.07}$ or from Figure G.2.2-20b) $\frac{C_1}{C_\phi} =$	$C_1 =$ $\frac{C_1}{r} =$ $\frac{L_e}{r} =$	From Figure G.2.2-18 or Figure G.2.2-19 $\frac{\delta_1 Er}{W} =$ $E =$	$\delta_1 =$ $\theta = \frac{3\delta_1}{C_\theta} =$

NOTE 1 Position A corresponds to quadrants Q1 and Q4 in Figure G.2.3-4.

NOTE 2 To ensure correct summation in suggested working form G3, letters have been inserted here for the stress components and their signs.

*NOTE 3 This suggested working form is provided for guidance when performing manual calculations in accordance with **G.2.3**. It does not cover additional recommendations such as those given in **G.2.2.1** and **G.2.7**.*

G.2.3.6 Summation of maximum stresses due to local loads on a cylindrical shell

G.2.3.6.1 General

Although the exact location of the stresses calculated in **G.2.3.2**, **G.2.3.3**, **G.2.3.4** and **G.2.3.5** is not known the stresses may be considered to lie within the 180° sectors shown in Figure G.2.3-3. The sign of the stress in one sector is known to be reversed in the opposite sector.

By dividing the loaded area into quadrants and summing the maximum stresses in each quadrant, a maximum combined stress is obtained. The method for this is shown in suggested working form G3. The stresses due to pressure are combined with those due to the local loads. The combined stresses and stress intensities are assessed against the allowable values specified in **A.3.3**.

The stress components should be inserted into the table according to the correct convention. To define this convention, each stress calculated in suggested working forms G1 and G2, including its algebraic sign, has been assigned a letter. These numbers should be entered into suggested working form G3, example A, in accordance with the convention shown.

NOTE 1 The signs of F_R, F_C, M_L, M_C and M_T are positive when they act in the direction shown in Figure G.2.3-4.

*NOTE 2 N_x and N_ϕ are positive for tensile membrane stresses and M_x and M_ϕ are positive when they cause compressive stresses on the outer surface of the shell. Stresses f_x and f_ϕ are positive when tensile and negative when compressive. This is in accordance with **G.2.2.2.3**.*

NOTE 3 The letters A to D apply to the stresses resulting from a radial load F_R. When F_R is positive, A and C represent positive numbers in quadrant Q_1 on the inside and B and D represent negative numbers in quadrant Q_1 on the inside.

NOTE 4 Absolute values of shear stress are used in the table. This is because the actual shear stress pattern is complex and because the formulae for shear stress due to shear force are approximate.

*NOTE 5 At the nozzle o.d. where a compensation pad is fitted, or at the edge of a load on an attachment or support, distribute N_ϕ, M_ϕ, N_x and M_x as in **G.3.1.5**. For a nozzle with a pad, an additional hoop moment should be added to M_ϕ as in **G.2.7**.*

NOTE 6 In the calculation of total stress intensity (lines 27 to 29 and 32 to 34 of the table) the pressure term has been omitted for simplicity.

Figure G.2.3-3 **Sector stresses**

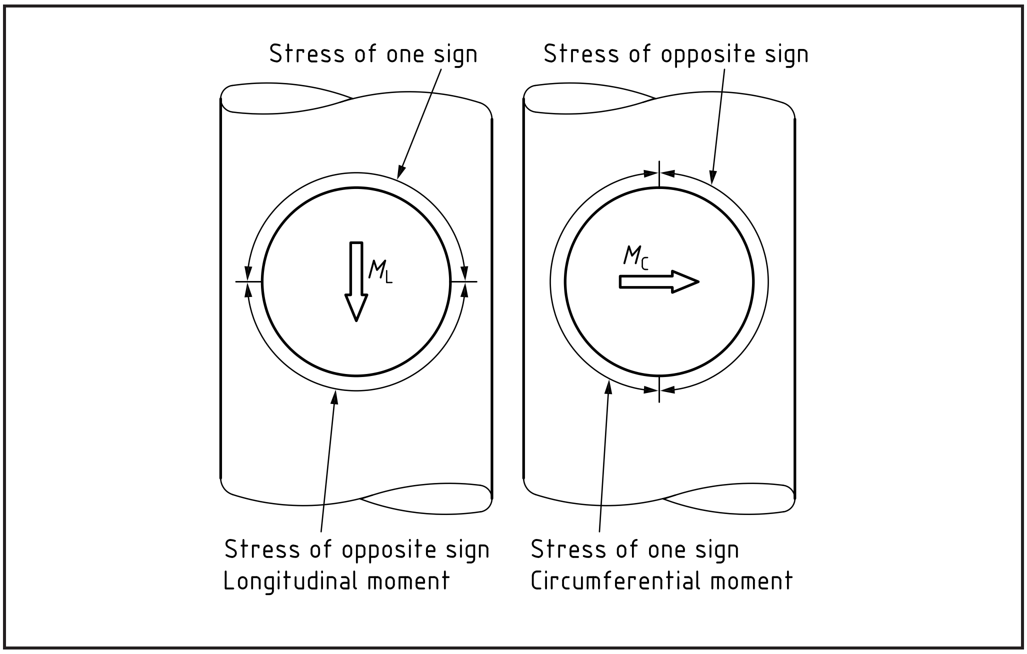
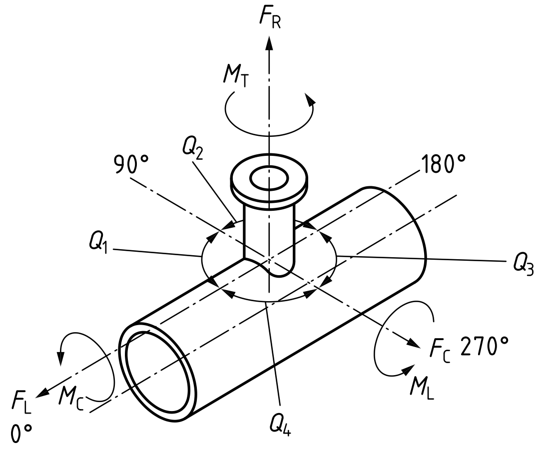


Figure G.2.3-4 **Notation for external loads at a nozzle or attachment on a cylindrical shell**



Suggested working form G3	Load Case:		Nozzle o.d. / pad o.d. / loaded area dimensions[a]	
Clause **G.2.3.6** Summation of maximum stresses due to local loads on a cylindrical shell	Radial load F_R		Shell thickness / Shell + pad thickness[a]	
	Shear force F_C			
	Shear force F_L		Shell i.d.	
	Torsional moment M_T		Design pressure	
	Circumferential moment M_C		Design stress (f)	
	Longitudinal moment M_L		Yield stress	

	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Circumferential stresses								
	Membrane component (N_ϕ/t) due to:								
1	Radial load	A	A	A	A	A	A	A	A
2	Circumferential moment	E	E	E	E	– E	– E	– E	– E
3	Longitudinal moment	I	I	– I	– I	– I	– I	I	I
4	Sub-total due to local loads								
5	Pressure (f_P from **G.2.3.6.2**)								
6	Sub-total ($f_{\phi m}$)								
	Bending component ($6M_\phi/t^2$) due to:								
7	Radial load	B	– B	B	– B	B	– B	B	– B
8	Circumferential moment	F	– F	F	– F	– F	F	– F	F
9	Longitudinal moment	J	– J	– J	J	– J	J	J	– J
10	Sub-total ($f_{\phi b}$)								
11	Total circumferential stress (f_ϕ)								
	Longitudinal stresses								
	Membrane component (N_x/t) due to:								
12	Radial load	C	C	C	C	C	C	C	C
13	Circumferential moment	G	G	G	G	– G	– G	– G	– G
14	Longitudinal moment	K	K	– K	– K	– K	– K	K	K
15	Sub-total due to local loads								
16	Pressure (f_P from **G.2.3.6.2**)								
17	Sub-total (f_{xm})								
	Bending component ($6M_x/t^2$) due to:								
18	Radial load	D	– D	D	– D	D	– D	D	– D
19	Circumferential moment	H	– H	H	– H	– H	H	– H	H
20	Longitudinal moment	L	– L	– L	L	– L	L	L	– L
21	Sub-total (f_{xb})								
22	Total longitudinal stress (f_x)								

[a] Delete as appropriate

Suggested working form G3 (*continued*)									
	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Shear stresses (from **G.2.3.6.3**) due to:								
23	Torsion moment								
24	Circumferential shear force								
25	Longitudinal shear force								
26	Total shear stress (τ)								
	Check of total stress intensity (membrane + bending) to **A.3.3.1** and **A.3.3.2**)								
27	$f_1 = \left[f_\phi + f_x + \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right]/2$								
28	$f_2 = \left[f_\phi + f_x - \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right]/2$								
29	$f_2 - f_1$								
	Maximum total stress intensity = maximum absolute value in rows 27, 28 and 29 = Allowable stress at nozzle [a] = 2.25*f* = or at edge of a compensation pad, attachment or support [a] = 2*f* =								
	Check of buckling stress to **A.3.3.3**								
30	Row 4 + row 10 if row 4 is compressive (–)								
31	Row 15 + row 21 if row 15 is compressive								
	Maximum compressive stress in rows 30 and 31 = Allowable compressive stress = –0.9 x yield stress =								
	At edge of compensation pad, attachment or support, check of membrane stress intensity to **A.3.3.1**[a]								
32	$f_{1m} = \left[f_{\phi m} + f_{xm} + \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right]/2$								
33	$f_{2m} = \left[f_{\phi m} + f_{xm} - \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right]/2$								
34	$f_{2m} - f_{1m}$								
	Maximum membrane stress intensity = maximum absolute value in rows 32, 33 and 34 = Allowable membrane stress = 1.2*f* =								
[a] Delete as appropriate									

G.2.3.6.2 Pressure stress formulae

The applicable pressure stress formulae are as follows.

a) At nozzle o.d.

Calculate $\rho = \frac{d}{D}\sqrt{\frac{D}{2e_{as}}}$ and $\frac{e_{ab}}{e_{as}}$.

Using Figure 3.5-9, Figure 3.5-10 or Figure 3.5-11 with e_{ab} and e_{as} in place of e_{rb} and e_{rs}, obtain a value for Ce_{as}/e_{ps}.

In the case of flush nozzles in cylindrical shells, if $d/D > 0.3$ calculate $Y = \frac{e_{ab}}{d}\sqrt{\frac{2D}{e_{as}}}$ and obtain a value for Ce_{as}/e_{ps} from Figure 3.5-12.

For flush nozzles in cylindrical shells, if $0.2 < d/D < 0.3$ calculate ρ and obtain a value for $(Ce_{as}/e_{ps})_1$ from Figure 3.5-11, and calculate Y and obtain a value for $(Ce_{as}/e_{ps})_2$ from Figure 3.5-12. The value of Ce_{as}/e_{ps} is calculated as follows:

$$\frac{Ce_{as}}{e_{ps}} = \left(\frac{Ce_{as}}{e_{ps}}\right)_1 + 10\left(\frac{d}{D} - 0.2\right)\left[\left(\frac{Ce_{as}}{e_{ps}}\right)_2 - \left(\frac{Ce_{as}}{e_{ps}}\right)_1\right]$$

Use the value of Ce_{as}/e_{ps} in the following expression:

$$f_p = \left(\frac{2.25}{1.1}\right)\left(\frac{Ce_{as}}{e_{ps}}\right)\left(\frac{pD}{2e_{as}}\right) \quad \text{(G.2.3-3)}$$

NOTE 1 The formula may be used for both the longitudinal and circumferential directions.

where d, D, e_{as} and e_{ab} are as defined in **3.5.4.1**.

NOTE 2 The thicknesses e_{as} and e_{ab} are the analysis thicknesses local to the nozzle to shell attachment, and include local thickening of the shell or nozzle and the thickness of a reinforcing pad if fitted.

b) In the shell at the edge of a loaded area, excluding nozzles.

1) Circumferentially

$$f_p = \frac{pD}{2e_{as}} \quad \text{(G.2.3-4)}$$

2) Longitudinally

$$f_p = \frac{pD}{4e_{as}} \quad \text{(G.2.3-5)}$$

where

e_{as} = analysis thicknesses of shell at edge of attachment.

NOTE The thickness e_{as} includes local thickening of the shell but excludes the thickness of any reinforcing pad.

c) In the shell at the edge of a pad.

1) Circumferentially

$$f_p = \frac{pD}{2e_{asp}} \quad \text{(G.2.3-6)}$$

2) Longitudinally

$$f_p = \frac{pD}{4e_{asp}} \quad \text{(G.2.3-7)}$$

where

e_{asp} = shell plate analysis thickness (see **1.6**).

G.2.3.6.3 Shear stress formulae

Due to:

a) torsion (M_T)

$$\tau = \frac{2M_T}{\pi d_o^2 T_1} \quad \text{(G.2.3-8)}$$

b) circumferential shear force (F_C)

$$\tau = \frac{2F_C}{\pi d_o T_1} \quad \text{(G.2.3-9)}$$

c) longitudinal shear force (F_L)

$$\tau = \frac{2F_L}{\pi d_o T_1} \quad \text{(G.2.3-10)}$$

where

d_o is the outside diameter of the nozzle or pad;

T_1 is the shell plus pad analysis thickness at the nozzle or attachment o.d. or the shell analysis thickness at the pad o.d.

NOTE In general the shear forces may be neglected but where required the formulae shown may be used. Equation (G.2.3-9) and Equation (G.2.3-10) are from WRC 107 [30].

G.2.3.7 Nozzle strength

For local loads on nozzles the longitudinal stresses in the nozzle should be verified as follows:

a) Longitudinal stress verification:

$$\frac{pr_o}{2e_{ab}} + \frac{\sqrt{M_C^2 + M_L^2}}{\pi r_o^2 e_{ab}} + \frac{F_R}{2\pi r_o e_{ab}} \leq f_b \quad \text{(G.2.3-11)}$$

where

e_{ab} and f_b are as defined in **3.5.4.1**;

F_R, M_C and M_L are as shown in Figure G.2.3-4.

b) Longitudinal instability check (with $p = 0$):

$$M_{max} = \pi r_o{}^2 e_{ab} \sigma_{z,allow} \quad \text{(G.2.3-12)}$$

$$F_{max} = 2\pi r_o e_{ab} \sigma_{z,allow} \quad \text{(G.2.3-13)}$$

$$\frac{\sqrt{M_C^2 + M_L^2}}{M_{max}} + \frac{|F_R|}{F_{max}} \leq 1.0 \quad \text{(G.2.3-14)}$$

where

$\sigma_{z,\ allow}$ is the maximum allowable longitudinal compressive stress in the nozzle (see **A.3.5**).

If F_R is positive it should be set to zero.

c) Reinforced nozzle wall:

For reinforcement provided as shown in Figures 3.5-19, 3.5-20 or 3.5-21, a longitudinal stress verification and longitudinal instability check should also be performed using values of r_o and e_{ab} at the thinnest part of the nozzle.

G.2.4 Local loads on spherical shells, rigid attachments

G.2.4.1 Applicability

The methods in this clause are not considered applicable in cases where the ratio r_o/r is larger than one-third.

G.2.4.2 Initial development

This clause is concerned with the stresses and deflections due to local radial loads or moments on spherical shells. Because these are local in character and die out rapidly with increasing distance from the point of application, the data can be applied to local loads on the spherical parts of pressure vessel ends as well as to complete spheres.

For pressure vessel ends with spherical portions, the method in this clause assumes that the local load on the spherical portion, the loading connection and any associated local reinforcement are located within a concentric circle of diameter no larger than 0.8 times the mean diameter of the end (see Figure 3.5-8).

For convenience, the loads are considered as acting on a pipe of radius r_o which is assumed to be a rigid body fixed to the sphere. This is the condition for the majority of practical cases.

Loads applied through square fittings of side $2C_x$ can be treated approximately as distributed over a circle of radius $r_o = C_x$.

Loads applied through rectangular brackets of sides $2C_x$ and $2C_\phi$ can be treated approximately as distributed over a circle of radius $r_o = \sqrt{C_x C_\phi}$.

The following forces and moments are set up in the wall of the vessel by any local load or moment.

a) Meridional moment M_x: acting per unit width on a normal section, formed by the intersection of shell with a cone of semi-vertex angle.

   $\phi = \sin^{-1}\dfrac{x}{r}$ (see Figure G.2.4-2 and Figure G.2.4-8)

b) Circumferential moment M_ϕ: acting per unit width on a meridional section passing through the axis of the shell and the axis of the branch.

c) Meridional membrane force: acting per unit width on a normal section as for the meridional moment M_x.

d) Circumferential membrane force: acting per unit width on a meridional section as defined for the circumferential moment M_ϕ.

A moment is considered as positive if it causes compression at the outside of the vessel.

A membrane force is considered as positive if it causes tension in the vessel wall.

A deflection is considered positive if it is away from the centre of the sphere.

These forces and moments and the deflection of the shell due to the load can be found in terms of the non-dimensional parameters:

$$s = \frac{1.82x}{\sqrt{rt}} \qquad \text{(G.2.4-1)}$$

and

$$u = \frac{1.82r_o}{\sqrt{rt}} \qquad \text{(G.2.4-2)}$$

These two factors can be found quickly from the chart in Figure G.2.4-1, given x, r_o and the ratio r/t.

The charts in **G.2.4.3** and **G.2.4.4** (Figure G.2.4-3 to Figure G.2.4-13) give graphs of non-dimensional functions of these deflections, forces and moments plotted against the parameter s for given values of u which have been derived from [3] and [9].

The full curves in each set of graphs give conditions at the edge of the loaded area where $u = s$. The most unfavourable combination of bending and direct stresses is usually found here.

The dotted curves for particular values of u give conditions at points in the shell away form the edge of the loaded area where x is greater than r_o and u is therefore less than s.

Since the charts are non-dimensional they can be used in any consistent system of units.

The stresses and deflections found from these charts will be reduced by the effect of internal pressure but this reduction is small and can usually be neglected in practice. (See [8] and [9].)

G.2.4.3 Stresses and deflections due to radial loads

Figure G.2.4-2 shows a radial load applied to a spherical shell through a branch of radius r_o.

The deflections, moments and membrane forces due to the load W can be found as follows from Figure G.2.4-3, Figure G.2.4-4, Figure G.2.4-5, Figure G.2.4-6 and Figure G.2.4-7. For explanation of these curves see **G.2.4.4**. For an example of their use see Annex W.

a) Deflection from Figure G.2.4-3 and the relation:

$$\delta = \text{ordinate of curve} \times \frac{W_r}{Et^2}$$

b) Meridional moment M_x per unit width from Figure G.2.4-4 and the relation:

$$M_x = \text{ordinate of } M_x \text{ curve} \times W$$

c) Circumferential moment M_ϕ per unit width from Figure G.2.4-5 and the relation:

$$M_\phi = \text{ordinate of } M_\phi \text{ curve} \times W$$

d) Meridional membrane force N_x per unit width from Figure G.2.4-6 and the relation:

$$N_x = \text{ordinate of } N_x \text{ curve} \times W/t$$

e) Circumferential moment N_ϕ per unit width from Figure G.2.4-7 and the relation:

$$N_\phi = \text{ordinate of } N_\phi \text{ curve} \times W/t$$

Figure G.2.4-1 **Chart for finding *s* and *u***

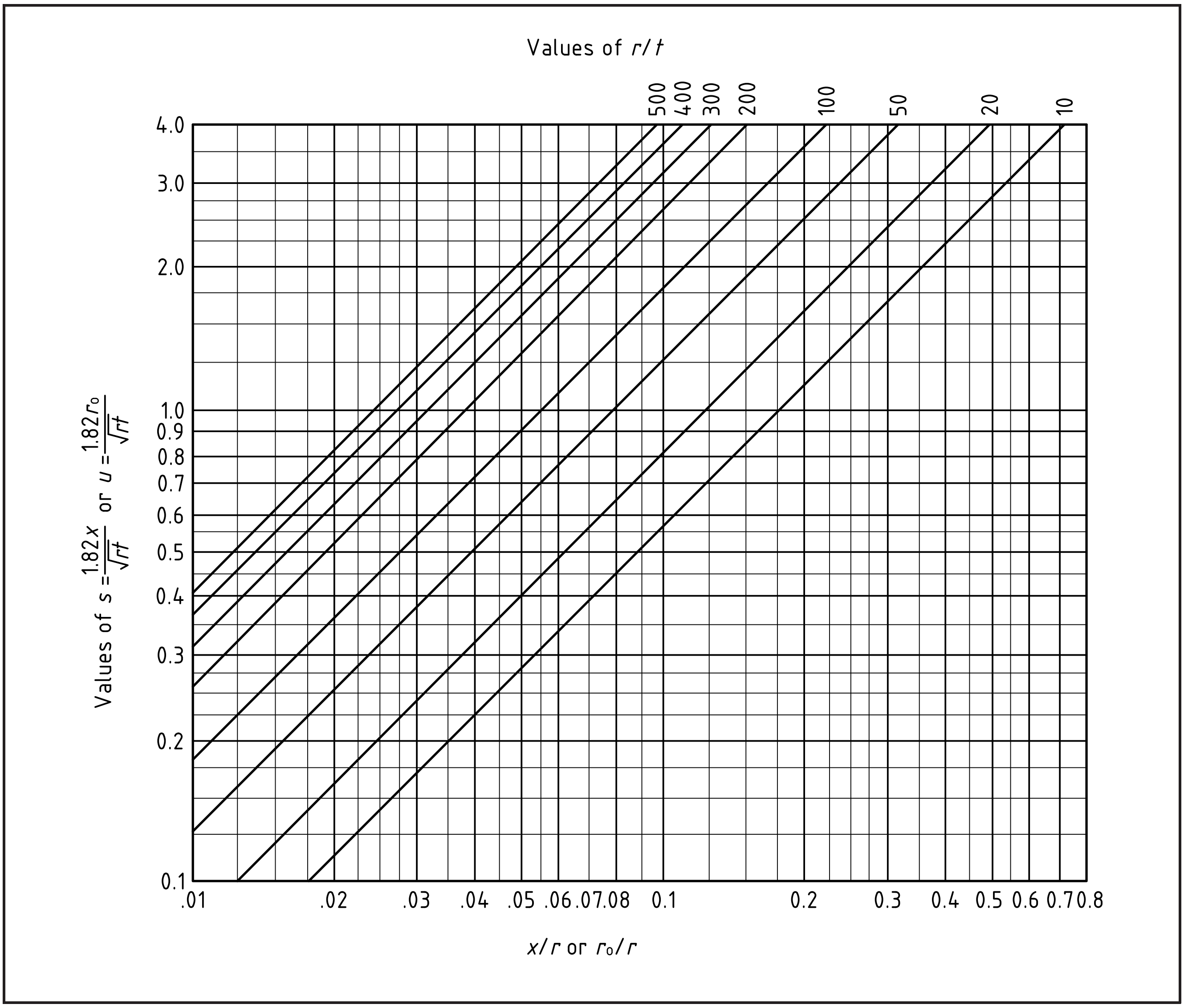


Figure G.2.4-2 **Spherical shell subjected to a radial load**

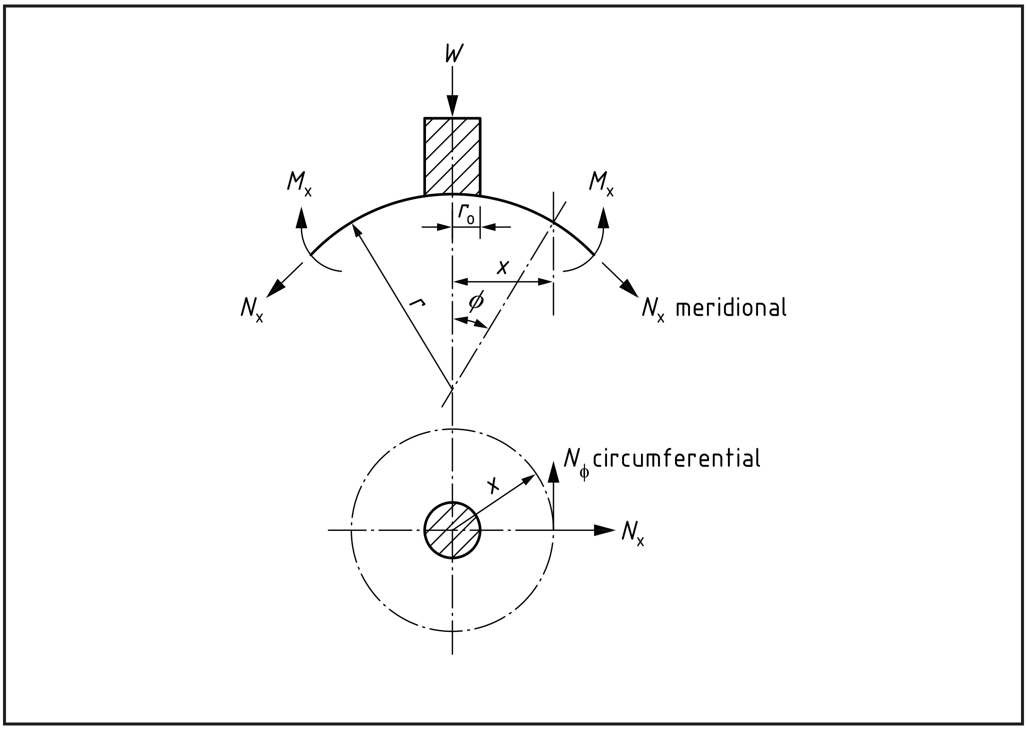


Figure G.2.4-3 **Deflections of a spherical shell subjected to a radial load *W***

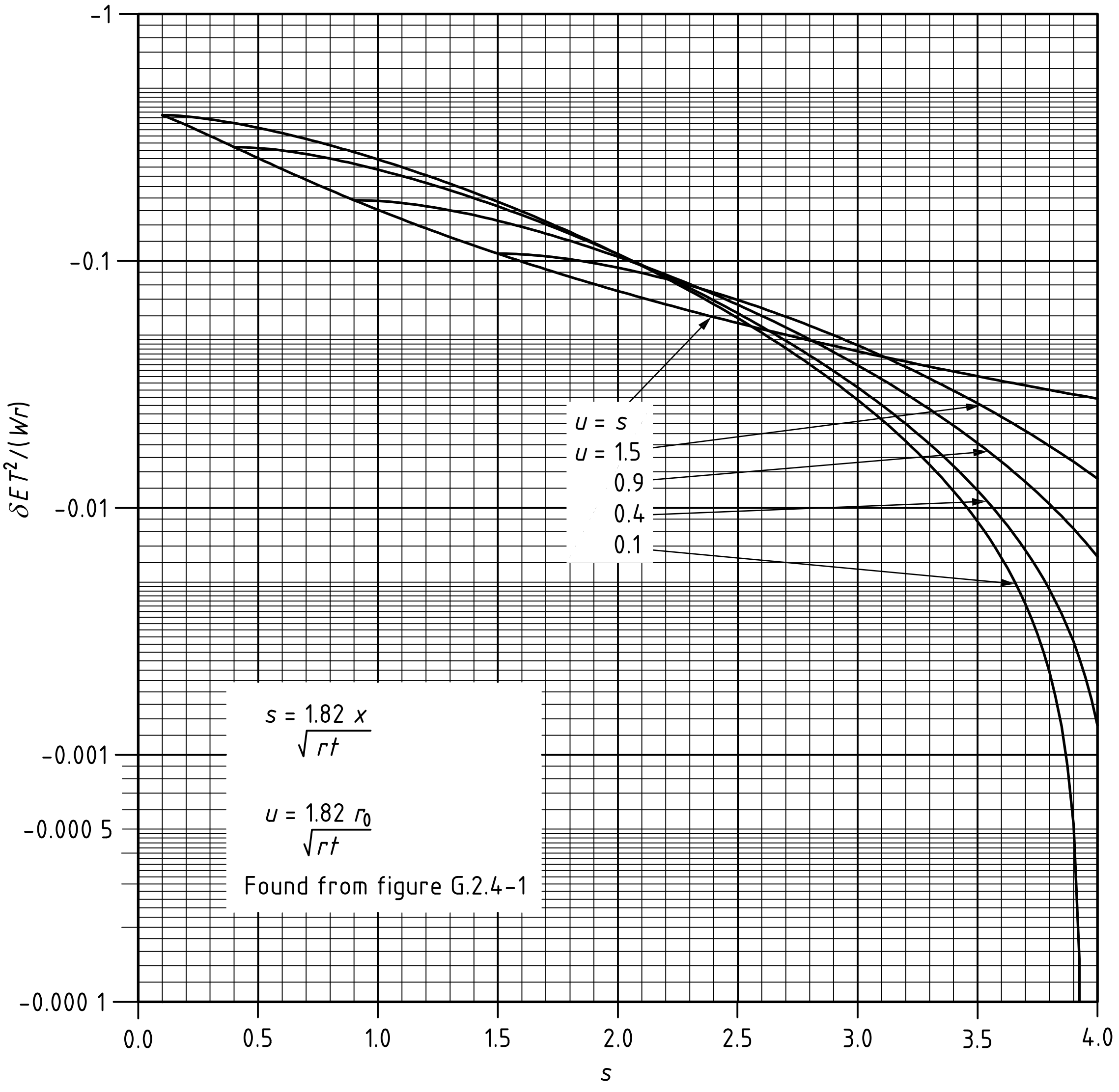


Figure G.2.4-4 **Meridional moment M_x in a spherical shell subjected to radial load W**

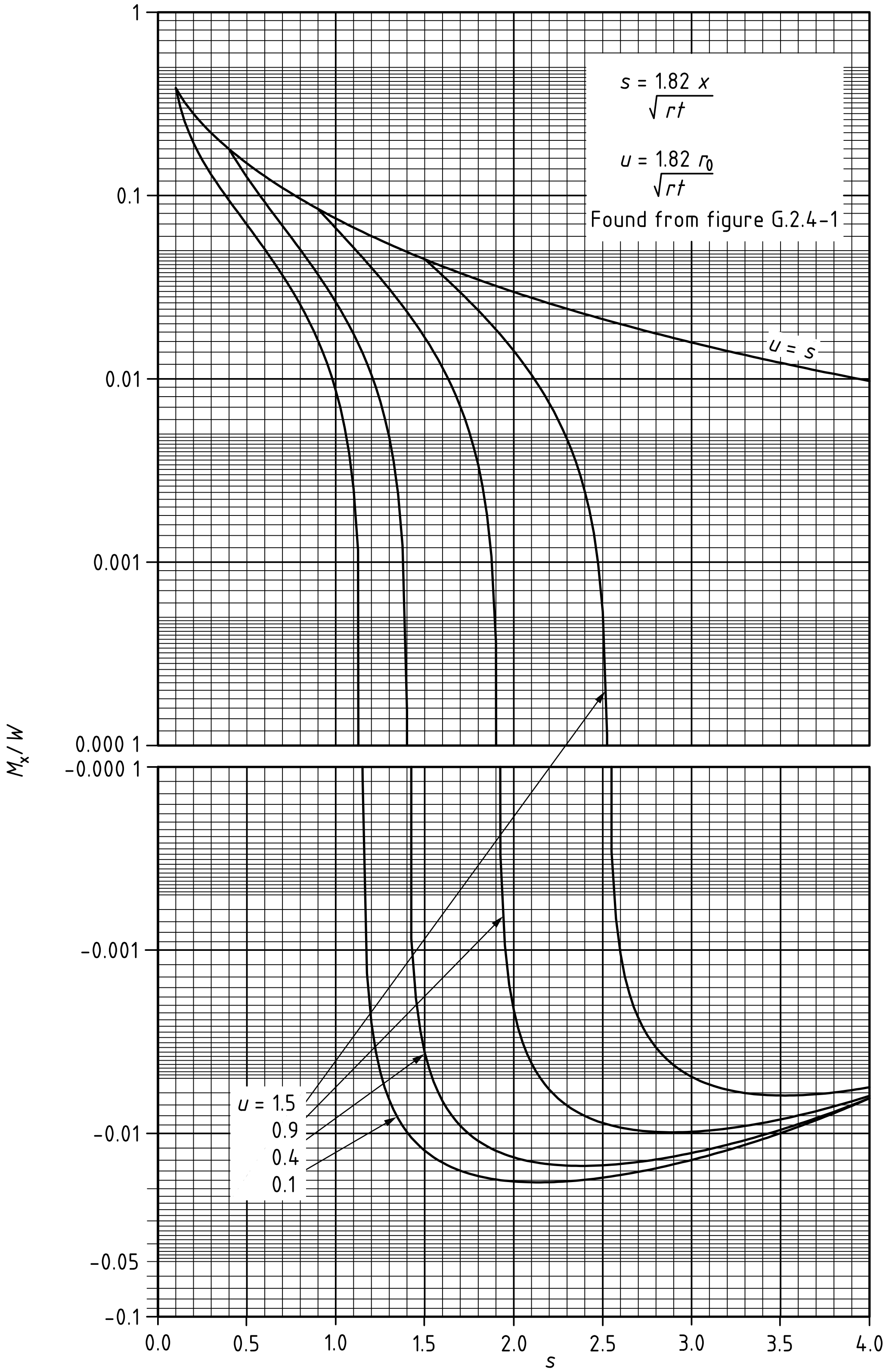


Figure G.2.4-5 **Circumferential moment M_ϕ in a spherical shell subjected to a radial load W**

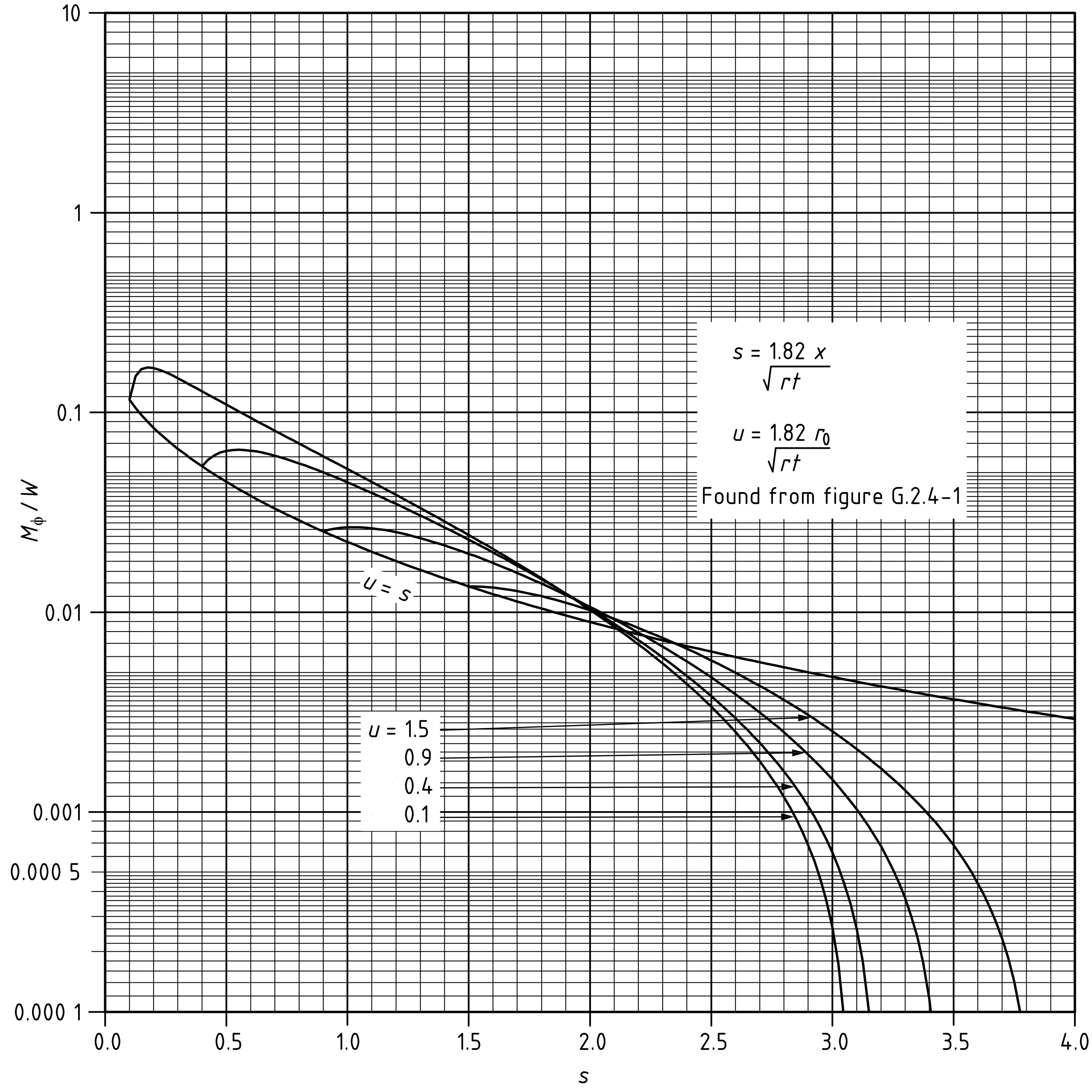


Figure G.2.4-6 **Meridional force N_x in a spherical shell subjected to a radial load *W***

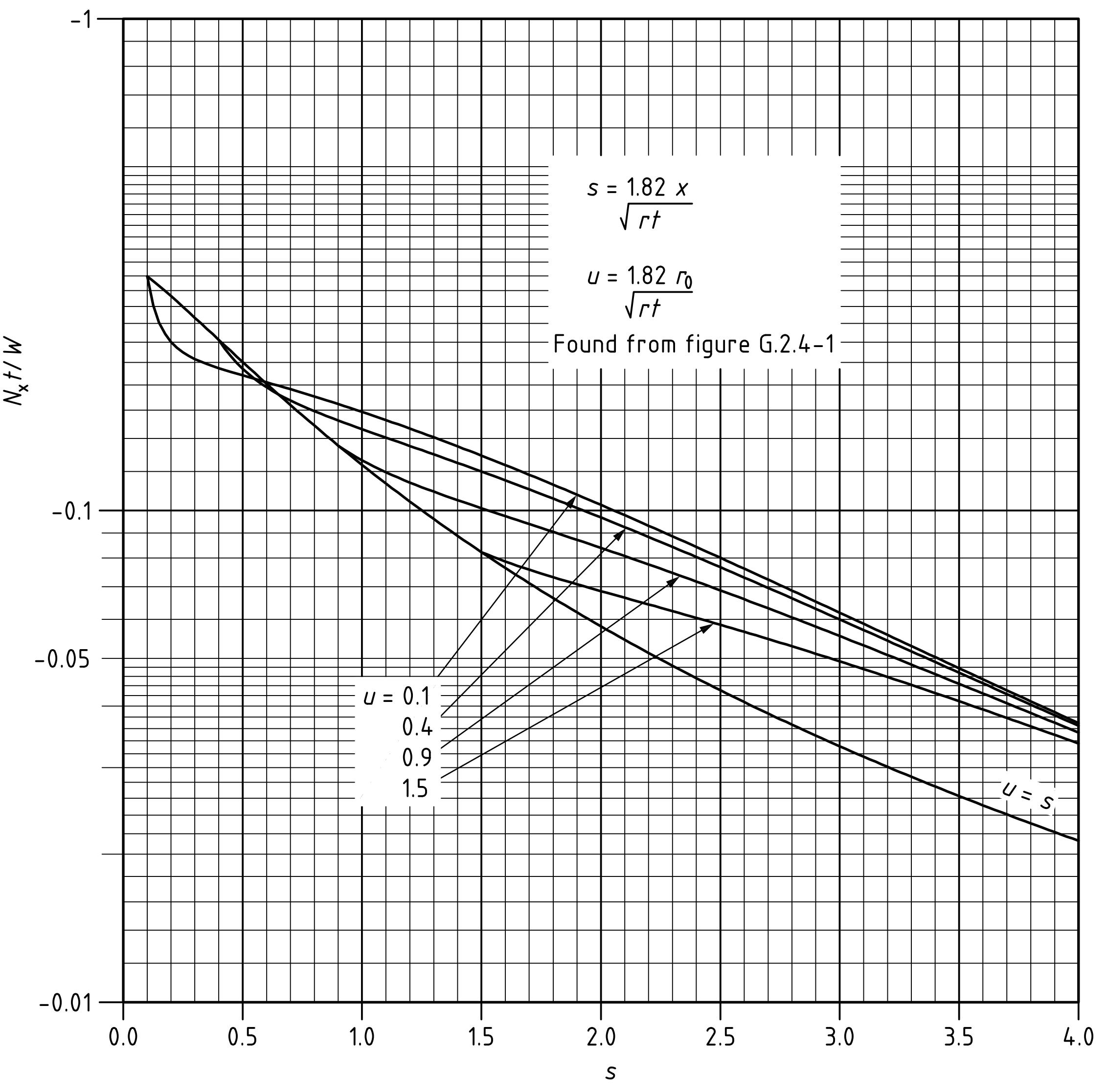
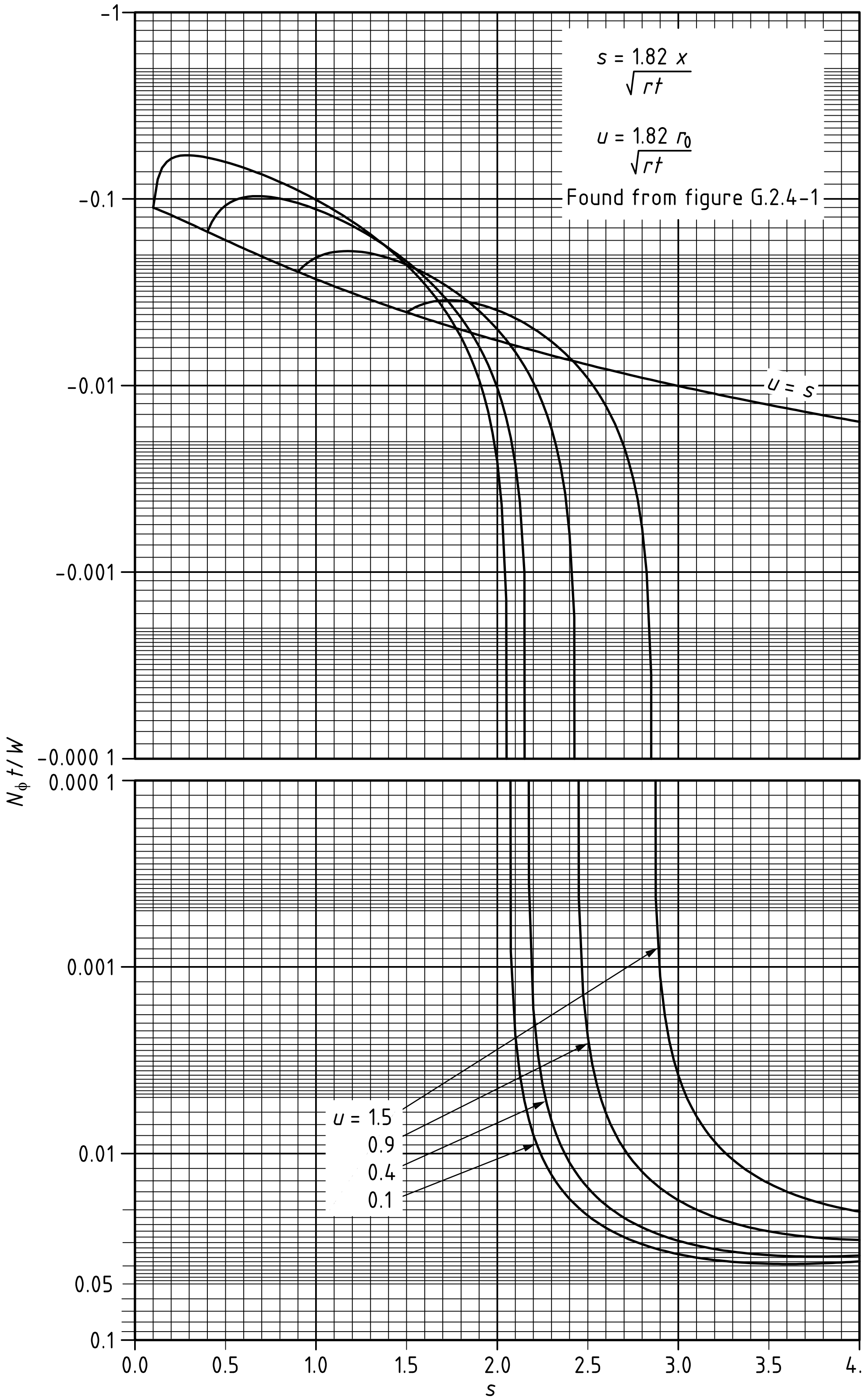


Figure G.2.4-7 **Circumferential force N_ϕ in a spherical shell subjected to a radial load W**



G.2.4.4 Stresses, deflections and slopes due to an external moment

Figure G.2.4-8 shows an external moment applied to a spherical shell through a branch of radius r_o.

In this case the deflections, moments and membrane forces depend on the angle θ as well as on the distance x from the axis of the branch. They can be found as follows from Figure G.2.4-9, Figure G.2.4-10, Figure G.2.4-11, Figure G.2.4-12 and Figure G.2.4-13. For explanations of these curves see **G.2.4.2**.

a) Deflections from Figure G.2.4-9 and the relation:

$$\delta = \text{ordinate of curve} \times \frac{M\cos\theta\sqrt{\frac{r}{t}}}{Et^2}$$

b) Meridional moment M_x per unit width from Figure G.2.4-10 and the relation:

$$M_x = \text{ordinate of } M_x \text{ curve} \times \frac{M\cos\theta}{\sqrt{rt}}$$

c) Circumferential moment M_ϕ per unit width from Figure G.2.4-11 and the relation:

$$M_\phi = \text{ordinate of } M_\phi \text{ curve} \times \frac{M\cos\theta}{\sqrt{rt}}$$

d) Meridional membrane force N_x per unit width from Figure G.2.4-12 and the relation:

$$N_x = \text{ordinate of } N_x \text{ curve} \times \frac{M\cos\theta}{t\sqrt{rt}}$$

e) Circumferential membrane force N_ϕ per unit width from Figure G.2.4-13 and the relation:

$$N_\phi = \text{ordinate of } N_\phi \text{ curve} \times \frac{M\cos\theta}{t\sqrt{rt}}$$

Equal and opposite maximum values of all the above quantities occur in the plane of the moment, i.e. where θ (see Figure G.2.4-8) = 0° and θ = 180°.

The slope of the branch due to the external moment is found from:

$$i_b = \frac{\delta_1}{r_o} \qquad \text{(G.2.4-3)}$$

where δ_1 is the maximum deflection at the edge of the branch for θ = 0 and $u = s$, i.e.:

$$\delta_1 = \frac{M\sqrt{\frac{r}{t}}}{Et^2} \times \text{ordinate of full curve in Figure G.2.4-9 for } x = r_o \qquad \text{(G.2.4-4)}$$

G.2.4.5 Pressure stress formula

In the shell at the edge of a loaded area or pad:

$$f_p = \frac{pr}{2e_{as}} \qquad \text{(G.2.4-5)}$$

where

e_{as} = analysis thicknesses of shell at edge of attachment or pad.

NOTE The thickness e_{as} includes local thickening of the shell but excludes the thickness of any reinforcing pad.

G.2.4.6 Shear stress formulae

Due to:

a) torsion (M_T)

$$\tau = \frac{M_T}{2\pi {r_o}^2 T_1} \tag{G.2.4-6}$$

b) shear force (F_S)

$$\tau = \frac{F_S}{\pi r_o T_1} \tag{G.2.4-7}$$

where

T_1 is the shell plus pad analysis thickness at the attachment o.d. or the shell analysis thickness at the pad o.d.

NOTE In general the shear forces may be neglected but where required the formulae shown may be used. The equations are from WRC 107 [30].

Figure G.2.4-8 **Spherical shell subjected to an external moment**

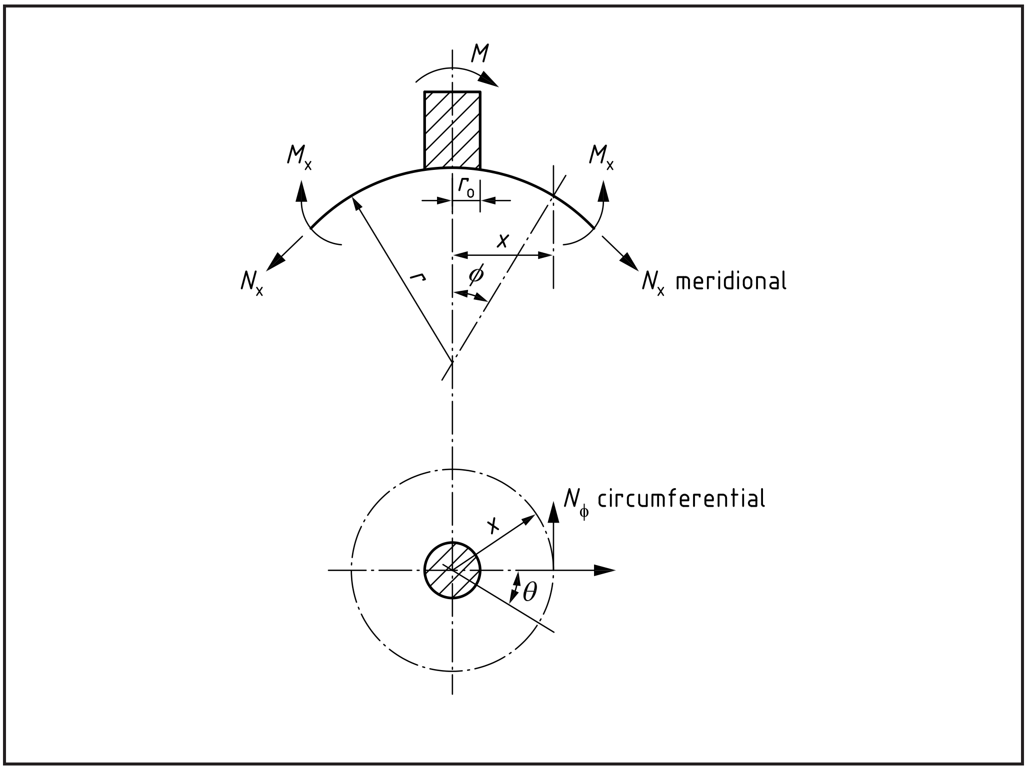


Figure G.2.4-9 **Deflections of a spherical shell subjected to an external moment *M***

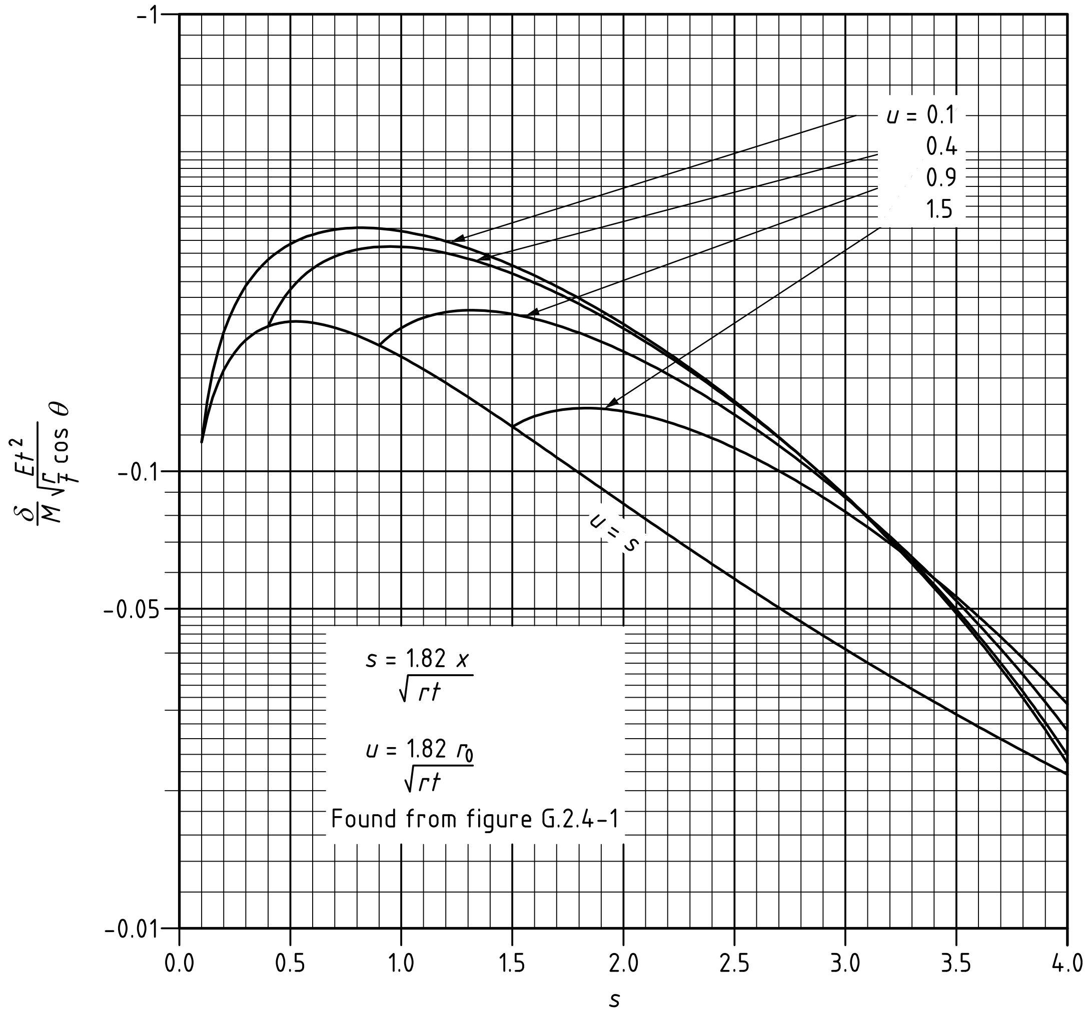


Figure G.2.4-10 **Meridional moment M_x in a spherical shell subjected to an external moment M**

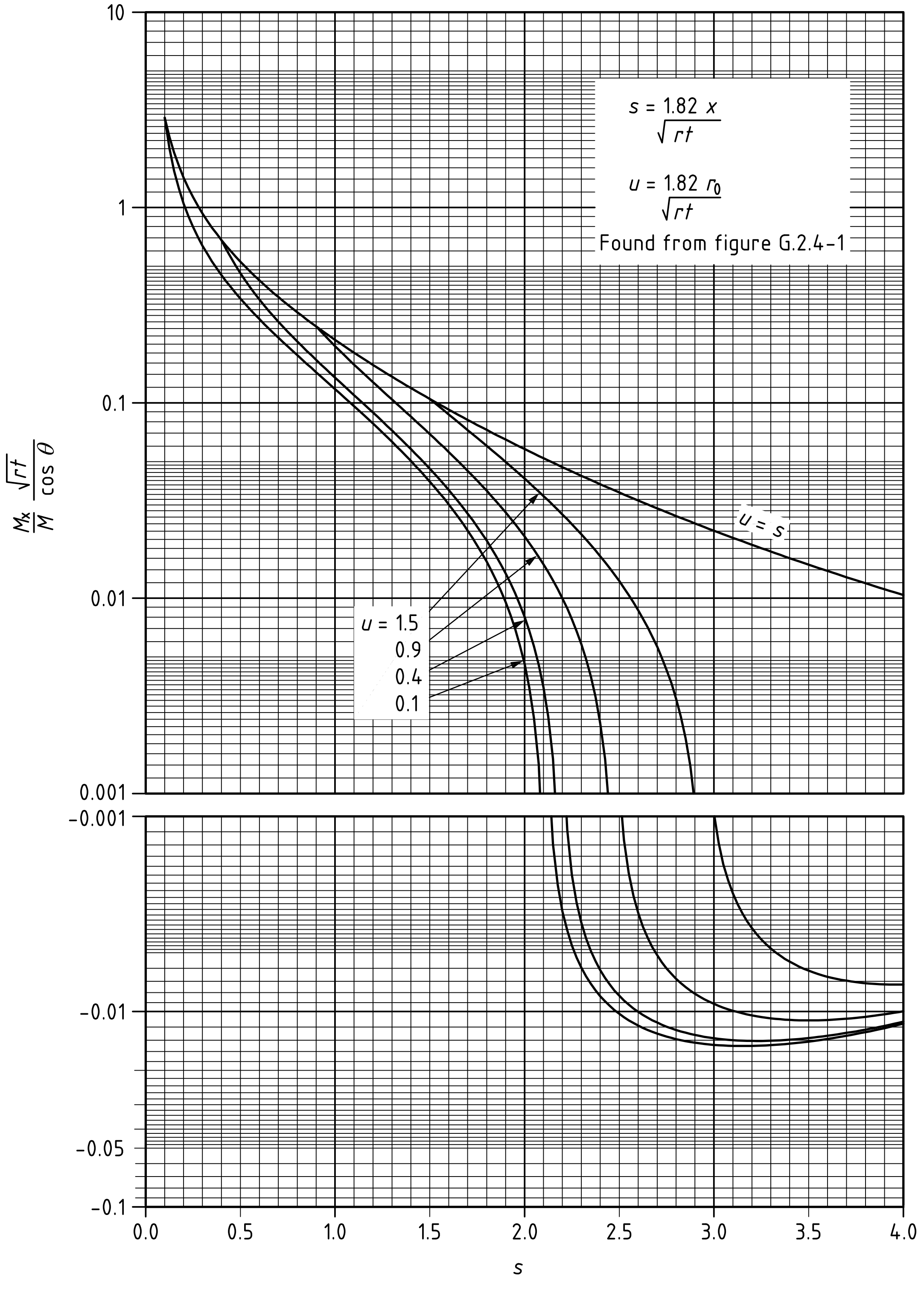


Figure G.2.4-11 **Circumferential moment M_ϕ in a spherical shell subjected to an external moment M**

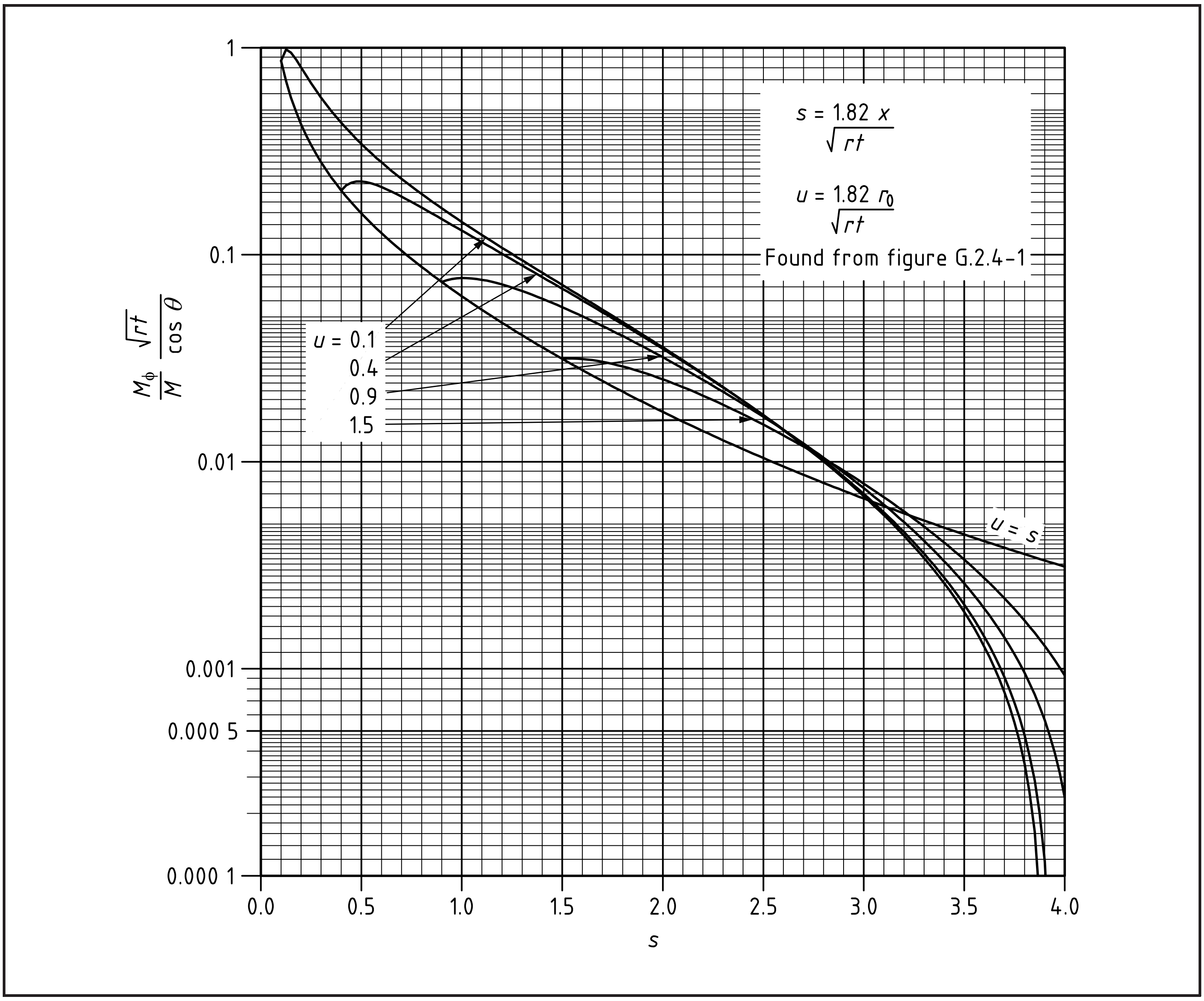


Figure G.2.4-12 **Meridional force N_x in a spherical shell subjected to an external moment M**

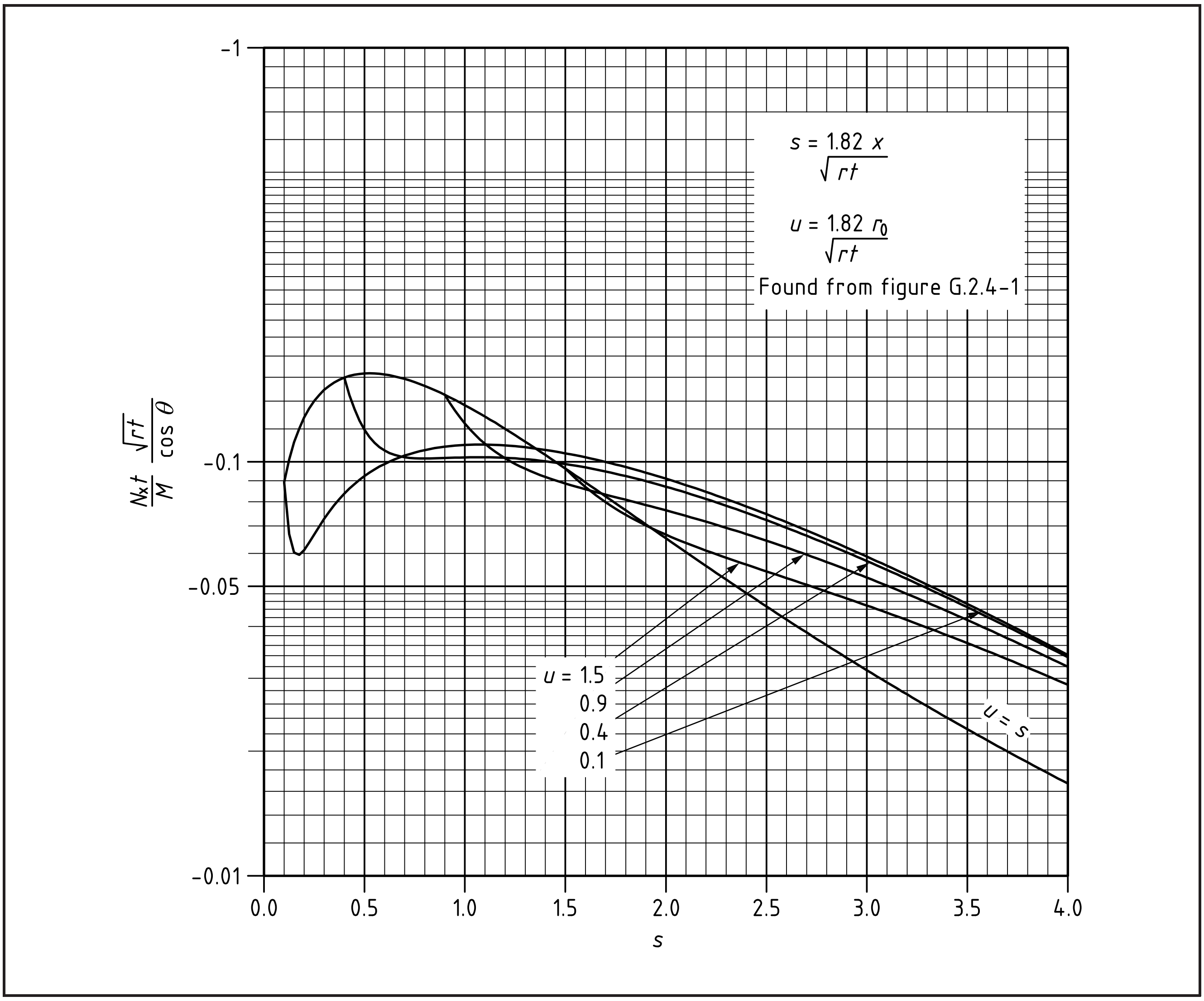
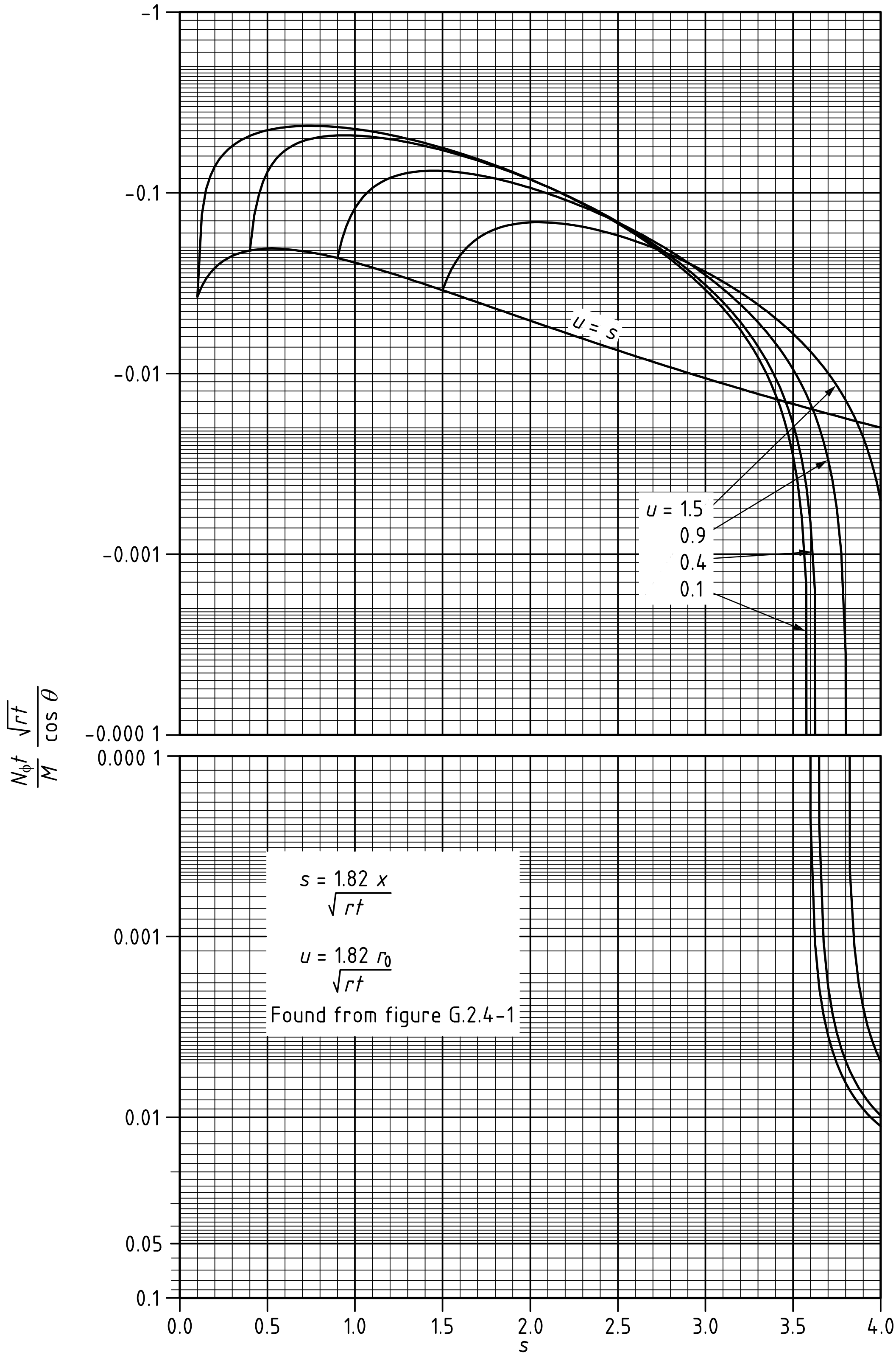


Figure G.2.4-13 **Circumferential force N_ϕ in a spherical shell subjected to an external moment M**



G.2.5 Local loads on spherical shell/nozzle attachments

G.2.5.1 General

G.2.5.1.1 Introduction

The method of calculating local stress levels at a nozzle junction is based on data given in [25]. Using this data it is possible to estimate the maximum stress which can occur at a sphere/nozzle attachment due to the application of internal pressure, thrust, external moment and shear force. The method covers both flush and protruding nozzles. In the original work the nozzle length is treated as semi-infinite without any restriction on its length. It is, however, considered necessary to stipulate a lower limit on the internal protrusion equal to $\sqrt{2rt}$. Nozzles with internal protrusion less than $\sqrt{2rt}$ should be treated as flush nozzles. In this way some additional conservatism will be introduced for those protruding nozzles where the internal projection does not satisfy this recommendation.

All the stress concentration factors given in Figure G.2.5-1 to Figure G.2.5-8 inclusive are based on the maximum principal stress theory.

The stress concentration factors given in **G.2.5.2** to **G.2.5.7** are based on data obtained for a sphere of constant thickness T', whereas in practice T' is looked upon as the local shell thickness adjacent to the nozzle, the main vessel being of a smaller thickness T. For these curves to be valid the thickness of the shell should not be reduced to T within a distance L_s as defined in **3.5.4.3.4**.

Work in progress shows that when the vessel thickness is reduced from T' to T at a distance L_s from the nozzle, higher stresses than those given in Figure G.2.5-1 to Figure G.2.5-8 inclusive may occur for small values of ρ and high values of t/T'. Further guidance cannot be given at the present stage.

This procedure provides a method of computing maximum stresses which occur in the shell rather than in the nozzle. In some instances calculated stresses may be higher in the nozzle wall than in the vessel shell, especially for very thin nozzles. These are not considered for the reasons stated in [31].

The stress intensities due to the combined pressure and external loads should be assessed in accordance with the limits given in **A.3.3**.

G.2.5.1.2 Notation

For the purposes of **G.2.5** and **G.2.6**, which are applicable to radial nozzles only, the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**):

K	is a factor;
M	is the external moment applied at nozzle (in N·mm);
M_T	is the torsional moment applied at nozzle (in N·mm);
P	is the internal pressure (in N/mm^2);
Q	is the radial thrust applied at nozzle (in N);
R	is the mean radius of spherical shell (in mm);
r	is the mean radius of nozzle (in mm);
S	is the shear load applied at nozzle (in N);
T'	is the local wall analysis thickness of shell, adjacent to nozzle (in mm) (see **1.6**);
t	is the wall analysis thickness of nozzle (in mm) (see **1.6**);
ρ	is the non-dimensional parameter $= \frac{r}{R}\sqrt{\frac{R}{T'}}$

σ_{max} is the maximum stress due to local loading (in N/mm^2) [see Equations (G.2.5-1) to (G.2.5-4)];

σ_θ is the circumferential stress (in N/mm^2);

σ_z is the meridional stress (longitudinal in a cylindrical shell) (in N/mm^2);

σ_y is the yield stress in simple tension (in N/mm^2);

$\overline{m}$ is the external moment shakedown factor [see Equation (G.2.6-3)];

$\overline{m}_o$ is the overall external moment shakedown factor (see **G.2.6.5**);

$\overline{p}$ is the internal pressure shakedown factor [see Equation (G.2.6-1)];

$\overline{p}_o$ is the overall internal pressure shakedown factor (see **G.2.6.5**);

$\overline{q}$ is the radial thrust shakedown factor [see Equation (G.2.6-2)];

$\overline{q}_o$ is the overall radial thrust shakedown factor (see **G.2.6.5**).

G.2.5.2 Maximum stress at a sphere/nozzle junction due to application of internal pressure

Figure G.2.5-1 gives plots of stress concentration factors (s.c.f.s) against the non-dimensional parameter ρ for various nozzle/shell wall t/T' ratios for flush nozzles. Figure G.2.5-2 gives similar plots for protruding nozzles. The maximum stress, σ_{max}, is then calculated by multiplying the s.c.f. thus obtained by the nominal pressure stress given by $\frac{PR}{2T'}$, i.e.:

$$\sigma_{max} = \text{s.c.f} \times \frac{PR}{2T'} \quad \text{(G.2.5-1)}$$

Before using Figure G.2.5-2 a check should be made to ensure that the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure G.2.5-1 should be used as for a flush nozzle for obtaining the s.c.f.

G.2.5.3 Maximum stress at a sphere/nozzle junction due to application of radial load or thrust

Figure G.2.5-3 gives plots of s.c.f. against the non-dimensional parameter ρ for flush nozzles. Figure G.2.5-4 gives similar plots for protruding nozzles. The maximum stress is calculated by multiplying the s.c.f. thus obtained by

$\frac{Q}{2\pi rT'}\sqrt{\frac{R}{T'}}$ i.e.:

$$\sigma_{max} = \text{s.c.f.} \times \frac{Q}{2\pi rT'}\sqrt{\frac{R}{T'}} \quad \text{(G.2.5-2)}$$

Before using Figure G.2.5-4 a check should be made to ensure that the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure G.2.5-3 should be used as for a flush nozzle for obtaining the s.c.f.

*NOTE It is not necessary to add the thrust due to the internal pressure acting over the cross-section area of the nozzle bore to the radial load acting on the nozzle. This pressure thrust load is taken into account in the calculation of the stresses due to internal pressure in **G.2.5.2**.*

G.2.5.4 Maximum stress at a sphere/nozzle junction due to application of external moment

For flush nozzles the maximum stress at a sphere/nozzle junction can be determined by using Figure G.2.5-5. Figure G.2.5-6 gives similar plots for protruding nozzles. The first step is to read off the s.c.f. for the appropriate vessel nozzle geometry. The maximum stress is then obtained by multiplying the s.c.f. thus obtained by the factor $\frac{M}{\pi r^2 T'}\sqrt{\frac{R}{T'}}$ i.e.:

$$\sigma_{\max} = \text{s.c.f.} \times \frac{M}{\pi r^2 T'}\sqrt{\frac{R}{T'}} \quad \text{(G.2.5-3)}$$

Before using Figure G.2.5-6 a check should be made to ensure that the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure G.2.5-5 should be used as for a flush nozzle for obtaining the s.c.f.

G.2.5.5 Maximum stress at a sphere/nozzle junction due to application of shear load and torsional moment

Figure G.2.5-7 should be used for determining the s.c.f. for flush nozzles. Figure G.2.5-8 gives similar plots for protruding nozzles. The maximum stress, due to the shear force S is then calculated by multiplying the s.c.f. thus obtained by the factor $S/(2\pi rT')$. The maximum stress, $\sigma_{\max}$, due to the shear force S and torsional moment M_T is given by:

$$\sigma_{\max} = \text{s.c.f} \times \frac{S}{2\pi rT'} + \frac{M_T}{2\pi r^2 T'} \quad \text{(G.2.5-4)}$$

Before using Figure G.2.5-8 a check should be made to ensure that the length of the internal nozzle protrusion is equal to or greater than $\sqrt{2rt}$; if it is not, Figure G.2.5-7 should be used as for a flush nozzle for obtaining the s.c.f.

G.2.5.6 Maximum stress at a sphere/nozzle junction under combined loading

For a conservative estimate of the stresses occurring under the action of combined loading the maximum stresses obtained from each of the individual loadings should be added together. This will always be conservative because the maximum stresses for individual loadings may occur at different locations and different directions (σ_θ and/or σ_z).

G.2.5.7 Stresses away from the loaded area

The method given in **G.2.5.1** and **G.2.5.6** for calculating local stresses at a sphere/nozzle junction caters for the maximum stress levels only. No information is given on stresses away from the loaded area.

Stress distributions in the vicinity of the sphere/nozzle junction are required in cases where other loaded areas are in proximity to the one under consideration. It is proposed to use the data already available in **G.2.4** to determine these stresses. The assumption here is that, although the magnitudes of local stresses may differ, the plot of stress level versus distance from the loaded area remains basically similar.

Thus **G.2.4** can be used to calculate the die away of stress, and the reduction factor, at the required distance from the loaded area for the application of radial loads or external moments. This reduction factor can then be applied to the maximum stress calculated in **G.2.5** to obtain the stresses away from the loaded area.

NOTE An alternative method may be used, see [23].

If the loaded nozzle area is less than 2.5 $\sqrt{Re}$ from another stress concentrating feature, stresses as calculated in accordance with Annex G become unreliable and some other method of assessing the total stress, for example finite element stress analysis or proof test, is required.

G.2.5.8 Nozzle strength

The longitudinal stresses in the nozzle should be verified as follows:

a) Longitudinal stress verification:

$$\frac{Pr}{2t} + \frac{M}{\pi r^2 t} + \frac{Q}{2\pi rt} \leq f_b \qquad \text{(G.2.5-5)}$$

where

f_b is as defined in **3.5.4.1**.

b) Longitudinal instability check (with P = 0):

$$M_{max} = \pi r^2 t\, \sigma_{z,allow} \qquad \text{(G.2.5-6)}$$

$$Q_{max} = 2\pi rt\, \sigma_{z,allow} \qquad \text{(G.2.5-7)}$$

$$\frac{M}{M_{max}} + \frac{|Q|}{Q_{max}} \leq 1.0 \qquad \text{(G.2.5-8)}$$

where

$\sigma_{z,\ allow}$ is the maximum allowable longitudinal compressive stress in the nozzle (see **A.3.5**).

If Q is positive it should be set to zero.

c) Reinforced nozzle wall:

For reinforcement provided as shown in Figures 3.5-15, 3.5-16 or 3.5-17, a longitudinal stress verification and longitudinal instability check should also be performed using values of r and t at the thinnest part of the nozzle.

G.2.6 Spherical shells: shakedown loads for radial nozzles

G.2.6.1 General

G.2.6.1.1 Introduction

All the shakedown loads given in **G.2.6.2**, **G.2.6.3**, **G.2.6.4** and **G.2.6.5** are based on the maximum shear stress criteria.

For vessels subjected to cyclic loading, be it pressure, radial load, external moment or any combination of these, it is essential to have a knowledge of the shakedown limit in order to prevent plastic cycling or incremental collapse. By keeping the cyclic loadings within the shakedown limits it ensures that, after initial plastic deformation, further deformation will be in the elastic range, i.e. the vessel has “shaken down” to purely elastic behaviour. The method given does not necessarily imply a limited plastic deformation before shakedown is achieved.

The shakedown conditions can occur after different numbers of cycles depending on the cyclic conditions and stress level; in certain cases, the plastic deformation before shakedown might be significant.

The method of predicting shakedown factors for internal pressure, radial nozzle thrust and external moment at a vessel/nozzle junction in **G.2.6.2**, **G.2.6.3, G.2.6.4** and **G.2.6.5** is based on data given in [27]. From the data shakedown factors for flush and protruding nozzles can be estimated for each of the aforementioned individual loading conditions.

Where the various loading conditions occur simultaneously a simple formula is given that considers the interaction between any of these loading conditions (see [27]).

No clear distinction between a flush and a protruding nozzle is given. It is considered necessary to stipulate a lower limit on the length of the nozzle internal protrusion equal to $\sqrt{2rt}$. Nozzles with internal protrusion less than $\sqrt{2rt}$ should be treated as flush nozzles. By doing so, some additional conservatism will be introduced for those protruding nozzles where the internal projection does not satisfy these recommendations.

The shakedown factors given in **G.2.6.2**, **G.2.6.3**, **G.2.6.4** and **G.2.6.5** are based on data obtained for a sphere of constant thickness *T* ′, whereas in practice *T* ′ is looked upon as the local shell thickness adjacent to the nozzle, the main vessel being of smaller thickness *T*. For these curves to be valid the thickness of the shell should not be reduced to *T* ′ within a distance *H* as defined in **3.5.4.3.4**.

Figure G.2.5-1 **Maximum stress in sphere for internal pressure (flush nozzles)**

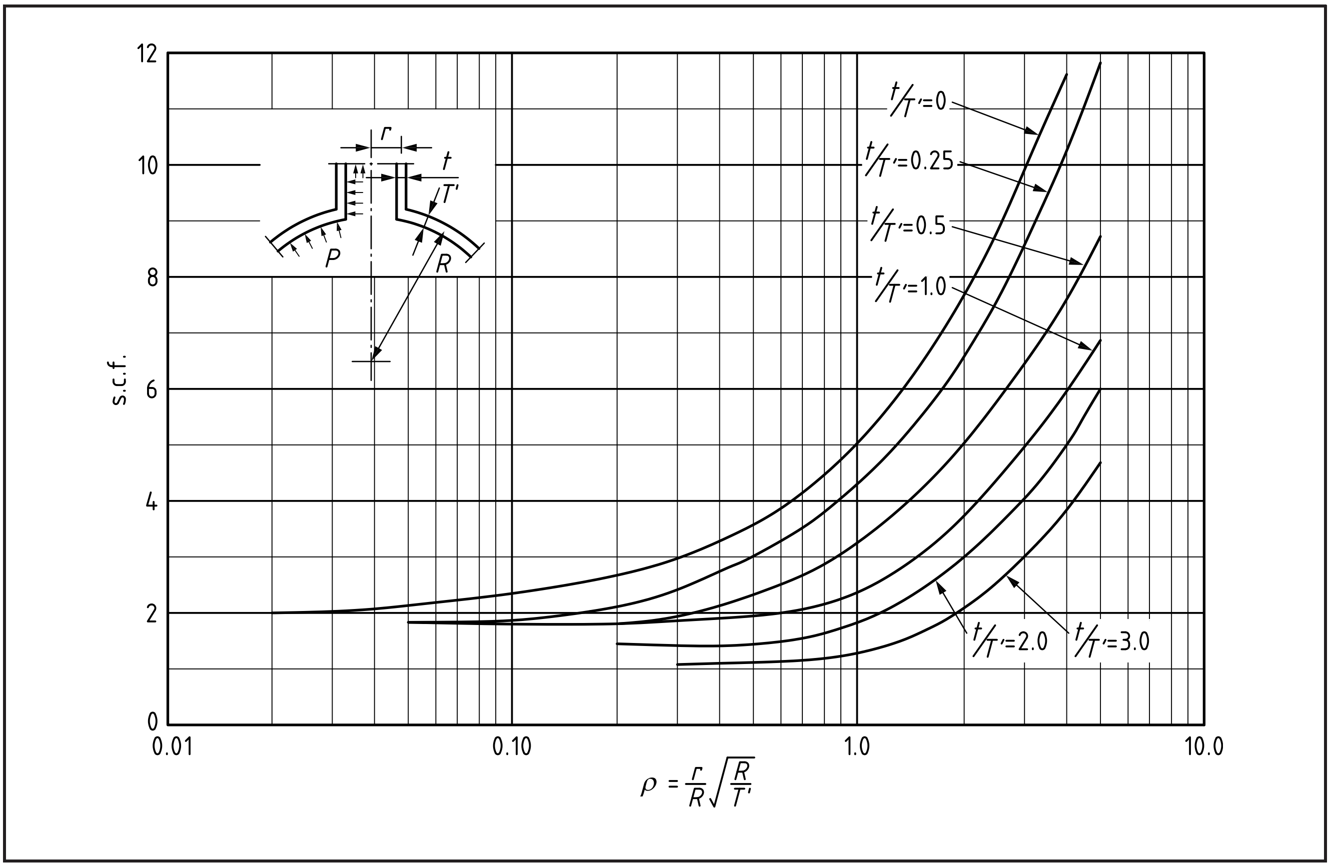


Figure G.2.5-2 **Maximum stress in sphere for internal pressure (protruding nozzles)**

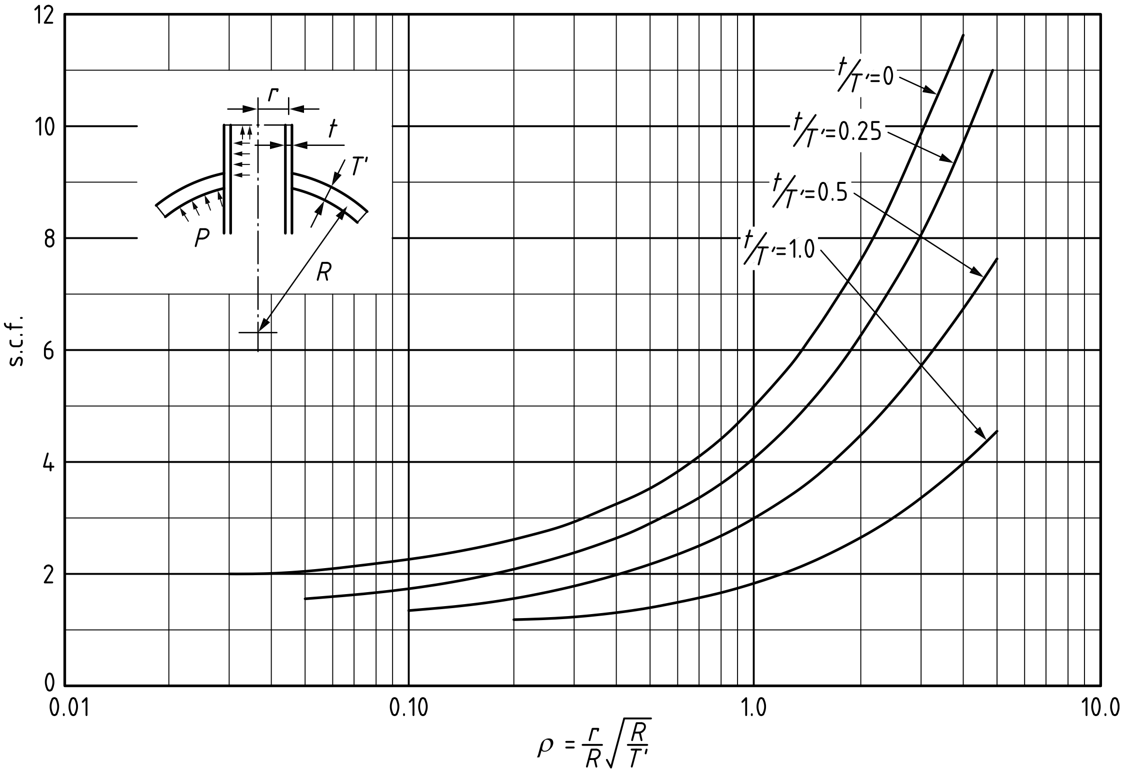


Figure G.2.5-3 **Maximum stress in sphere for thrust loading (flush nozzles)**

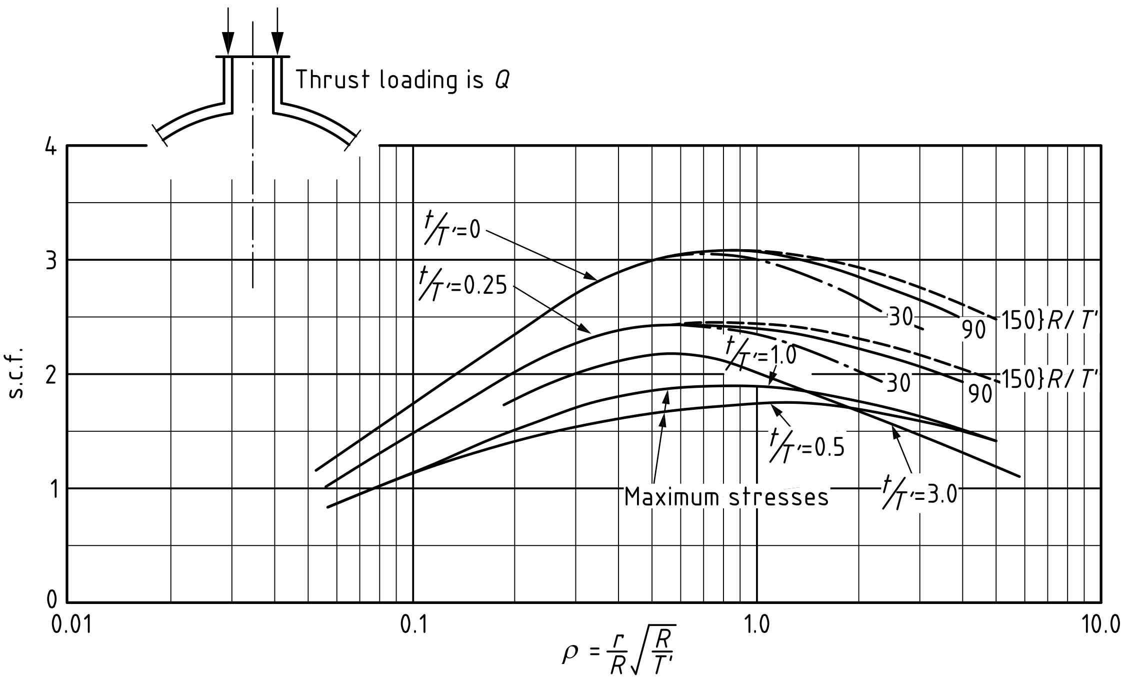


Figure G.2.5-4 **Maximum stress in sphere for thrust loading (protruding nozzles)**

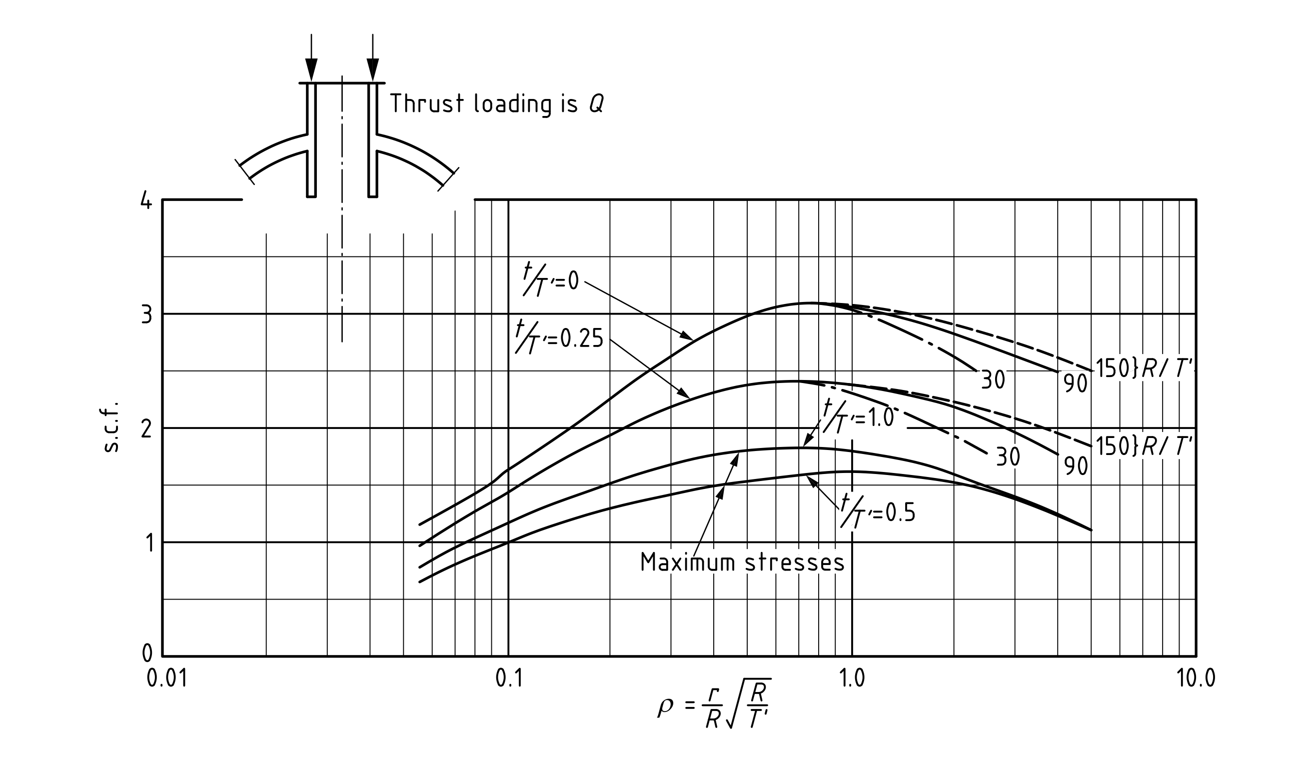


Figure G.2.5-5 **Maximum stress in sphere for moment loading (flush nozzles)**

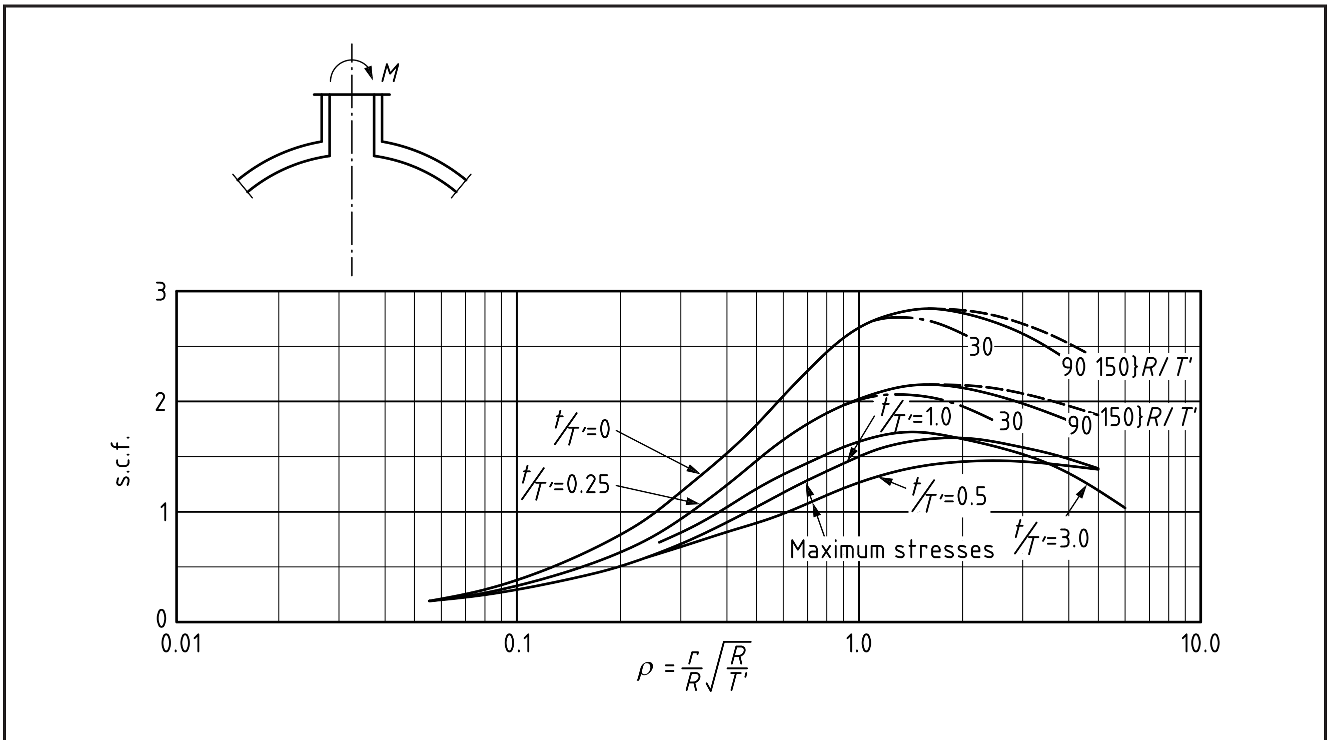


Figure G.2.5-6 **Maximum stress in sphere for moment loading (protruding nozzles)**

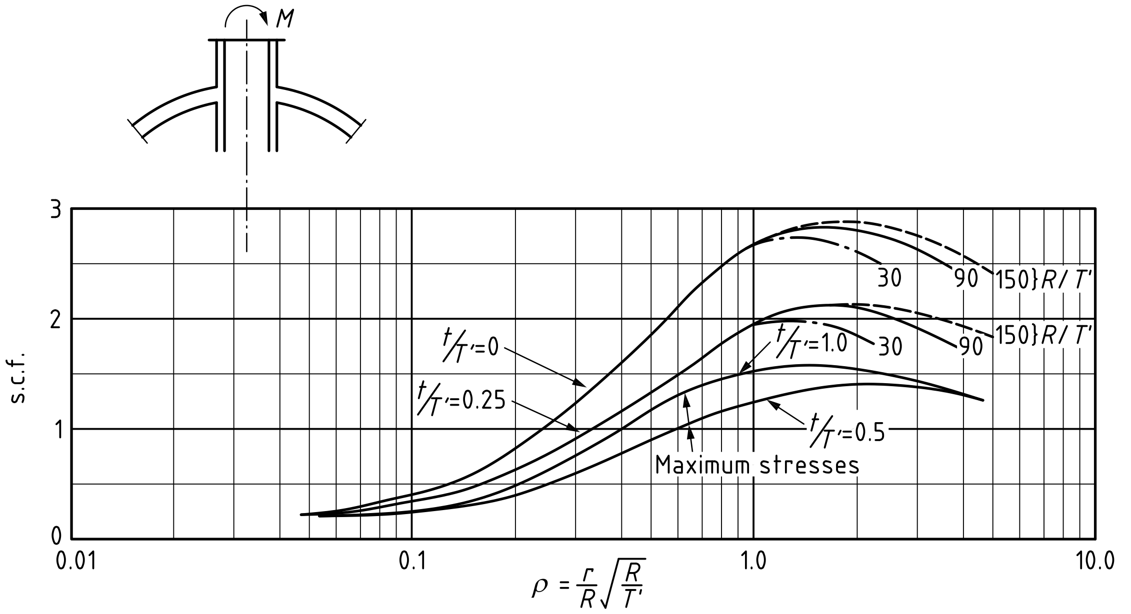


Figure G.2.5-7 **Maximum stress in sphere for shear loading (flush nozzles)**

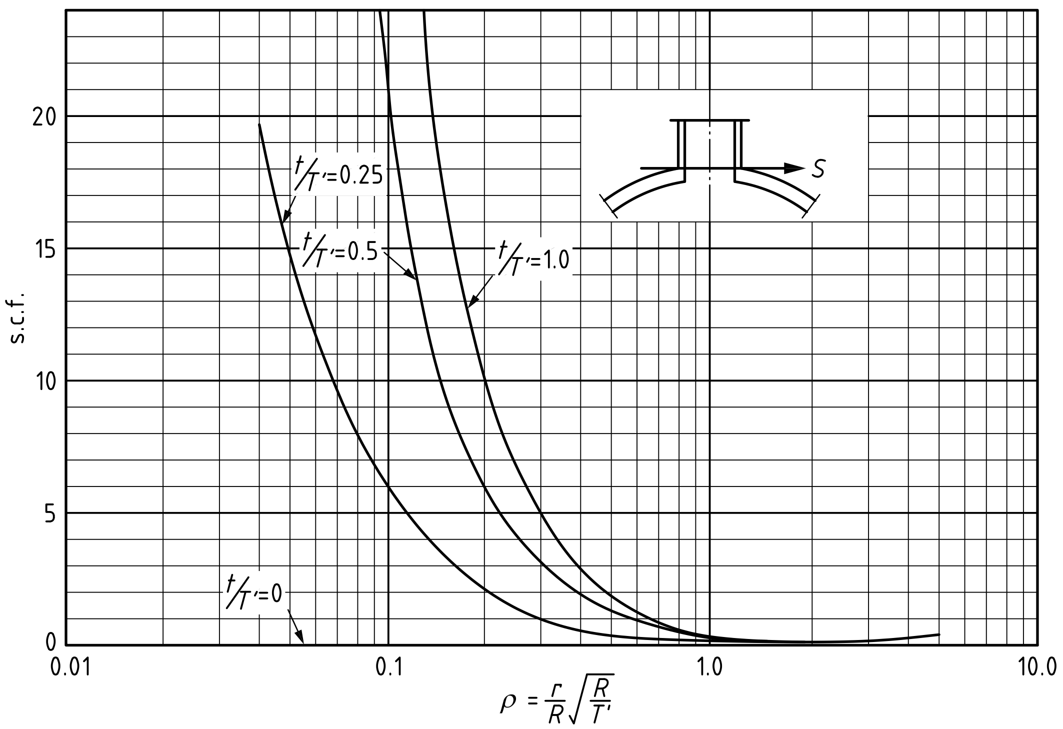
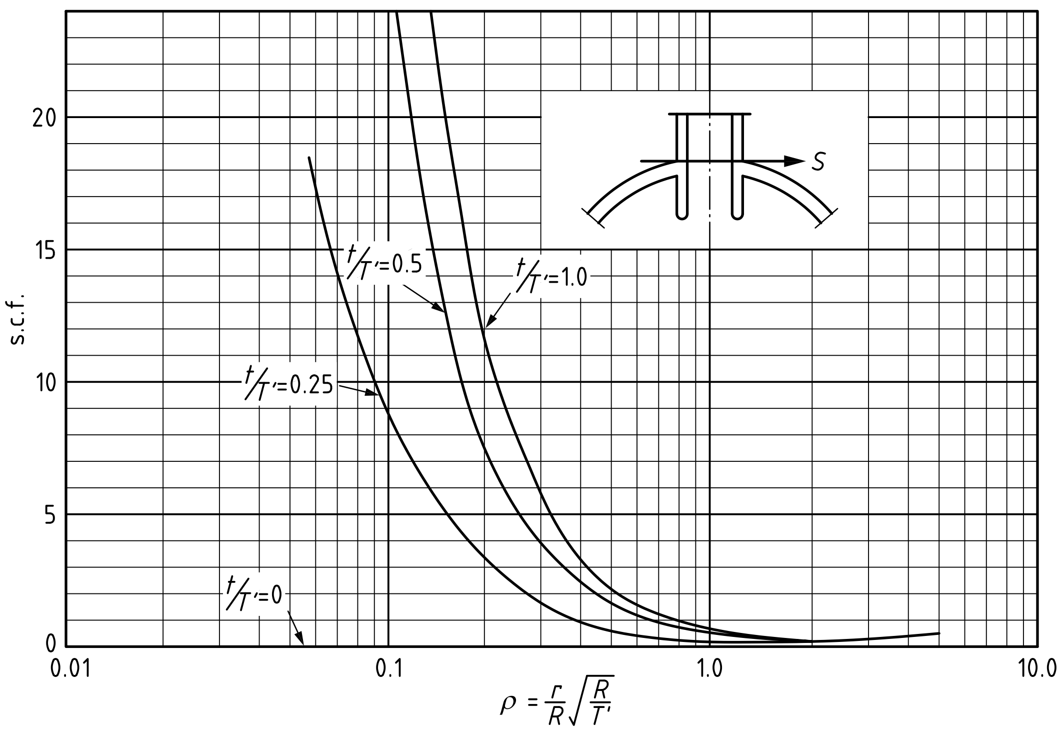


Figure G.2.5-8 **Maximum stress in sphere for shear loading (protruding nozzles)**



G.2.6.1.2 Notation

For the purposes of **G.2.6** the symbols are as defined in **G.2.5**.

G.2.6.2 Shakedown factor for internal pressure loading

Figure G.2.6-1 and Figure G.2.6-2 should be used for determining the shakedown factors under internal pressure conditions for flush and protruding nozzles respectively.

The pressure shakedown factor can be defined as the ratio of the nominal pressure stress in the spherical shell to the value of yield stress in simple tension, i.e.:

$$\overline{p} = \frac{PR}{2T'\sigma_y} \tag{G.2.6-1}$$

G.2.6.3 Shakedown factor for radial thrust at a nozzle

The relevant shakedown factors for flush and protruding nozzles subjected to radial loads (radial with respect to the vessel) should be determined from Figure G.2.6-3, Figure G.2.6-5 and Figure G.2.6-7 and from Figure G.2.6-4, Figure G.2.6-6 and Figure G.2.6-8 respectively.

The radial thrust shakedown factor can be defined as:

$$\overline{q} = \frac{1}{2\pi r}\frac{Q}{T'\sigma_y}\sqrt{\frac{R}{T'}} \tag{G.2.6-2}$$

*NOTE It is not necessary to add the thrust due to the internal pressure acting over the cross-section area of the nozzle bore to the radial load acting on the nozzle. This pressure thrust load is taken into account in the calculation of the stresses due to internal pressure in **G.2.6.2**.*

G.2.6.4 Shakedown factor for external moment

Figure G.2.6-3, Figure G.2.6-5, and Figure G.2.6-7 should be used for calculating the moment shakedown factor *m* for flush nozzles. For protruding nozzles the corresponding plots for the shakedown factor are given in Figure G.2.6-4, Figure G.2.6-6 and Figure G.2.6-8.

The moment shakedown factor can be defined as:

$$\overline{m} = \frac{M}{\pi r^2 T' \sigma_y}\sqrt{\frac{R}{T'}} \tag{G.2.6-3}$$

Before using the relevant figures for the protruding nozzles, a check should be carried out on the nozzle inner projection. If this is less than $\sqrt{2rt}$ then the corresponding plots for flush nozzles should be used in determining the necessary shakedown factor.

G.2.6.5 Interaction between shakedown factors under combined loading conditions

For the case of the combined loading condition, [27] gives the following equation so that the overall shakedown condition is obtained:

$$\frac{\overline{p}}{\overline{p}_o} + \frac{\overline{q}}{\overline{q}_o} + \frac{\overline{m}}{\overline{m}_o} \leq 1 \tag{G.2.6-4}$$

In this equation the values of $\overline{p}_o$, $\overline{q}_o$ and $\overline{m}_o$ are read off from Figure G.2.6-1 to Figure G.2.6-8 inclusive for the appropriate vessel/nozzle geometry, while $\overline{p}$, $\overline{q}$ and $\overline{m}$ are as derived from the relevant Equations (G.2.6-1), (G.2.6-2) and (G.2.6-3).

Where the conditions are such that the relationship given by Equation (G.2.6-4) is not satisfied then a revised nozzle/shell geometry (increased vessel shell or branch wall thickness) should be used and the procedure repeated until Equation (G.2.6-4) is fulfilled.

Figure G.2.6-1 **Shakedown values for pressure loading (flush nozzle)**

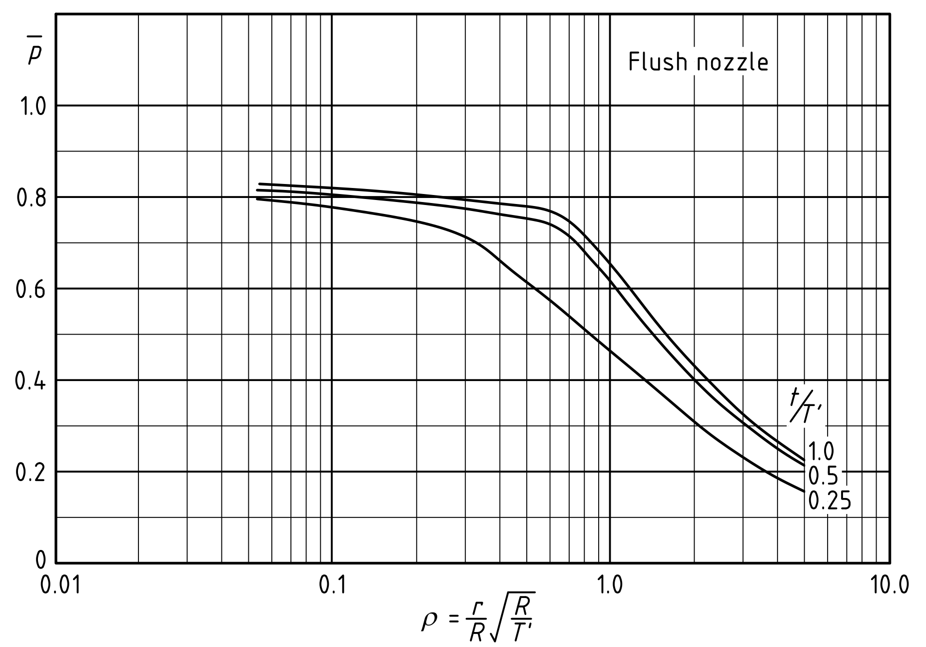


Figure G.2.6-2 **Shakedown values for pressure loading (protruding nozzle)**

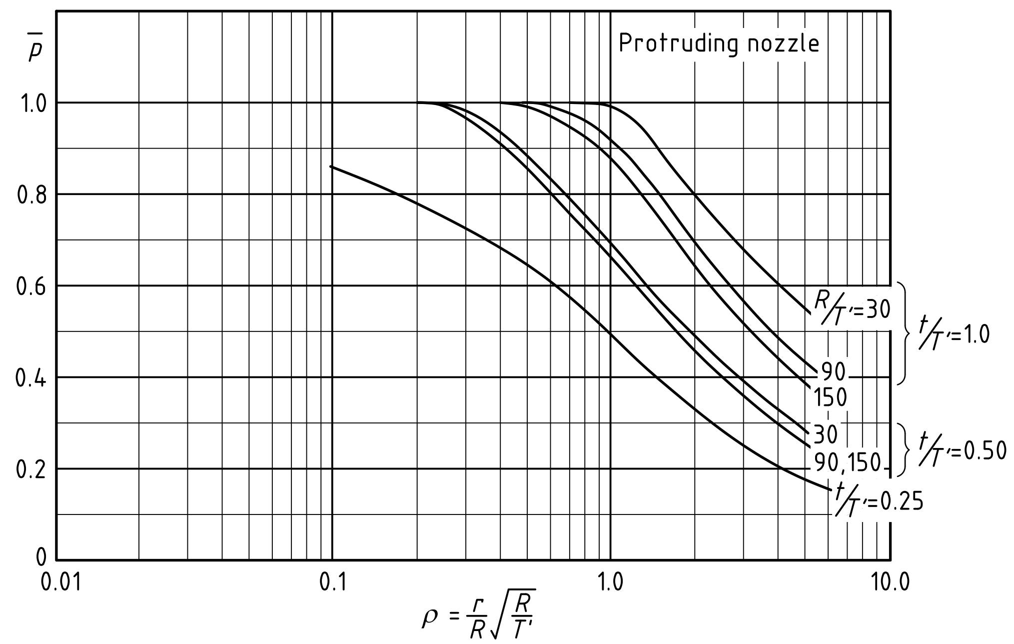


Figure G.2.6-3 **Shakedown values for thrust and moment loadings (flush nozzle)**

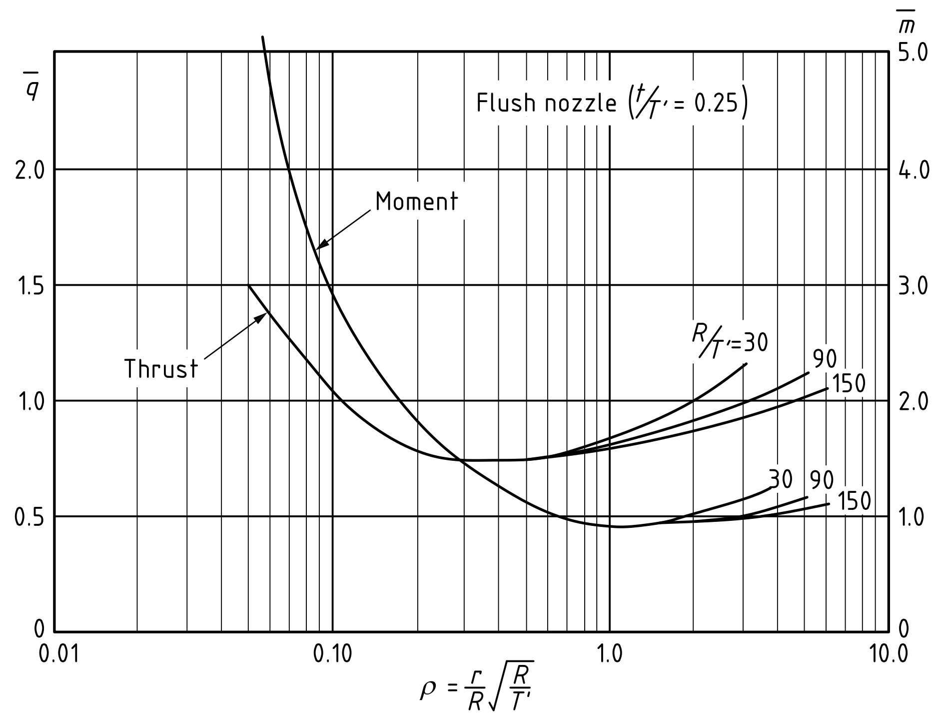


Figure G.2.6-4 **Shakedown values for thrust and moment loadings (protruding nozzle)**

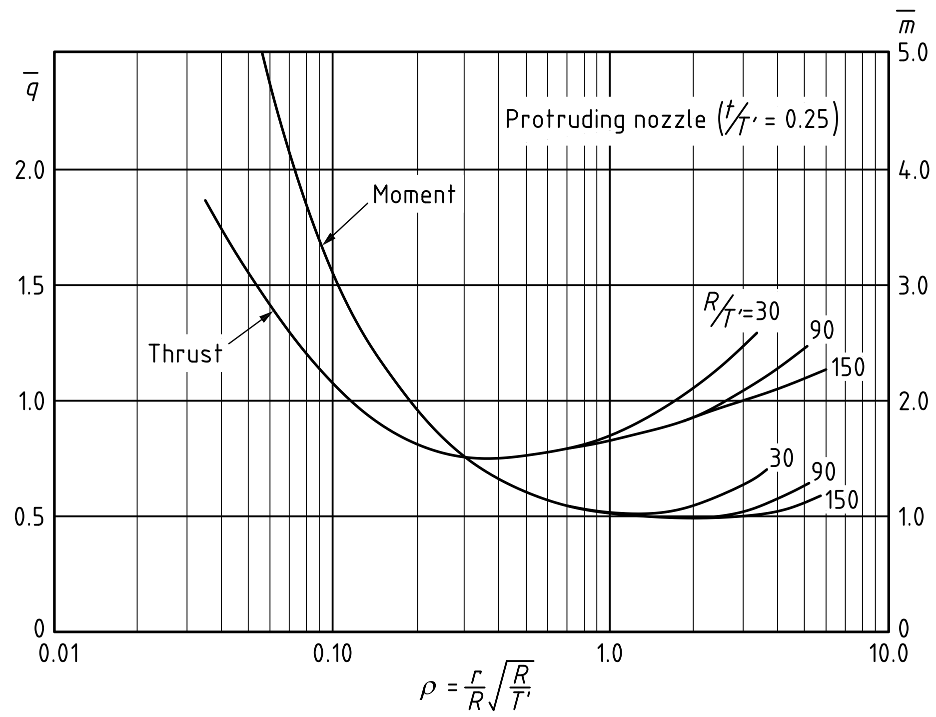


Figure G.2.6-5 **Shakedown values for thrust and moment loadings (flush nozzle)**

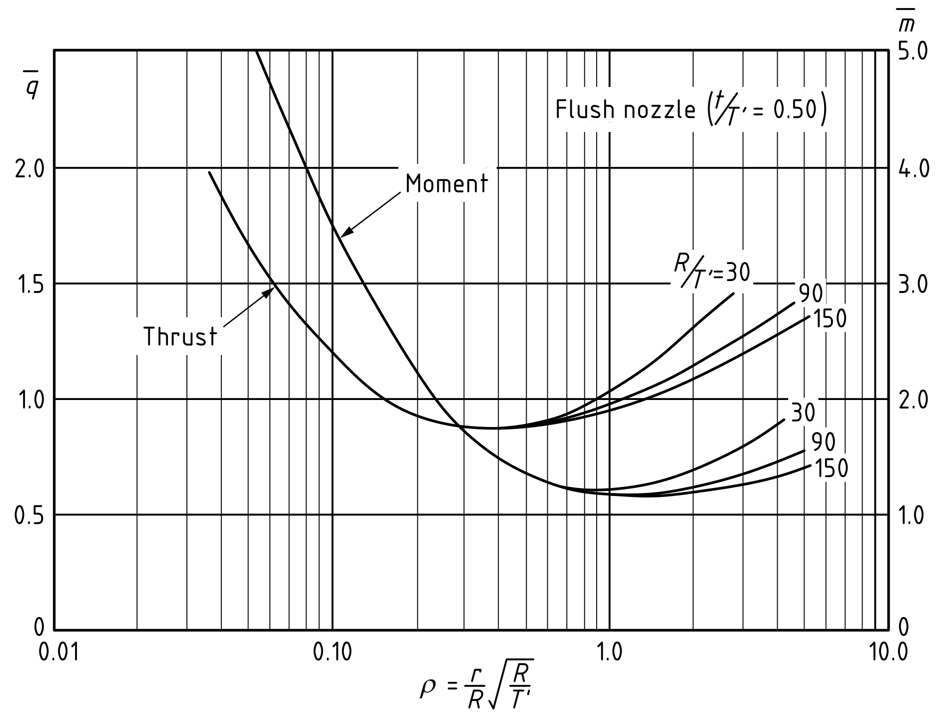


Figure G.2.6-6 **Shakedown values for thrust and moment loadings (protruding nozzle)**

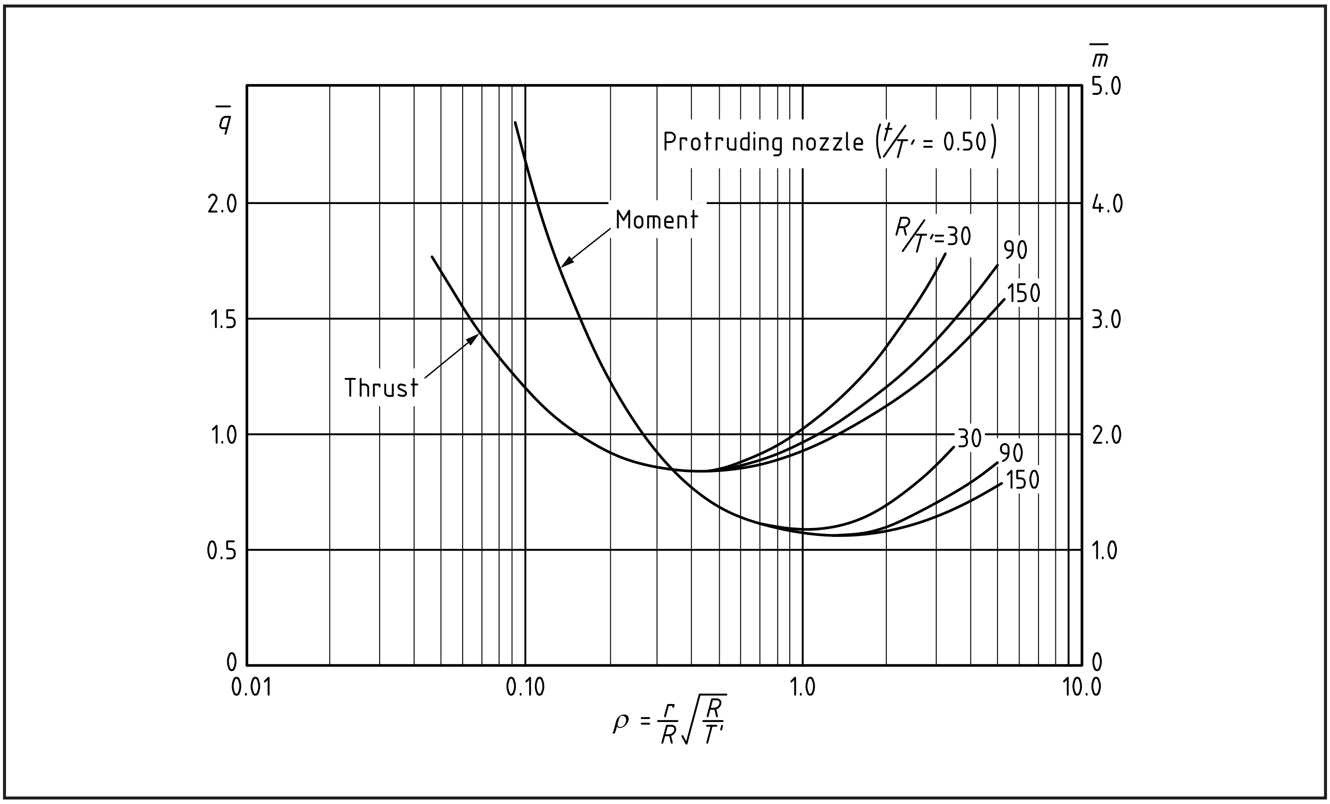


Figure G.2.6-7 **Shakedown values for thrust and moment loadings (flush nozzle)**

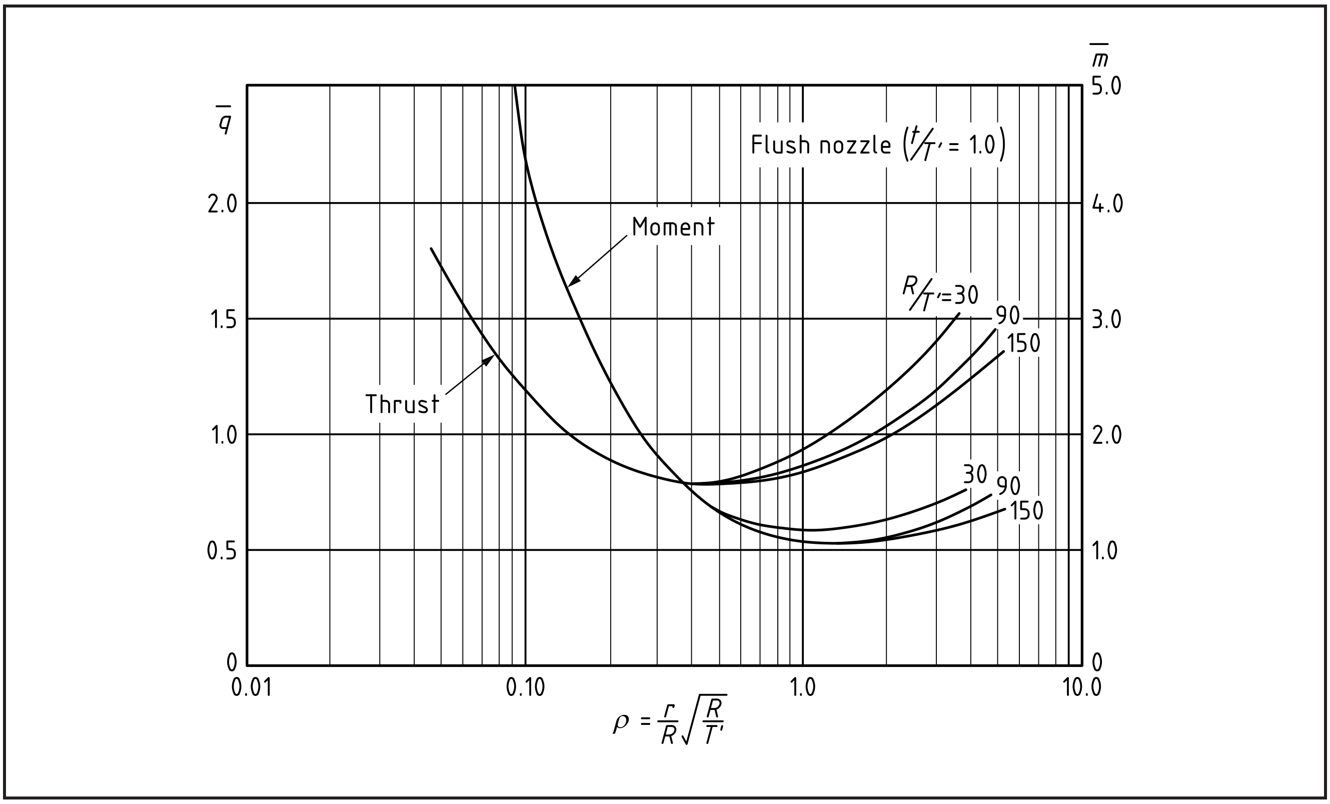
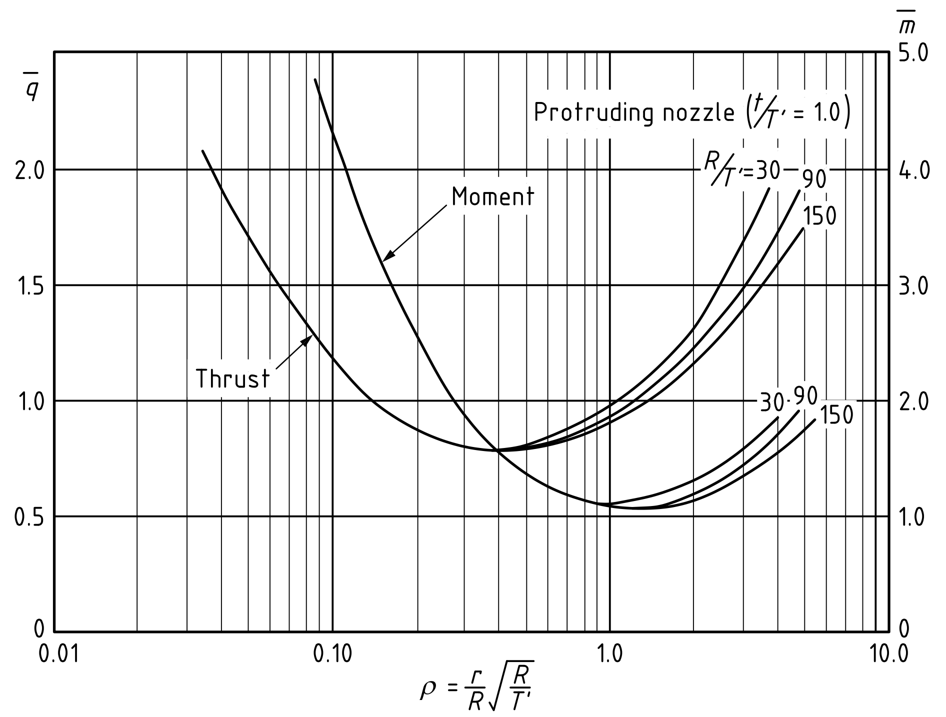


Figure G.2.6-8 **Shakedown values for thrust and moment loadings (protruding nozzle)**



G.2.7 The effect of external forces and moments at branches

Large external forces and moments can be applied to the branches of vessels by the thermal movements of pipework.

The stresses due to these are likely to be greatly overestimated if the forces in the pipe system are determined by assuming that the connection to the vessel is equivalent to an anchor in the pipe system.

More accurate values of the terminal forces and moments can be found if the deflection due to a unit radial load and the slopes due to unit longitudinal and circumferential moments distributed over the area of the branch and its reinforcement are known.

These can be found for a given cylindrical vessel and branch by the methods given in **G.2.2.4** and **G.2.3**. Experiments in the USA, discussed in [17], have shown that slopes and deflections calculated in this way are sufficiently accurate for practical purposes except that the slope of a branch on a cylindrical vessel due to a circumferential moment is about 75% of the calculated value because of the effect of local stiffening by the metal of the branch.

When the loads from the pipework are known, the local stresses in the cylindrical shell can be found by the methods given in **G.2**, except that, in a branch with an external compensating ring of thickness t_2 subject to a *circumferential* moment there is an additional circumferential moment in the shell at the edge of the reinforcing ring to $N_\phi t_2/4$ and [17] recommends that this amount should be added to the value of M_ϕ calculated in **G.2.3**.

These corrections apply only to circumferential moments and are due to the effect of the rigidity of the attachment of the branch which has little influence on the effect of longitudinal moments.

The tension at the inside of the shell due to the local circumferential bending moment M_ϕ is added to the circumferential membrane stress due to internal pressure, but this stress will not be present when the vessel is under hydraulic test.

Deflections and stresses at branches in spherical vessels can be found by the methods given in **G.2.4.3** and **G.2.4.4**.

Where nozzle branches with reinforcing plates are attached with full penetration welds, i.e. in accordance with Figure E.24, Figure E.25a) and Figure E.28a), they may be assumed (for the purpose of local stresses evaluation) to be integral with the shell and the stresses evaluated in accordance with **G.2**. Where nozzle branches with reinforcing plates are attached, with partial penetration welds, i.e. in accordance with Figure E.25b), Figure E.26, Figure E.27 and Figure E.28b), they may be analysed in accordance with **G.3.1.5**.

G.2.8 Alternative rules for local loads on circular nozzles in spherical and cylindrical shells

The following rules provide alternative methods for checking local loads on circular nozzles in spherical and cylindrical shells. They provide an alternative to the calculation of stresses given in **G.2.2**, **G.2.3**, **G.2.4**, **G.2.5** and **G.2.6**. These rules are also applicable to circular attachments that are welded to shells without an opening, e.g. lifting trunnions.

*NOTE It is not necessary to add the thrust due to the internal pressure acting over the cross-section area of the nozzle bore to the axial force acting on the nozzle. This pressure thrust is taken into account in the calculation of the maximum allowable pressure in **G.2.8.2.3** c) and **G.2.8.3.3** b), and the stresses due to internal pressure in **G.2.8.2.5** a) and **G.2.8.3.5** a).*

G.2.8.1 General

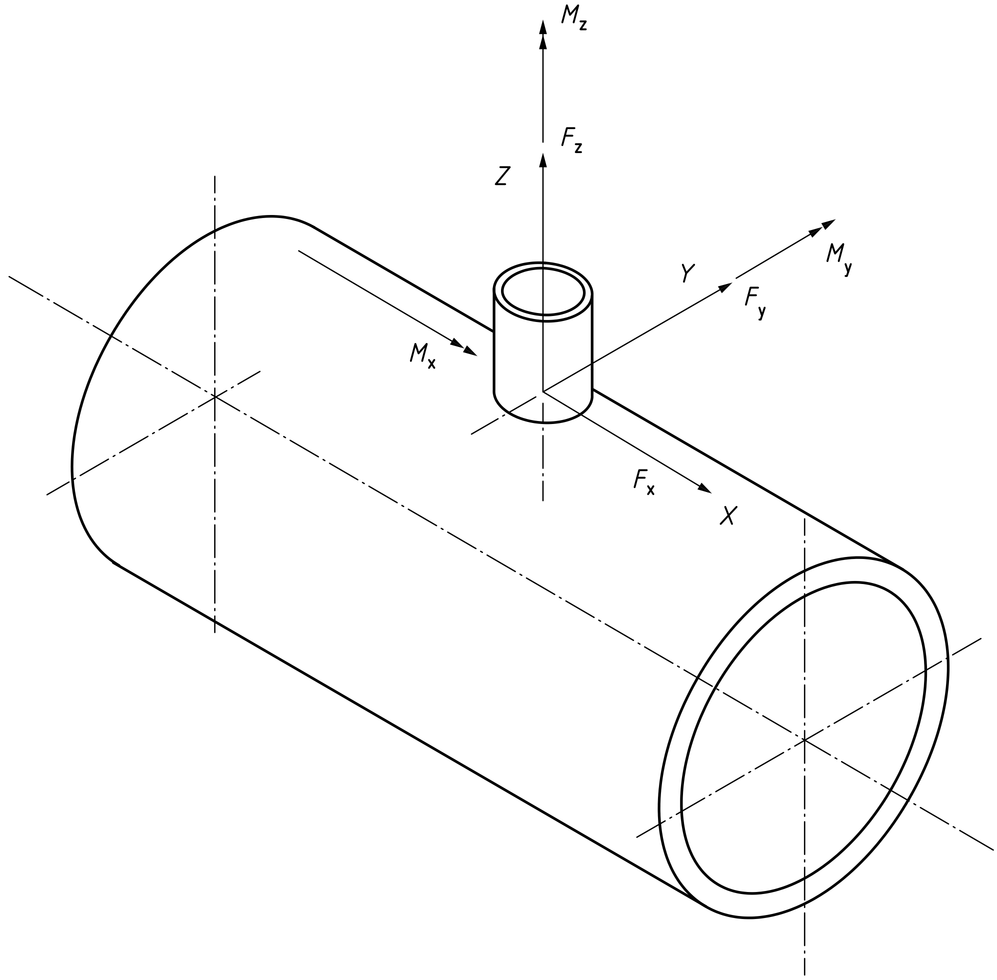
Geometrical limitations are given separately for spherical and cylindrical shells in **G.2.8.2.1** and **G.2.8.3.1** respectively.

For the purposes of **G.2.8** the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**).

a_0 to a_4	are polynomial coefficients;
C_1	is a nozzle factor;
C_2 to C_4	are calculation factors;
d	is a diameter used in the analysis;
d_i	is the inside nozzle diameter;
d_2	is the outside diameter of the reinforcing ring;
D	is the mean shell diameter at the nozzle;
e_{ab}	is the nozzle analysis thickness;
e_{ac}	is the shell calculation thickness;
e_{as}	is the shell analysis thickness;
e_{ap}	is the reinforcing pad analysis thickness;
e_{eq}	is the equivalent shell thickness;
f_n	is the allowable design stress of nozzle material;
f_s	is the allowable design stress of the shell material;
f_2	is the allowable design stress of a reinforcing plate;
F_S	is the shear force on the spherical shell nozzle (at the shell junction);
F_X	is the longitudinal shear force on the cylindrical shell nozzle (at the shell junction) (see Figure G.2.8-1);
F_Y	is the circumferential shear force on the cylindrical shell nozzle (at the shell junction) (see Figure G.2.8-1);

F_Z is the axial force on the nozzle (tensile force >0) (see Figure G.2.8-1);
$F_{Z,max}$ is the maximum allowable axial force on the nozzle;
L_p is the width of the reinforcing plate;
M_B is the bending moment on the spherical shell nozzle (at the shell junction);
$M_{B,max}$ is the maximum allowable bending moment on the spherical shell nozzle (at the shell junction);
M_X is the circumferential moment on the cylindrical shell nozzle (at the shell junction) (see Figure G.2.8-1);
$M_{X,max}$ is the maximum allowable circumferential moment on the cylindrical shell nozzle (at the shell junction);
M_Y is the longitudinal moment on the cylindrical shell nozzle (at the shell junction) (see Figure G.2.8-1);
$M_{Y,max}$ is the maximum allowable longitudinal moment on the cylindrical shell nozzle (at the shell junction);
M_Z is the torsional moment on the nozzle (at the shell junction) (see Figure **G.2.8-1**);
p is the design pressure;
p_{max} is the maximum allowable pressure;
R is the mean shell radius at the nozzle;
λ_c is a geometric parameter for cylindrical shells;
λ_s is a geometric parameter for spherical shells;
Φ is a load ratio;
σ_P is the stress range due to pressure;
σ_{FZ} is the stress range due to axial nozzle loads;
σ_{MB} is the stress range due to moments on the spherical shell nozzle;
σ_{MX} is the stress range due to circumferential moments on the cylinder nozzle;
σ_{MY} is the stress range due to longitudinal moments on the cylinder nozzle;
σ_T is the thermal stress due to temperature differences through the wall thickness;
κ is a reinforcement factor;
τ_F is the shear stress in the spherical shell due to the shear force;
τ_X is the shear stress in the cylindrical shell due to the longitudinal shear force;
τ_Y is the shear stress in the cylindrical shell due to the circumferential shear force;
τ_Z is the shear stress in the shell due to the torsional moment;
τ is the total shear stress range in the shell.

Figure G.2.8-1 **Moment and force vectors**



G.2.8.2 External loads on nozzles in spherical shells

G.2.8.2.1 Limitations

These rules for external loads on nozzles in spherical shells apply within the following limitations.

a) $0.001 \leq e_{as}/R \leq 0.1$.

NOTE Values of $e_{as}/R < 0.001$ are acceptable provided that the shell wall deflection does not exceed half the wall thickness.

b) Distances to any other local load in all directions should be not less than $\sqrt{Re_{ac}}$.

c) The nozzle thickness should be maintained over a distance of not less than $\sqrt{de_{ab}}$.

G.2.8.2.2 Design procedure

Loads and stresses are calculated in accordance with the following procedure given in steps a) to f), at the nozzle outer diameter. If a reinforcing plate is fitted and the width $L_p < \sqrt{R(e_{as} + e_{ap})}$ the maximum allowable loads are also calculated at the reinforcing plate outer diameter, in accordance with steps a) to c).

a) Select shell and nozzle (and reinforcing plate) thicknesses, calculate e_{ac} and d.

   1) For a nozzle with integral reinforcement, $e_{ac} = e_{as}$ and d is the mean nozzle diameter.

   2) For a nozzle with a reinforcing plate:

      i) at the nozzle outer diameter, $e_{ac} = e_{as} + e_{ap} \times \min(f_2/f_s;1.0)$ and d is the mean nozzle diameter;

      ii) at the reinforcing plate outer diameter, $e_{ac} = e_{as}$ and $d = d_2$.

b) Calculate the maximum allowable individual loads (see **G.2.8.2.3**).

c) Verify the load ratios and the interaction of the loads (see **G.2.8.2.4**).

d) At the nozzle outer diameter only; define the equivalent shell thickness e_{eq} and verify the allowable load range (see **G.2.8.2.5**).

e) Verify the nozzle strength (see **G.2.8.2.6**).

f) If any of the criteria in b) to e) above are not satisfied, repeat the procedure from a) with increased thicknesses.

G.2.8.2.3 Maximum allowable individual loads

To determine the maximum allowable values of pressure, axial load and bending moment, which may be independently applied to a nozzle, the following procedure should be applied.

a) Determine the reinforcement factor κ.

$$\kappa = \min\left(\frac{2f_n e_{ab}}{f_s e_{ac}}\sqrt{\frac{e_{ab}}{d}}; 1.0\right) \quad \text{(G.2.8-1)}$$

For the calculation of the allowable loads at the edge of the reinforcing plate, or for a circular attachment on a shell without an opening, the reinforcement factor κ is equal to 1.0.

b) Determine the geometric parameter λ_s.

$$\lambda_s = \frac{d}{\sqrt{Re_{ac}}} \quad \text{(G.2.8-2)}$$

c) At the nozzle outer diameter the permissible pressure p_{max} is derived using the rules of **3.5.4.3** taking C = 1.1 or is given by the following expression, taken from **3.5.4.9**. See **3.5.4.9.1** for definitions of the various terms.

$$p_{max}\left[A_p + 0.5\left(A_{fs} + A_{fb} + A_{fp}\right)\right] = f_s A_{fs} + f_p A_{fp} + f_b A_{fb} \quad \text{(G.2.8-3)}$$

At the reinforcing plate outer diameter the permissible pressure $p_{max} = 2f_s e_{as}/R$.

d) Allowable axial nozzle load $F_{Z,max}$ is given by the equation:

$$F_{Z,max} = f_s e_{ac}^2 \left\{ 1.82 + 2.4\left(\sqrt{1+\kappa}\right)\lambda_s + 0.91\kappa\lambda_s^2 \right\} \quad \text{(G.2.8-4)}$$

Non-dimensional upper and lower bounds are given in Figure G.2.8-2.

e) Allowable bending moment $M_{B,max}$ is given by the equation:

$$M_{B,max} = f_s e_{ac}^2 \frac{d}{4} \left\{ 4.9 + 2.0\left(\sqrt{1+\kappa}\right)\lambda_s + 0.91\kappa{\lambda_s}^2 \right\} \quad \text{(G.2.8-5)}$$

Non-dimensional upper and lower bounds are given in Figure G.2.8-3.

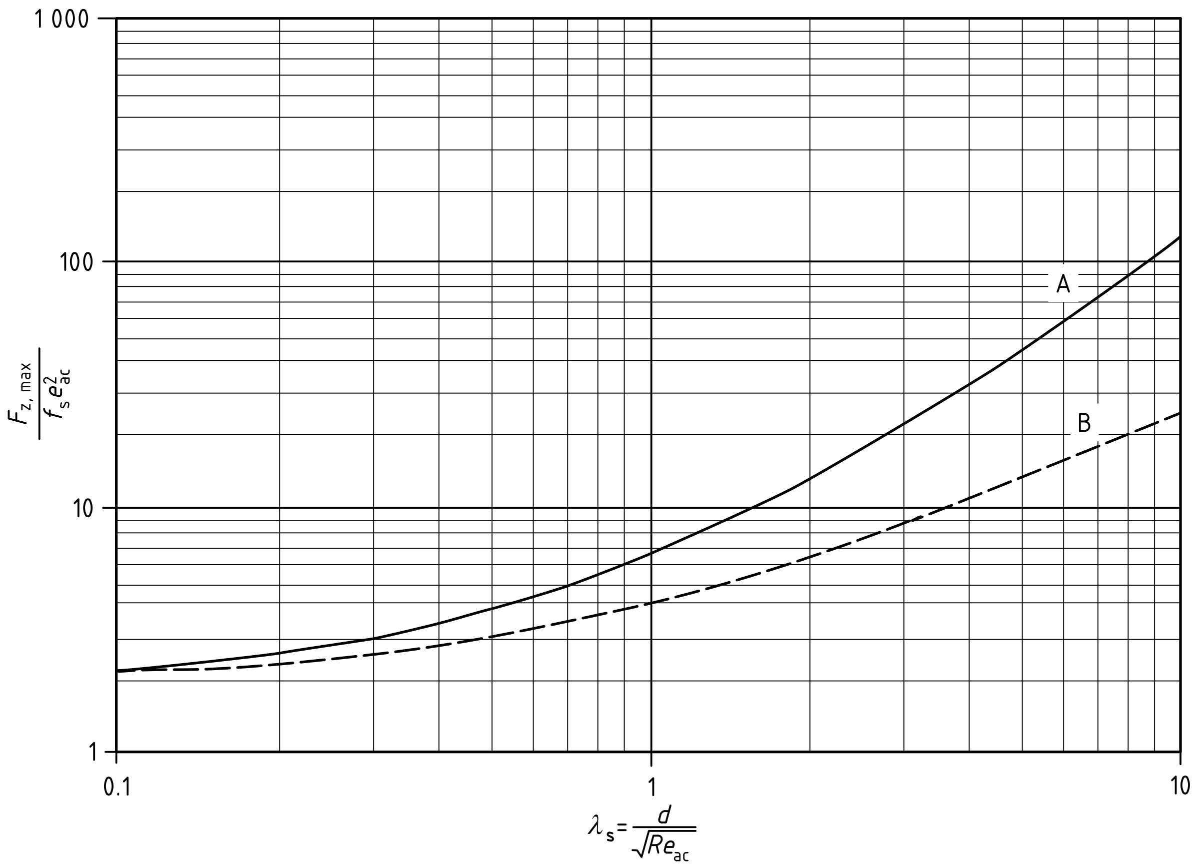
f) Calculate the shear stresses:

$$\tau_F = \text{s.c.f} \times \frac{F_S}{\pi d e_{ac}} \quad \text{(G.2.8-5a)}$$

where the s.c.f for flush nozzles is obtained from Figure G.2.5-7 and the s.c.f for protruding nozzles is obtained from Figure G.2.5-8, taking $r = d/2$, $T' = e_{ac}$ and $t = e_{ab}$.

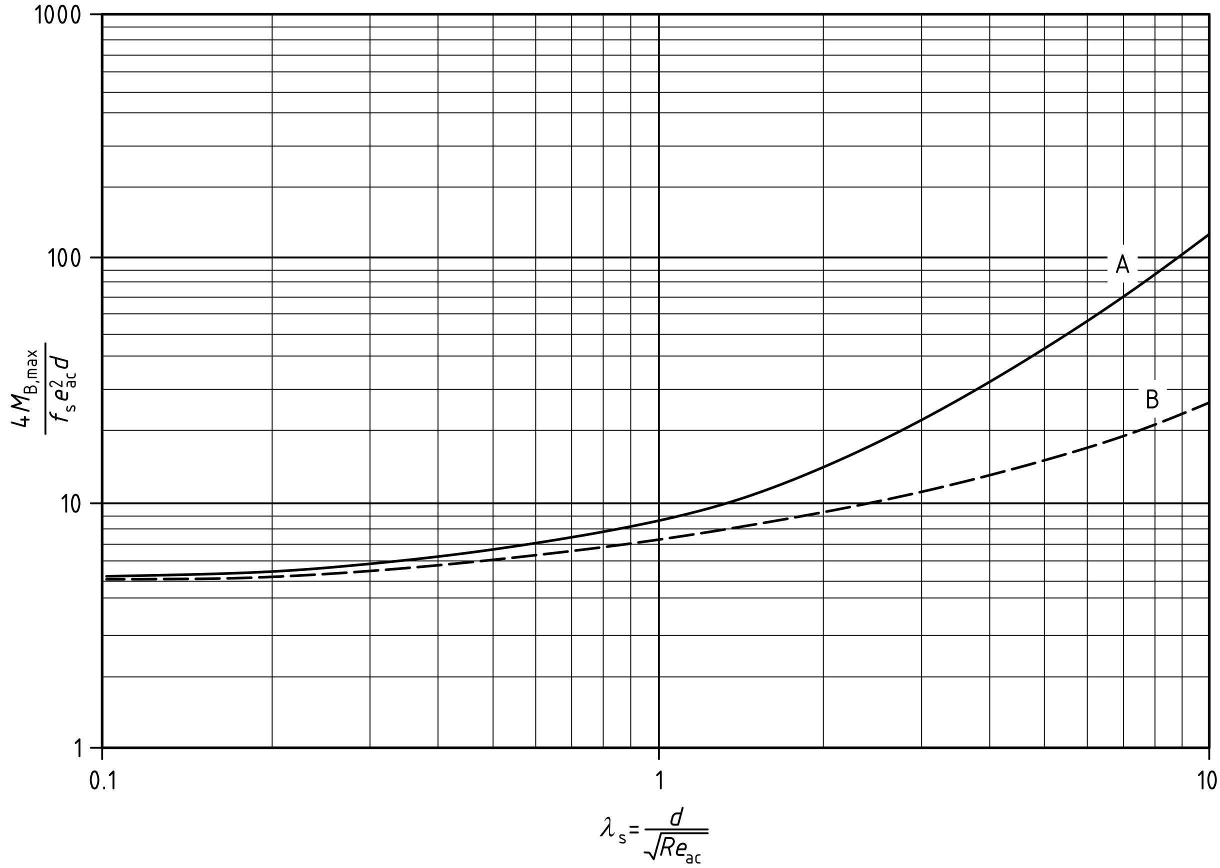
$$\tau_Z = \frac{2M_Z}{\pi d^2 e_{ac}} \quad \text{(G.2.8-5b)}$$

Figure G.2.8-2 **Allowable axial nozzle load**



NOTE Curve A, fully reinforced. Curve B, unreinforced.

Figure G.2.8-3 **Allowable nozzle moment**



NOTE Curve A, fully reinforced. Curve B, unreinforced.

G.2.8.2.4 Allowable simultaneous loads

To determine the effects of the combination of pressure, axial load and bending moment acting simultaneously the following procedure should be applied.

a) Calculate the individual load ratios as follows:

$$\Phi_p = \frac{p}{p_{max}} \quad \text{(G.2.8-6)}$$

$$\Phi_z = \frac{F_Z}{F_{Z,max}} \quad \text{(G.2.8-7)}$$

$$\Phi_B = \frac{M_B}{M_{B,max}} \quad \text{(G.2.8-8)}$$

$$\Phi_S = \frac{|\tau_F| + |\tau_Z|}{0.5f} \quad \text{(G.2.8-8a)}$$

b) Each individual load ratio should be ≤1.0, i.e.

$$|\Phi_p| \leq 1.0 \quad \text{(G.2.8-9)}$$

$$|\Phi_Z| \leq 1.0 \quad \text{(G.2.8-10)}$$

$$|\Phi_B| \leq 1.0 \quad \text{(G.2.8-11)}$$

$$|\Phi_S| \leq 1.0 \quad \text{(G.2.8-11a)}$$

c) The result of the interaction of loads should be checked so that:

where p_{max} was calculated using **3.5.4.3** (see **G.2.8.3**) then:

$$\left(|\Phi_p| + |\Phi_Z| + |\Phi_B| + |\Phi_S|\right) \leq 1.0 \qquad \text{(G.2.8-12)}$$

or where p_{max} was calculated using Equation (G.2.8-3), or at the reinforcing plate outer diameter, then:

$$\sqrt{\left\{\max\left(\left|\frac{\Phi_p}{C_1} + \Phi_Z\right|; |\Phi_Z|; \left|\frac{\Phi_p}{C_1} - 0.2\Phi_Z\right|\right) + |\Phi_B|\right\}^2 + \Phi_S^2} \leq 1.0 \qquad \text{(G.2.8-13)}$$

Where C_1 = 1.1 for calculations at the nozzle outer diameter, and C_1 = 1.0 at the reinforcing plate outer diameter.

NOTE Equation (G.2.8-12) is based on a linear interaction and Equation (G.2.8-13) is based on a linear interaction of pressure and axial load with the bending moment, and both yield conservative results. Although a circular interaction might be less conservative, at the present time this could only be shown by applying a design by analysis approach to individual cases.

G.2.8.2.5 Maximum allowable load range

For all load cases which can occur in service, the minimum and the maximum values of the internal pressure and each considered external load should be defined, such that Equations (G.2.8-14), (G.2.8-15) and (G.2.8-16) are satisfied.

$$\Delta p = \max(p;0) - \min(p;0) \qquad \text{(G.2.8-14)}$$

$$\Delta F_Z = \max(F_Z;0) - \min(F_Z;0) \qquad \text{(G.2.8-15)}$$

$$\Delta M_B = \max(M_B;0) - \min(M_B;0) \qquad \text{(G.2.8-16)}$$

$$\Delta F_S = \max(F_S;0) - \min(F_S;0) \qquad \text{(G.2.8-16a)}$$

$$\Delta M_Z = \max(M_Z;0) - \min(M_Z;0) \qquad \text{(G.2.8-16b)}$$

The stresses are to be considered only at the nozzle edge with the equivalent shell thickness e_{eq} set equal to e_{ac}. However, when a reinforcing ring of width $L_p < \sqrt{R(e_{as} + e_{ap})}$ is used (see **G.2.8.2.2**), the calculation thickness e_{eq} in Equations (G.2.8-18), (G.2.8-19) and (G.2.8-20) should be set equal to:

$$e_{eq} = e_{as} + \min\left(\frac{e_{ap}L_p}{\sqrt{R(e_{as} + e_{ap})}}; e_{ap}\right)\min\left(\frac{f_2}{f_s}; 1.0\right) \qquad \text{(G.2.8-17)}$$

a) Stress σ_P due to pressure range is given by:

$$\sigma_P = scf\frac{\Delta pR}{2e_{eq}} \qquad \text{(G.2.8-18)}$$

b) Stress σ_{FZ} due to axial nozzle load range is given by:

$$\sigma_{FZ} = scf\frac{\Delta F_z}{\pi d e_{eq}}\sqrt{\frac{R}{e_{eq}}} \qquad \text{(G.2.8-19)}$$

c) Stress σ_{MB} due to external moment range is given by:

$$\sigma_{MB} = scf\frac{4\Delta M_B}{\pi d^2 e_{eq}}\sqrt{\frac{R}{e_{eq}}} \qquad \text{(G.2.8-20)}$$

d) The shear stress due to the shear force and torsional moment ranges is given by:

$$\tau = scf\frac{\Delta F_S}{\pi d e_{ac}} + \frac{2\Delta M_Z}{\pi d^2 e_{ac}} \quad \text{(G.2.8-20a)}$$

NOTE The scf factors are obtained from Figure G.2.5-1 to Figure G.2.5-8 for the relevant load.

e) The thermal stress σ_T which is due to the temperature differences between the nozzle wall and shell wall including the through thickness temperature distribution effects, should be calculated by an appropriate design method such as that in **G.4**.

f) The combination of the stress ranges should be restricted to $3f_s$.

$$\left|\sigma_T + \sqrt{\left(\sigma_p + \sigma_{FZ}\right)^2 + \sigma_{MB}^2 + 4\tau^2}\right| \le 3f_s \quad \text{(G.2.8-21)}$$

G.2.8.2.6 Nozzle strength

For all load conditions the longitudinal stresses should be verified as follows.

a) Longitudinal stress verification:

$$\frac{pd}{4e_{ab}} + \frac{4M_B}{\pi d^2 e_{ab}} + \frac{F_Z}{\pi d e_{ab}} \le f_n \quad \text{(G.2.8-22)}$$

If F_Z is negative it should be set to zero.

b) Longitudinal instability verification (with $p = 0$):

$$\frac{M_B}{M_{max}} + \frac{|F_Z|}{F_{max}} \le 1.0 \quad \text{(G.2.8-23)}$$

If F_Z is positive it should be set to zero.

Where:

M_{max} $= 0.25\pi d^2\, e_{ab}\sigma_{z,allow}$;

F_{max} $= \pi d e_{ab}\sigma_{z,allow}$;

$\sigma_{z,allow}$ is the maximum permitted compressive longitudinal stress in the nozzle (see **A.3.5**).

c) Reinforced nozzle wall:

For reinforcement provided as shown in Figures 3.5-15, 3.5-16 or 3.5-17, a longitudinal stress verification and longitudinal instability check should also be performed using values of d and e_{ab} at the thinnest part of the nozzle.

G.2.8.3 External loads on nozzles in cylindrical shells

G.2.8.3.1 Limitations

The rules for external loads on nozzles in cylindrical shells apply with the following limitations.

a) $d/D \le 0.5$.

b) $e_{as}/D \le 0.1$.

c) The parameter $\lambda_c = \dfrac{d}{\sqrt{De_{ac}}}$ should be ≤10. If this range is extended the effects of a twisting moment should be considered.

d) Distances to any other local load in all directions should be not less than $\sqrt{De_{ac}}$.

e) The nozzle thickness should be maintained over a distance not less than $\sqrt{de_{ab}}$.

G.2.8.3.2 Design procedure

Loads and stresses are calculated in accordance with the following procedure given in steps a) to f), at the nozzle outer diameter. If a reinforcing plate is fitted and the width $L_p < \sqrt{D(e_{as} + e_{ap})}$ the maximum allowable loads are also calculated at the reinforcing plate outer diameter, in accordance with steps a) to c).

a) Select shell, nozzle (and reinforcing plate) thicknesses, calculate e_{ac} and d.

1) For a nozzle with integral reinforcement $e_{ac} = e_{as}$ and d is the mean nozzle diameter.

2) For a nozzle with a reinforcing plate:

i) at the nozzle outer diameter $e_{ac} = e_{as} + e_{ap} \times \min(f_2/f_s;1.0)$ and d is the mean nozzle diameter; and

ii) at the reinforcing plate outer diameter $e_{ac} = e_{as}$ and $d = d_2$.

b) Calculate the maximum allowable individual loads (see **G.2.8.3.3**).

c) Verify the load ratios and the interaction of the loads (see **G.2.8.3.4**).

d) At the nozzle outer diameter only; define the equivalent shell thickness e_{eq} and verify the allowable load range (see **G.2.8.3.5**).

e) Verify the nozzle strength (see **G.2.8.3.6**).

f) If any of the criteria in b) to e) above are not satisfied, repeat the procedure from a) with increased thicknesses.

G.2.8.3.3 Maximum allowable individual loads

To determine the maximum allowable values of pressure, axial load and bending moment, which may be independently applied to a nozzle, the following procedure should be applied.

a) Determine the geometric parameter λ_c.

$$\lambda_c = \frac{d}{\sqrt{De_{ac}}} \tag{G.2.8-24}$$

b) At the nozzle outer diameter the permissible pressure p_{max} is derived using the rules of **3.5.4.3** taking C = 1.1, or is given by the following expression taken from **3.5.4.9**. See **3.5.4.9.1** for definitions of the various terms.

$$p_{max}\left[A_p + 0.5\left(A_{fs} + A_{fb} + A_{fp}\right)\right] = f_s A_{fs} + f_p A_{fp} + f_b A_{fb} \tag{G.2.8-25}$$

At the reinforcing plate outer diameter the permissible pressure $p_{max} = 2f_s e_{as}/D$

c) The allowable axial nozzle load $F_{Z,max}$ is given by:

$$F_{Z,max} = f_s e_{ac}^2 \max[C_2; 1.81] \tag{G.2.8-26}$$

with

$$C_2 = a_0 + a_1\lambda_c + a_2{\lambda_c}^2 + a_3{\lambda_c}^3 + a_4{\lambda_c}^4 \tag{G.2.8-27}$$

and coefficients a_0 to a_4 from Table G.2.8-1 or C_2 from Figure G.2.8-4.

d) The allowable circumferential moment $M_{X,max}$ is given by:

$$M_{X,max} = f_s e_{ac}^2 \frac{d}{4} \max[C_3; 4.90] \quad \text{(G.2.8-28)}$$

with

$$C_3 = a_0 + a_1\lambda_c + a_2{\lambda_c}^2 + a_3{\lambda_c}^3 + a_4{\lambda_c}^4. \quad \text{(G.2.8-29)}$$

and with coefficients a_0 to a_4 from Table G.2.8-1 or C_3 from Figure G.2.8-5

e) The allowable longitudinal moment $M_{Y,max}$ is given by:

$$M_{Y,max} = f_s e_{ac}^2 \frac{d}{4} \max[C_4; 4.90] \quad \text{(G.2.8-30)}$$

with

$$C_4 = a_0 + a_1\lambda_c + a_2{\lambda_c}^2 + a_3{\lambda_c}^3 + a_4{\lambda_c}^4 \quad \text{(G.2.8-31)}$$

and coefficients a_0 to a_4 from Table G.2.8-1 or C_4 from Figure G.2.8-6.

f) Calculate the shear stresses:

$$\tau_X = \frac{2F_X}{\pi d e_{ac}} \quad \text{(G.2.8-31a)}$$

$$\tau_Y = \frac{2F_Y}{\pi d e_{ac}} \quad \text{(G.2.8-31b)}$$

$$\tau_Z = \frac{2M_Z}{\pi d^2 e_{ac}} \quad \text{(G.2.8-31c)}$$

NOTE 1 The curves of Figure G.2.8-4, Figure G.2.8-5 and Figure G.2.8-6 are derived from Welding Research Council Bulletin No. 297 [49], while the allowable loads are based on a maximum stress of $2.25f_s$.

NOTE 2 For $0.2 < e_{ab}/e_{ac} < 0.5$ the value of C_4 is obtained by linear interpolation.

NOTE 3 For the calculations at the reinforcing plate outer diameter the value of C_4 is obtained from Figure G.2.8-6 using the curve for $e_{ab}/e_{ac} \geq 0.5$, or is calculated using equation (G.2.8-31) with coefficients a_0 to a_4 for $e_{ab}/e_{ac} \geq 0.5$ from Table G.2.8-1.

Table G.2.8-1 **Coefficients to define factors C_2, C_3 or C_4**

Factor	a_0	a_1	a_2	a_3	a_4
C_2	0.60072181	0.95196257	0.0051957881	—0.001406381	0
C_3	4.526315	0.064021889	0.15887638	—0.021419298	0.0010350407
C_4 with $e_{ab}/e_{ac} \leq 0.2$	4.8517511	0.0251012	0.7428624	–0.0153153	0
C_4 with $e_{ab}/e_{ac} \geq 0.5$	4.8588639	2.1870887	1.4567053	–0.3316430	0.0253850

Figure G.2.8-4 **Calculation factor C_2**

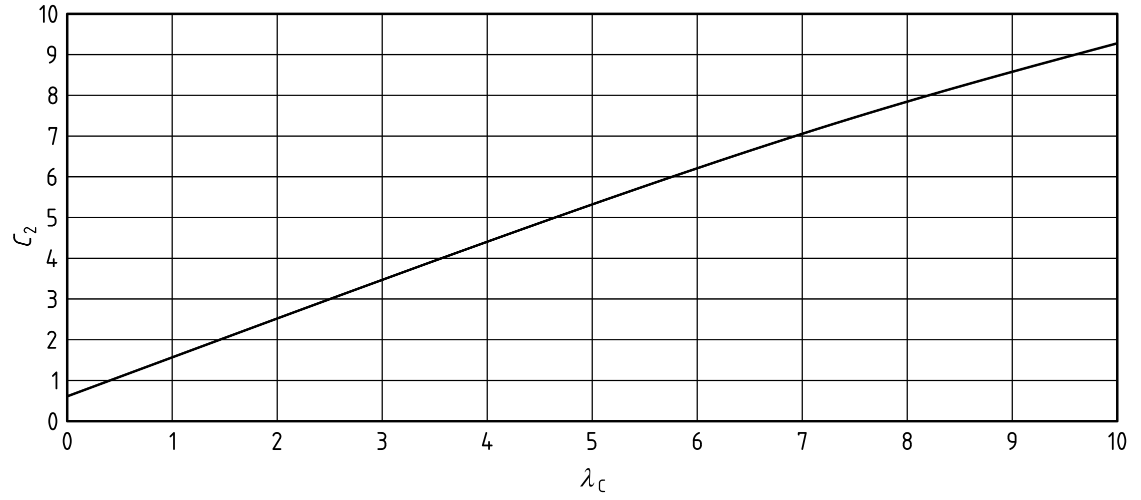


Figure G.2.8-5 **Calculation factor C_3**

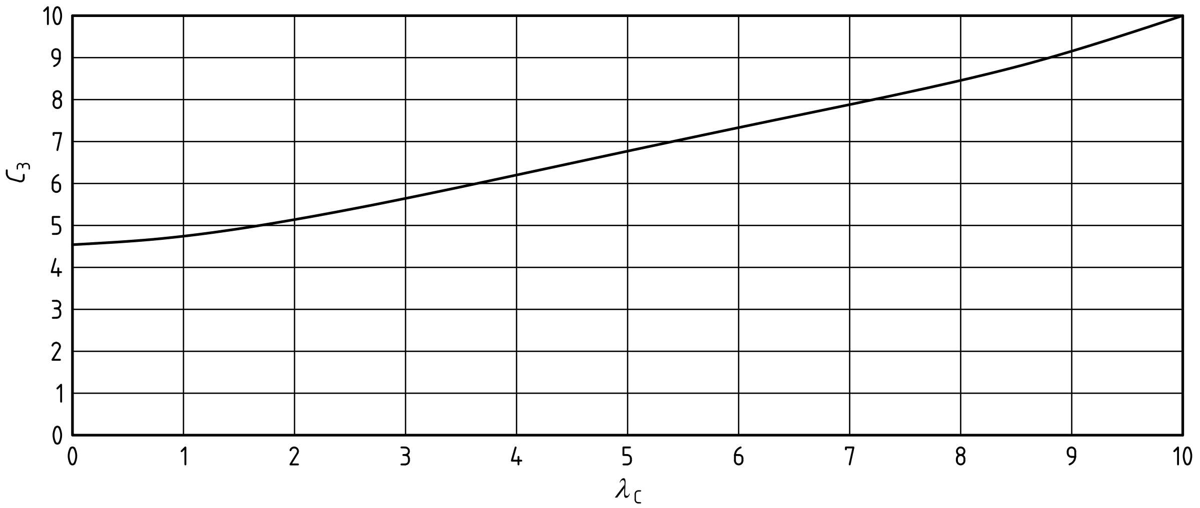
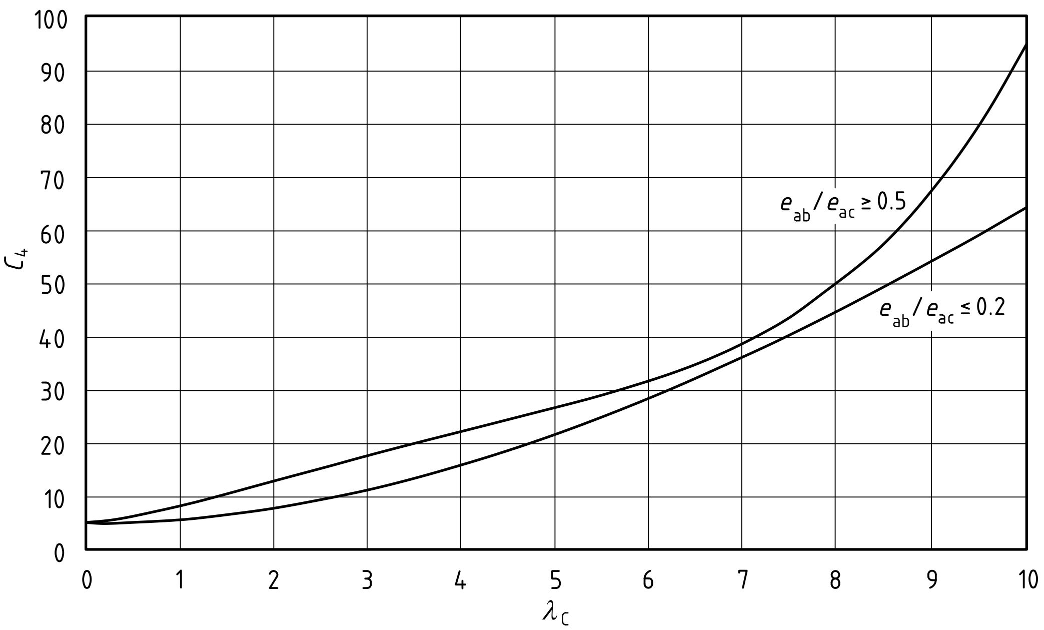


Figure G.2.8-6 **Calculation factor C_4**



G.2.8.3.4 Combination of external loads and internal pressure

To determine the effects of the combination of pressure, axial load and bending moments acting simultaneously the following procedure should be applied.

a) Calculate the individual load ratios as follows:

$$\Phi_P = \frac{p}{p_{max}} \quad \text{(G.2.8-32)}$$

$$\Phi_Z = \frac{F_Z}{F_{Z,max}} \quad \text{(G.2.8-33)}$$

$$\Phi_B = \sqrt{\left(\frac{M_X}{M_{X,max}}\right)^2 + \left(\frac{M_Y}{M_{Y,max}}\right)^2} \quad \text{(G.2.8-34)}$$

$$\Phi_S = \frac{|\tau_X| + |\tau_Y| + |\tau_Z|}{0.5f} \quad \text{(G.2.8-34a)}$$

b) Each individual load ratio should be ≤1.0, i.e.

$$|\Phi_P| \leq 1.0 \quad \text{(G.2.8-35)}$$

$$|\Phi_Z| \leq 1.0 \quad \text{(G.2.8-36)}$$

$$|\Phi_B| \leq 1.0 \quad \text{(G.2.8-37)}$$

$$|\Phi_S| \leq 1.0 \quad \text{(G.2.8-37a)}$$

c) The result of the interaction of loads should be checked so that where p_{max} was calculated using **3.5.4.3** (see **G.2.8.2.3**) then:

$$(|\Phi_P| + |\Phi_Z| + |\Phi_B| + |\Phi_S|) \leq 1.0 \quad \text{(G.2.8-38)}$$

or where p_{max} was calculated using Equation (G.2.8-25), or at the reinforcing plate outer diameter, then:

$$\sqrt{\max\left(\left|\frac{\Phi_P}{C_1}+\Phi_Z\right|;|\Phi_Z|;\left|\frac{\Phi_P}{C_1}-0.2\Phi_Z\right|\right)^2+{\Phi_B}^2+\Phi_S^2}\leq 1.0 \qquad \text{(G.2.8-39)}$$

Where C_1 = 1.1 for calculations at the nozzle outer diameter, and C_1 = 1.0 at the reinforcing plate outer diameter.

NOTE In Equation (G.2.8-39) a circular interaction with the bending moment load is accepted on the grounds of a conservative estimate of the stress concentration factors in Welding Research Council Bulletin No. 297 [49].

G.2.8.3.5 Maximum allowable load range

For all load cases which may occur in service the minimum and the maximum values of the internal pressure and each considered external load should be defined, such that Equations (G.2.8-40) to (G.2.8-43) are satisfied.

$$\Delta p = \max(p;0) - \min(p;0) \qquad \text{(G.2.8-40)}$$

$$\Delta F_Z = \max(F_Z;0) - \min(F_Z;0) \qquad \text{(G.2.8-41)}$$

$$\Delta M_X = \max(M_X;0) - \min(M_X;0) \qquad \text{(G.2.8-42)}$$

$$\Delta M_Y = \max(M_Y;0) - \min(M_Y;0) \qquad \text{(G.2.8-43)}$$

$$\Delta F_X = \max(F_X;0) - \min(F_X;0) \qquad \text{(G.2.8-43a)}$$

$$\Delta F_Y = \max(F_Y;0) - \min(F_Y;0) \qquad \text{(G.2.8-43b)}$$

$$\Delta M_Z = \max(M_Z;0) - \min(M_Z;0) \qquad \text{(G.2.8-43c)}$$

The stresses are to be considered only at the nozzle edge, with the equivalent shell thickness e_{eq} set equal to e_{ac}. However, when a reinforcing ring of width $L_p < \sqrt{D(e_{as}+e_{ap})}$ is used (see **G.2.8.3.2**), the calculation thickness e_{eq} in Equations (G.2.8-44) to (G.2.8-48) should be set equal to:

$$e_{eq} = e_{as} + \min\left(\frac{e_{ap}L_p}{\sqrt{D(e_{as}+e_{ap})}};e_{ap}\right)\min\left(\frac{f_2}{f_s};1.0\right) \qquad \text{(G.2.8-44)}$$

a) Stress σ_P due to pressure range:

$$\sigma_P = \left(\frac{\Delta pD}{2e_{eq}}\right)\left[\frac{2+2\dfrac{d}{D}\sqrt{\dfrac{de_{ab}}{De_{eq}}}+1.25\dfrac{d}{D}\sqrt{\dfrac{D}{e_{eq}}}}{1+\dfrac{e_{ab}}{e_{eq}}\sqrt{\dfrac{de_{ab}}{De_{eq}}}}\right] \qquad \text{(G.2.8-45)}$$

b) Stress σ_{FZ} due to axial nozzle load range:

$$\sigma_{FZ} = \frac{2.25}{C_2}\left(\frac{\Delta F_Z}{e_{eq}^2}\right) \qquad \text{(G.2.8-46)}$$

with C_2 from Equation (G.2.8-27) or Figure G.2.8-4.

c) Stress σ_{MX} due to circumferential moment range:

$$\sigma_{MX} = \frac{2.25}{C_3}\left(\frac{4\Delta M_X}{e_{eq}^2 d}\right) \quad \text{(G.2.8-47)}$$

with C_3 from Equation (G.2.8-29) or Figure G.2.8-5.

d) Stress σ_{MY} due to longitudinal moment range:

$$\sigma_{MY} = \frac{2.25}{C_4}\left(\frac{4\Delta M_Y}{e_{eq}^2 d}\right) \quad \text{(G.2.8-48)}$$

with C_4 from Equation (G.2.8-31) or Figure G.2.8-6.

e) The shear stress due to the shear force and torsional moment ranges is given by:

$$\tau = \frac{\Delta F_X}{\pi d e_{ac}} + \frac{\Delta F_Y}{\pi d e_{ac}} + \frac{2\Delta M_Z}{\pi d^2 e_{ac}} \quad \text{(G.2.8-48a)}$$

f) The thermal stress σ_T which is due to the temperature differences between the nozzle wall and shell wall, including any through thickness temperature distribution effects, should be calculated by an appropriate design method such as that given in **G.4**.

g) The combination of the stress ranges should be restricted to $3f_s$.

$$\left|\sigma_T + \sqrt{(\sigma_P + \sigma_{FZ})^2 + \sigma_{MX}^2 + \sigma_{MY}^2 + 4\tau^2}\right| \leq 3f_s \quad \text{(G.2.8-49)}$$

G.2.8.3.6 Nozzle strength

For all load conditions the longitudinal stresses should be verified as follows.

a) Longitudinal stress verification:

$$\frac{pd}{4e_{ab}} + \frac{4\sqrt{M_X^2 + M_Y^2}}{\pi d^2 e_{ab}} + \frac{F_Z}{\pi d e_{ab}} \leq f_n \quad \text{(G.2.8-50)}$$

If F_Z is negative it should be set to zero.

b) Longitudinal instability check (with p = 0):

$$\frac{\sqrt{M_X^2 + M_Y^2}}{M_{max}} + \frac{|F_Z|}{F_{max}} \leq 1.0 \quad \text{(G.2.8-51)}$$

where

M_{max} = 0.25 πd^2 $e_{ab}\sigma_{z,allow}$;

F_{max} = πd $e_{ab}\sigma_{z,allow}$;

$\sigma_{z,allow}$ = maximum permitted compressive longitudinal stress in nozzle (see **A.3.5**).

If F_Z is positive it should be set to zero.

c) Reinforced nozzle wall:

For reinforcement provided as shown in Figures 3.5-19, 3.5-20 or 3.5-21, a longitudinal stress verification and longitudinal instability check should also be performed using values of d and e_{ab} at the thinnest part of the nozzle.

G.3 Supports and mountings for pressure vessels

G.3.1 General considerations for supports

G.3.1.1 Introduction

This clause and **G.3.2** and **G.3.3** are concerned with the supports for pressure vessels and the supports for fittings carried from the shell or ends of the vessel, with regard to their effect on the vessel. The structural design of supports is not included because it can be dealt with by the usual methods of structural design. Convenient references for these are [40] and [41].

The supports of vessels and of fittings carried by the shell produce local moments and membrane forces in the vessel wall which can be treated by the methods given in **G.2**. Notes and cross-references for applying these to various types of support are included.

The supports of a vessel should be designed to withstand all the external loads likely to be imposed on it in addition to the dead weight of the vessel and its contents. These loads may include:

a) superimposed loads;

b) wind loads on exposed vessels;

c) thrusts or moments transmitted from connecting pipework;

d) shock loads due to liquid hammer or surging of the vessel contents; and

e) forces due to differential expansion between the vessel and its supports.

*NOTE Lifting attachments for vessels can be checked in accordance with **G.3.1.4** and **G.3.1.5**. However whilst Annex A stress levels are appropriate for the assessment of steady loads on attachments, they are not necessarily relevant when loading/dynamic factors are applied to take account of non-steady loads.*

G.3.1.2 *Notation*

For the purposes of **G.3** the following symbols apply. All dimensions are in the corroded condition unless otherwise indicated (see **3.1.5**):

a	is the effective cross-sectional area of stiffener (or stiffeners) and the portion of the shell that can be assumed to act with it (them) for a horizontal vessel (in mm^2);
a_1	is the effective cross-sectional area of extended saddle plate for a horizontal vessel (in mm^2);
A	is the distance from saddle support to adjacent end of cylindrical part (in mm);
b	is the mean depth of dished end of vessel (in mm);
b_1	is the axial width of saddle support (in mm);
b_2	$= b_1 + 10t$;
c	is the distance from centroid of effective area of stiffener and the portion of the shell that can be assumed to act with it to surface of shell (in mm);
c_1	is the distance from centroid of effective area of extended saddle plate to surface of saddle plate (in mm);
$C_1 \ldots C_5$	are constants;
C_x	is the half length of rectangular loading area in longitudinal direction (in mm);
C_ϕ	is the half length of rectangular loading area in circumferential direction (in mm);

d	is the distance from centroid of effective area of stiffener and the portion of the shell that can be assumed to act with it to tip of stiffener (in mm);
d_x	is the distance from centroid of effective area of stiffener to tip of stiffener in longitudinal direction (in mm);
d_ϕ	is the distance from centroid of effective area of stiffener to tip of stiffener in circumferential direction (in mm);
D	is the mean diameter of the vessel (in mm);
e	is the perpendicular distance from the line of the reaction to the centroid of the weld area (in mm);
E	is the modulus of elasticity (in N/mm^2) from Table 3.6-3;
f	is the nominal design stress (in N/mm^2);
$f_1 \ldots f_{10}$	are the resultant stresses in horizontal vessel due to mode of support (in N/mm^2) [see Equations (G.3.3-5) to (G.3.3-8), (G.3.3-17) to (G.3.3-33) and (G.3.3-35)];
f_n	is the nominal stress in dished end calculated as in Section 3 (in N/mm^2);
F	is the resultant of horizontal forces acting on vertical vessel (in N);
H	is the resultant horizontal force in least cross-section of saddle support (in N);
I	is the second moment of area of effective cross-section of stiffening ring and the portion of the shell that can be assumed to act with it (in mm^4);
I_1	is the second moment area of effective cross-section of extended saddle plate (in mm^4);
$K_1 \ldots K_{11}$	are constants;
L	is the length of cylindrical part of vessel (in mm);
l	is the length of part of shell of horizontal vessel assumed to act with a ring support (in mm);
M_1	is the bending moment in horizontal ring girder above its own support (in N·mm);
M_2	is the bending moment in horizontal ring girder midway between its supports (in N·mm);
M_3	is the longitudinal bending moment in horizontal vessel midway between its supports (in N·mm) [see Equations (G.3.3-1) and (G.3.3-3)];
M_4	is the longitudinal bending moment in horizontal vessel at its supports (in N·mm) [see Equations (G.3.3-2) and (G.3.3-4)];
M_x	is the longitudinal or meridional bending moment per unit circumference (in N·mm/mm);
M_ϕ	is the circumferential bending moment per unit length (in N·mm/mm);
N_x	is the longitudinal membrane force per unit circumference (in N/mm);
N_ϕ	is the circumferential membrane force per unit length (in N/mm);
p_m	is the internal pressure at equator (horizontal centre line of vessel) (in N/mm^2);
q	is the shear stress in vessel shell (in N/mm^2) (see **G.3.3.2.5**);
q_e	is the shear stress in vessel end (in N/mm^2) (see **G.3.3.2.5**);
r	is the mean radius of cylindrical part of vessel (in mm);
r_i	is the inside radius of cylindrical part of vessel (in mm);
r_1	is the radius of base of skirt support of vertical vessel (in mm);
r_2	is the mean radius of horizontal ring girder or of ring support (in mm);
t	is the analysis thickness of vessel shell (in mm) (see **1.6**);
t_1	is the analysis thickness of reinforcing plate (in mm) (see **1.6**);
t_2	is the analysis thickness of ring stiffeners (in mm) (see **1.6**);
t_e	is the analysis thickness of vessel end (in mm) (see **1.6**);
T	is the maximum twisting moment in horizontal ring girder (in N·mm);
w	is the average weight of vertical vessel per millimetre height (in N/mm);
W	is the weight of vessel (in N);
W_1	is the maximum reaction at support (in N);

x	is the distance from support of horizontal ring girder to nearest point of maximum twisting moment (in mm);
y	is the distance of the external load from the vessel wall (in mm);
$\bar{y}$	is the height of the resultant of horizontal forces acting on vessel above its supports (in mm);
Z	is the section modulus of effective cross-section of ring support for horizontal vessel (in mm^3);
ε	is the circumferential buckling strain;
θ	is the included angle of saddle support (in degrees);
ϕ_1	is the angle between radius drawn to position of support and vertical centre line of vessel (in degrees).

G.3.1.3 Reaction at the supports

The reactions at the supports of a vessel can be found by the ordinary methods of statics except in the case of long horizontal vessels supported at more than two positions.

The reactions at the supports of vessels subject to heavy external loads may need to be examined for the following conditions:

a) working conditions, including full wind load and loads due to pipework;

b) test conditions, including full wind load, if any, and forces due to "cold pull up" of any pipes that will remain connected to the vessel during tests;

c) shut-down conditions, vessel empty, and exposed to full wind load, if any, and the forces due to "cold pull up" in the pipe system connected to it. It is essential to provide anchor bolts if there is an upward reaction to any support under any of these conditions.

The theoretical reactions at the supports of long horizontal vessels supported at more than two positions can be found by the methods used for continuous beams but the calculated values are always doubtful because of settlement of the supports and initial errors of roundness of or straightness in the vessel.

G.3.1.4 Brackets

Brackets are fitted to the shells of pressure vessels to support either the vessel or some structure which has to be carried from it. Typical brackets are shown in Figure G.3.1-1.

The brackets themselves are designed by the ordinary methods used for brackets supporting beams in structural engineering.

A bracket always applies an external moment to the shell equal to $W_1 y$.

The effect of this moment on the shell can be found by the method given in **G.2.3**. If the local stresses found in this way are excessive, a reinforcing plate, designed as described in **G.3.1.5**, should be fitted between the bracket and the vessel wall.

In addition to the vertical loads, the brackets supporting a vertical vessel may be subject to tangential forces due to thrusts and moments transmitted from pipework. Such brackets impose a circumferential moment on the vessel wall in addition to the longitudinal moment. The stresses due to this can be calculated and added to the others but ring or skirt supports are preferable in cases of this type.

G.3.1.5 Reinforcing plates for brackets

G.3.1.5.1 Introduction

Reinforcing plates are required when the local stresses in the vessel shell, found as described in **G.2** for the connection of a support or mounting, are excessive. The form of reinforcement will depend upon the direction of load and whether a moment is applied.

NOTE Experimental work, discussed in [17], has shown that there is some stress concentration near the sharp corners of rectangular reinforcing plates. Rounded corners are therefore preferable.

Moments and radial loads on brackets attached to cylinders with partial penetration welds may be assessed in accordance with **G.3.1.5.2** and **G.3.1.5.3**. Where these welds are full penetration welds, the assembly can be assumed to be integral with the shell and the stresses evaluated in accordance with **G.2**.

G.3.1.5.2 Radially inward load on a cylinder

Figure G.3.1-2a) shows a typical simple reinforcing plate applied to a cylinder.

The stresses in the vessel shell at the edge of the reinforcing plate are approximately equal to those calculated by assuming the load to be distributed over the whole area of the reinforcing plate $2d_x \times 2d_\phi$ and proceeding as described in **G.2.2.2**.

A safe approximation for the maximum stresses in the reinforcing plate, which occurs at the edges of the actual loaded area $2C_x \times 2C_\phi$, is given by the following procedure.

a) Find the maximum moments M_ϕ and M_x and the maximum membrane forces N_ϕ and N_x, for the same loading applied to a cylinder of thickness $(t + t_1)$, from the charts in **G.2.2.2** for a radial load, applied over a loaded area $2C_x \times 2C_\phi$.

b) Find the resultant stresses due to these by assuming that the vessel shell and the reinforcing plate share the moments M_ϕ and M_x in proportion to the cubes of their thicknesses and the membrane forces N_ϕ and N_x in direct proportion to their thicknesses.

i.e.

$$M_{\phi\ reinforcing\ plate} = M_\phi t_1{}^3/(t_1{}^3 + t^3) \quad \text{(G.3.1-1)}$$

$$M_{\phi\ vessel\ shell} = M_\phi t^3/(t_1{}^3 + t^3) \quad \text{(G.3.1-2)}$$

$$N_{\phi\ reinforcing\ plate} = N_\phi t_1/(t_1 + t) \quad \text{(G.3.1-3)}$$

$$N_{\phi\ vessel\ shell} = N_\phi t/(t_1 + t) \quad \text{(G.3.1-4)}$$

G.3.1.5.3 Radially outward load and/or moment on a cylinder

These loads require gusset plates as shown on the typical arrangement in Figure G.3.1-2b) to achieve the load transference from the attachment to the vessel.

The stresses in the vessel shell at the edge of the reinforcing plate are approximately equal to those calculated by assuming the load or moment to be distributed over the whole area of the reinforcing plate $2d_x \times 2d_\phi$ and proceeding as described in **G.2.2.2** for a radial load or in **G.2.3** for a moment. Note that the sign of the radial load is reversed to that in **G.3.1.5.2**.

A safe approximation for the maximum stresses in the reinforcing plate, which occur at the edges of the actual loaded area $2C_x \times 2C_\phi$ is given by the following procedure.

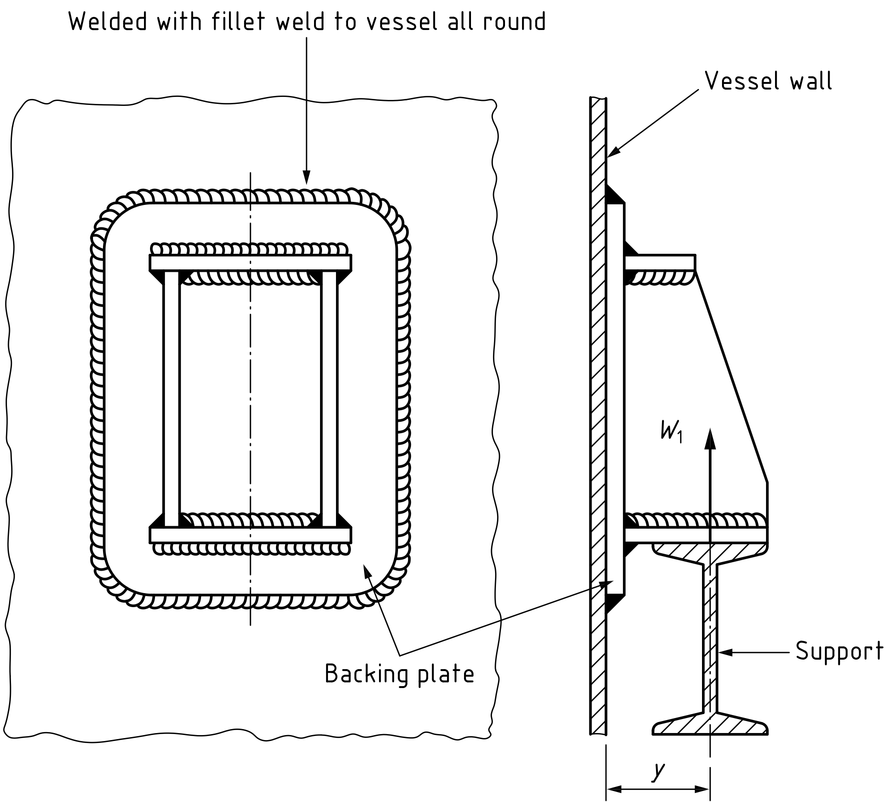
a) Find the maximum moments M_ϕ and M_x and the maximum membrane forces N_ϕ and N_x for the same loading applied to a cylinder of thickness $(t + t_1)$ from the charts in **G.2.2.2** for a radial load or from **G.2.3** for a moment, both applied over the loaded area $2C_x \times 2C_\phi$.

b) Find the resultant stresses due to these by assuming that the vessel wall and the reinforcing plate share the moments M_ϕ and M_x in proportion to the cubes of their thicknesses and the membrane forces N_ϕ and N_x in direct proportion to their thicknesses, as given in **G.3.1.5.2**b).

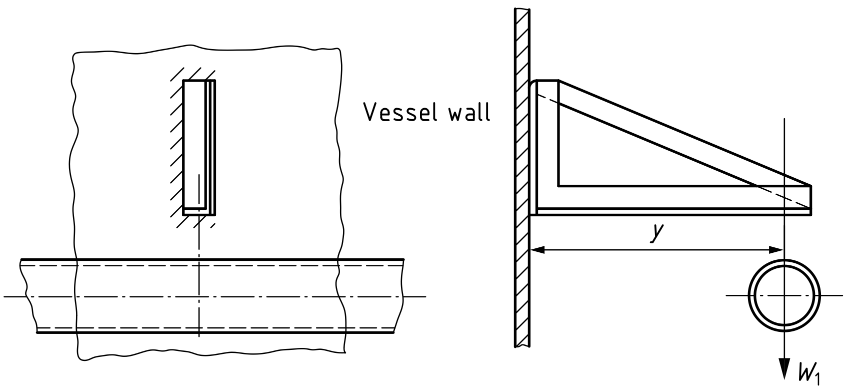
G.3.1.5.4 Loads on spherical vessels

The principles of **G.3.1.5.2** and **G.3.1.5.3** can be applied using the appropriate charts of **G.2.4.3** and **G.2.4.4**.

Figure G.3.1-1 **Typical brackets**

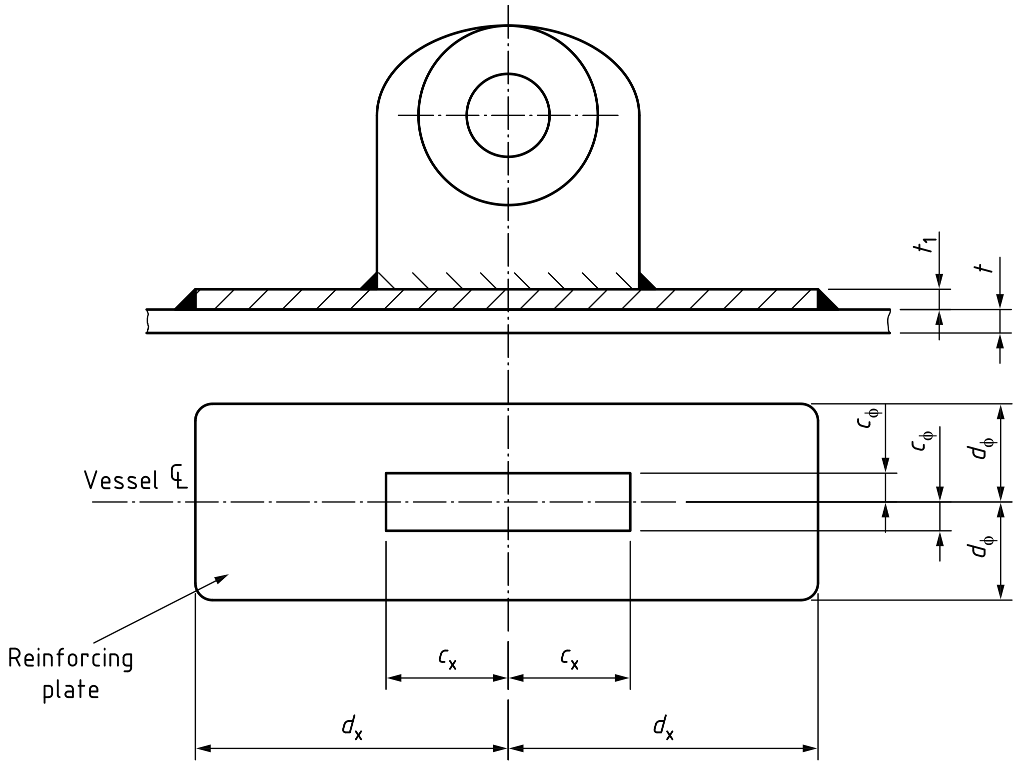


a) Bracket for vessel support

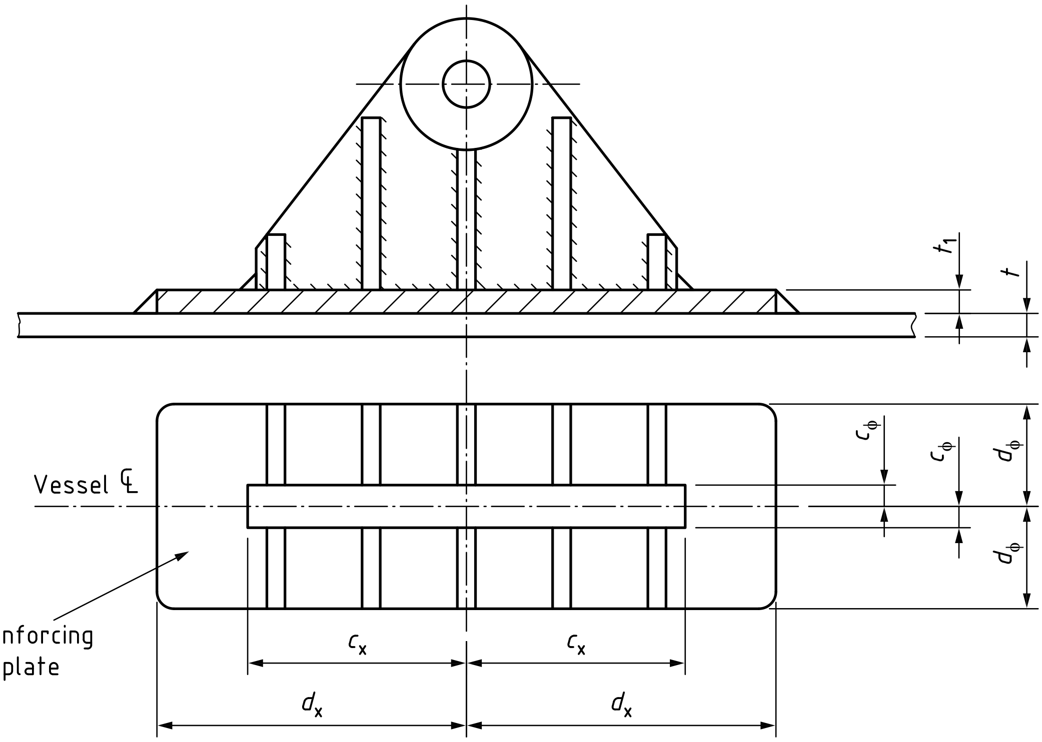


b) Bracket for external load

Figure G.3.1-2 **Typical reinforcing plates on cylindrical shells**



a) Simple arrangement for radially inward load



b) Gusset arrangement for radially outward load/movement

G.3.1.5.5 Deflections

The deflection at a support or fitting provided with a reinforcing plate is approximately equal to the sum of the deflections of the wall of a cylinder or sphere of thickness (t + t_1) loaded over the area of the reinforcing plate. These are found from **G.2.2.4** for cylinders or **G.2.4.3** and **G.2.4.4** for spheres and spherical parts of vessel ends.

The slope due to an external moment can be found from the deflection calculated in this way by the method given in **G.2.3** and **G.2.4**.

G.3.2 Supports for vertical vessels

G.3.2.1 Introduction

This clause is concerned with the design of supports for vertical vessels except where the conventional methods of simple applied mechanics can be used directly.

The design of brackets used to connect the vessel to its supports is given in **G.3.1.4**.

G.3.2.2 Skirt supports

Skirt supports are recommended for large vertical vessels because they do not lead to concentrated local loads on the shell, they offer less constraint against differential expansion between the part of the vessel under pressure and its supports, and they reduce the effect of discontinuity stresses at the junction of the cylindrical shell and the bottom (but see [18] and [22]).

Skirt supports should have at least one inspection opening to permit examination of the bottom of the vessel unless this is accessible from below through supporting framing. Such openings may need to be compensated.

Skirt supports may also be applied to spherical vessels and to the spherical parts of vessel ends. The local stresses due to skirt supports in these positions should be calculated as in **G.2.4**.

G.3.2.2.1 Overturning moments on skirt supports

At any horizontal section of a skirt support, the maximum load per unit length of the skirt circumference is given by:

$$N_x = \frac{W}{2\pi r} \pm \frac{F\bar{y}}{\pi r^2} = \text{stress} \times \text{thickness of skirt} \qquad \text{(G.3.2-1)}$$

If there is a negative value of N_x anchor bolts will be necessary because there will be a net moment of $M = Wr_1 - F\bar{y}$ tending to overturn the vessel about the leeward edge of the skirt support flange.

For small vessels the anchor bolts can be designed on the assumption that the neutral axis of the bolt group lies along a diameter of the support flange, but this assumption leads to overdesign in the case of tall vessels with large overturning moments because the effect of the elasticity of the foundation, which produces an additional resisting moment, is neglected.

Suitable design procedures for such cases are given in [16].

G.3.2.2.2 Discontinuity stresses at skirt supports

The presence of a skirt support reduces the discontinuity stresses at the junction of the bottom and the vessel wall.

A procedure for calculating the actual discontinuity stresses and also the design of skirt supports for vessels subject to severe cyclic loading due to thermal stresses is given in [18].

G.3.2.3 Ring supports for vertical vessels

It is often convenient to support vertical vessels from steelwork by means of a ring support in a convenient position on the shell as shown in Figure G.3.2-1.

Such a ring support corresponds to one flange of a bolted joint with the "hub" of the flange extending on both sides and with the couple due to the bolts replaced by that due to the eccentricity between the supporting force and the vessel wall. Its thickness can therefore be determined by adapting the equations in **3.8** and the associated figures.

The stresses should be determined as for an integral flange (see **3.8.3.4**) except that one-half of the flange design moment only should be used in calculating the longitudinal hub stress S_H.

The stresses calculated in this way should not exceed the allowable values for the stresses in flanges specified in **3.8.3.4.2**.

All ring supports of this type should rest on some form of continuous support or on steelwork as indicated in Figure G.3.2-2. They should not be used to connect vessels directly to leg or column supports, but should rest on a ring girder or other steelwork joining the tops of the columns.

G.3.2.4 Leg supports for vertical vessels

Leg supports for vertical vessels can, in general, be designed by the usual methods of applied mechanics, e.g. those described in chapter XXIII of [6].

They should always be arranged as close to the shell as the necessary clearance for insulation will permit.

If brackets are used to connect the legs to the vertical wall of the vessel as in Figure G.3.2-3 they should be designed as described in **G.3.1.4** and fitted with reinforcing plates if required.

Short legs, or legs braced to resist horizontal forces, may impose a severe constraint on a vessel wall due to differences in thermal expansion. This constraint can be avoided by using brackets on the vessel wall provided with slotted holes to allow for expansion. In addition, the mechanical loads at the points of support should be assessed and the local stresses due to these determined using the charts of **G.2.4**. Reinforcing pads designed as in **G.3.1.5** should be fitted if necessary.

Figure G.3.2-1 **Typical ring support**

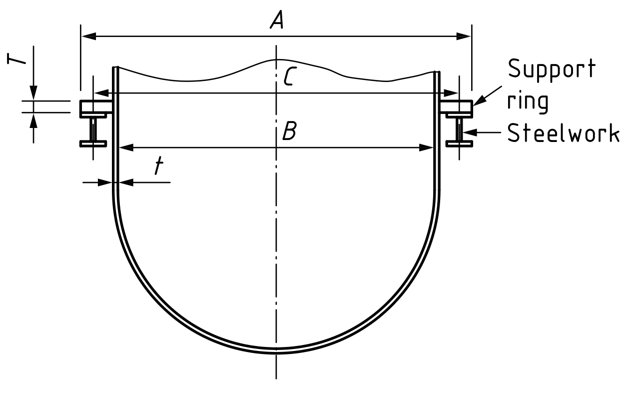


Figure G.3.2-2 **Typical steelwork under ring support**

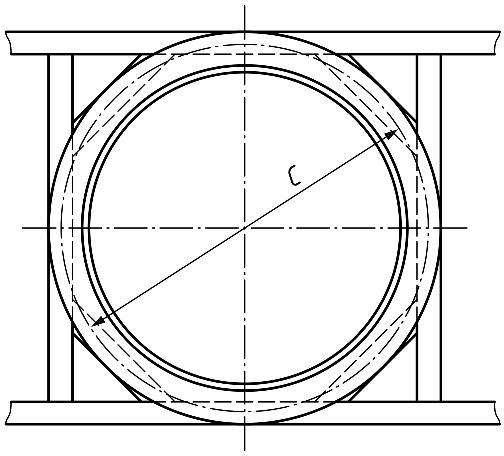
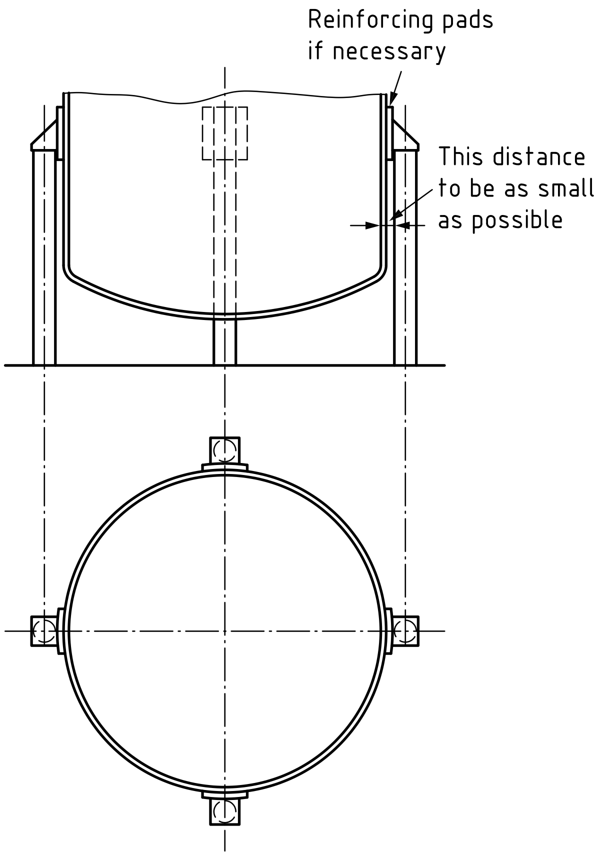


Figure G.3.2-3 **Leg supports for vertical vessels**



G.3.2.5 Ring girders

The supporting legs of large vertical vessels and spherical vessels are often connected to a ring girder that supports the vessel shell. In some designs the lower part of a skirt support is reinforced to form a ring girder. Figure G.3.2-4 shows a typical ring girder. Such girders are subject to torsion as well as bending and require special consideration.

When the supporting columns are equally spaced, the bending and twisting moments in the ring girder can be found from Table G.3.2-1, taken from [20].

A bending moment causing tension at the underside of the girder is taken as positive. The torsion in the girder is zero at the supports and midway between them and the bending moment is zero at the points of maximum torsion.

Table G.3.2-1 **Moments in a ring girder**

Number of legs	**4**	**6**	**8**	**12**
Load on each leg	$W/4$	$W/6$	$W/8$	$W/12$
Maximum shear in ring girder	$W/8$	$W/12$	$W/16$	$W/24$
$M_1/(Wr_2)$	– 0.034 2	– 0.014 8	– 0.008 27	– 0.003 65
$M_2/(Wr_2)$	+ 0.017 6	+ 0.007 51	+ 0.004 15	+ 0.001 90
x/r_2	0.335	0.222	0.166	0.111
$T/(Wr_2)$	0.005 3	0.001 5	0.000 63	0.000 185

G.3.3 Supports and mountings for horizontal vessels[4)]

G.3.3.1 General

Horizontal vessels are subject to longitudinal bending moments and local shear forces due to the weight of their contents, as well as to local stresses at supports and fittings.

They are conveniently supported on saddles, rings or leg supports (see Figure G.3.3-1).

When vessels are supported at more than two cross-sections the support reactions are significantly affected by small variations in the level of the supports, the straightness and local roundness of the vessel shell and the relative stiffness of different parts of the vessel against local deflections. Support at two cross-sections is thus to be preferred even if this requires stiffening of the support region of the vessel (see [19]).

Ring supports are preferable to saddle supports for vessels in which support at more than two cross-sections is unavoidable and for vacuum vessels. It may be necessary to provide ring supports for heavy fittings or structures supported from the vessel.

Vessels designed to contain gases or liquids lighter than water should be designed as vessels full of water when they are to be hydraulically tested.

Subclause **G.3.3.2** can be used to assess this design condition provided that the following three conditions are satisfied.

a) The stresses f_1 to f_{10} for the gas (or liquid) are to be limited to the values given in **G.3.3.2**, where the design stress f is derived at the design temperature. The stresses f_1 to f_{10} are calculated using the equations given in **G.3.3.2** where p_m is the design pressure at the equator (see **G.3.1.2**), the self weight includes both the vessel weight and the contents under the design conditions, with the wall thickness t equal to the analysis thickness.

b) The stresses f_1 to f_{10} for the hydraulic test when the vessel is just full of liquid with no internal pressure are to be limited to the values given in **G.3.3.2**, where the design stress f is derived at the test temperature (usually ambient). The stress f_1 to f_{10} are calculated using the equations given in **G.3.3.2** where the wall thickness t is the nominal thickness at the time of the test reduced by any allowance for under tolerance.

4) For a derivation of the basic equations and constants in this clause see [37].

c) The stresses f_1 to f_4, calculated by the following procedure, are to be limited to 90% of the minimum specified yield or proof stress (as in **5.8.5.2**). The value of the test pressure p_t is calculated using **5.8.5.1**, with due regard to the requirements for vessels comprising a number of interconnected sections, where appropriate. This test pressure and self weight (test liquid and vessel weight) are then used to recalculate the stresses f_1 to f_4 (considered to be membrane stresses) using the equations given in **G.3.3.2**. The thickness t for both the pressure term $p_m r/(2t)$ and the terms containing M_3 and M_4 should be the nominal thickness at the time of the test reduced by any allowance for under tolerance.

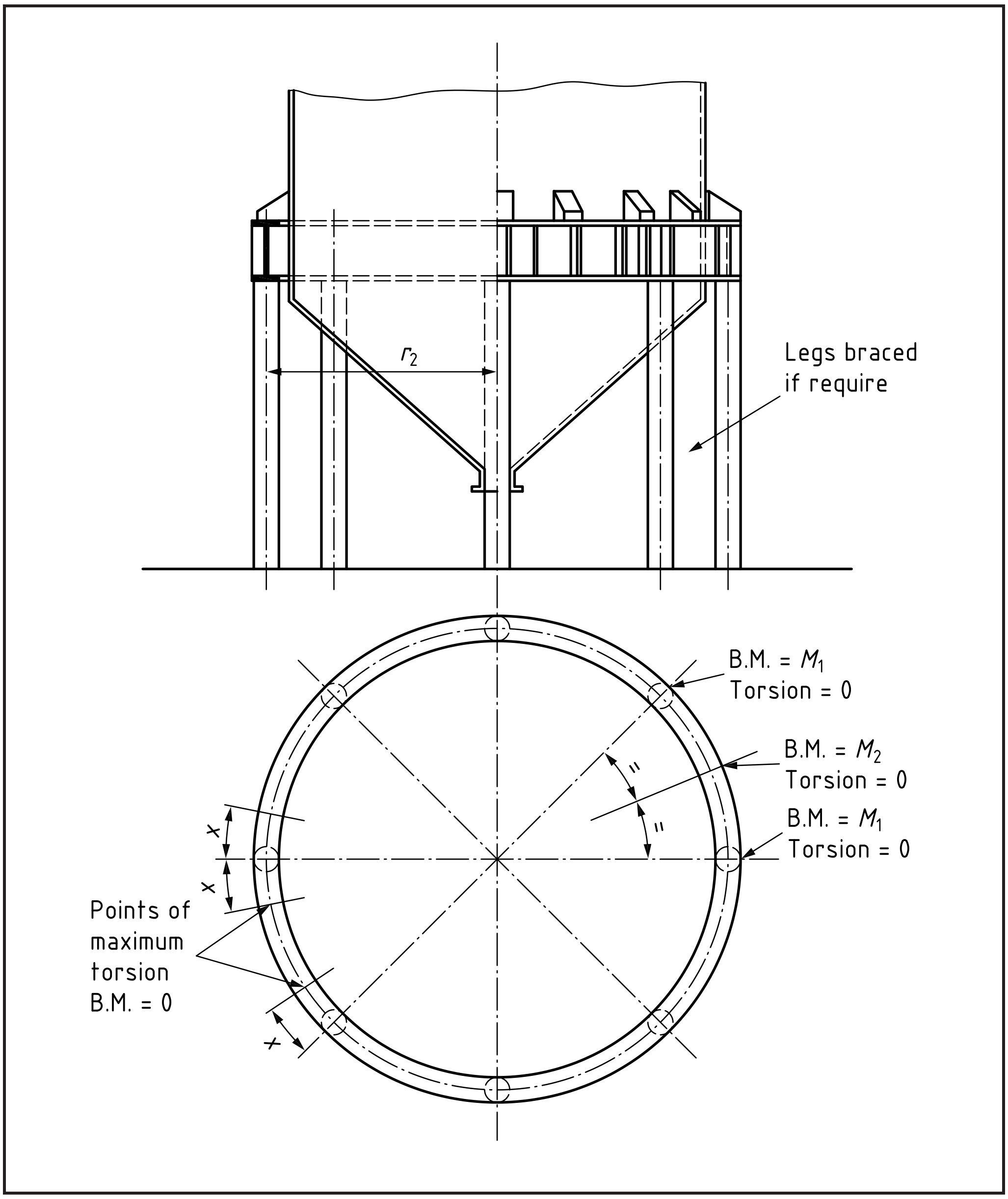
The use of leg supports only, as in Figure G.3.3-1c), should be confined to small vessels in which the longitudinal bending stresses are small compared with the axial stress due to pressure, and the local stresses due to the support reactions (found from **G.2**) can be kept within allowable limits.

Mountings and brackets fitted to the vessel to support external loads should be designed as described in **G.3.1**.

The shell thickness should not be less than that required for internal pressure in **3.5.1.2**a).

*NOTE Worked examples of the design of supports and mountings for horizontal vessels are given in **W.2**.*

Figure G.3.2-4 **Typical ring girder**



G.3.3.2 Saddle supports

G.3.3.2.1 Applicability

Figure G.3.3-1a) shows a horizontal vessel fitted with saddle supports. The methods given in **G.2** are not strictly applicable to loaded areas extending over the large proportion of the total circumference of the vessel which is usual for saddle supports.

The following treatment is based on an empirical analysis presented in [19] and extended based on experience with large-diameter thin-walled vessels with diameter to thickness ratios up to the order of 1 250:1. The analysis applies to saddles and rings welded to the vessel. For other cases where an alternative analysis is required, see **3.2.2**.

Loose rings or saddles depend critically upon fit for their effectiveness and require analysis by alternative computational methods (see [28]).

The method gives approximate values of stress which, together with the appropriate stress limits, provide a reasonable basis for design for non-cyclically loaded vessels.

In the case of vessels with significant cyclic loading, a rigorous analysis is required (see [28], [32], [38] and [39]). More accurate values of maximum stresses in the saddle support region of a horizontal vessel loaded by contents, for use in a fatigue assessment, can be calculated from **G.3.3.2.7**.

Maximum vessel stresses can occur when the vessel is full of liquid but not subject to internal pressure (see [19] and [21]) and this loading condition should be investigated.

In the case of large-diameter thin-walled vessels, the most arduous conditions can occur during filling. However, the methods presented, based on the full condition, produce designs which are satisfactory for the partially full condition.

The included angle of a saddle support [θ in Figure G.3.3-1a)] should normally be within the range $120° \leq \theta \leq 150°$. This limitation, which is imposed by most codes of practice, is an empirical one based on experience of large vessels. Saddle angles outside this range would require careful consideration.

When the supports are near the ends of the vessel ($A \leq r/2$) the stiffnesses of the ends tend to maintain circular support cross-sections and the shell is said to be stiffened by the ends.

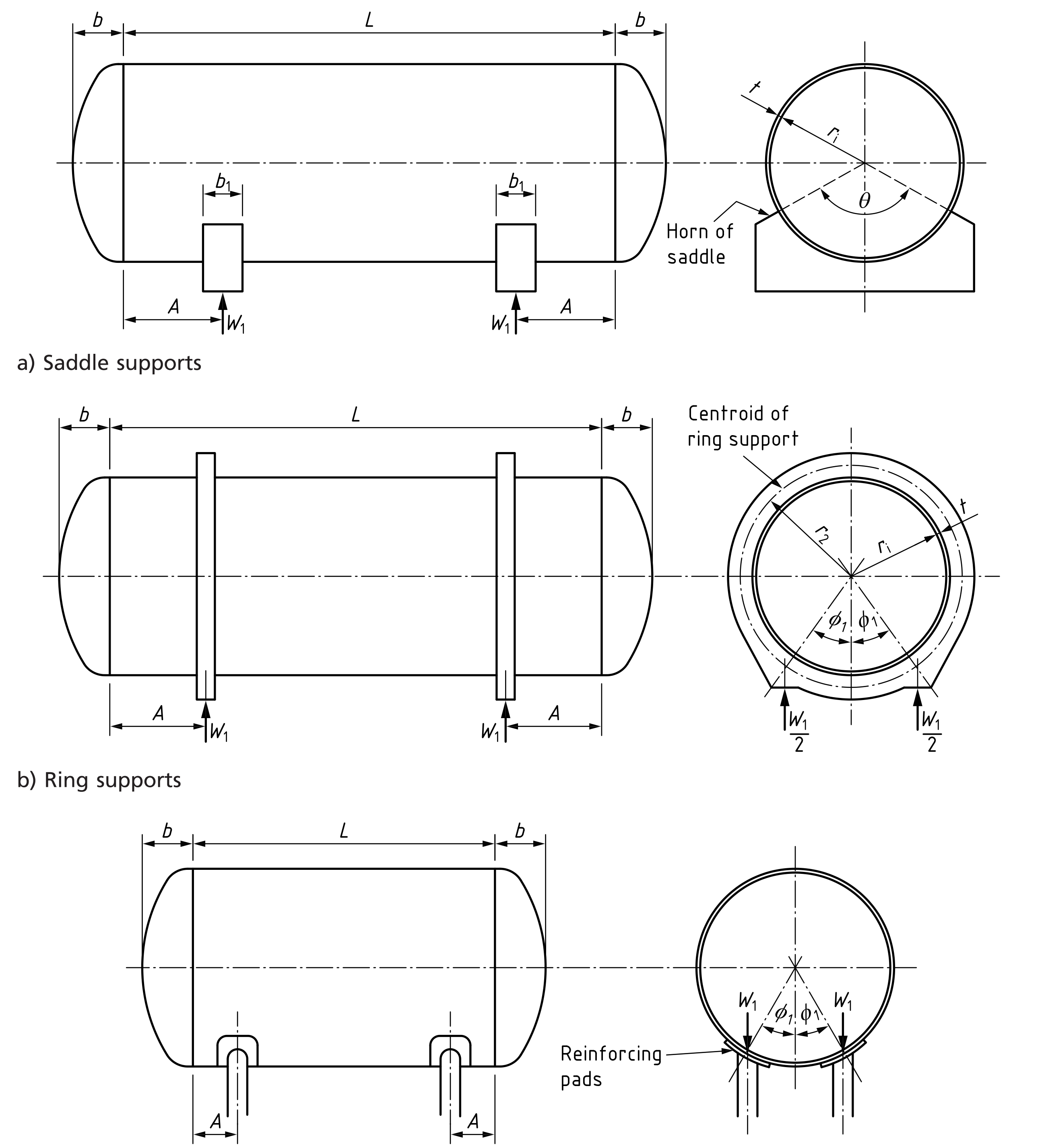
Where the stresses in the region of the support are found to exceed the allowable values a thickened strake may be used. The width of this should not be less than $\pm r/2$ about the centre saddle profile. That is, a total length equal to, or greater than, the radius of the vessel see [38] and [39].

NOTE 1 In providing a thickened strake in the region of the saddle it is assumed that the high stresses associated with the saddle have died away. The longitudinal and shear stresses at the stepped down thickness of the vessel may, therefore, be calculated using K_1 and K_2 = 1.0 and K_3 = 0.319.

Although the values of the bending moment and shear force at the stepped down thickness will be slightly less than M_4 and $W_1[(L - 2A)/(L + 4b/3)]$ respectively, it is recommended that the full values of these are used in Equations (G.3.3-7), (G.3.3-8) and (G.3.3-9) with the values of the constants quoted above.

NOTE 2 A range of standardized saddle supports welded to pressure vessels is included in [42].

Figure G.3.3-1 **Typical supports for horizontal vessels**



G.3.3.2.2 Longitudinal bending moments

Figure G.3.3-2 shows the loads, reactions and longitudinal bending moments in a vessel resting on two symmetrically placed saddle supports. The bending moments are given by the following equations (see [19] and [37]):

at mid-span

$$M_3 = \frac{W_1 L}{4}\left[\frac{1+\dfrac{2\left(r^2-b^2\right)}{L^2}}{1+\dfrac{4b}{3L}} - \frac{4A}{L}\right] \quad \text{(G.3.3-1)}$$

at supports

$$M_4 = -W_1 A\left[1-\frac{1-\dfrac{A}{L}+\left(\dfrac{r^2-b^2}{2AL}\right)}{1+\dfrac{4b}{3L}}\right] \quad \text{(G.3.3-2)}$$

A positive bending moment found from these equations is one causing tension at the lowest point of the shell cross-section. The moment M_4 may be positive in vessels of large diameter with supports near the ends because of the effect of hydrostatic pressure (see Figure G.3.3-2).

When L/r and b/r are known, these reduce to:

$$M_3 = W_1(C_1 L - A) \quad \text{(G.3.3-3)}$$

where

C_1 is a factor obtained from Figure G.3.3-3, and

$$M_4 = \frac{W_1 A}{C_2}\left[1-\frac{A}{L}+C_3\frac{r}{A}-C_2\right] \quad \text{(G.3.3-4)}$$

where C_2 and C_3 are factors obtained from Figure G.3.3-4.

Similar expressions for the longitudinal bending moments can be obtained by the ordinary methods of statics for vessels in which the supports are not symmetrically placed.

G.3.3.2.3 Longitudinal stresses at mid-span

The resultant longitudinal stresses at mid-span due to pressure and bending are given by the following equations:

at the highest point of the cross-section

$$f_1 = \frac{p_m r}{2t} - \frac{M_3}{\pi r^2 t} \quad \text{(G.3.3-5)}$$

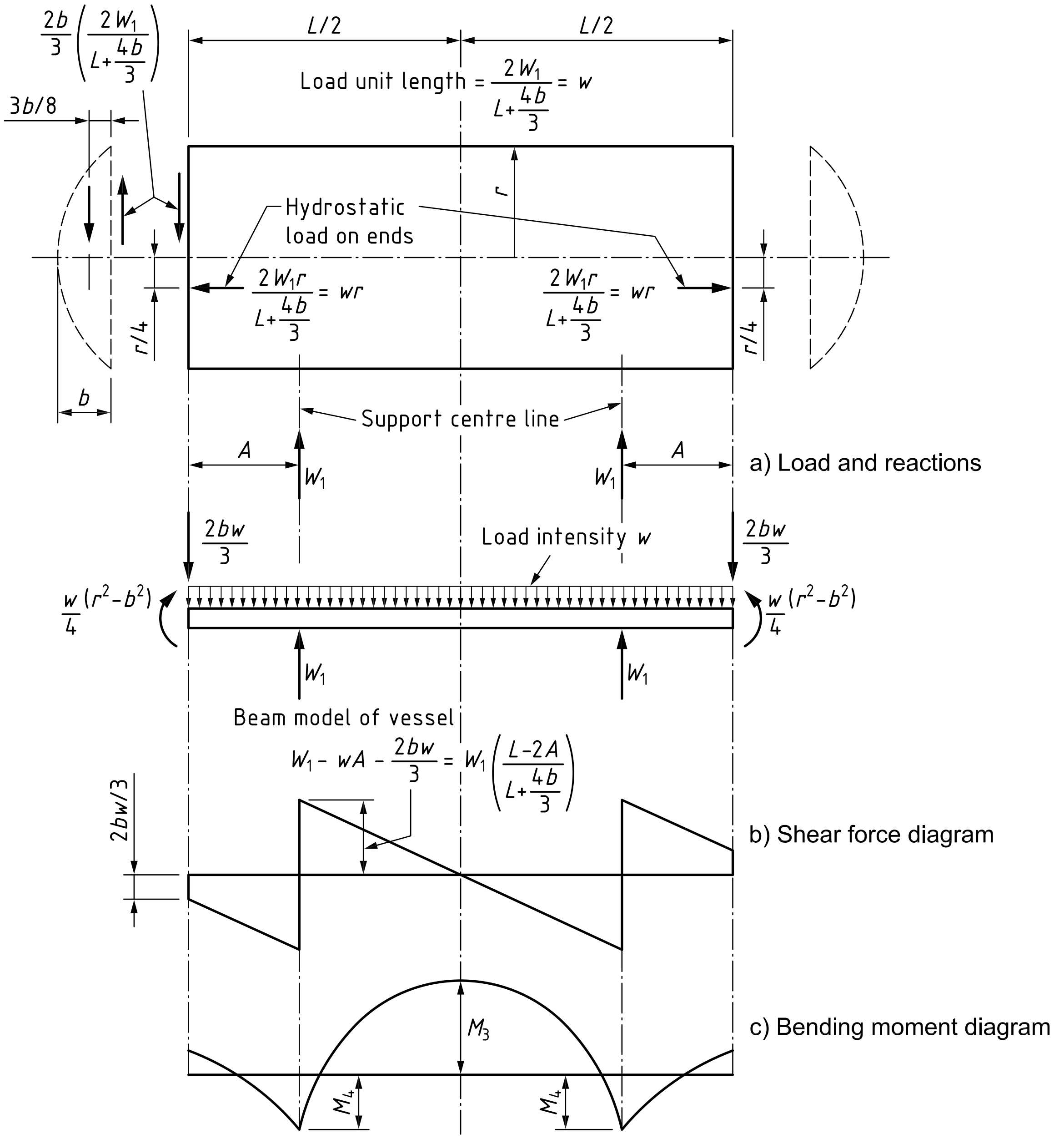
at the lowest point of the cross-section

$$f_2 = \frac{p_m r}{2t} + \frac{M_3}{\pi r^2 t} \quad \text{(G.3.3-6)}$$

These equations are based on simple beam theory which assumes that cross-sections remain circular.

The calculated tensile and compressive stresses should not exceed the values permitted in **A.3.4.2.1** and **A.3.5**.

Figure G.3.3-2 **Cylindrical shell acting as beam over supports**



Positive values of M_4 are obtained for the following forms and proportions:

flat ends	$A/r < 0.707$
ends with 10 % knuckle radius	$A/r < 0.44$
semi-ellipsoidal ends 2:1 ratio	$A/r < 0.363$

M_4 is always negative for hemispherical ends.

The dimension $3b/8$ is an approximation for the distance from the tangent plane to the centre of gravity of the dished end and its contents for all ranges of dished end covered by this specification.

Figure G.3.3-3 **Factor for bending moment at mid-span**

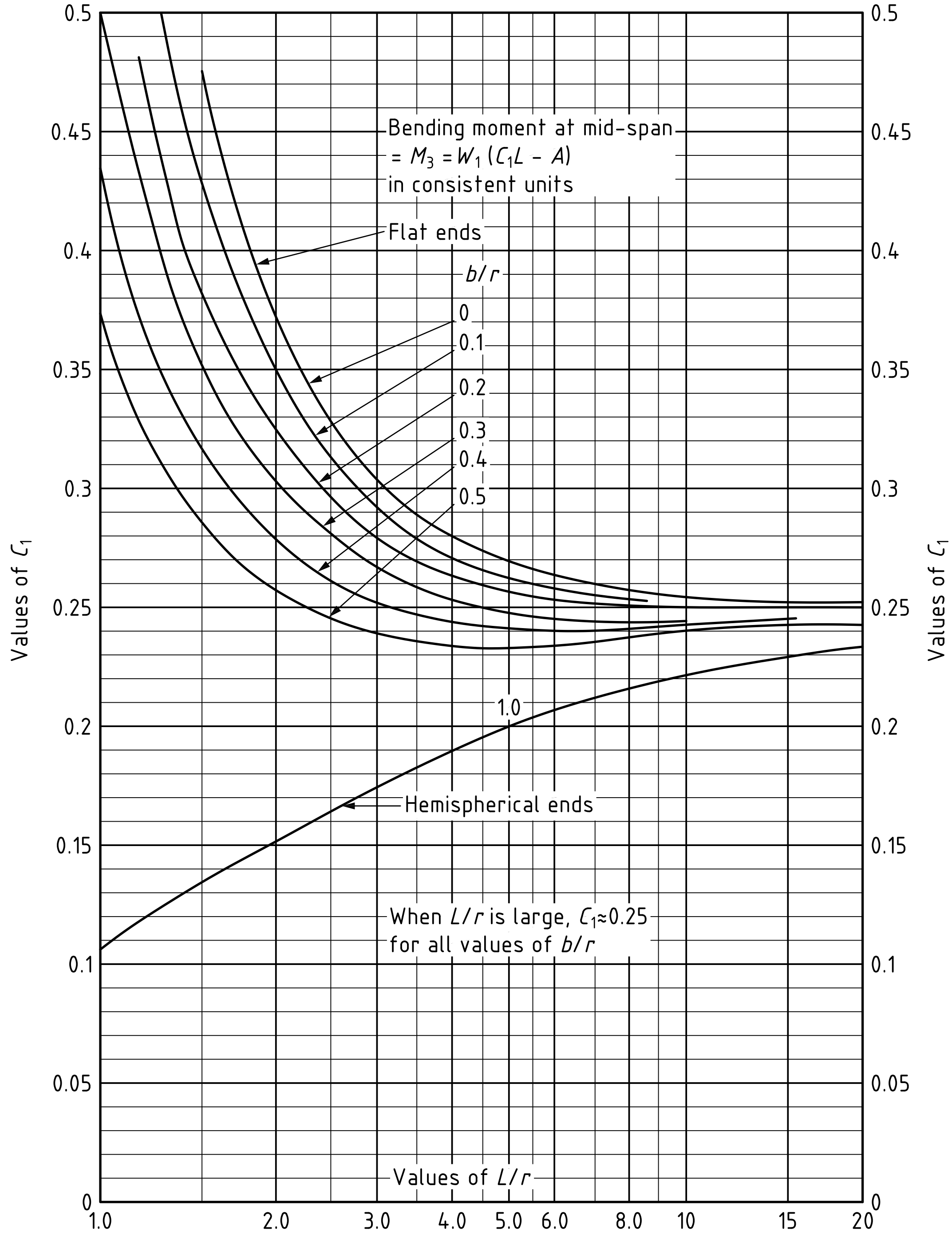
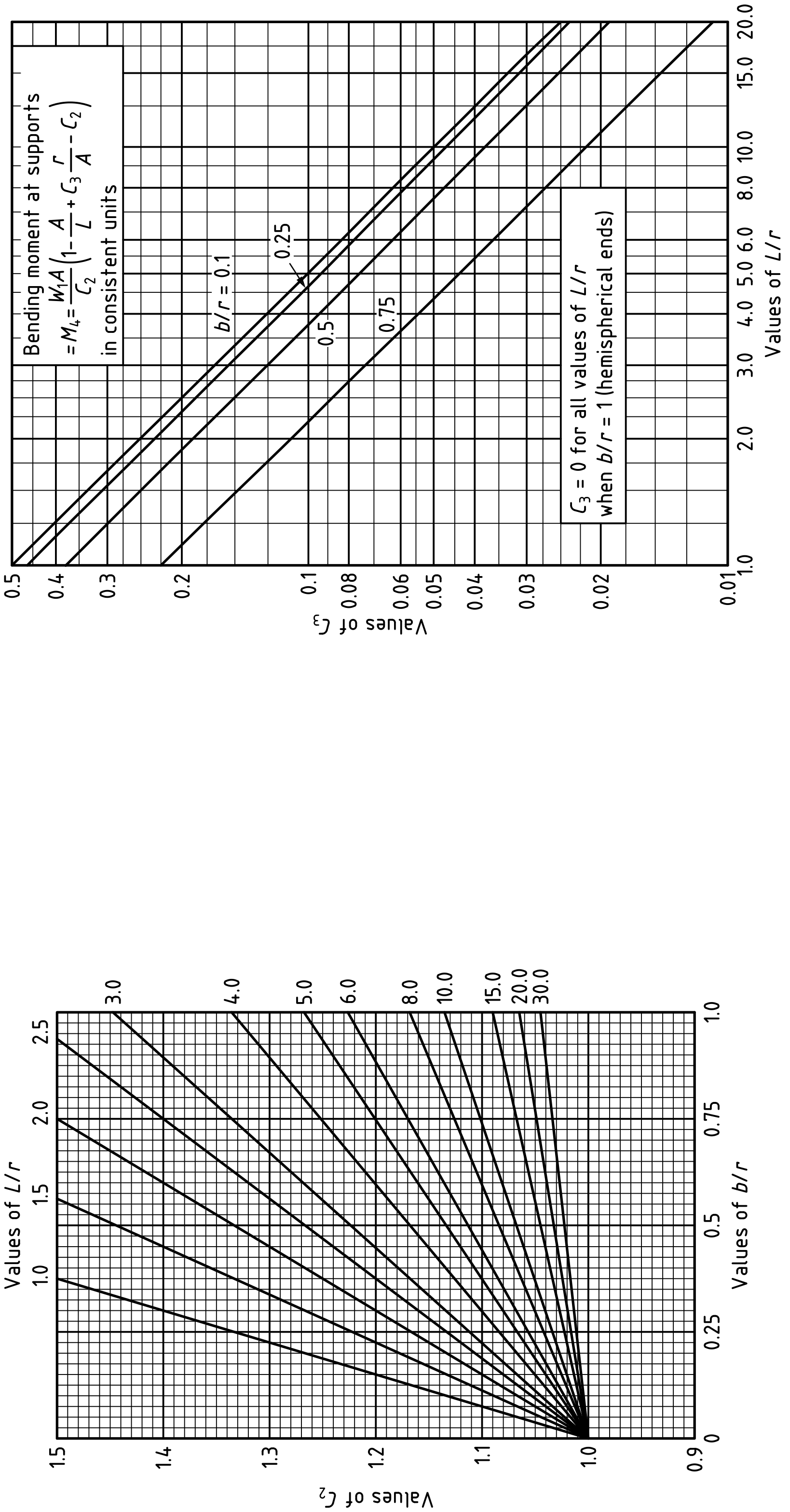


Figure G.3.3-4 **Factors for bending moment at supports**



a) Factor C_2

b) Factor C_3

G.3.3.2.4 Longitudinal stresses at the saddles

Longitudinal stresses at the saddles depend upon the local stiffness of the shell in the plane of the supports because, if the shell does not remain round under load, a portion of the upper part of its cross-section, as shown diagrammatically in Figure G.3.3-5, is ineffective against longitudinal bending (see [19]).

The resultant longitudinal stresses due to pressure and weight should be evaluated at two positions as follows.

a) *Either*

1) at the highest point of the cross-section when the shell is stiffened by rings or by proximity of the ends,

i.e. $A \le r/2$;

or

2) near the equator when the shell is unstiffened.

In both cases 1) and 2) the stress is given by:

$$f_3 = \frac{p_m r}{2t} - \frac{M_4}{K_1 \pi r^2 t} \qquad \text{(G.3.3-7)}$$

b) At the lowest point of the cross-section:

$$f_4 = \frac{p_m r}{2t} + \frac{M_4}{K_2 \pi r^2 t} \qquad \text{(G.3.3-8)}$$

Values of K_1 and K_2 are given in Table G.3.3-1.

The thickness of the saddle plate should not be included in the equations.

The calculated tensile and compressive stresses should not exceed the values permitted in **A.3.4.2.1** and **A.3.5**.

Figure G.3.3-5 **Portion of shell ineffective against longitudinal bending**

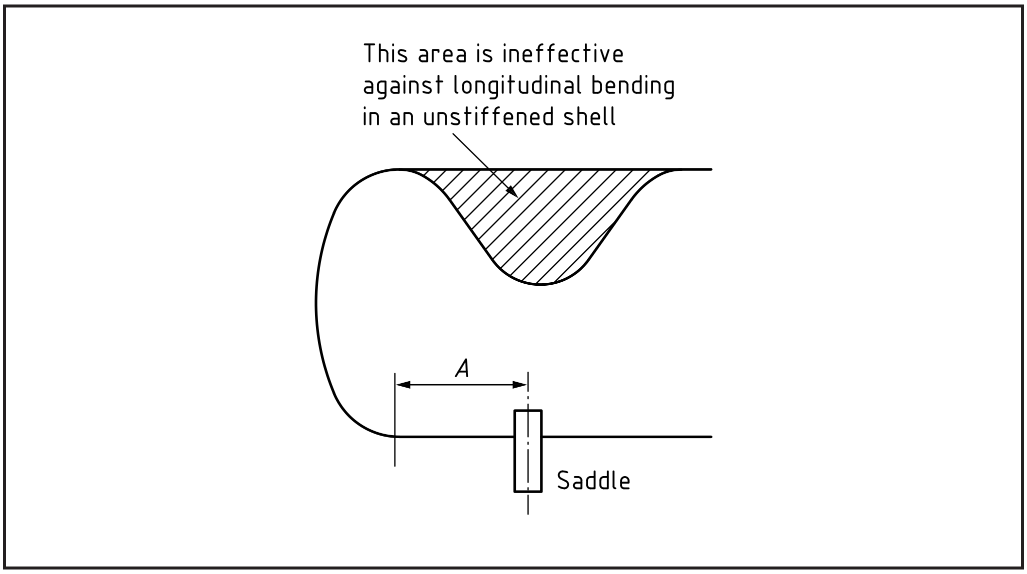


Table G.3.3-1 **Design factors K_1 and K_2**

Condition	Saddle angle θ (degrees)	K_1	K_2
Shell stiffened by end or rings, i.e. $A \le r/2$ or rings provided	120 135 150	1 1 1	1 1 1
Shell unstiffened by end or rings, i.e. $A > r/2$ and no rings provided	120 135 150	0.107 0.132 0.161	0.192 0.234 0.279

G.3.3.2.5 Tangential shearing stresses

Tangential shearing stresses are given by the following equations.

The values of K_3, K_4 and the allowable tangential shearing stress values are given in Table G.3.3-2.

The thickness of the saddle plate should not be included when using Equations (G.3.3-9) to (G.3.3-11).

a) Saddle not near vessel end ($A > r/2$), with or without rings added

$$q = \frac{K_3 W_1}{rt}\left[\frac{L - 2A}{L + \frac{4b}{3}}\right] \tag{G.3.3-9}$$

This equation does not apply when $A > L/4$, but such proportions are unusual.

b) Saddle near vessel end ($A \le r/2$), without rings added.

In this case there are shearing stresses in both the shell and vessel end. They are given by:

1) in the shell

$$q = \frac{K_3 W_1}{rt} \tag{G.3.3-10}$$

2) in the end

$$q_e = \frac{K_4 W_1}{rt_e} \tag{G.3.3-11}$$

c) Saddle near vessel end ($A \le r/2$), with rings in the plane of the saddle.

When the shell is stiffened by rings in the plane of the saddle and $b_1 < A \le r/2$ or $b_1/2 < A \le b_1$, shear stresses are:

1) in the shell adjacent to the ring

$$q = \frac{W_1}{\pi rt} \sin \phi \tag{G.3.3-12}$$

where

ϕ is measured form the zenith (top) of the cylinder;

q is a maximum at $\phi = \pi/2$ in which case:

$$q = \frac{K_3 W_1}{rt}$$

2) in the end

$$q_e = \frac{K_4 W_1}{rt_e} \tag{G.3.3-13}$$

d) Saddle near vessel end ($A \leq r/2$), with rings adjacent to the saddle.

When rings are placed adjacent to the saddle, it is assumed that the shear stresses do not benefit from the rings, see Table G.3.3-2. In view of this, the most appropriate equations for the case when $b_1 < A \leq r/2$ or $b_1/2 < A \leq b_1$ are therefore those given for the shell stiffened by the dished end to the vessel.

1) In the shell

In the region of $0 < \phi < \alpha$

$$q = \frac{W_1}{\pi rt} \sin \phi \tag{G.3.3-14}$$

In the region of $\alpha < \phi < \pi$

$$q = \frac{W_1(\alpha - \sin\alpha\cos\alpha)\sin\phi}{\pi rt(\pi - \alpha + \sin\alpha\cos\alpha)} \tag{G.3.3-15}$$

where

$$\alpha = \frac{19}{20}\left(\pi - \frac{\theta}{2}\right)$$

ϕ is measured from the zenith (top) of the cylinder;

θ is the included angle of the saddle support [see Figure G.3.3-1a)];

q is a maximum at $\phi = \alpha$, in which case $q = K_3 W_1/(rt)$.

2) In the end

$$q_e = \frac{K_4 W_1}{rt_e} \tag{G.3.3-16}$$

NOTE When rings are present, the shear stress in the end, q_e, for cases c) and d) will be reduced from that given by Equation (G.3.3-11). However, for simplicity, it is recommended that the procedure given by Equation (G.3.3-11) be adopted for these cases.

G.3.3.2.6 Circumferential stresses

G.3.3.2.6.1 Introduction

Figure G.3.3-6 shows the circumferential bending moments diagrammatically.

Circumferential stresses should be calculated using the equations given in **G.3.3.2.6.2** for shells not stiffened by rings, or **G.3.3.2.6.3** for shells stiffened by rings.

These stresses may be reduced if necessary by extending the saddle plate as shown in Figure G.3.3-7. It is recommended that the thickness of the saddle plate in this case should be equal to the thickness of the shell plate. The width of the saddle plate should be not less than $b_2 = b_1 + 10t$ and the angle it subtends should not be less than (θ + 12°).

The saddle plate should be welded to the vessel shell or the saddle, or both. If the saddle plate is welded to both the vessel and the saddle then the saddle is treated as being welded to the vessel, otherwise the saddle is treated as not being welded to the vessel.

When the saddle is welded to the vessel the absolute value of f_5 should not exceed f. For saddles without an extended saddle plate f is the nominal design stress for the shell material and for saddles with an extended saddle plate f is the smaller of the nominal design stresses for the shell and saddle plate materials.

When the saddle is not welded to the vessel, the value of f_5 should not exceed $\varepsilon E/3$, where E is the modulus of elasticity of the vessel shell and ε is the circumferential buckling strain which is obtained from Figure 3.6-2 or from Equation (3.6.2-9), which in turn uses n_{cyl} from Figure 3.6-3. In this derivation the value of $L/(2R)$ always equals 0.2 in Figure 3.6-2 and Figure 3.6-3, and in Equation (3.6.2-9), i.e. $Z = 2.5\pi$. In Figure 3.6-2 and Figure 3.6-3 the value of $e/(2R)$ is calculated taking the thickness $e = t$, the analysis thickness of the vessel shell.

NOTE The background to this design method is given in [44].

The allowable values of f_6, f_7 and f_8 are given separately for each saddle configuration in the following calculation procedures.

It has recently been shown that peak stresses in the shell at the horn of the saddle can be reduced by introducing some flexibility into the saddle design in the region of the saddle horn (see [28] and [32]).

Table G.3.3-2 **Design factors K_3 and K_4 and allowable tangential shearing stresses**

Component	**Condition**	**Saddle angle θ (degrees)**	**Factor**	
Vessel shell K_3			$A > r/2$	$A \le r/2$
	Shell unstiffened by rings	120 135 150	1.171 0.958 0.799	0.880 0.654 0.485
	Shell stiffened by rings in plane of saddles	120 135 150	0.319 0.319 0.319	0.319 0.319 0.319
	Shell stiffened by rings adjacent to saddles	120 135 150	1.171 0.958 0.799	0.880 0.654 0.485
Vessel end K_4			$b_1/2 < A \le b_1$	$b_1 < A \le r/2$
	Shell stiffened by end of vessel	120 135 150	0.880 0.654 0.485	0.401 0.344 0.295
Allowable tangential shearing stresses (see Note 1)	**Vessel shell**	**Vessel end**		
Design	min ($0.8f$; $0.06Et/r$)	$1.25f - f_{n(d)}$ (see Note 2)		

NOTE 1 Allowable tangential shearing stress values are derived from strain gauge tests on large vessels (see [19]) and experience with large diameter thin walled vessels.

NOTE 2 The nominal maximum tensile stress in dished end due to internal pressure, $f_{n(d)}$ can be found from Figure 3.5-2 using appropriate values of h_e/D and e/D to give p/f and hence $f_{n(d)} = p/(p/f)$ where e is the vessel end analysis thickness.

G.3.3.2.6.2 Shell not stiffened by rings

The circumferential stresses are calculated as follows.

a) Shell not provided with an extended saddle plate, as shown in Figure G.3.3-7a)

1) At the lowest point of the cross-section:

$$f_5 = \frac{-K_5 W_1}{t b_2} \qquad \text{(G.3.3-17)}$$

When the saddle is not welded to the vessel, the value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle. When the saddle is welded to the vessel, K_5 may be taken as one-tenth of this value.

2) At the horn of the saddle [see Figure G.3.3-1a)]:

for $\frac{L}{r} \geq 8$, then $$f_6 = \frac{-W_1}{4 t b_2} - \frac{3 K_6 W_1}{2 t^2} \qquad \text{(G.3.3-18)}$$

for $\frac{L}{r} < 8$, then $$f_6 = \frac{-W_1}{4 t b_2} - \frac{12 K_6 W_1 r}{L t^2} \qquad \text{(G.3.3-19)}$$

Values for K_6 are given in Table G.3.3-3.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the nominal design stress for the shell material.

If the stresses are unacceptable then the width and/or the included angle of the saddle should be increased and the calculations repeated, or alternatively an extended saddle plate may be provided and calculations carried out in accordance with b) below or rings may be provided and an analysis carried out in accordance with **G.3.3.2.6.3**.

b) Shell provided with an extended saddle plate, as shown in Figure G.3.3-7b).

If the width of the saddle plate is not less than b_2 and it subtends an angle not less than (θ + 12°), the circumferential stresses at the edge of the saddle and at the edge of the saddle plate are calculated as follows.

At the edge of the saddle when the saddle plate is welded to the vessel

1) At the lowest point of the cross-section:

$$f_5 = \frac{-K_5 W_1}{(t + t_1) b_2} \qquad \text{(G.3.3-20)}$$

When the saddle is not welded to the vessel, the value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle. When the saddle is welded to the vessel, K_5 may be taken as one-tenth of this value.

2) At the horn of the saddle [see Figure G.3.3-1 a)]

for $\frac{L}{r} \geq 8$, then $$f_6 = \frac{-W_1}{4(t + t_1) b_2} - \frac{3 K_6 W_1}{2(t + t_1)^2} \qquad \text{(G.3.3-21)}$$

for $\frac{L}{r} < 8$, then $$f_6 = \frac{-W_1}{4(t + t_1) b_2} - \frac{12 K_6 W_1 r}{L(t + t_1)^2} \qquad \text{(G.3.3-22)}$$

Values for K_6 are given in Table G.3.3-3.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the minimum of the design stress values for the shell and for the saddle plate.

At the edge of the saddle when the saddle plate is not welded to the vessel

NOTE In this case the saddle is assumed to be welded to the saddle plate but is not classed as being welded to the vessel.

3) At the lowest point of the cross-section:

The stress f_5 in the shell at the edge of the saddle is calculated using Equation (G.3.3-17).

The value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to the saddle.

4) Stress in the shell at the horn of the saddle [see Figure G.3.3-1a)]:

for $\frac{L}{r} \geq 8$, then $$f_6 = \frac{-W_1}{4tb_2} - \frac{3K_6 W_1 t}{2(t^3 + t_1^3)} \quad \text{(G.3.3-23)}$$

for $\frac{L}{r} < 8$, then $$f_6 = \frac{-W_1}{4tb_2} - \frac{12K_6 W_1 rt}{L(t^3 + t_1^3)} \quad \text{(G.3.3-24)}$$

Values for K_6 are given in Table G.3.3-3.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the shell.

5) Stress in the saddle plate at the horn of the saddle [see Figure G.3.3-1a)]:

for $\frac{L}{r} \geq 8$, then $$f_6 = \frac{-3K_6 W_1 t_1}{2(t^3 + t_1^3)} \quad \text{(G.3.3-25)}$$

for $\frac{L}{r} < 8$, then $$f_6 = \frac{-12K_6 W_1 rt_1}{L(t^3 + t_1^3)} \quad \text{(G.3.3-26)}$$

Values for K_6 are given in Table G.3.3-3.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the saddle plate.

At the edge of the extended saddle plate

The stresses in the shell at the edge of the saddle plate are calculated using Equations (G.3.3-17) to (G.3.3-19). The value of t should be taken equal to the shell thickness; b_2 is assumed to be unchanged.

The value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to the saddle and values for K_6 are given in Table G.3.3-3. The saddle angle θ may now include the angle of the saddle plate up to but not exceeding +12°.

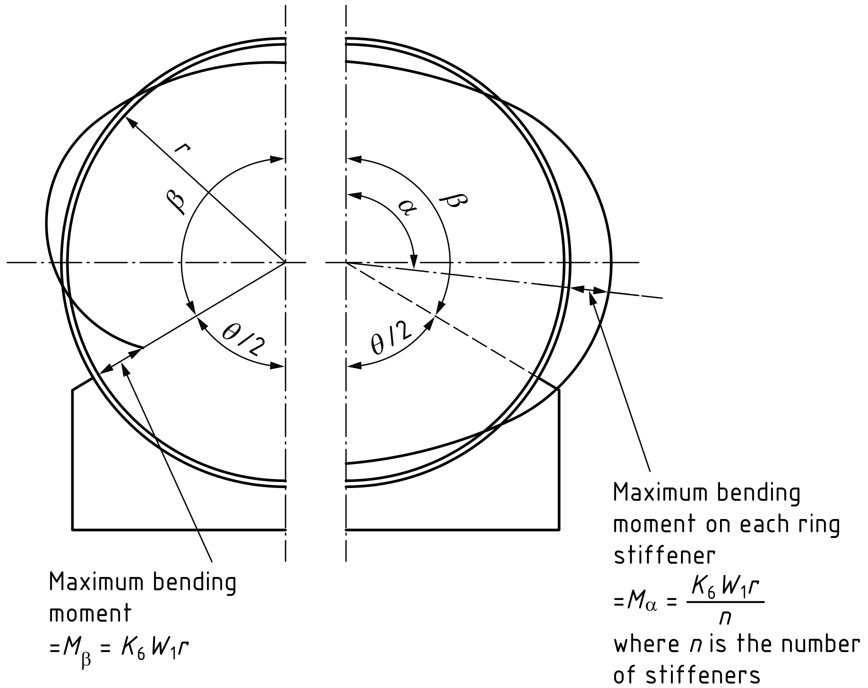
The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the shell.

Table G.3.3-3 **Design factor K_6**

A*/*r	**θ (degrees)**			
	120	**135**	**150**	**165**
≤0.50	0.0132	0.0103	0.0079	0.0059
≥1.00	0.0528	0.0413	0.0316	0.0238

NOTE For 0.50 < A/r < 1.00 values of K_6 should be obtained by linear interpolation of the values in this table.

Figure G.3.3-6 **Circumferential bending moment diagrams**



a) For no stiffener or for ring stiffener in plane of saddle

b) For ring stiffeners adjacent to saddle

G.3.3.2.6.3 Shell stiffened by rings (see Figure G.3.3-8)

Circumferential stresses should be calculated using the equations given in a) or b) following for a stiffening ring in the plane of the saddle [see Figure G.3.3-8a)], or using the equations given in c) or d) following for stiffening rings adjacent to the saddle [see Figure G.3.3-8b and Figure G.3.3-8c)]. Positive values denote tensile stresses and negative values denote compression.

Values of C_4, C_5, K_7, and K_8 are given in Table G.3.3-4. The effective cross-sectional area, *a*, of the stiffener (or stiffeners) and the portion of the shell that can be assumed to act with it (them) is indicated by the shaded areas in Figure G.3.3-8.

The second moment of area, *I*, is taken about the x-x axis parallel to the axis of the shell and through the centroid of the shaded area.

The stiffeners shown in Figure G.3.3-8 are of rectangular section. Stiffeners of other sections may be used if preferred.

When several stiffeners are used, as in Figure G.3.3-8b) and Figure G.3.3-8c), the values of *I* and *a* are for the sum of the shaded areas.

When two ring stiffeners are being used, it is essential that these be placed adjacent to the saddle and can be welded to either the inside or the outside of the shell as shown in Figure G.3.3-8b) and Figure G.3.3-8c).

It is essential that the axial length of shell between the stiffeners be not less than b_1 plus 10 times the shell thickness and not more than the mean radius of the shell. In this case, it is essential that a further check on the magnitude of f_6 be made assuming the value of K_6, from Table G.3.3-3, is that for $A/r \leq 0.50$.

a) A ring in the plane of the saddle and without an extended saddle plate

Stresses f_5 and f_6 are not applicable for a ring in the plane of the saddle.

Stresses f_7 and f_8 are calculated using Equations (G.3.3-27) and (G.3.3-28) with constants K_7 and K_8 based on an angle of θ. A thickness of t and axial length of shell of $t_2 + 10t$ are used to calculate I, a, c and d.

1) At the horn of the saddle, in the shell:

$$f_7 = \frac{C_4 K_7 W_1 rc}{I} - \frac{K_8 W_1}{a} \quad \text{(G.3.3-27)}$$

The absolute value of the circumferential stress f_7 should not exceed $1.25f$, where f is the design stress value for the shell.

2) At the horn of the saddle in the flange or tip of the ring remote from the shell:

$$f_8 = \frac{C_5 K_7 W_1 rd}{I} - \frac{K_8 W_1}{a} \quad \text{(G.3.3-28)}$$

The absolute value of the circumferential stress f_8 should not exceed $1.25f$, where f is the design stress value for the stiffening ring.

b) A ring in the plane of the saddle and with an extended saddle plate

Stresses f_5 and f_6 are not applicable for a ring in the plane of the saddle.

If the width of the saddle plate is not less than b_2 and it subtends an angle not less than (θ + 12°), stresses f_7 and f_8 are calculated as follows.

At the edge of the saddle when the saddle plate is welded to the vessel

If the saddle plate is welded to the vessel the stresses f_7 and f_8 are calculated using Equations (G.3.3-27) and (G.3.3-28) with constants K_7 and K_8 based upon an angle θ. A combined thickness equal to $(t + t_1)$, a length of shell of $t_2 + 10(t + t_1)$ and a length of saddle plate equal to the smaller of the actual width of the saddle plate or $t_2 + 10(t + t_1)$ may be used to calculate I, a, c and d.

The absolute value of the circumferential stress f_7 should not exceed $1.25f$, where f is the minimum of the design stress values for the shell and saddle plate.

The absolute value of the circumferential stress f_8 should not exceed $1.25f$, where f is the design stress value for the stiffening ring.

At the edge of the saddle when the saddle plate is not welded to the vessel

If the saddle plate is not welded to the vessel the stresses f_7 and f_8 are calculated using Equations (G.3.3-29) to (G.3.3-31) with constants K_7 and K_8 based upon an angle θ.

For the shell and internal stiffening ring I, a, c and d are calculated using a shell thickness of t and axial length of shell of $t_2 + 10t$. For the saddle plate I_1, a_1 and c_1 are calculated using the saddle plate thickness t_1 and an axial length of saddle plate equal to the smaller of the actual width of the saddle plate or $t_2 + 10t$.

1) At the horn of the saddle, in the shell:

$$f_7 = \frac{C_4 K_7 W_1 rc}{I + I_1} - \frac{K_8 W_1}{a + a_1} \quad \text{(G.3.3-29)}$$

2) At the horn of the saddle, in the saddle plate:

$$f_7 = \frac{C_4 K_7 W_1 rc_1}{I + I_1} - \frac{K_8 W_1}{a + a_1} \quad \text{(G.3.3-30)}$$

3) At the horn of the saddle in the flange or tip of the ring remote from the shell:

$$f_7 = \frac{C_5 K_7 W_1 rd}{I + I_1} - \frac{K_8 W_1}{a + a_1} \qquad \text{(G.3.3-31)}$$

At the edge of the extended saddle plate

The stresses at the edge of the saddle plate are calculated using Equations (G.3.3-27) and (G.3.3-28) with constants K_7 and K_8 based upon an angle θ + 12° and using a thickness t and a length of shell of $t_2 + 10t$ to calculate I, a, c and d.

The absolute value of the circumferential stress f_7 should not exceed 1.25f, where f is the design stress value for the shell.

The absolute value of the circumferential stress f_8 should not exceed 1.25f, where f is the design stress value for the stiffening ring.

c) Rings adjacent to the saddle and without an extended saddle plate

At the edge of the saddle

The stresses f_5 and f_6 are calculated using Equations (G.3.3-17) to (G.3.3-19) with constants K_5 and K_6 based upon an angle θ.

When the saddle is not welded to the vessel, the value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle. When the saddle is welded to the vessel, K_5 may be taken as one-tenth of this value.

The value of K_6 should correspond to that given in Table G.3.3-3 for $A/r \leq 0.50$.

The absolute value of the circumferential stress f_6 should not exceed 1.25f, where f is the design stress value for the shell.

In the stiffening ring centre profile

The stresses f_7 and f_8 are calculated using Equations (G.3.3-32) and (G.3.3-33) with constants K_7 and K_8 based on an angle of θ. A thickness of t and axial length of shell of $t_2 + 10t$ are used to calculate I, a, c and d.

1) Near the equator, in the shell:

$$f_7 = \frac{C_4 K_7 W_1 rc}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-32)}$$

The absolute value of the circumferential stress f_7 should not exceed 1.25f, where f is the design stress value for the shell.

2) Near the equator, in the flange or tip of the ring remote from the shell:

$$f_8 = \frac{C_5 K_7 W_1 rd}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-33)}$$

The absolute value of the circumferential stress f_8 should not exceed 1.25f, where f is the design stress value for the stiffening ring.

d) Rings adjacent to the saddle and with an extended saddle plate

If the width of the saddle plate is not less than b_2 and it subtends an angle not less than (θ + 12°), the circumferential stresses at the edge of the saddle and at the edge of the saddle plate are calculated as follows.

At the edge of the saddle when the saddle plate is welded to the vessel

The stresses f_5 and f_6 are calculated using Equations (G.3.3-20) to (G.3.3-22) with constants K_5 and K_6 based upon an angle θ.

When the saddle is not welded to the vessel, the value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle. When the saddle is welded to the vessel, K_5 may be taken as one-tenth of this value.

The value of K_6 should correspond to that given in Table G.3.3-3 for $A/r \leq 0.50$.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the minimum of the design stress values for the shell and for the saddle plate.

At the edge of the saddle when the saddle plate is not welded to the vessel

NOTE In this case the saddle is assumed to be welded to the saddle plate but is not classed as being welded to the vessel.

1) At the lowest point of the cross-section:

The stress f_5 in the shell at the edge of the saddle is calculated using Equation (G.3.3-17).

The value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle.

2) Stress in the shell at the horn of the saddle:

The stress f_6 in the shell at the edge of the saddle is calculated using Equations (G.3.3-23) and (G.3.3-24).

The value of K_6 should correspond to that given in Table G.3.3-3 for $A/r \leq 0.50$.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the shell.

3) Stress in the saddle plate at the horn of the saddle:

The stress f_6 in the saddle plate at the edge of the saddle is calculated using Equations (G.3.3-25) and (G.3.3-26).

The value of K_6 should correspond to that given in Table G.3.3-3 for $A/r \leq 0.50$.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the saddle plate.

At the edge of the extended saddle plate

The stresses in the shell at the edge of the saddle plate are calculated using Equations (G.3.3-17) to (G.3.3-19). The value of t should be taken equal to the shell thickness.

The value of K_5 should correspond to that given in Table G.3.3-4 for rings adjacent to saddle and the value of K_6 should correspond to that given in Table G.3.3-3 for $A/r \leq 0.50$. The saddle angle θ may now include the angle of the saddle plate up to but not exceeding +12°.

The absolute value of the circumferential stress f_6 should not exceed $1.25f$, where f is the design stress value for the shell.

In the stiffening ring centre profile

The stresses f_7 and f_8 are calculated using Equations (G.3.3-32) and (G.3.3-33) with constants K_7 and K_8 based on an angle of θ + 12°. A thickness of t and axial length of shell of $t_2 + 10t$ are used to calculate I, a, c and d.

The absolute value of the circumferential stress f_7 should not exceed $1.25f$, where f is the design stress value for the shell.

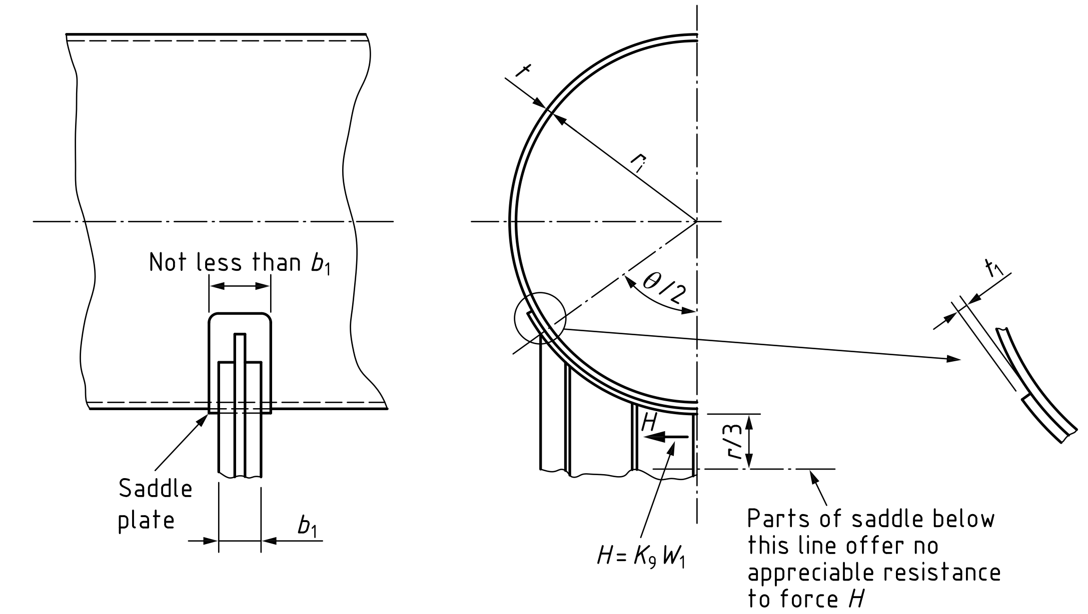
The absolute value of the circumferential stress f_8 should not exceed 1.25f, where f is the design stress value for the stiffening ring.

Table G.3.3-4 **Values of constants C_4, C_5, K_5, K_7, and K_8**

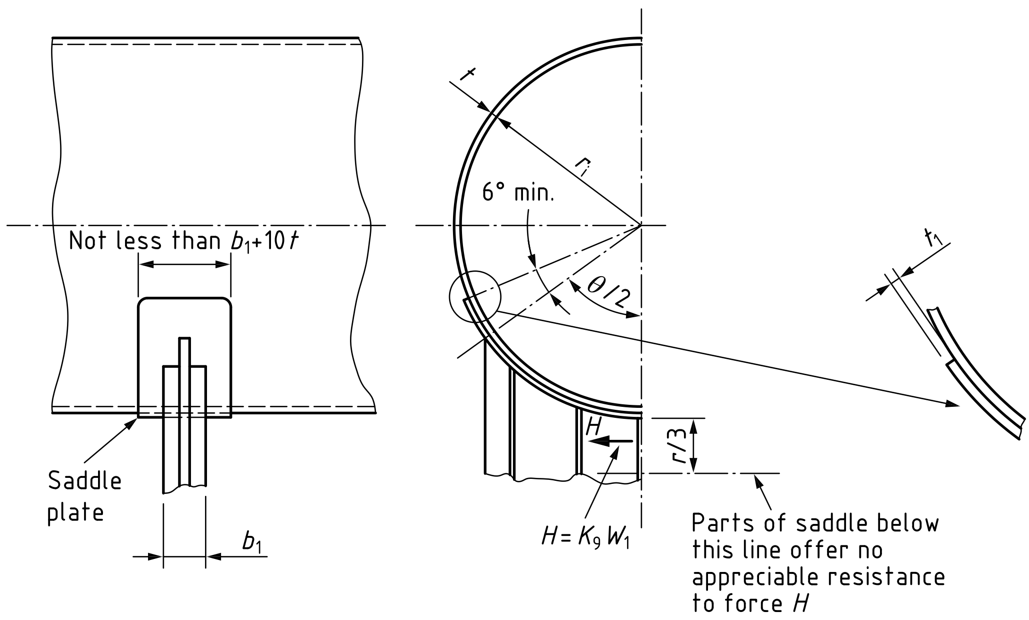
	Ring in plane of saddle				Rings adjacent to saddle							
	Internal ring [see Figure G.3.3-8a)]				Internal rings [see Figure G.3.3-8b)]				External rings [see Figure G.3.3-8c)]			
θ°	**120**	**135**	**150**	**165**	**120**	**135**	**150**	**165**	**120**	**135**	**150**	**165**
C_4	–1	–1	–1	–1	+1	+1	+1	+1	–1	–1	–1	–1
C_5	+1	+1	+1	+1	–1	–1	–1	–1	+1	+1	+1	+1
K_5	—	—	—	—	0.760	0.711	0.673	0.645	0.760	0.711	0.673	0.645
K_7	0.0528	0.0413	0.0316	0.0238	0.0581	0.0471	0.0355	0.0242	0.0581	0.0471	0.0355	0.0242
K_8	0.340	0.323	0.303	0.277	0.271	0.248	0.219	0.184	0.271	0.248	0.219	0.184

NOTE Intermediate values of K_5, K_7 and K_8 may be obtained by linear interpolation.

Figure G.3.3-7 **Saddle supports**

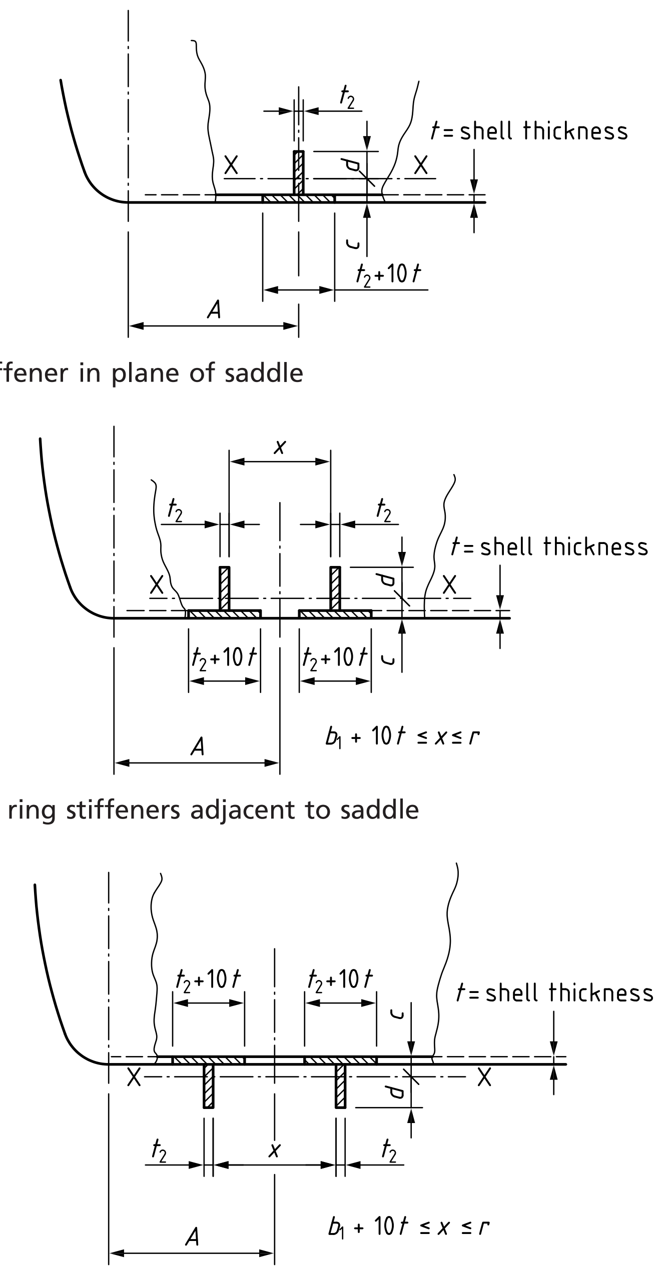


a) Simple saddle support



b) Saddle support with extended plate

Figure G.3.3-8 **Typical ring stiffeners**



a) Ring stiffener in plane of saddle

b) Internal ring stiffeners adjacent to saddle

c) External ring stiffeners adjacent to saddle

G.3.3.2.7 Design of saddles

The width b_1 of steel saddles [see Figure G.3.3-1a)] should be chosen to satisfy the circumferential stress limits as defined by Equations (G.3.3-17) to (G.3.3-33), where applicable. For most cases a width equal to $\sqrt{30D}$ (where D is the mean diameter of the vessel in mm) will be satisfactory.

The minimum section at the low point of a saddle (see Figure G.3.3-7) has to resist a force H equal to the sum of the horizontal components of the reactions on one-half of the saddle. The effective cross-section resisting this load should be limited to the metal cross-section within a distance equal to r/3 below the shell and the average direct stress on this cross-section should be limited to two-thirds of the allowable design stress.

$H = K_9 W_1$

where the values of K_9 are given in Table G.3.3-5.

The upper and lower flanges of a steel saddle should be thick enough to resist the longitudinal bending over the web or webs due to the bearing loads as in any machine support. The web should be stiffened against buckling due to vertical shear forces as for structural beams, and against bending due to longitudinal external loads on the vessel.

One saddle of each vessel should be provided with some form of sliding bearing or rocker in the following cases:

a) when steel saddles are welded to the vessel shell;

b) when large movements due either to thermal expansion or to axial strain in a long vessel are expected.

Table G.3.3-5 **Values of K_9**

$\theta°$	120	135	150	165
K_9	0.204	0.231	0.259	0.288

G.3.3.2.8 Maximum stresses for use in a fatigue assessment

G.3.3.2.8.1 General

The highest stressed region in a horizontal vessel supported on saddles and loaded by contents, where the shell is not stiffened by rings, is on the outside surface of the saddle centre profile at the uppermost point of the saddle, see references [47] and [48]. The membrane plus bending stress in this region is identified as f_6 in **G.3.3.2.6.1**, however this stress f_6 is not appropriate to be used in a fatigue assessment in accordance with Annex C. For this fatigue assessment the following procedure is used for the determination of a maximum stress, that when added to the stress due to pressure, is used in Annex C.

For the purposes of **G.3.3.2.8** the following symbols apply.

- A is the distance from the saddle support to the adjacent end of the cylinder (see Figure G.3.3-1);
- b_1 is the axial width of the saddle support (see Figure G.3.3-7);
- f_b is the "basic stress" [see Equation (G.3.3-34)];
- f_f is the maximum stress due to contents [see Equation (G.3.3-35)];
- F_A is a rigid end factor from Figure G.3.3-9 or Table G.3.3-6;
- F_B is a saddle width factor from Figure G.3.3-10 or Table G.3.3-7;
- F_I is a saddle interaction factor from Figure G.3.3-11 or Table G.3.3-8;
- F_L is a length change factor from Figure G.3.3-12 or Table G.3.3-9;
- F_θ is a saddle wrap-around factor from Figure G.3.3-13 or Table G.3.3-10;
- L is the length of the cylindrical part of the vessel;
- L_s is the distance between the saddle supports;
- r is the mean radius of the cylindrical part of the vessel;
- t is the analysis thickness of the cylinder;
- W is the reaction load at an individual saddle;
- x is a polynomial variable in Equation (G.3.3-36);
- θ is the included angle of the saddle support (see Figure G.3.3-1).

The procedure is subject to the following limitations:

a) 500 mm $\leq r \leq$ 4 000 mm;

b) 4 mm $\leq t \leq$ 30 mm;

c) $25 \leq (r/t) \leq 300$;

d) $0.1 \leq b_1/r \leq 0.5$;

e) $A \leq L/4$;

f) the vessel is full of contents.

G.3.3.2.8.2 Calculation of maximum stresses

A maximum stress in the saddle region has been derived using references [47] and [48] for a vessel of such length so as to isolate the influences of the dished end and the other saddle. This stress is referred to as the "basic stress" f_b, and is calculated from Equation (G.3.3-34). The "basic stress" is then multiplied by various factors to give the maximum stress, f_f, due to contents in an actual vessel W.

$$f_b = 1.17\frac{W}{t^2}\left(\frac{r}{t}\right)^{-0.32} \tag{G.3.3-34}$$

The maximum stress f_f is calculated from Equation (G.3.3-35).

$$f_f = f_b F_A F_B F_l F_L F_\theta \tag{G.3.3-35}$$

The geometric factors F_A, F_B, F_l, F_L, F_θ are given in Figure G.3.3-9 to Figure G.3.3-13 or can be derived from a fourth order polynomial of the form given in Equation (G.3.3-36).

$$F = a_0 + a_1x + a_2x^2 + a_3x^3 + a_4x^4 \tag{G.3.3-36}$$

The coefficients for the appropriate factor F, together with the definition of the relevant x, are given in Table G.3.3-6 to Table G.3.3-10. Linear interpolation in these tables, and between r/t values in the figures, is permissible. Extrapolation of the curves given in Figure G.3.3-9 to Figure G.3.3-13, within the limits specified in **G.3.3.2.8.1**, is possible using Equation (G.3.3-36).

The maximum stress due to contents, f_f, is used in a fatigue assessment in accordance with Annex C. This assessment should include the addition of any relevant stresses due to pressure. According to Annex C the fillet welds at the horn of the saddle are classified as class "G" or "G2" welds; see Table C.2d). The appropriate fatigue curve from Figure C.3 and Table C.1 is then used to derive the fatigue life.

Table G.3.3-6 **Coefficients for F_A, with $x = A/r$**

r/t	a_0	a_1	a_2	a_3	a_4
25	0.14621	0.55498	—0.13428	0.014198	—5.51E-4
50	0.08909	0.44585	—0.09151	0.009231	—3.69E-4
75	0.06572	0.38390	—0.07136	0.006990	—2.70E-4
100	0.05054	0.34826	—0.06336	0.006414	—2.64E-4
125	0.04093	0.32083	—0.05656	0.005812	—2.42E-4
150	0.03417	0.29953	—0.05098	0.005278	—2.21E-4
175	0.02947	0.28258	—0.04626	0.004787	—2.01E-4
200	0.02737	0.26214	—0.03867	0.003750	—1.51E-4
250	0.02098	0.24866	—0.03581	0.003600	—1.50E-4
300	0.01788	0.23465	—0.03126	0.003066	—1.27E-4

Table G.3.3-7 **Coefficients for F_B, with $x = b_1/r$**

r/t	a_0	a_1	a_2	a_3	a_4
25	1.733	−5.919	15.111	−21.742	12.875
50	1.757	−6.091	15.298	−21.308	12.208
75	1.789	−6.486	17.077	−24.850	14.833
100	1.805	−6.647	17.598	−25.533	15.167
125	1.828	−6.921	18.769	−27.675	16.625
150	1.838	−6.996	18.854	−27.358	16.125
175	1.855	−7.215	19.884	−29.408	17.625
200	1.873	−7.447	20.935	−31.400	19.000
250	1.890	−7.612	21.377	−31.617	18.750
300	1.917	−7.994	23.283	−35.550	21.667

Table G.3.3-8 **Coefficients for F_l, with $x = L_s/r$**

r/t	a_0	a_1	a_2	a_3	a_4
25	0.63783	0.016572	−0.002274	0.000266	−7.16E-6
50	0.66755	0.019146	−0.001266	0.000092	−1.33E-6
75	0.69860	0.008906	0.001551	−0.000140	4.36E-6
100	0.68000	0.007285	0.002474	−0.000214	5.87E-6
125	0.67898	0.012773	0.001992	−0.000183	4.89E-6
150	0.64491	0.021468	0.000935	−0.000112	3.04E-6
175	0.60577	0.030931	−0.000267	−0.000033	1.08E-6
200	0.56838	0.038209	−0.001084	0.000020	−0.21E-6
250	0.50338	0.046902	−0.001904	0.000076	−1.64E-6
300	0.45423	0.049776	−0.001978	0.000086	−2.02E-6

Table G.3.3-9 **Coefficients for F_L, with $x = L/r$**

r/t	a_0	a_1	a_2	a_3	a_4
25	−0.053702	0.065643	−0.002351	5.17E-5	−4E-7
50	−0.084688	0.065522	−0.001856	2.83E-5	−2E-7
75	−0.032241	0.041286	0.000617	−5.75E-5	8E-7
100	0.027463	0.018664	0.002856	−1.317E-4	16E-7
125	0.066444	0.003717	0.004032	−1.659E-4	19E-7
150	0.089579	−0.003488	0.004514	−1.767E-4	20E-7
175	0.100234	−0.005717	0.004505	−1.709E-4	19E-7
200	0.095531	−0.001799	0.003912	−1.455E-4	16E-7
250	0.084560	0.005450	0.002825	−9.97E-5	10E-7
300	0.059512	0.017537	0.001403	−4.57E-5	4E-7

Table G.3.3-10 **Coefficients for F_θ, with $x = \theta$ (in degrees)**

r/t	a_0	a_1	a_2	a_3	a_4
25	3.83590E+00	−3.79137E-02	1.57839E-04	−3.78768E-07	4.62000E-10
50	3.90484E+00	−3.87574E-02	1.59261E-04	−3.71204E-07	4.56000E-10
75	4.04122E+00	−4.31288E-02	2.15622E-04	−7.07569E-07	1.21700E-09
100	4.68385E+00	−6.82825E-02	5.71067E-04	−2.87133E-06	6.02800E-09
125	4.72200E+00	−7.10954E-02	6.41871E-04	−3.53371E-06	8.06900E-09
150	4.40982E+00	−5.76894E-02	4.39663E-04	−2.24354E-06	5.10500E-09
175	2.09028E+00	4.03490E-02	−1.04513E-03	7.38111E-06	−1.75450E-08
200	2.94342E+00	5.75797E-03	−5.45410E-04	4.31655E-06	−1.07920E-08
250	2.37512E+00	2.98110E-02	−9.11896E-04	6.70152E-06	−1.63930E-08
300	1.74637E+00	5.65080E-02	−1.32162E-03	9.38830E-06	−2.27430E-08

Figure G.3.3-9 **Graph of rigid end factor, F_A**

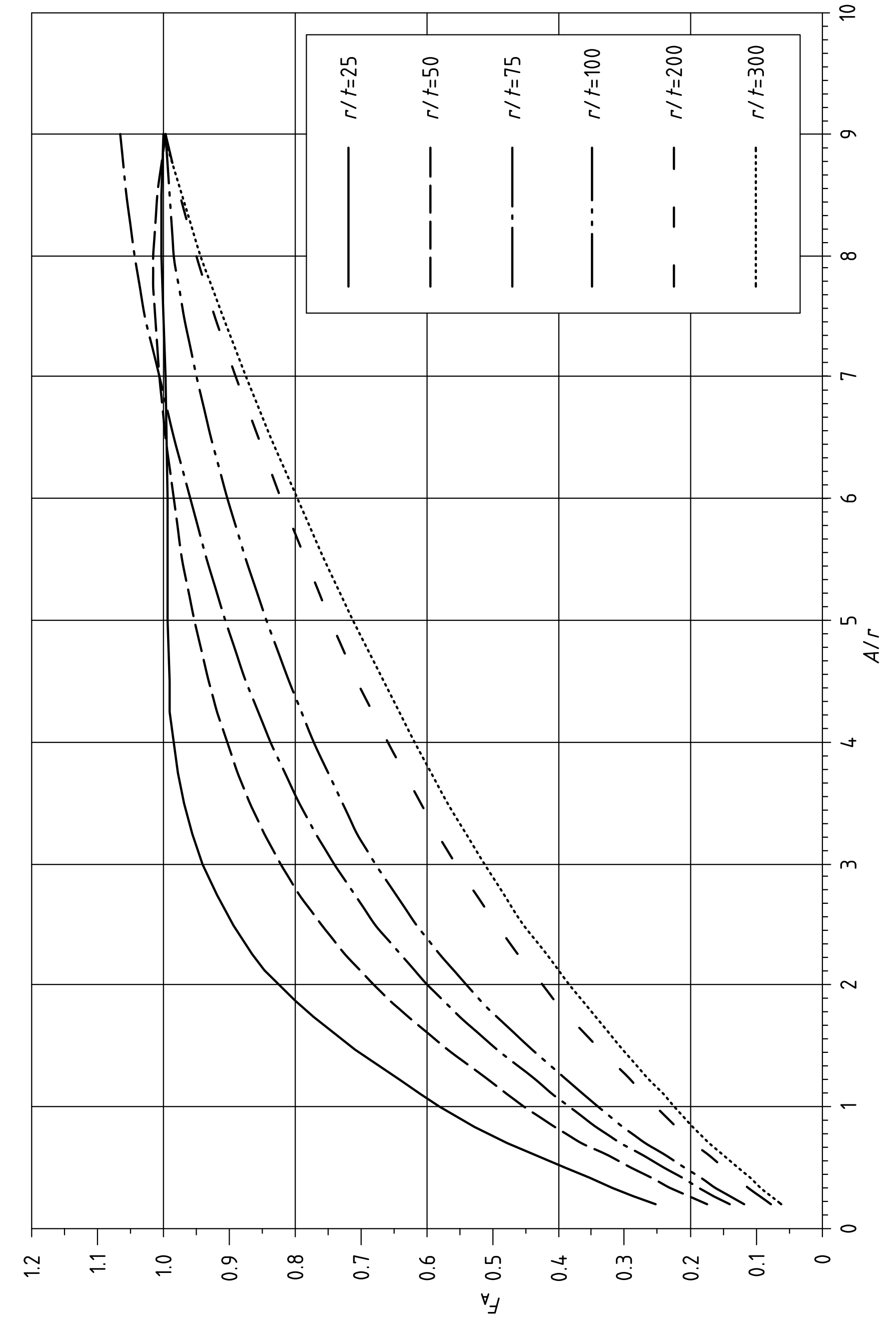


Figure G.3.3-10 **Graph of saddle width factor, F_B**

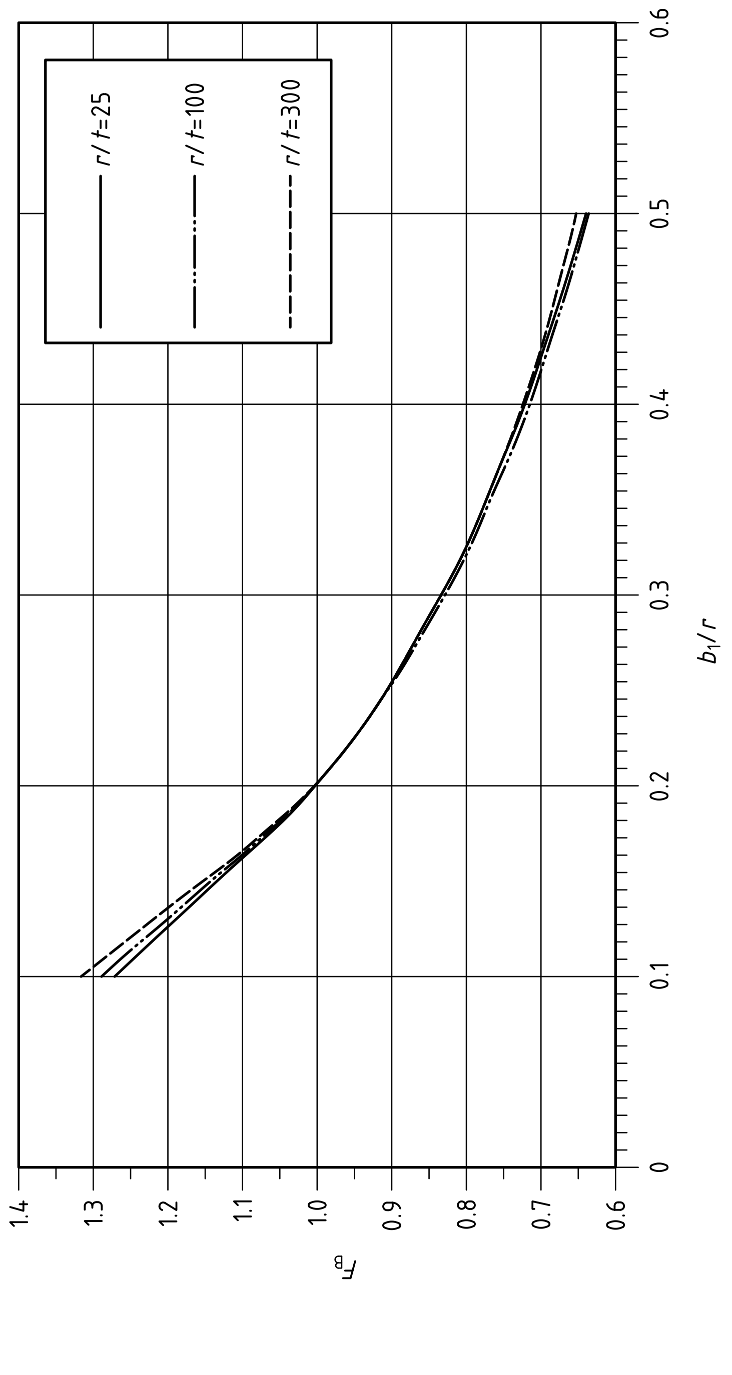


Figure G.3.3-11 **Graph of saddle interaction factor, F_I**

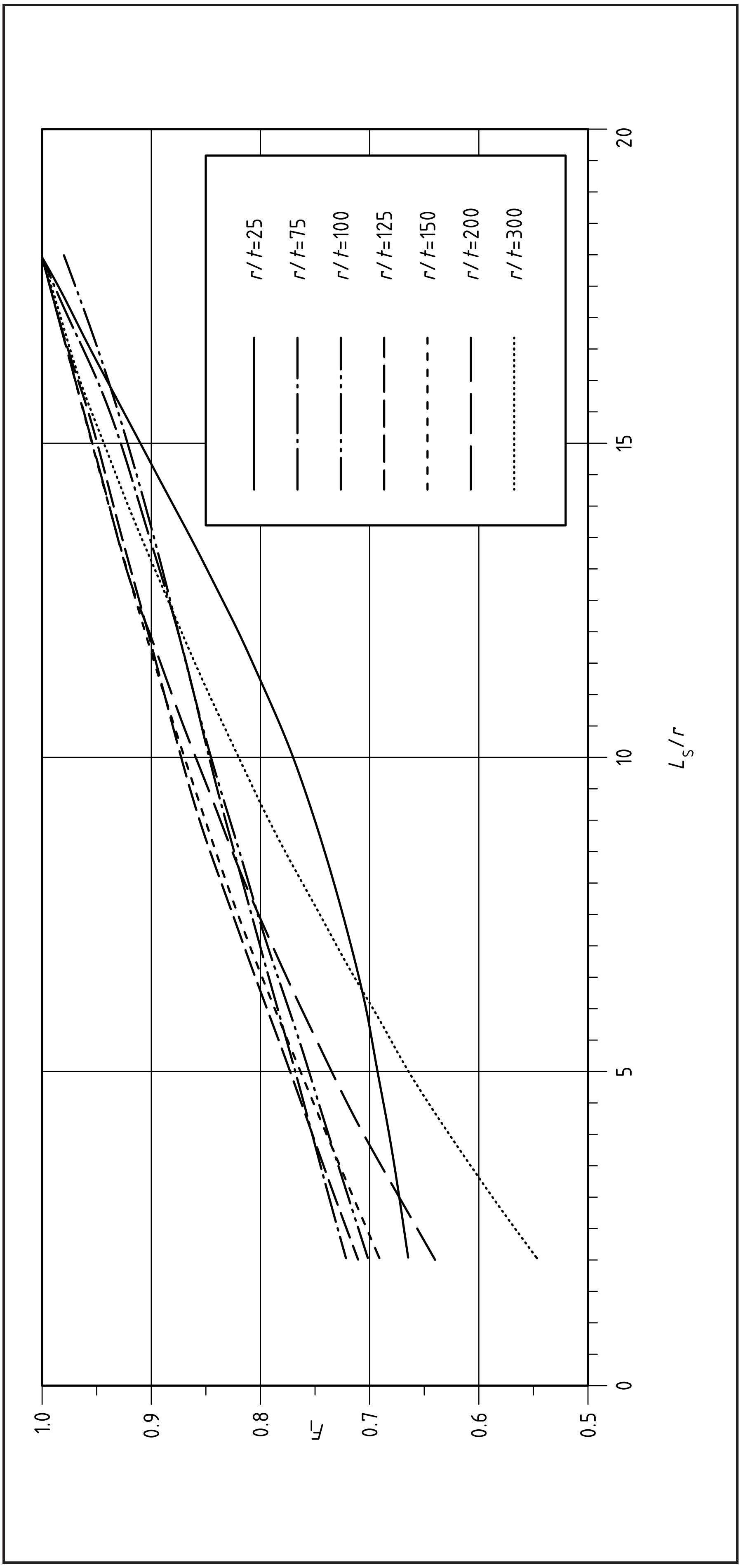


Figure G.3.3-12 **Graph of length change factor, F_L**

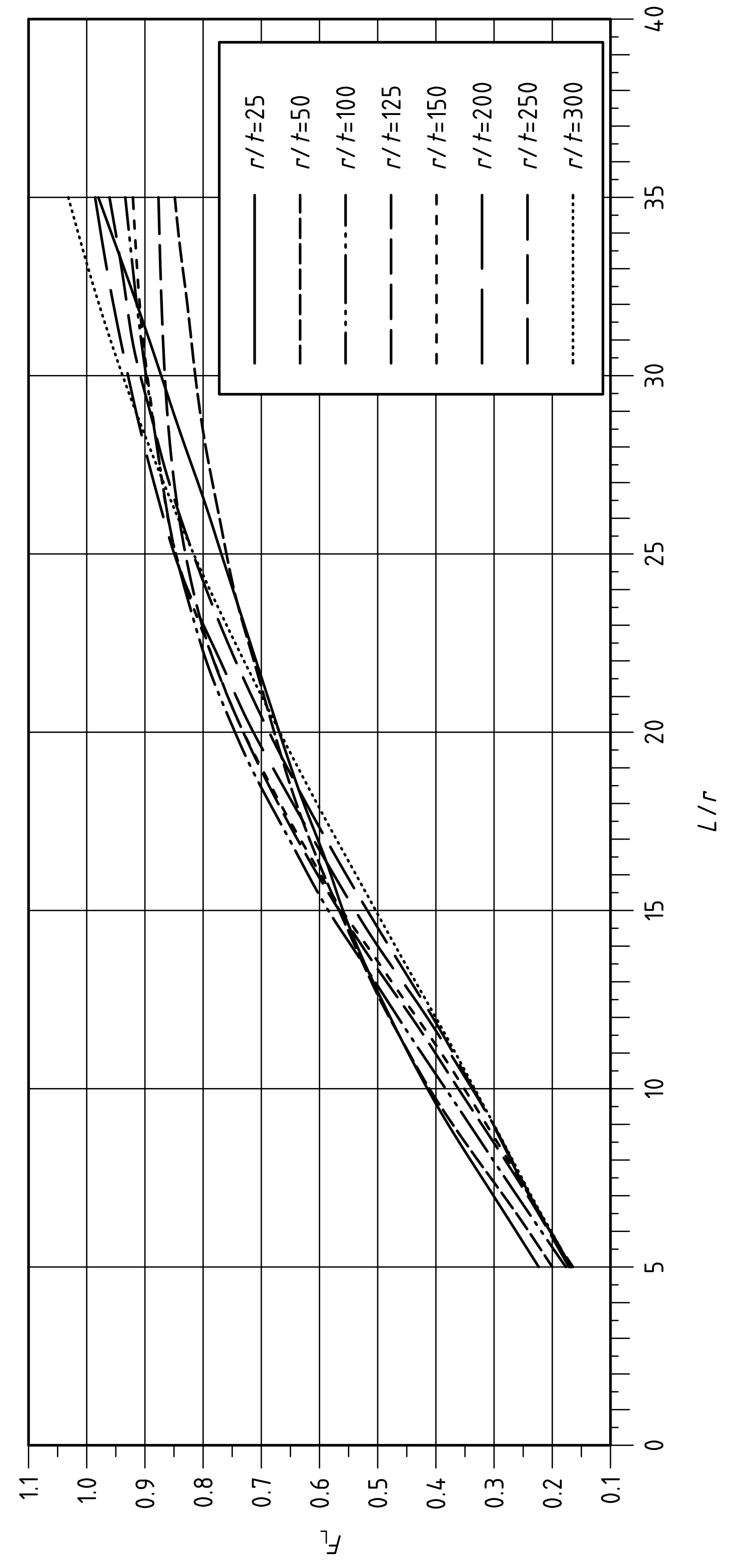
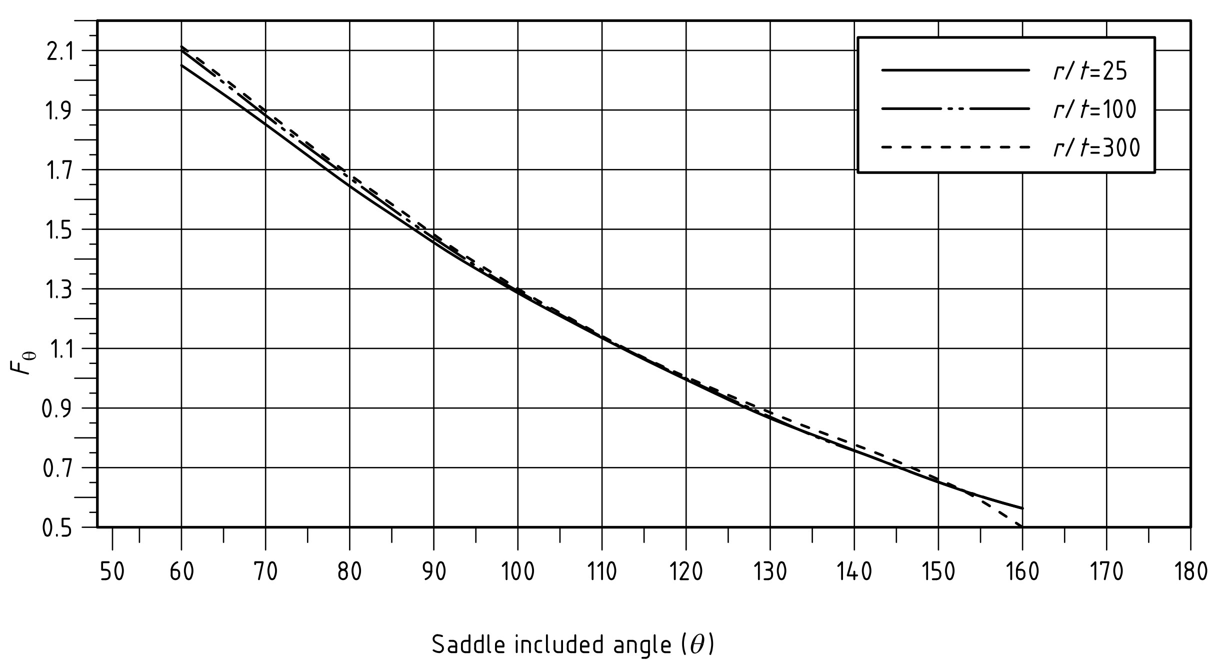


Figure G.3.3-13 **Graph of saddle wrap-round factor, F_θ**



G.3.3.3 Ring supports for horizontal vessels

Ring supports for horizontal vessels, as shown in Figure G.3.3-1b), are used where it is important to ensure that the shell of the vessel close to the supports remains round under load. This is usually the case for:

a) thin-walled vessels likely to distort excessively due to their own weight;

b) long vessels requiring support at more than two positions.

The longitudinal bending moments in the shell and the corresponding stresses can be found in the same way as for saddle supports from Equations (G.3.3-1) to (G.3.3-8).

The tangential shear stresses in the shell adjacent to the ring support are given by:

$$q = \left(\frac{0.319W_1}{rt}\right)\left(\frac{L-2A}{L+\frac{4b}{3}}\right) \tag{G.3.3-37}$$

The allowable tangential shearing stress values are given in Table G.3.3-2.

The maximum circumferential stress in the ring, due to dead loads is given by:

$$f_{10} = \frac{K_{10}W_1r_2}{Z} + \frac{K_{11}W_1}{a} \tag{G.3.3-38}$$

It can be assumed that a length of shell l (= $\sqrt{rt}$ + contacting width of support) acts with the ring support to form a combined section and that r_2 is the radius through the centroid of this section, Z is the least section modulus and a is an effective area of the section. The constants K_{10} and K_{11} are found from Table G.3.3-11.

The stress in the ring f_{10} should not exceed f. In the case of category 1 and 2 vessels the rings are in general of the same material as the vessel and constructed to the same category as the vessel with the same f value as the vessel. In the case of the rings associated with category 3 vessels, it is considered acceptable to use the corresponding category 1 and 2 vessel f values provided the radial weld seams joining the segments of the rings are located in the region of low bending stress in the rings. The distribution of the bending moment in a typical ring support is shown in [37].

Where the ring is made of a different material from that of the vessel, the f value for the weaker material should be used. For mild steel ring girders used on category 3 vessels and not subject to above ambient temperatures, it is acceptable to use the allowable stresses from [40]. In this case the ring should be designed as a separate structure *without* the benefit of the length of the shell.

Unless a vessel with ring supports works at atmospheric temperature and pressure, at least one ring support has to be provided with some form of sliding bearing at its connection to the foundation or supporting structure.

NOTE The values of K_{10} and K_{11} are derived from the absolute maximum circumferential moment and the absolute maximum direct force in a ring support as shown in Figure G.3.3-1b). The influence of shear forces in the ring due to reactions $W_1/2$ is not taken into account and the designer should satisfy himself that the ring section is sufficient in cross-sectional area and lateral stiffness to resist these forces. It is not necessary to take into consideration secondary shell bending stresses induced by the rigidity of, for example, a support ring, when evaluating, except where fatigue is a governing criterion when the permissible stress is a matter for individual consideration.

Table G.3.3-11 **Values of K_{10} and K_{11}**

Angle ϕ_1 degrees	K_{10}	K_{11}
30	0.075	0.41
35	0.065	0.40
40	0.057	0.39
45	0.049	0.38
50	0.043	0.37
55	0.039	0.36
60	0.035	0.35
65	0.030	0.34
70	0.025	0.32
75	0.020	0.31
80	0.017	0.29
85	0.015	0.27
90	0.015	0.25

G.3.3.4 Ring stiffeners or supports for horizontal vessels subject to external pressure and dead load

In the absence of a precise method, the following conservative linear interaction approach should be used to assess horizontal vessels subject to simultaneous external pressure and dead load, and fitted with ring stiffeners (see Figure G.3.3-8) or ring supports [see Figure G.3.3-1b)].

When the vessel is fitted with ring stiffeners:

$$\frac{\sigma_s}{sf_s} + \frac{|f_8|}{1.25f_s} \leq 1.0 \qquad \text{(G.3.3-39)}$$

When the vessel is fitted with ring supports:

$$\frac{\sigma_s}{sf_s}+\frac{f_{10}}{f_s}\leq 1.0 \tag{G.3.3-40}$$

where

σ_s is the maximum stress in the ring stiffener due to external pressure [Equation (3.6.2-15)];

f_8 is the maximum circumferential stress in the ring stiffener [Equation (G.3.3-28), (G.3.3-31) or (G.3.3-33)];

f_{10} is the maximum circumferential stress in the ring support [Equation (G.3.3-38)];

f_s is the allowable stress in the ring support/stiffener (not greater than the allowable shell stress).

G.4 Simplified method for assessing transient thermal stress at a pressure vessel nozzle

G.4.1 Introduction

It is often necessary to consider the stresses that will arise at the junction of a nozzle with a cylindrical or spherical shell when the fluid contained in the vessel is subject to a rise or fall in temperature. The value of these stresses may decide the number of temperature transients which can be accommodated without the risk of fatigue failure or, alternatively, the stress levels may dictate the rates of temperature variation which can safely be permitted.

During such variation in operating conditions, shell and branch material will be subject to stresses developed by transient through-thickness temperature distribution. The intensity of these stresses will be dependent upon the rate of fluid temperature rise or fall, the surface heat-transfer coefficient and also upon the metal thicknesses and properties.

Since the thickness of branch and shell will usually be dissimilar, there will be differential expansion of the branch and shell during the transient, which will produce additional discontinuity stress.

A rigorous stress analysis would need the use of finite element computer methods which, in the case of a branch on a cylindrical shell, would involve a complex three-dimensional approach. It would be difficult to be equally precise in specifying the heat transfer rates operating, which have been shown experimentally to vary considerably around the circumference of branches in cylinders. The cost of one such rigorous analysis would be prohibitive in most cases and usually the designer will need to consider several transient operating conditions.

Of more value in general pressure vessel work are more simple methods which give realistically conservative maximum stress levels for use in a fatigue assessment.

G.4.2 Outline of the suggested design method

The method described in **G.4.3**, **G.4.4** and **G.4.5** first uses well known analytical methods for determining through-thickness temperature distribution and stresses in the branch and shell material during a fluid transient. The average temperature of each component is then used in a thin-shell discontinuity analysis at the junction of branch and shell. The total stress is taken to be the sum of the temperature and discontinuity stress.

The solution yields a conservative estimate of the gross section stresses from which the maximum equivalent stress intensity can be calculated. In applying the results in a fatigue analysis, stress concentration factors would be applied to allow for the effect of welds or local geometry.

Graphs and tables are included which reduce the overall solution to the simple use of thermal and stress factors which are applied in a final set of stress equations.

G.4.3 Notation and derivation of method

G.4.3.1 Notation

For the purposes of **G.4** the following symbols apply.

a_1, a_2, a_3	are branch influence coefficients;
A_1, A_2, A_3	are shell influence coefficients;
C_1, C_2, C_3, C_4	are stress factors (from Table G.4.3-1, Table G.4.3-2, Table G.4.3-3, and Table G.4.3-4);
c	is the specific heat of material [in J/(kg·K)];
d	is the diffusivity of material (in m^2/s) [see Equation (G.4.3-5)];
E	is the modulus of elasticity (in N/m^2), i.e. the value from Table 3.6-3 multiplied by 10^6;
F	shear force (in N);
h	is the surface heat transfer coefficient [in $W/(m^2 \cdot K)$];
h_b	is the surface heat transfer coefficient at the branch inner surface [in $W/(m^2 \cdot K)$];
h_s	is the surface heat transfer coefficient at the shell inner surface [in $W/(m^2 \cdot K)$];
k	is the conductivity of vessel material [in $W/(m^2 \cdot K)$];
k_1, k_2	are branch thermal factors (from Figure G.4.3-3 and Figure G.4.3-4);
K_1, K_2	are shell thermal factors (from Figure G.4.3-3 and Figure G.4.3-4);
K_b, K_s	are branch and shell mean temperature factors (from Figure G.4.3-5);
K_d	is the mean temperature difference factor;
m	is a thermal parameter [see Equation (G.4.3-4)];
M	shear moment (in N·m);
N	is a thermal parameter [see Equation (G.4.3-3)];
r	is the mean radius of branch (in m);
r_o	is the outer radius of branch (in m);
r_i	is the inner radius of branch (in m);
R	is the mean radius of shell (in m);
S	is stress (in N/m^2) (see text for specific symbols);
t	is the nominal branch thickness (in m) (see **1.6**);
T	is the nominal shell thickness (in m) (see **1.6**);
T_f	is the fluid temperature rise from start of transient (in K);
T_i	is the inner surface temperature (in K);
T_o	is the outer surface temperature (in K);
T_m	is the mean temperature (in K);
V	is the discontinuity of edge rotation [see Equation (G.4.3-8)];
α	is the coefficient of linear expansion [in m/(m·K)];
δ	is the radial discontinuity [see Equations (G.4.3-6) and (G.4.3-7)];
θ	is the time from start of transient (in s);
ρ	is the density of the material (in kg/m^3).

G.4.3.2 Derivation of method

Consider a cylinder-to-sphere assembly as shown in Figure G.4.3-1 with a fluid subject to a rise in temperature on the inside. Assume that heat transfer coefficients (h_b and h_s) apply at the branch and shell inner surfaces. The fluid velocity in the branch will usually be greater than that in the shell and h_b may be several times larger than h_s. During a ramp rise in temperature the time-temperature behaviour of branch and shell material will be similar to that shown in Figure G.4.3-2.

Branch and shell material away from the discontinuity will be subject to thermal stress proportional to the difference between the surface temperature (T_i or T_o) and the mean temperature (T_m). These through-thickness temperature stresses will generally be different in branch and shell.

Solutions are given in [29] for stresses in a flat plate subject to a ramp rise in fluid temperature at one surface.

Taking Poisson's ratio as equal to 0.3, these solutions may be plotted in the form of Figure G.4.3-3 and Figure G.4.3-4,

where

$$S_i = -K_1 E \alpha T_f \tag{G.4.3-1}$$

$$S_o = K_2 E \alpha T_f \tag{G.4.3-2}$$

S_i and S_o are the thermal stresses at inner and outer surfaces.

The values of K_1 and K_2 are plotted in Figure G.4.3-3 and Figure G.4.3-4 against the parameters:

$$N = \frac{d\theta}{t^2} \tag{G.4.3-3}$$

$$m = \frac{k}{ht} \tag{G.4.3-4}$$

where

$$d = k/(c\rho) \tag{G.4.3-5}$$

Also, from the solutions given in [1], curves may be drawn as shown in Figure G.4.3-5 which give the ratio of rise in mean metal temperature to the rise in fluid temperature (T_m/T_f) using the same parameters N and m.

Assuming that the thermal expansion of the branch and the shell opening is proportional to the respective average metal temperatures, the radial discontinuity introduced at the junction would be

$$\delta = (K_b - K_s)\alpha r T_f \tag{G.4.3-6}$$

where K_b and K_s are obtained from Figure G.4.3-5.

In addition to the relative horizontal displacement of the two parts, a rotational discontinuity (V) will also be produced by edge rotation of the shell opening.

In an actual construction these discontinuities will be removed by shear forces (F) and moments (M) acting at the junction and their values would be given by the equations:

$$(a_1 + A_1)F + (a_2 + A_2)M = \delta \tag{G.4.3-7}$$

$$(a_2 + A_2)F + (a_3 + A_3)M = V \tag{G.4.3-8}$$

where a_n and A_n are deflections and rotation influence coefficients for branch and shell respectively. The values of a_n may be obtained from simple thin-cylinder bending theory; values of A_n referring to a pierced hemisphere may be more conveniently obtained from thin-shell computer analysis.

In practice the "free" rotation at the edge of the shell opening would be small and would in any case tend to reduce the values of discontinuity force and moment. If the value V is therefore taken to be zero, the equations offer a more simple solution, giving somewhat conservatively high values of F and M.

Taking Poisson's ratio as 0.3, inserting equations for a_n and letting $C = R/T$, $S = r/R$, $Z = T/t$, a non-dimensional solution of the equations will be given by:

$$\frac{F}{E\delta} = \frac{D_1}{D} \quad \text{(G.4.3-9)}$$

$$\frac{M}{E\delta T} = \frac{D_2}{D} \quad \text{(G.4.3-10)}$$

where

$$D_1 = A_3/C^2 + 8.54(CS)^{0.5}Z^{2.5} \quad \text{(G.4.3-11)}$$

$$D_2 = A_2/C - 3.33(CS)Z^2 \quad \text{(G.4.3-12)}$$

$$D = D_1\left(2.6(CSZ)^{1.5} + A_1\right) - D_2^2 \quad \text{(G.4.3-13)}$$

Using the calculated values of junction force and moment, equations for stresses in the branch and shell at the junction may also be written, and stresses will be directly proportional to the difference in mean temperature between the two parts. A general expression for discontinuity stress may therefore be written as

$$S_n = K_d C_n E \alpha T_f \quad \text{(G.4.3-14)}$$

where K_d is equal to the difference between the temperature factors K_b and K_s given by Figure G.4.3-5 and C_n represents factors for the various component stresses in the assembly. The values of C_n for a range of branch/shell geometries have been computed and are given in Table G.4.3-1, Table G.4.3-2, Table G.4.3-3 and Table G.4.3-4.

The total stresses will be given by combining the discontinuity stresses with those due to through-thickness transient temperature distribution (S_i and S_o) and may be represented by a general set of stress equations as given in **G.4.4**.

Figure G.4.3-1 **Nozzle geometry**

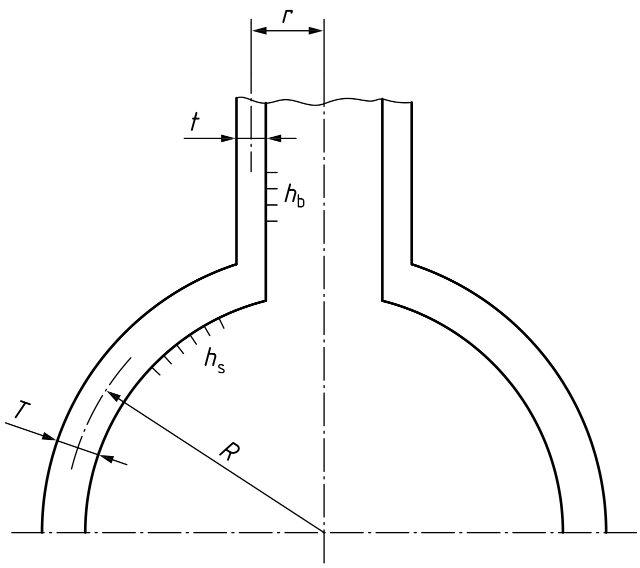
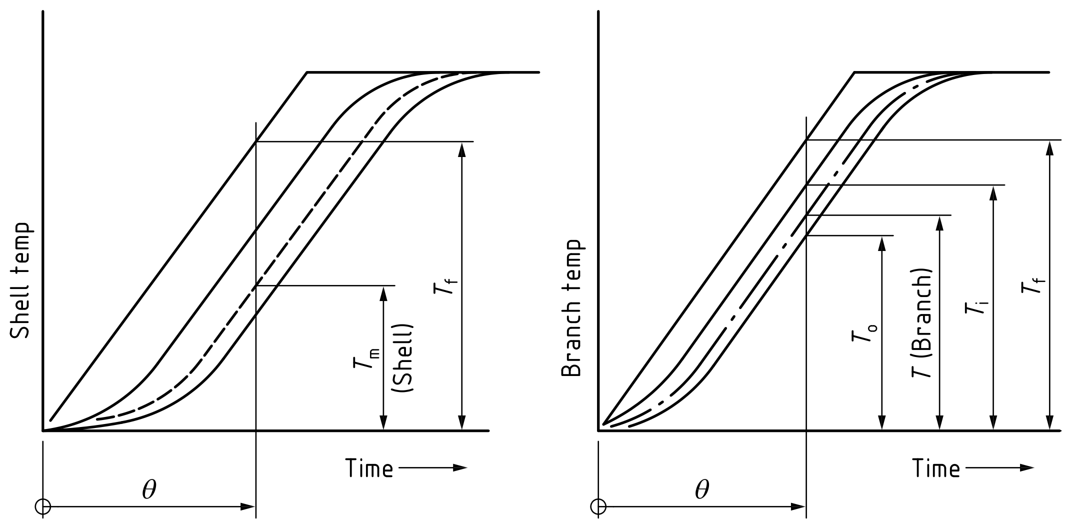


Figure G.4.3-2 **Transient fluid and metal temperatures**



$$K_1 = \frac{T_i - T_m}{(1 - \mu)\,T_f} \qquad K_2 = \frac{T_m - T_o}{(1 - \mu)\,T_f} \qquad K_m = \frac{T_m}{T_f}$$

Figure G.4.3-3 **Inner surface thermal stress factors K_1 and k_1**

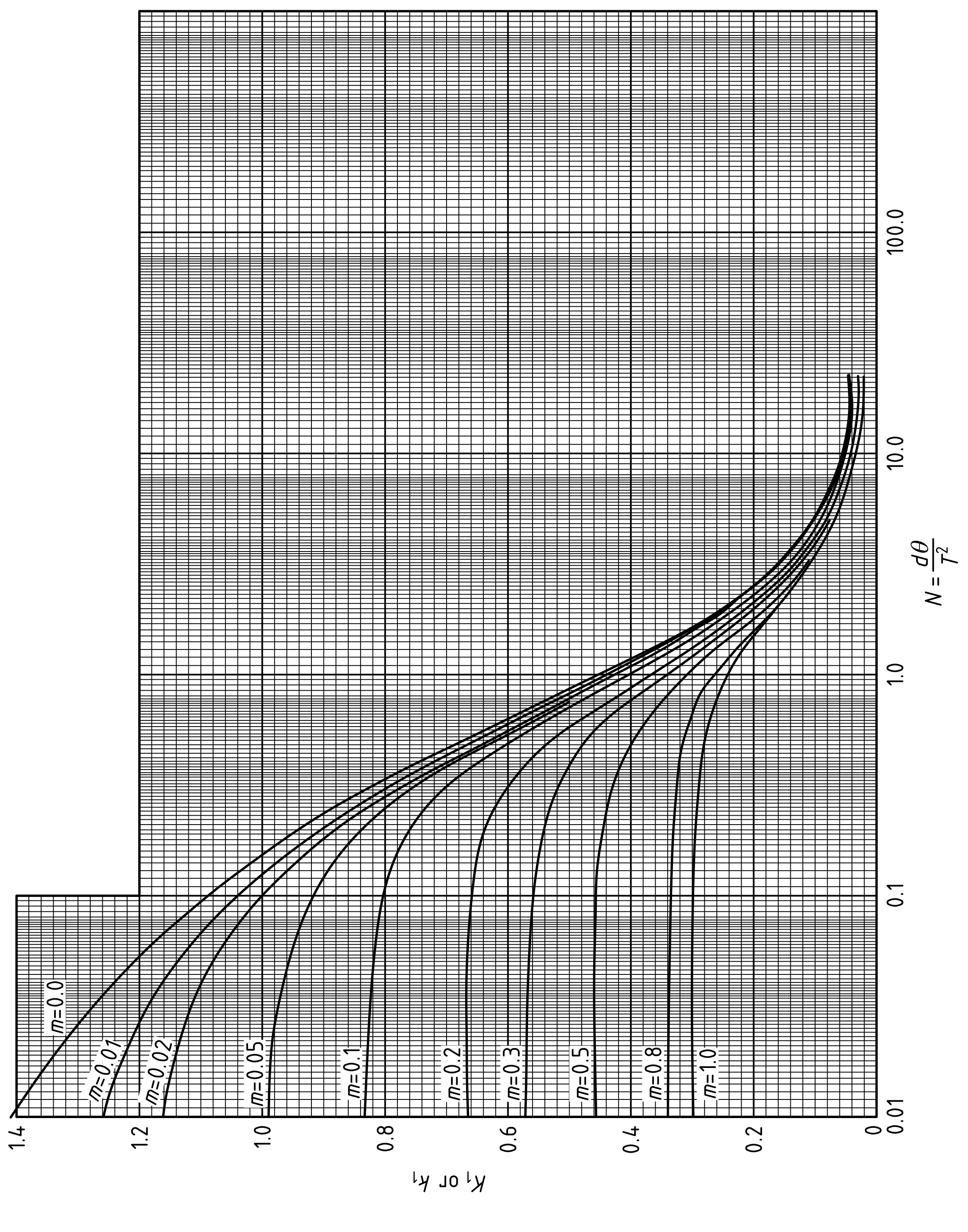


Figure G.4.3-4 Outer surface thermal stress factors K_2 and k_2

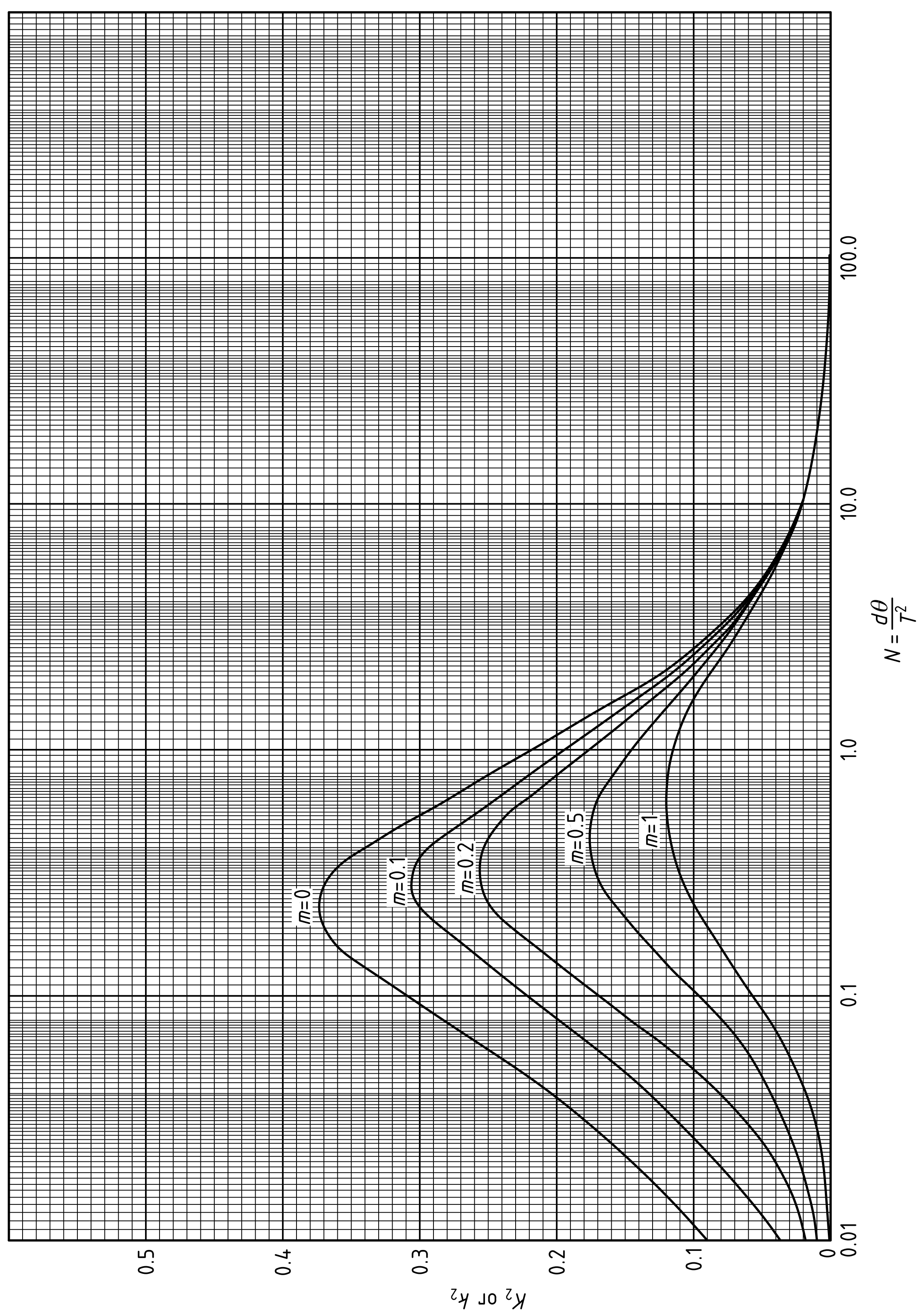
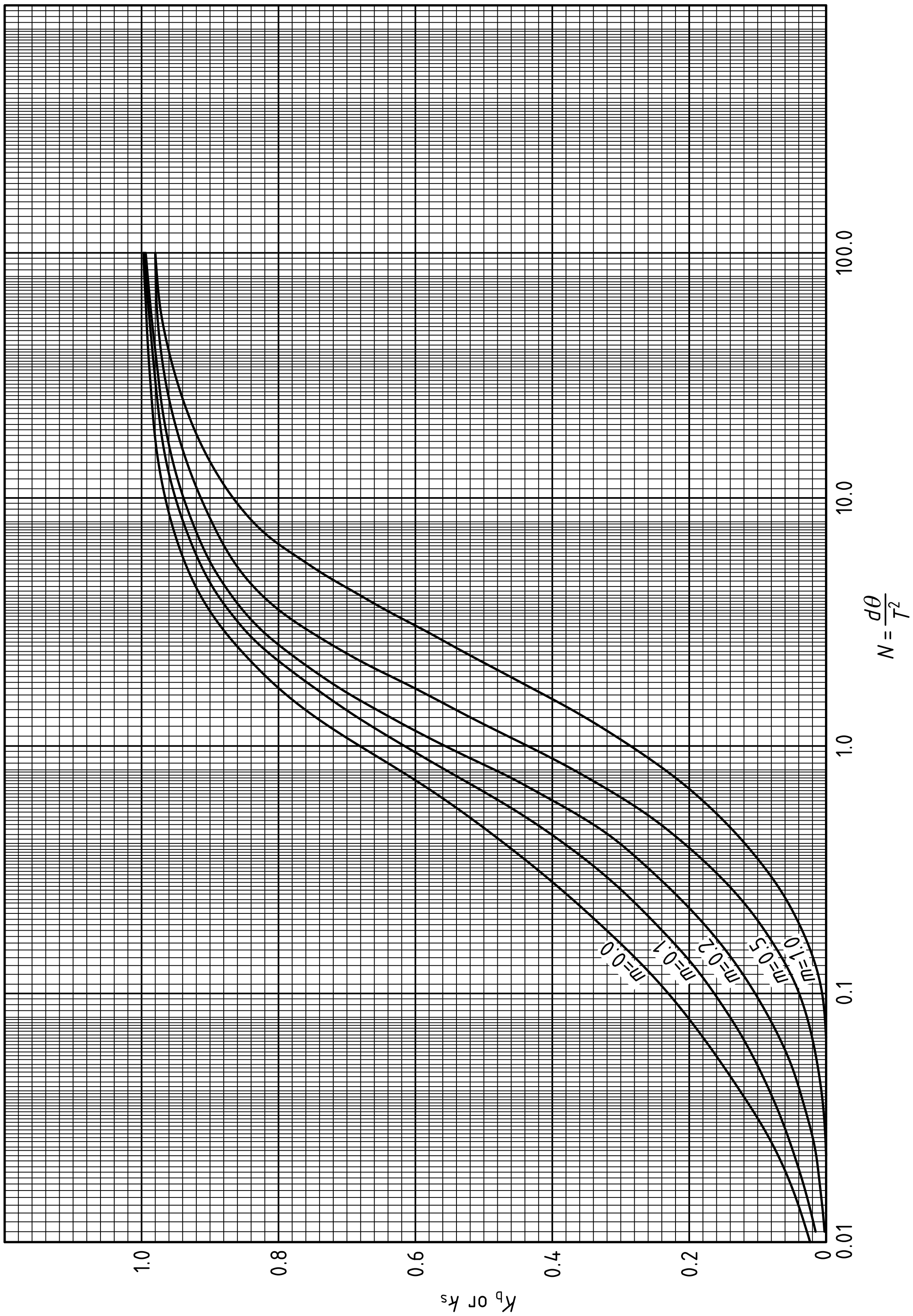


Figure G.4.3-5 Mean temperature factors K_b and k_s



G.4.4 Total stress equations

G.4.4.1 Junction stresses

a) *Shell*

Inner surface

circumferential

$$S_{hi} = \left[K_d\left(C_1 + \frac{0.3C_2}{Z^2}\right) - K_1\right]E\alpha T_f \tag{G.4.4-1}$$

meridional

$$S_{mi} = \left[K_d\left(\frac{C_2}{Z_2} - C_3\right) - K_1\right]E\alpha T_f \tag{G.4.4-2}$$

Outer surface

circumferential

$$S_{ho} = \left[K_2 + K_d\left(C_1 - \frac{0.3C_2}{Z^2}\right)\right]E\alpha T_f \tag{G.4.4-3}$$

meridional

$$S_{mo} = \left[K_2 - K_d\left(C_3 + \frac{C_2}{Z^2}\right)\right]E\alpha T_f \tag{G.4.4-4}$$

b) *Branch*

Inner surface

circumferential

$$S'_{hi} = [K_d(C_1 + 0.3C_2 - 1.0) - k_1]E\alpha T_f \tag{G.4.4-5}$$

longitudinal

$$S'_{li} = [K_dC_2 - k_1]E\alpha T_f \tag{G.4.4-6}$$

Outer surface

circumferential

$$S'_{ho} = [k_2 + K_d(C_1 - 0.3C_2 - 1.0)]E\alpha T_f \tag{G.4.4-7}$$

longitudinal

$$S'_{lo} = [k_2 - K_dC_2]E\alpha T_f \tag{G.4.4-8}$$

where

$K_d = K_b - K_s$ (read from Figure G.4.3-5);

K_1, K_2, k_1, k_2 are temperature factors from Figure G.4.3-3 and Figure G.4.3-4 for shell (K) and branch (k);

C_1, C_2, C_3 are stress factors from Table G.4.3-1, Table G.4.3-2 and Table G.4.3-3.

The maximum equivalent stress intensity will usually occur at the junction between branch and shell to which point the above stress equations refer.

Maximum bending stress in the branch may occur at a distance $0.62\sqrt{rt}$ from the junction. At this point the total thermal stresses will be given by equations in **G.4.4.2**.

G.4.4.2 Branch stresses

Inner surface

circumferential

$$S'_{\mathrm{hi}} = [K_{\mathrm{d}}(0.322(C_1 - 1) + 0.192C_2 - 0.3C_1) - k_1]E\alpha T_{\mathrm{f}} \tag{G.4.4-9}$$

longitudinal

$$S'_{\mathrm{li}} = [K_{\mathrm{d}}(0.644C_2 - C_4) - k_1]E\alpha T_{\mathrm{f}} \tag{G.4.4-10}$$

Outer surface

circumferential

$$S'_{\mathrm{ho}} = [K_{\mathrm{d}}(0.322(C_1 - 1) + 0.3C_1 - 0.192C_2) + k_2]E\alpha T_{\mathrm{f}} \tag{G.4.4-11}$$

longitudinal

$$S'_{\mathrm{lo}} = [K_{\mathrm{d}}(C_4 - 0.644C_2) - k_2]E\alpha T_{\mathrm{f}} \tag{G.4.4-12}$$

where

C_4 is a stress factor from Table G.4.3-4.

G.4.5 Use and limitations of the method

The final stress equations should provide a designer with a simple means of estimating stress ranges in a branch due to thermal cycling. Although based upon the analysis of the rotationally symmetric cylinder-to-sphere connection, the results should be sufficiently accurate for use in the fatigue assessment of branches in cylindrical shells.

The analysis for through-thickness temperature stress is based upon a flat-plate solution and is reasonably accurate for cylinders where $r_{\mathrm{o}}/r_{\mathrm{i}}$ is less than about 1.1. For branches thicker than this, the stress factor (k_1) may be multiplied by the ratio r/r_{i} for a conservative result.

The analysis for discontinuity stress will give conservative results since it neglects the effects of edge rotation produced by any axial temperature gradient on the branch and by temperature gradient through the shell wall. Both effects would in practice tend to reduce the junction forces and moments.

Such a simple type of analysis cannot, of course, predict the peak stresses which would occur due to local changes in geometry at the junction and the designer would need to apply appropriate stress concentration factors before applying stress results in a fatigue analysis.

In practical use the tabulated stress factors C_1, C_2, C_3 and C_4 will be found to plot as fairly straight lines against the various parameters, and interpolation for intermediate geometric ratios can be made with reasonable accuracy.

Table G.4.3-1 **Circumferential stress factor C_1**

***R/T* = 15**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.11	0.15	0.20	0.29	0.36	0.46	0.57	0.66
0.1	0.10	0.13	0.18	0.26	0.32	0.40	0.50	0.59
0.2	0.11	0.14	0.20	0.27	0.32	0.39	0.48	0.56
0.3	0.12	0.16	0.22	0.30	0.35	0.41	0.49	0.57
0.4	0.13	0.17	0.23	0.31	0.37	0.43	0.50	0.57
0.5	0.14	0.18	0.24	0.33	0.38	0.44	0.51	0.58
***R/T* = 50**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.08	0.11	0.15	0.22	0.26	0.34	0.48	0.53
0.1	0.09	0.11	0.16	0.22	0.26	0.33	0.41	0.49
0.2	0.10	0.14	0.19	0.26	0.30	0.35	0.43	0.50
0.3	0.11	0.15	0.21	0.27	0.33	0.38	0.45	0.52
0.4	0.12	0.17	0.22	0.30	0.35	0.39	0.47	0.53
0.5	0.14	0.18	0.24	0.32	0.38	0.43	0.49	0.56
***R/T* = 100**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.07	0.10	0.13	0.19	0.23	0.30	0.39	0.47
0.1	0.08	0.11	0.15	0.21	0.25	0.30	0.38	0.45
0.2	0.10	0.13	0.18	0.24	0.29	0.33	0.41	0.48
0.3	0.12	0.15	0.21	0.26	0.32	0.35	0.44	0.50
0.4	0.12	0.17	0.22	0.28	0.35	0.39	0.46	0.53
0.5	0.13	0.18	0.24	0.32	0.37	0.42	0.49	0.55

Table G.4.3-2 **Bending stress factor C_2**

***R/T* = 15**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.51	1.39	1.18	0.80	0.52	0.21	0.05	0.01
0.1	1.48	1.35	1.11	0.70	0.42	0.13	0	−0.03
0.2	1.44	1.29	1.04	0.62	0.34	0.07	−0.04	−0.06
0.3	1.41	1.25	1.00	0.58	0.31	0.04	−0.07	−0.08
0.4	1.39	1.24	0.98	0.56	0.29	0.03	−0.08	−0.09
0.5	1.38	1.22	0.97	0.55	0.28	0.02	−0.09	−0.10
***R/T* = 50**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.50	1.36	1.12	0.70	0.41	0.13	0.01	−0.02
0.1	1.45	1.30	1.04	0.60	0.32	0.07	0	−0.05
0.2	1.41	1.26	0.99	0.56	0.29	0.03	−0.07	−0.08
0.3	1.38	1.23	0.97	0.54	0.28	0.01	−0.09	−0.10
0.4	1.38	1.23	0.97	0.54	0.28	0.01	−0.09	−0.10
0.5	1.37	1.21	0.96	0.54	0.27	0.01	−0.10	−0.11
***R/T* = 100**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.47	1.32	1.06	0.63	0.35	0.09	−0.01	−0.03
0.1	1.43	1.26	0.99	0.55	0.28	0.04	−0.05	−0.06
0.2	1.40	1.24	0.97	0.54	0.27	0.02	−0.08	−0.09
0.3	1.39	1.23	0.97	0.54	0.27	0.02	−0.08	−0.10
0.4	1.38	1.22	0.96	0.54	0.27	0.01	−0.09	−0.11
0.5	1.37	1.21	0.95	0.54	0.27	0.01	−0.10	−0.11

Table G.4.3-3 **Meridional stress factor C_3**

***R/T* = 15**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.07	0.09	0.12	0.18	0.22	0.29	0.37	0.44
0.1	0.05	0.06	0.09	0.13	0.16	0.21	0.29	0.35
0.2	0.03	0.04	0.06	0.08	0.10	0.14	0.20	0.25
0.3	0.03	0.03	0.05	0.06	0.08	0.11	0.15	0.19
0.4	0.02	0.03	0.04	0.05	0.06	0.09	0.12	0.15
0.5	0.02	0.02	0.03	0.04	0.05	0.07	0.10	0.13
***R/T* = 50**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.04	0.05	0.07	0.10	0.13	0.18	0.25	0.31
0.1	0.03	0.03	0.05	0.07	0.09	0.12	0.18	0.23
0.2	0.02	0.02	0.03	0.04	0.06	0.08	0.12	0.15
0.3	0.01	0.02	0.02	0.03	0.04	0.06	0.09	0.11
0.4	0.01	0.01	0.02	0.03	0.03	0.05	0.07	0.09
0.5	0.01	0.01	0.02	0.02	0.03	0.04	0.06	0.07
***R/T* = 100**								
r/R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	0.03	0.04	0.05	0.07	0.09	0.13	0.19	0.25
0.1	0.02	0.02	0.03	0.05	0.06	0.09	0.13	0.18
0.2	0.01	0.02	0.02	0.03	0.04	0.06	0.09	0.11
0.3	0.01	0.01	0.02	0.02	0.03	0.04	0.06	0.08
0.4	0.01	0.01	0.01	0.02	0.02	0.03	0.05	0.06
0.5	0.01	0.01	0.01	0.02	0.02	0.03	0.04	0.05

Table G.4.3-4 **Branch bending stress factor C_4**

***R*/*T* = 15**								
r*/*R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.00	0.94	0.85	0.67	0. 54	0.38	0.27	0.20
0.1	1.00	0.94	0.84	0.66	0.53	0.39	0.29	0.23
0.2	0.98	0.91	0.80	0.62	0.50	0.37	0.29	0.24
0.3	0.97	0.89	0.78	0.59	0.48	0.35	0.27	0.23
0.4	0.94	0.88	0.76	0.58	0.46	0.34	0.26	0.22
0.5	0.94	0.87	0.75	0.57	0.45	0.33	0.25	0.21
***R*/*T* = 50**								
r*/*R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.02	0.96	0.86	0.68	0.56	0.43	0.33	0.26
0.1	1.00	0.93	0.82	0.65	0.53	0.41	0.33	0.28
0.2	0.98	0.91	0.79	0.61	0.50	0.39	0.31	0.26
0.3	0.96	0.89	0.77	0.59	0.48	0.36	0.29	0.25
0.4	0.95	0.88	0.76	0.58	0.46	0.35	0.27	0.23
0.5	0.94	0.88	0.76	0.57	0.45	0.33	0.26	0.22
***R*/*T* = 100**								
r*/*R	***Z* = 5**	***Z* = 4**	***Z* = 3**	***Z* = 2**	***Z* = 1.5**	***Z* = 1**	***Z* = 0.66**	***Z* = 0.5**
0.05	1.01	0.95	0.84	0.67	0.56	0.44	0.35	0.30
0.1	0.99	0.92	0.81	0.64	0.53	0.42	0.35	0.30
0.2	0.97	0.90	0.79	0.61	0.50	0.39	0.32	0.27
0.3	0.97	0.89	0.77	0.59	0.48	0.37	0.28	0.25
0.4	0.94	0.89	0.75	0.58	0.46	0.35	0.28	0.24
0.5	0.94	0.85	0.75	0.57	0.45	0.34	0.26	0.22

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Annex H Recommendations for post-weld heat treatment of dissimilar ferritic steel joints

NOTE It is essential that individual clauses of this annex are not read in isolation.

H.1 Basic conditions

The recommendations in this annex are based on the following conditions.

a) Post-weld heat treatment should be compatible with the parent materials being welded.

b) Post-weld heat treatment should be compatible with the relative importance of the pressure parts being welded.

c) The weld metal should be compatible with the post-weld heat treatment. The materials have been classified into five groups (as shown in Table H.1), the minimum temperature for post-weld heat treatment in each group being constant. Alternative post-weld heat treatments to those listed in Table H.1 should take into account the basic conditions a), b) and c).

Table H.1 **Classification of materials**

Material		Post-weld heat treatment temperature range		Group
Group or sub-group	**Type**	T_1 °C	T_2 °C	
1, 2	Carbon and carbon manganese steel	580	620	HT1
9.1, 9.2	3½Ni	580	620	
1 CMo	CMo	630	670	HT2
4.1, 4.2	MnCrMoV[a]	630	670	
5.1	1Cr½Mo[b]	630	670	
	1¼Cr½Mo[b]	630	670	
5.2	2¼Cr1Mo[c]	630	670	
5.1	1Cr½Mo[d]	650	700	HT3
	1¼Cr½Mo[d]	650	700	
6.1	½Cr½Mo¼V	680	720	HT4
5.2	2¼Cr1Mo[b]	680	720	
5.2	2¼Cr1Mo[d]	710	750	HT5
5.3	5Cr½Mo	710	750	

[a] Refers to 271 and 281 type steels (see Table K.1-3, Table K.1-5 and Table K.1-6).
[b] For optimum high temperature properties.
[c] For high tensility.
[d] For maximum softening.

H.2 Welds between material grades within a group

H.2.1 The post-weld heat treatment of welds between material grades within a group is permissible.

H.2.2 Where a weld is made between dissimilar pressure parts within the same group, the consumables are to be appropriate to either of the materials.

H.3 Welds between material grades from different groups

H.3.1 The post-weld heat treatment of welds between material grades in different groups should be permissible where $T_1' - T_2''$ is not greater than 10 °C, T_1' being the lower temperature of the material grade requiring the higher temperature post-weld heat treatment and T_2'' being the higher temperature of the material grade requiring the lower temperature post-weld heat treatment.

EXAMPLE Post-weld heat treatment following welding group HT1 to group HT2 materials:

T_1' = 630 °C, T_2'' = 620 °C

$T_1' - T_2''$ = 10 °C which is permissible.

H.3.2 Where the temperature difference $T_1' - T_2''$ is greater than 10 °C, such joints should be the subject of agreement between the manufacturer, purchaser and Inspecting Authority.

H.3.3 Where a weld is made between pressure parts in materials belonging to different groups, the consumables should be chosen from the range of consumables appropriate to the group that controls the post-weld heat treatment.

H.4 Pressure part controlling post-weld heat treatment temperature range for materials from different groups

H.4.1 Where a weld is between pressure parts of equal importance, post-weld heat treatment should be in the higher temperature range (see **H.5.1**).

H.4.2 Where a weld is between pressure parts of differing importance, the post-weld heat treatment should be as for the major pressure part (see **H.5.1** and **H.5.2**).

H.4.3 Where a weld is between a structural part and a pressure part, the post-weld heat treatment should be as for the pressure part.

H.4.4 Where the major pressure part does not require post-weld heat treatment but the minor pressure part does, then special consideration should be given to ensure technical acceptability, e.g. buttering of the minor component with the consumable adopted for the buttered component and, separate post-weld heat treatment of the buttered component.

H.5 General considerations

H.5.1 Where the post-weld heat treatment is being carried out between different material groups in the higher temperature range, the average temperature of the assembly should be held as near to the minimum as is practicable.

Using the example given in **H.3.1**, if group HT2 is the ruling component, then the target is approximately 640 °C.

H.5.2 Where post-weld heat treatment is being carried out between different material groups in the lower temperature range, the average temperature of the assembly should be held as near to the maximum as is practicable.

Using the example given in **H.3.1**, if group HT1 is the ruling component, then the target is approximately 610 °C.

H.5.3 Where the time at temperature of a part of lower alloy content being post-weld heat treated at a higher temperature is greater than 60 min more than it would normally receive if heat treated in its normal temperature range, this should be the subject of agreement between the manufacturer, purchaser and Inspecting Authority.

H.5.4 The manufacturing sequence and post-weld heat treatment operations should be so arranged as to minimize the amount of degraded material.

H.5.5 Assemblies involving welds from three or more different groups requiring simultaneous treatment are to be avoided.

Annex J Recommendations for pressure relief protective devices

When considering the safety valve characteristics and the system requirements, the relationship between the design pressure and the permitted accumulated pressure in a vessel (or system) will be dictated by the requirements of **3.13.2** and **3.13.3**. The relationship with the set pressure and overpressure at which the safety valve attains its certified capacity is illustrated in Figure J.1.

For direct operated safety valves in gas or vapour service (see Figure J.1) the required discharge capacity should be achieved at an overpressure not exceeding 10% above the set pressure. Following discharge the valve will reseat within the range 5% to 10% below the set pressure providing that it is correctly adjusted. The normal operating pressure of the system should be below the reseat pressure, the difference being chosen on the basis of the probable variations in operating pressure due to process factors and the tolerance on cold differential test pressure.

With assisted and supplementary loaded safety valves, failure of the assist mechanism, or of the supplementary load to be released may result in the valve remaining closed until a pressure higher than the desired set pressure is reached. If the integrity of the assist mechanism or release of the supplementary load cannot be assured, the set pressure of the valve should be such that, in the event of failure, the required capacity will still be achieved at the desired accumulation. Alternatively, this risk may, by agreement between the appropriate parties, be covered by the fitting of additional valve(s).

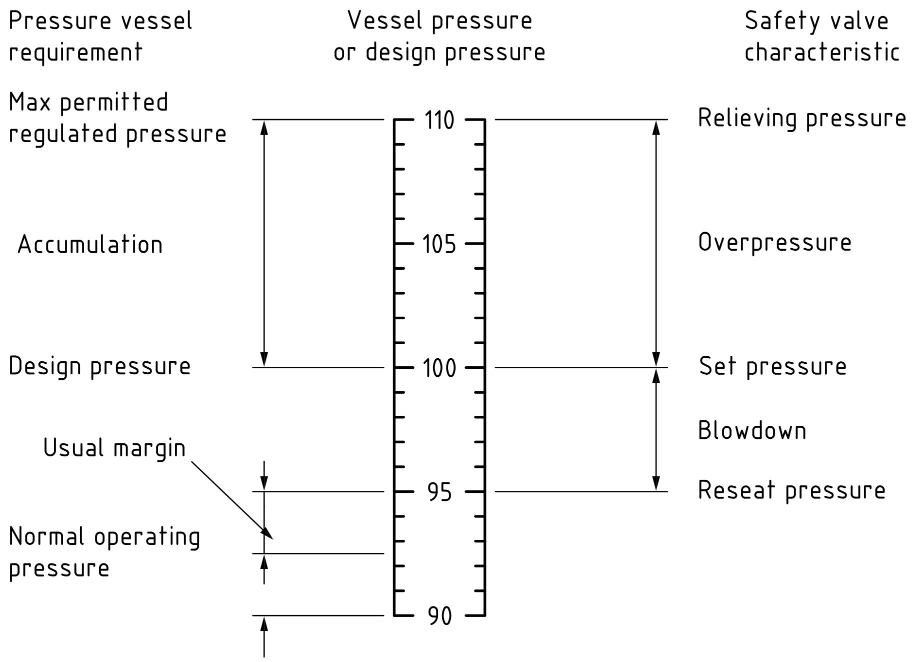
For direct operated safety valves in liquid service (see Figure J.1) the required discharge capacity of the valve may not be reached until an overpressure of 25% above the set pressure is reached when the valve will achieve full lift. To ensure that the maximum accumulated pressure given in **3.13.2** is not exceeded, valves in liquid service should be set at a lower pressure than those in gas or vapour service. A reasonable margin is required between the normal operating pressure of the vessel and the reseat pressure of the valve and as a result the normal operating pressure may be as much as 22% below the design pressure of the vessel. If this pressure margin is unavailable, it may be possible to install a larger capacity valve to give the required discharge capacity at an overpressure of less than 25% of the set pressure. This larger valve would not achieve full lift and its selection would require discussion with the valve manufacturer. Safety valves certified at 10% overpressure may be considered as an alternative.

NOTE Further information may be found in the following American Petroleum Institute publications, which are available from Customer Services, Sales Department, BSI, 389 Chiswick High Road, London W4 4AL.

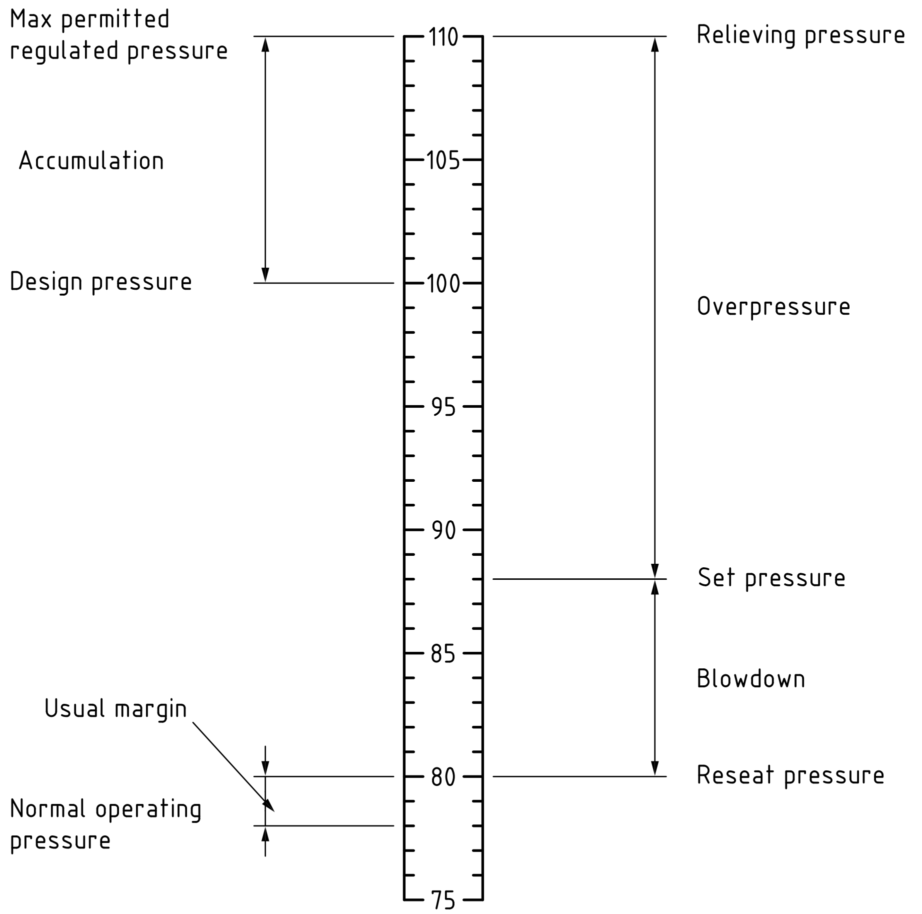
API STD 520	*Sizing, selection and installation of pressure relieving devices*
API STD 521	*Pressure-relieving and depressuring systems*

See also BS 5908 which calls up these API publications and other reference documents.

Figure J.1 **Typical pressure term relationships**



a) Safety valve with 10% overpressure (gas or vapour service)



b) Safety valve with chosen 25% overpressure (liquid service)

Annex K Requirements for design stresses for British Standard materials

K.1 General

K.1.1 The British Standards listed in Table K.1-2 to Table K.1-12 that have been revised in or after 1978 specify minimum elevated temperature yield/proof stress values derived, in most cases, in accordance with the procedures specified in BS 3920-1:1973.

K.1.2 These values show some difference from the properties specified in previous standards, which were based on individual assessments of the data then available. The procedure described in BS 3920-1 is essentially empirical and properties derived by it are regarded as characteristic values (to be used for quality control purposes as specified in the relevant materials standards) rather than as critical properties in the design context. Nevertheless, it is reasonable and convenient to base permissible design strengths directly on these characteristic yield/proof stress values unless this would result in design strengths for which there is no justification in terms of previous experience and current understanding of structural behaviour. This has been done, except in a few cases which are identified in Table K.1-2 to Table K.1-12, where design strengths based on the simple relationships specified in **2.3.3** and **2.3.4** would have resulted in an unwarranted reduction or increase in the strength levels that have previously been established for the materials in question.

K.1.3 Design strengths in the creep range are given for a range of design lifetimes that may be extended, on expiry, on the basis of periodic "fitness-for-continued-service reviews" based upon inspection and consideration of actual load-temperature history. This approach recognizes the limitations inherent in any simple design method for vessels operating in the creep range. It also provides a flexible basis that may be used in cases where the design strength values that have been derived from ISO data are significantly different from those used with success in the past. Specific requirements for these reviews are not given in this specification.

*NOTE 1 For time independent values in Table K.1-2 to Table K.1-12, values of $R_{e(T)}$ (see **2.3.2**) have been taken as equal to those specified for otherwise similar material having specified elevated temperature values, except that where no such $R_{e(T)}$ values are available design strength values have been based on conservative interpretation of other available information.*

*NOTE 2 For time-dependent values in Table K.1-2 to Table K.1-12, the appropriate S_{Rt} (see **2.3.2**) properties agreed by Subcommittee 10 of Technical Committee 17 of ISO have been used wherever possible. These do not necessarily correspond to those specified in the British Standards listed in Table K.1-2 to Table K.1-12. In general, time-dependent values are not given for materials that are unsuitable, or are unlikely to be used, in the creep range (see, however, **K.1.4.1.2**).*

*NOTE 3 In most cases, the S_{Rt} properties agreed by ISO for lifetimes in excess of 100 000 h have been obtained by extended extrapolation of time (more than three times on actual data), and those towards the upper end of the temperature range by extended stress extrapolation. Tabulated design strengths that are significantly lower than values well established by experience are identified by Note 6 to Table K.1-2 to Table K.1-12 which permit values up to 10% higher to be used provided that fitness for continued service reviews (see **3.2.4**) are instituted at two-thirds of the agreed design lifetime.*

NOTE 4 Some of the British Standards listed in Table K.1-2 to Table K.1-12 have been superseded but have been retained in the table for information in circumstances where their use may be required.

Table K.1-1 **Design strength values: index of steels**

	Standard	Table number
Steel plates	BS 1501-1 BS 1501-2 BS 1501-3	Table K.1-2 Table K.1-3 Table K.1-4
Steel sections and bars	BS 1502	Table K.1-5
Steel forgings	BS 1503	Table K.1-6
Steel castings	BS 1504	Table K.1-7
Steel pipes and tubes	BS 3059-1 and BS 3059-2 BS 3601, BS 3602-1 and BS 3602-2 BS 3603 BS 3604-1 and BS 3604-2 BS 3605-1 and BS 3605-2 BS 3606	Table K.1-8 Table K.1-9 Table K.1-9 Table K.1-10 Table K.1-11 Table K.1-12

K.1.4 Notes to Table K.1-2 to Table K.1-12

K.1.4.1 General notes

K.1.4.1.1 Table K.1-2 to Table K.1-12 give the nominal design strength (f_N) of various British Standard materials for design temperatures (°C) not exceeding those stated at the head of each column. Time-dependent values which are given for design lifetimes of 100 000, 150 000, 200 000 and 250 000 h are italicized. Values at intermediate temperatures shall be obtained by linear interpolation; values obtained by linear interpolation involving only one italicized value may be regarded as time-independent.

K.1.4.1.2 Except in cases where special note (1d) applies (see **K.1.4.2**), the design temperature as defined in Section 3 should not exceed the upper temperature for which a value of f_N is given; where extrapolation of the values given is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

K.1.4.1.3 If required for stress analysis purposes, the transition from linear elastic to linear plastic behaviour for all materials except austenitic steels may be assumed to occur at a stress of $1.5 \times f_N$, unless f_N is time-dependent; for austenitic steels the corresponding stress may be taken as $1.35 \times f_N$.

K.1.4.1.4 Overall thickness limits are as defined in the relevant material specification. In the case of forgings the term ruling section shall be interpreted in accordance with BS 5046.

K.1.4.1.5 R_m and R_e are defined as the minimum tensile strength and minimum yield strength, respectively, for the material concerned at room temperature, tested in accordance with BS EN 10002-1.

K.1.4.2 Special notes

The designations in the "Notes" column of Table K.1-2 to Table K.1-12 have the following meanings.

1b The values for forgings may be increased up to (but not greater than) the values permitted for plate in the equivalent material grade and equivalent ruling section on provision by the forgemaster of appropriate supporting data showing that the minimum acceptance criteria for equivalent plate are satisfied.

1c An appropriate casting quality factor as specified in **3.4** should be applied to these values.

1d Design strength values at higher temperatures are subject to agreement between the manufacturer, purchaser and Inspecting Authority.

1e Design strengths for these materials shall be derived using **2.3**.

2 At 580 °C and above the effect of scaling may be significant.

3 This material is available only in a limited range of thickness and in the normalized condition. It has a restricted carbon content to obviate the necessity for stress relief subsequent to fabrication. A nominal design strength of 129 N/mm^2 may be assumed at temperatures below 350 °C.

6 Subject to continued fitness for service reviews (see **3.2.4**) being instituted at ⅔ of the design lifetime indicated, time-dependent values may be increased by up to 10% providing the resulting value does not exceed the lowest time-independent value given.

8 Design strengths relate to material in the fully annealed condition; higher values may be justified where material is subject to an alternative heat treatment.

9 Design strengths are not strictly based on specified yield/proof stress values in the relevant material standard (see **K.1.2**).

10 The use of longitudinally welded piping and tubing and the associated design strength values are conditional on the longitudinal weld being subjected to the same NDT requirements as for the pressure vessel shell longitudinal weld.

13 Nickel based and some austenitic filler materials undermatch the parent material yield strength and might also undermatch the parent metal tensile strength. The weld metal properties of these consumables should satisfy a minimum 0.2% proof strength of 360 N/mm^2 (see **4.3.2.3**).

14 See **2.1.2.2.2.1** (restriction on carbon content of ferritic steels intended for welding).

15 Material will (or may be) supplied in the quenched and tempered condition in which case properties will be particularly susceptible to degradation as a result of subsequent fabrication processes (see **3.4.2.1**).

16 For thickness values between those asterisked, design strengths may be obtained by linear interpolation.

17 Design strengths of tubes manufactured from slit rimming steel are subject to agreement between the manufacturer, purchaser and Inspecting Authority.

18 This material is not to be used for vessels designed to operate below 0 °C.

19 This material is not to be used without impact tests for vessels designed to operate below 0 °C.

20 Time-dependent design values apply only if the minimum carbon content equals or exceeds 0.04%.

Table K.1-2 **Design strength values (N/mm²)**

Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Plates: Values of f_N for design temperatures (°C) not exceeding																	Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	h	
Carbon steel (rimmed) to BS 1501-1																							
141, 360A	360	205	1.1	3 to 16*	137	126	115	112	100	86	79												9, 16, 18
		195		19*	130	122	113	108															
Carbon steel (semi-killed with aluminium) to BS 1501-1																							
154, 360A	360	205	1.1	3 to 9.5	137	126	115	112	100	86	79												9
154, 400A	400	235	1.1	3 to 9.5	156	144	133	130	116	97	91												
154, 430A	430	250	1.1	3 to 9.5	167	157	148	143	128	107	102												
Carbon steel (semi-killed or fully-killed) to BS 1501-1																							
151, 161, 360	360	205	1.1	3 to 16*	137	126	115	112	100	86	81		78		77	76	69	60	52	44	36	100 000	9, 16
(A and B)		195		40*	130	119	113	108	96							74	65	56	48	40	32	150 000	
		185		63*	123	114	105	101	95							71	62	53	45	37	28	200 000	
		175		100*	117	108	99	96	91							68	59	51	42	35	26	250 000	
		170		150*	113	105	96	93	88														
151, 161, 400	400	225	1.1	3 to 16*	150	141	133	130	117	97	92		89	88	87	79	69	60	52	44	36	100 000	9, 16
(A and B)		215		40*	143	135	128	123	110					88	83	74	65	56	48	40	32	150 000	
		205		63*	137	129	121	117	108					88	80	71	62	53	45	37	28	200 000	
		200		100*	133	123	113	111	105					87	78	68	59	51	42	35	26	250 000	
		195		150*	130	119	110	108	102														
151, 161, 430	430	250	1.1	3 to 16*	167	157	148	143	128	107	102	100	99	98	88	79	69	60	52	44	36	100 000	9, 16
(A and B)		240		40*	160	150	140	135	121				99	93	83	74	65	56	48	40	32	150 000	
		230		63*	153	143	133	128	119				99	90	80	71	62	53	45	37	28	200 000	
		220		100*	147	135	124	121	115				97	87	78	68	59	51	42	35	26	250 000	
		210		150*	140	129	118	115	110														
Carbon steel (fully-killed aluminium treated) to BS 1501-1																							
164, 360A	360	255	1.1	3 to 16*	153	138	124	110	97	86	79		73	73	72	71	69	60	52	44	36	100 000	9, 16
		235		40*	153	137	122	109								71	65	56	48	40	32	150 000	
		220		63*	147	130	115	106								71	62	53	45	37	28	200 000	
		202		100*	135	122	110	104								68	59	51	42	35	26	250 000	
		180		150*	120	112	105	102															
164, 360B	360	255	1.1	3 to 16*	153	145	137	122	107	95	87		83	82	81	79	69	60	52	44	36	100 000	9, 16
		235		40*	153	143	134	121							81	74	65	56	48	40	32	150 000	
		220		63*	147	137	127	117							80	71	62	53	45	37	28	200 000	
		202		100*	135	129	122	115							78	68	59	51	42	35	26	250 000	
		180		150*	120	118	117	112															

Table K.1-2 **Design strength values (N/mm²)** *(continued)*

Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Thickness mm	Plates: Values of f_N for design temperatures (°C) not exceeding 50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	Design lifetime h	Notes[a]
Carbon steel (fully-killed aluminium treated) to BS 1501-1 *(continued)*																							
164, 400A	400	275	1.1	3 to 16*	170	155	140	125	111	98	91		86	85	84	79	69	60	52	44	36	100 000	9, 16
		265		40*	170	154	138	125							83	74	65	56	48	40	32	150 000	
		245		63*	163	147	131	122							80	71	62	53	45	37	28	200 000	
		225		100*	150	138	126	120							78	68	59	51	42	35	26	250 000	
		201		150*	134	127	121	117															
164, 400B	400	275	1.1	3 to 16*	170	161	153	137	121	108	101		95	94	88	79	69	60	52	44	36	100 000	9, 16
		265		40*	170	160	149	135						93	83	74	65	56	48	40	32	150 000	
		245		63*	163	152	144	133						90	80	71	62	53	45	37	28	200 000	
		225		100*	150	143	138	130						87	78	68	59	51	42	35	26	250 000	
		201		150*	134	132	132	128															
Carbon manganese steel (fully-killed niobium treated) to BS 1501-1																							
223, 460A	460	340	1.2	3 to 16*	195	182	169	155	143	130	124	117	115	114	105	90	77	65	56	48	42	100 000	9, 16
		330		40*	195	182							115	111	95	81	68	58	50	43	38	150 000	
		320		63*	195	182							115	104	88	75	63	54	46	40	34	200 000	
		294		100*	195	182							115	98	83	70	59	51	43	37	32	250 000	
		262		150*	175	172																	
223, 460B	460	340	1.2	3 to 16*	195	187	179	173	158	139	131	125	124	121	105	90	77	65	56	48	42	100 000	9, 16
		330		40*	195	187	179						124	111	95	81	68	58	50	43	38	150 000	
		320		63*	195	187	179						121	104	88	75	63	54	46	40	34	200 000	
		294		100*	195	187	179						115	98	83	70	59	51	43	37	32	250 000	
		262		150*	175	175	175																
223, 490A	490	355	1.2	3 to 16*	208	193	179	173	158	139	131	125	124	121	105	90	77	65	56	48	42	100 000	9, 16
		345		40*	208	193	179						124	111	95	81	68	58	50	43	38	150 000	
		340		63*	208	193	179						121	104	88	75	63	54	45	40	34	200 000	
		313		100*	208	193	179						115	98	83	70	59	51	43	37	32	250 000	
		279		150*	186	180	175																
223, 490B	490	355	1.2	3 to 16*	208	198	189	184	168	147	137	133	132	121	105	90	77	65	56	48	42	100 000	9, 16
		345		40*	208	198	189						128	111	95	81	68	58	50	43	38	150 000	
		340		63*	208	198	189						121	104	88	75	63	54	46	40	34	200 000	
		313		100*	208	198	189						115	98	83	70	59	51	43	37	32	250 000	
		279		150*	186	186	186																

Table K.1-2 **Design strength values (N/mm²)** *(continued)*

Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Thickness mm	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	Design lifetime h	Notes[a]
Plates																							
					Values of f_N for design temperatures (°C) not exceeding																		
Carbon manganese steel (fully-killed aluminium treated) to BS 1501-1																			
224, 400A	400	275	1.1	3 to 16*	170	155	140	125	111	98	91		86	85	84	83	77	65	56	48	42	100 000	9, 16
		265		40*	170	154	138	125							84	81	68	58	50	43	38	150 000	
		245		63*	163	147	131	122							84	75	63	54	46	40	34	200 000	
		225		100*	150	138	126	120							83	70	59	51	43	37	32	250 000	
		201		150*	134	127	121	117															
224, 400B	400	275	1.1	3 to 16*	170	161	153	137	121	108	101		95	94	93	90	77	65	56	48	42	100 000	9, 16
		265		40*	170	161	149	135							93	*81*	68	58	50	43	38	150 000	
		245		63*	163	154	144	133							*88*	*75*	63	54	46	40	34	200 000	
		225		100*	150	144	138	130							*83*	*70*	59	51	43	37	32	250 000	
		201		150*	134	133	132	128															
224, 430A	430	305	1.2	3 to 16*	183	168	153	137	121	108	101		95	94	93	90	77	65	56	48	42	100 000	9, 16
		285		40*	183	166	149	135							93	81	68	58	50	43	38	150 000	
		275		63*	183	164	144	133							*88*	75	63	54	46	40	34	200 000	
		253		100*	168	153	138	130							*83*	70	59	51	43	37	32	250 000	
		225		150*	150	141	132	128															
224, 430B	430	305	1.2	3 to 16*	183	174	165	149	132	117	111		105	104	103	90	77	65	56	48	42	100 000	9, 16
		285		40*	183	172	161	147						104	*95*	81	68	58	50	43	38	150 000	
		275		63*	183	170	157	145						104	*88*	75	63	54	46	40	34	200 000	
		253		100*	168	160	151	142						*98*	83	70	59	51	43	37	32	250 000	
		225		150*	150	148	144	139															
224, 460A	460	325	1.2	3 to 16*	196	180	165	149	132	117	111		105	104	103	90	77	65	56	48	42	100 000	9, 16
		315		40*	196	178	161	147						104	*95*	81	68	58	50	43	38	150 000	
		305		63*	196	176	157	145						104	88	75	63	54	46	40	34	200 000	
		281		100*	187	169	151	142						98	83	70	59	51	43	37	32	250 000	
		250		150*	167	155	144	139															
224, 460B	460	325	1.2	3 to 16*	196	186	177	160	142	128	121	116	115	114	*105*	90	77	65	56	48	42	100 000	
		315		40*	196	184	173	158					115	*111*	*95*	81	68	58	50	43	38	150 000	
		305		63*	196	183	171	156					115	104	*88*	75	63	54	46	40	34	200 000	
		281		100*	187	176	164	153					*115*	98	*83*	70	59	51	43	37	32	250 000	
		250		150*	167	162	157	150															

[a] See **K.1.4** which includes an explanation of the "Notes" column.

Table K.1-2 **Design strength values (N/mm²)** *(continued)*

Plates																							
Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	**Material group**	**Thickness** mm	**Values of f_N for design temperatures (°C) not exceeding** 50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	**Design lifetime** h	**Notes**[a]
Carbon manganese steel (fully-killed aluminium treated) to BS 1501-1 *(continued)*																							
224, 490	490	325	1.2	3 to 16*	208	192	177	160	142	128	121	116	115	114	*105*	90	77	65	56	48	42	100 000	9, 16
(A and B)		315		40*	208	190	173	158					115	*111*	95	81	68	58	50	43	38	150 000	
		305		63*	204	187	171	156					115	104	88	75	63	54	46	40	34	200 000	
		281		100*	187	176	164	153					*115*	98	*83*	70	59	51	43	37	32	250 000	
		250		150*	167	162	157	150															
Carbon manganese steel (fully-killed niobium and aluminium treated) to BS 1501-1																							
225, 460A	460	340	1.2	3 to 16*	196	175	154	142	131	119	111	105	104	104	103	*90*	*77*	*65*	*56*	*48*	*42*	100 000	9, 16
		330		40*	196	175								104	*95*	*91*	*68*	*58*	*50*	*43*	*38*	150 000	
		320		63*	196	175								104	*88*	*75*	*63*	*54*	*46*	*40*	*34*	200 000	
		294		100*	196	175								*98*	*83*	*70*	*59*	*51*	*43*	*37*	*32*	250 000	
		262		150*	175	165																	
225, 460B	460	340	1.2	3 to 16*	196	180	164	151	140	127	119	112	111	111	*105*	*90*	*77*	*65*	*56*	*48*	*42*	100 000	9, 16
		330		40*	196	180								111	*95*	*81*	*68*	*58*	*50*	*43*	*38*	150 000	
		320		63*	196	180								*104*	*88*	*75*	*63*	*54*	*46*	*40*	*34*	200 000	
		294		100*	196	180								98	*83*	*70*	*59*	*51*	*43*	*37*	*32*	250 000	
		262		150*	175	170																	
225, 490A	490	355	1.2	3 to 16*	208	193	179	173	158	139	131	125	124	121	105	90	77	65	56	48	42	100 000	9, 16
		345		40*	208	193	179						124	*111*	*95*	*81*	*68*	*58*	*50*	*43*	*38*	150 000	
		340		63*	208	193	179						*121*	*104*	*88*	*75*	*63*	*54*	*46*	*40*	*34*	200 000	
		313		100*	208	193	179						*115*	*98*	*83*	*70*	*59*	*51*	*43*	*37*	*32*	250 000	
		279		150*	186	180	175																
225, 490B	490	355	1.2	3 to 16*	208	198	189	184	168	147	137	133	132	*121*	*105*	*90*	*77*	*65*	*56*	*48*	*42*	100 000	9, 16
		345		40*	208	198	189						128	*111*	*95*	*81*	*68*	*58*	*50*	*43*	*38*	150 000	
		340		63*	208	198	189						*121*	*104*	*88*	*75*	*63*	*54*	*46*	*40*	*34*	200 000	
		313		100*	208	198	189						*115*	*98*	*83*	*70*	*59*	*51*	*43*	*37*	*32*	250 000	
		279		150*	186	186	186																

[a] See **K.1.4** which includes an explanation of the "Notes" column.

Table K.1-3 **Design strength values (N/mm²)**

Plates																										
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																				Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	450	480	490	500	510	520	530	540	550	560	570	580	h	
Low alloy steel to BS 1501-2																										
243A	440	275	1.1	≤16*	183	164	145	136	125	104	96	93														1d, 16
0.3Mo	440	270		40*	180	163																				
	440	260		60*	173	159																				
	430	240		100*	160	151	141	133	121	100	91	88														
243B	440	275	1.1	≤16*	183	169	155	145	133	111	102	99														1d, 16
0.3Mo	440	270		40*	180	167																				
	440	260		60*	173	164																				
	430	240		100*	160	155	151	141	129	107	97	93														
660A	460	310	6.1	≤100	196	184	173	167	151	141	135	131	127	126												
CrMoV																										
660B	460	310	6.1	≤100	196	190	184	178	161	150	144	139	135		134	133	*131*	115	100	87	76	66	56		100 000	
CrMoV																133	*122*	106	92	80	69	59	47		150 000	
																132	*115*	100	87	75	65	55			200 000	
																128	*111*	96	83	72	62	50			250 000	
503	450	275	9.2	≤30	183																					13, 16
3½N	450	265		50*	175																					
510	690	590	9.3	≤50	295																					13
510N	690	525																								
9Ni																										
828	640	550		≤100																						1e, 13
3NiCrMo																										

Table K.1-3 **Design strength values (N/mm²)** *(continued)*

Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Plates: Values of f_N for design temperatures (°C) not exceeding																							Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	410	420	430	440	450	460	470	480	490	500	510	520	530	540	550	h	
Low alloy steel to BS 1501-2 *(continued)*																													
271	640	500	4.2	≤25*	272	271	271	265	259	249	242	234					231	*209*	181	155	129	107	87	69	54	41	30	100 000	16
MnCrMoV																	*229*	*202*	173	145	118	95	76	60	44	32	23	150 000	
																	221	*192*	160	132	108	87	68	51	38	26	18	200 000	
																	216	*188*	157	129	105	83	65	48	35	23	14	250 000	
	610	460		75*	260	259	259	253	248	236	229	221					219	*209*	181	155	129	107	87	69	54	41	30	100 000	16
																	219	*202*	173	145	118	95	76	60	44	32	23	150 000	
																	219	*192*	160	132	108	87	68	51	38	26	18	200 000	
																	216	*188*	*157*	*129*	*105*	*83*	*65*	*48*	*35*	*23*	*14*	250 000	
	590	430		150*	251	242	233	227	221	207	201	195					193	190	*181*	155	129	107	87	69	54	41	30	100 000	16
																		190	*173*	145	118	95	76	60	44	32	23	150 000	
																		190	160	132	108	87	68	51	38	26	18	200 000	
																		188	157	129	105	83	65	48	35	23	14	250 000	
281	640	500	4.2	≤25*	272	271	271	265	259	249	242	234	233	233	*226*	209	188	168	145	124	103	85	69	55	44	33	24	100 000	16
NiCrMoV													233	*232*	*218*	202	182	158	135	112	93	77	62	47	35	26		150 000	
													233	226	213	196	177	152	128	105	87	69	55	40	31	22		200 000	
													232	222	207	191	172	147	123	98	82	65	50	35	27			250 000	
	610	460		75*	260	259	259	253	248	236	229	221			220	*209*	188	168	145	124	103	85	69	55	44	33	24	100 000	16
															218	*202*	182	158	135	112	93	77	62	47	35	26		150 000	
															213	*196*	177	152	128	105	87	69	55	40	31	22		200 000	
															207	191	172	147	123	98	82	65	50	35	27			250 000	
	590	430		150*	251	242	233	227	221	207	201	195				193	*188*	168	145	124	103	85	69	55	44	33	24	100 000	16
																193	182	158	135	112	93	77	62	47	35	26		150 000	
																193	177	152	128	105	87	69	55	40	31	22		200 000	
																191	*172*	147	123	98	82	65	50	35	27			250 000	

Table K.1-3 **Design strength values (N/mm²)** *(continued)*

Plates																											
Grade, type and method of manufacture	R_m	R_e	**Material group**	**Thickness**	**Values of f_N for design temperatures (°C) not exceeding**																					**Design lifetime**	**Notes**[a]
	N/mm²	N/mm²		mm	**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**450**	**460**	**470**	**480**	**490**	**500**	**510**	**520**	**530**	**540**	**550**	**560**	**570**	h	
Low alloy steel to BS 1501-2 *(continued)*																											
620A	480	340	5.1	≤ 75*	204	186	169	161	154	143	134	132	129		127												16
1Cr½Mo	450	280		100*	187	168	150	144	136	121	113	110	108			106											
	430	255		150*	170	154	138	131	125	108	99	96	94				92										
620B	480	340	5.1	≤ 75*	204	192	180	172	165	152	143	141	137		136	135	134	*112*	93	76	62	52	42	33	27	100 000	16
1Cr½Mo																135	*124*	*102*	83	67	55	44	35	29	24	150 000	
																135	114	*94*	76	61	49	40	32	26	22	200 000	
																131	*107*	*88*	70	57	45	37	30	25	20	250 000	
	450	280		100*	187	173	160	153	145	129	121	117	115			113	112	111	93	76	62	52	42	33	27	100 000	16
																	112	102	83	67	55	44	35	29	24	150 000	
																	112	*94*	76	61	49	40	32	26	22	200 000	
																	107	*88*	70	57	45	37	30	25	20	250 000	
	430	255		150*	170	158	147	140	133	115	105	103	100				98	*97*	93	76	62	52	42	33	27	100 000	16
																		97	83	67	55	44	35	29	24	150 000	
																		94	76	61	49	40	32	26	22	200 000	
																		88	70	57	45	37	30	25	20	250 000	
621A	515	340	5.1	≤ 75*	219	205	191	182	175	166	159	157	153	152													16
1¼Cr½Mo	500	320		100*	213	197	181	173	166	156	149	146	143		141												
	490	310		150*	207	191	175	168	160	149	142	139	136		134												
621B	515	340	5.1	≤ 75*	219	211	203	194	187	177	170	167	163	162	161	160	136	112	93	76	62	52	42	33	27	100 000	16
1¼Cr½Mo															161	*149*	124	102	*83*	*67*	*55*	*44*	*35*	*29*	*24*	150 000	
															161	138	*114*	*94*	76	61	49	40	32	26	22	200 000	
															155	131	*107*	*88*	70	57	45	37	30	25	20	250 000	
	500	320		100*	213	203	193	185	177	167	159	156	152		150	149	136	112	93	76	62	52	42	33	27	100 000	16
																149	124	102	83	67	55	44	35	29	24	150 000	
																138	*114*	94	76	61	49	40	32	26	22	200 000	
																131	*107*	88	70	57	45	37	30	25	20	250 000	
	490	310		150*	207	197	187	179	171	159	151	148	145		142	142	*136*	112	93	76	62	52	42	33	27	100 000	16
																142	*124*	102	83	67	55	44	35	29	24	150 000	
																138	*114*	94	76	61	49	40	32	26	22	200 000	
																131	*107*	88	70	57	45	37	30	25	20	250 000	

Table K.1-3 **Design strength values (N/mm²)** *(continued)*

Plates																															
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	340	350	360	400	440	450	460	470	480	490	500	510	520	530	540	550	560	570	580	h	
Low alloy steel to BS 1501-2 *(continued)*																															
622-515A	515	310	5.2	≤ 100*	207	192	177	172	166	163		158		153	146																16
2¼Cr1Mo	500	285		150*	190	178	166	161	155	151		147		143		134	132														
622-515B	515	310	5.2	≤ 100*	207	198	189	183	177	173		168		163	156	154	152	143	131	118	105	94	82	72	61	53	45	39	34	100 000	2, 16
2¼Cr1Mo																154	148	135	122	108	97	85	73	63	56	48	42	36	31	150 000	
																154	143	130	117	104	92	79	68	59	52	45	38	33	28	200 000	
																152	139	126	113	100	87	75	65	57	49	42	36	32	27	250 000	
	500	285		150*	190	183	177	171	165	161		157		153		143	141	139	131	118	105	94	82	72	61	52	45	39	34	100 000	
																	141	135	122	108	97	85	73	63	56	48	42	36	31	150 000	
																	141	130	117	104	92	79	68	59	52	45	38	33	28	200 000	
																	139	126	113	100	87	75	65	57	49	42	36	32	27	250 000	
622-690A	690	555	5.2	≤50	294	294	294	294	294	292	284																				
2¼Cr1Mo																															
622-690B	690	555	5.2	≤50	294	294	294	294	294	294	294	294	287	235		170	157	143	131	118	105	94	82	72	61	53	45	39	34	100 000	2
2¼Cr1Mo												290	277	225		161	148	135	122	108	97	85	73	63	56	48	42	36	31	150 000	
												286	273	221		156	143	130	117	104	92	79	68	59	52	45	38	33	28	200 000	
												281	268	216		152	139	126	113	100	87	75	65	57	49	42	36	32	27	250 000	

Table K.1-4 **Design strength values (N/mm²)**

Plates																															
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	740	750	h	
High alloy steel to BS 1501-3																															
304-S11	480	215	8.1	Up to 100	143	127	111	101	95	90	86	81	80																		
304-S31	500	230	8.1	Up to 100	153	136	119	109	103	98	93	89	87	85			83	82	76	65	55	45	42	38	33	28				100 000	20
																		82	*72*	61	51	42	39	36	31					150 000	
																		81	*69*	58	48	40	37	34	30					200 000	
																		78	66	55	46	38	35	33	29					250 000	
304-S51	490	230	8.1	Up to 100	153	136	119	109	103	98	93	89	87	85			83	82	82	*76*	65	55	50	45	38	33	28			100 000	
																			82	72	61	51	46	42	36	31				150 000	
																			81	69	58	48	44	40	34	30				200 000	
																			78	*66*	55	46	42	38	33	29				250 000	
304-S61	550	305	8.1	Up to 100	203	174	149	135	127	121	116	110	107	104			101														1d
321-S31	510	235		Up to 100	157	145	133	127	121	117	113	110	107	104	103	103	102	102	*86*	71	57	42	36	31						100 000	20
																103	*101*	*93*	*78*	64	49	36	32	27						150 000	
																103	*97*	89	*74*	58	45	33	28	25						200 000	
																100	*93*	85	*71*	55	42	31	26	22						250 000	
321-S51	490	210	8.1	Up to 100	140	127	115	109	104	99	96	93	90	87				86	85	*82*	68	55	49	44	34	26				100 000	
																			85	*76*	62	50	44	38	29					150 000	
																			85	*72*	58	46	41	35	27					200 000	
																			84	68	55	44	38	33	25					250 000	
347-S31	510	240	8.1	Up to 100	160	151	142	135	127	123	120	118	116	115	114	114	*109*	99	82	66	53	41	35	31	23					100 000	20
																112	*102*	93	75	61	48	37	32	28	22					150 000	
																106	*98*	88	72	58	45	34	29	25	19					200 000	
																102	*94*	85	68	55	42	32	28	24	19					250 000	
347-S51	510	240	8.1	Up to 100	160	151	142	135	127	123	120	118	116	115		114	113	113	*99*	82	66	53	47	41	31	23				100 000	
																	113	*112*	*93*	75	61	48	42	37	28	22				150 000	
																	113	106	88	72	58	45	39	34	25	19				200 000	
																	112	*102*	*85*	68	55	42	37	32	24	19				250 000	

Table K.1-4 **Design strength values (N/mm^2)** *(continued)*

Plates																															
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
	N/mm^2	N/mm^2		mm	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	740	750	h	
High alloy steel to BS 1501-3 *(continued)*																															
316-S11, S13	490	225	8.1	Up to 100	150	134	119	110	103	99	94	91	88																		
316-S31, S33	510	240	8.1	Up to 100	160	143	127	118	111	106	101	99	96	93			90	89	89	74	58	46	41	35	28	23				100 000	20
																			85	*66*	52	41	36	32	25	19				150 000	
																			79	62	48	38	33	29	22	18				200 000	
																			75	58	45	35	31	27	22	18				250 000	
316-S51, S53	510	240	8.1	Up to 100	160	143	127	118	111	106	101	99	96	93			90		89	88	74	58	53	46	35	28	23			100 000	6
																				85	66	52	46	41	32	25	19			150 000	
																				79	62	48	43	38	29	22	18			200 000	
																				75	58	45	40	35	27	22	18			250 000	
316-S61, S63	580	315	8.1	Up to 100	210	180	154	142	133	127	123	119	116	113			110			107											1d
320-S31	510	245	8.1	Up to 100	163	147	130	121	114	109	104	101																			
309-S16	510	240	8.2	Up to 100	160	142	124	114	108	103	98	93	91	90			87														
310-S16	510	240	8.2	Up to 100	160	142	124	114	108	103	98	93	91	90			87														1d
318-S13	680	450	10.1	Up to 20	289	249	209	194	184	178																					
	680	480		80	289	249																									
	640	480		100	272	241																									
904-S13	520	255	8.2	Up to 100																											1e

Table K.1-5 **Design strength values (N/mm²)**

Sections and bars																							
Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Thickness mm	Diameter mm	Values of f_N for design temperatures (°C) not exceeding 50	100	150	200	250	300	350	400	410	420	430	440	450	460	470	480	Design lifetime h	Notes[a]
Carbon steel (semi- or silicon-killed) to BS 1502																							
151, 161	430	250	1.1	Up to 16*	Up to 25*	167	157	148	143	128	107	102	99	*98*	88	79	69	60	52	44	36	100 000	16, 19
		240		Up to 40*	Up to 50*	160	150	140	135	121			99	*93*	83	74	65	56	48	40	32	150 000	
		230		Up to 63*	Up to 100*	153	143	133	128	119			99	*90*	80	71	62	53	45	37	28	200 000	
		220		Up to 100*	Up to 160*	147	135	124	121	115			*97*	*87*	78	68	59	51	42	35	26	250 000	
Carbon manganese steel (semi- or silicon-killed) to BS 1502																							
211, 221	430	250	1.1	Up to 16*	Up to 25*	167	157	148	143	128	107	102	99	98	97	*90*	77	65	56	48	42	100 000	16
		240		Up to 40*	Up to 50*	160	150	140	135	121					95	*81*	68	58	50	43	38	150 000	
		230		Up to 63*	Up to 100*	153	143	133	128	119					*88*	*75*	63	54	46	40	34	200 000	
		220		Up to 100*	Up to 160*	147	135	124	121	115					83	*70*	59	51	43	37	32	250 000	
Carbon manganese steel (silicon-killed, aluminium treated) to BS 1502																							
224, 430	430	275	1.1	Up to 16*	Up to 25*	183	168	153	137	121	108	101	95	94	93	*90*	77	65	56	48	42	100 000	16
		265		Up to 40*	Up to 50*	176	163	149	136						93	*81*	68	58	50	43	38	150 000	
		245		Up to 63*	Up to 100*	163	154	144	133						*88*	75	63	54	46	40	34	200 000	
		240		Up to 100*	Up to 160*	160	151	141	131						*83*	70	59	51	43	37	32	250 000	
224, 490	490	325	1.2	Up to 16*	Up to 25*	208	193	177	160	142	128	121	115	114	*105*	90	77	65	56	48	42	100 000	16
		315		Up to 40*	Up to 50*	208	191	173	158					*111*	*95*	81	68	58	50	43	38	150 000	
		305		Up to 63*	Up to 100*	203	187	171	156					*104*	*88*	75	63	54	46	40	34	200 000	
		300		Up to 100*	Up to 160*	200	184	168	152					98	*83*	70	59	51	43	37	32	250 000	

Table K.1-5 **Design strength values (N/mm²)** *(continued)*

Sections and bars																													
Grade, type and method of manufacture	R_m	R_e	**Material group**	**Thickness**	**Values of f_N for design temperatures (°C) not exceeding**																							**Design lifetime**	**Notes**[a]
	N/mm²	N/mm²		mm	**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**440**	**450**	**460**	**470**	**480**	**490**	**500**	**510**	**520**	**530**	**540**	**550**	**560**	**570**	**580**	h	
Low alloy steel to BS 1502																													
271-560	560	370	4.2	≤ 160	238	229	220	215	208	204	199	192		188	186	*181*	*155*	*129*	107	87	69	54	41	30				100 000	15, 19
MnCrMoV																*173*	145	118	95	76	60	44	32	23				150 000	
																160	132	108	87	68	51	38	26	18				200 000	
																157	129	105	83	65	48	35	23	14				250 000	
620-440	440	265	5.1	≤ 160	177	163	153	147	139	123	113	109		107				105	104	*93*	76	62	52	42	33	27		150 000	15
1Cr½Mo																			*102*	*83*	67	55	44	35	29	24		150 000	
																			94	76	*61*	49	40	32	26	22		200 000	
																			88	*70*	57	45	37	30	25	20		250 000	
620-470	470	300	5.1	≤ 160	200	187	173	166	159	144	136	133		130			128	127	*112*	93	76	62	52	42	33	27		100 000	15
1Cr½Mo																		*124*	*102*	*83*	*67*	*55*	*44*	35	29	24		150 000	
																		114	*94*	76	61	49	40	32	26	22		200 000	
																		107	*88*	70	57	45	37	30	25	20		250 000	
620-540	540	375	5.1	≤ 160	230	224	219	210	202	196	189	186		182	181	180	*162*	136	112	93	76	62	52	42	33	27		100 000	15
1Cr½Mo															181	*174*	*149*	124	102	83	67	55	44	35	29	24		150 000	
															181	162	*138*	114	94	76	61	49	40	32	26	22		200 000	
															179	155	*131*	107	88	70	57	45	37	30	25	20		250 000	
622-490	490	275	5.2	≤ 160	183	176	169	163	157	153	149	145		137	135	132	*131*	118	105	94	82	72	61	53	45	39	34	100 000	6, 15
2¼Cr1Mo																132	*122*	108	97	85	73	63	56	48	42	36	31	150 000	
																130	*117*	104	92	79	68	59	52	45	38	33	28	200 000	
																126	*113*	100	87	75	65	57	49	42	36	32	27	250 000	
625-590	590	450	5.3	≤ 160																									1e, 15
5Cr½Mo																													
625-640	640	500	5.3	≤ 160																									1e, 15
5Cr½Mo																													
629-590	590	400	5.3	≤ 160	251	241	232	223	220	217	215	211	208	207	*195*	176	159	144	129	115	103	91	80	68	58	48	38	100 000	
9Cr1Mo														207	*188*	169	152	137	123	110	97	85	74	62	52	42	34	150 000	
														204	*183*	165	148	132	118	105	93	81	69	58	48	38	31	200 000	
														199	*179*	161	144	129	115	102	89	78	66	55	45	35	28	250 000	
509-650	650	490	9.3	≤ 75	277																								1d, 13
9Ni																													
509-690 9Ni	690	580	9.3	≤ 100	294																								1d, 13, 15

Table K.1-5 **Design strength values (N/mm²)** *(continued)*

Sections and bars																														
Grade, type and method of manufacture	R_m	R_e	Material group	Ruling section	Values of f_N for design temperatures (°C) not exceeding																								Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	740	750	h	
Austenitic stainless steel to BS 1502																														
304-S11	480	215	8.1	160	143	127	111	101	95	90	86	81	80																		
304-S31	490	230	8.1	160	153	136	119	109	103	98	93	89	87																		
304-S51	490	230	8.1	160	153	136	119	109	103	98	93	89	87	85			83		82	76	65	55	50	45	38	33	28			100 000	
																			82	72	61	51	46	42	36	31				150 000	
																			81	69	58	48	44	40	34	30				200 000	
																			78	66	55	46	42	38	33	29				250 000	
304-S61	550	305	8.1	160	203	174	149	135	127	121	116	110	107	104																	
304-S71	550	305	8.1	160	203	174	149	135	127	121	116	110	107	104																	
316-S11, S13	490	225	8.1	160	150	134	119	110	103	99	94	91	88																		
316-S31, S33	510	240	8.1	160	160	143	127	118	111	106	101	99	96																		
316-S51, S53	510	240	8.1	160	160	143	127	118	111	106	101	99	96	93			90		89	88	74	58	53	46	35	28	23			100 000	6
																				85	66	42	46	41	32	25	19			150 000	
																				79	62	48	43	38	29	22	18			200 000	
																				75	58	45	40	35	27	22	18			250 000	
316-S61, S63-S65, S67	580	315	8.1	160	210	180	154	142	133	127	123	119	116	113																	
321-S31	510	235	8.1	160	157	145	133	127	121	117	113	110	107																		
321-S51	490	190	8.1	160	127	115	103	97	93	87	84	81	79	78			77		76	76	*68*	55	49	44	34	26				100 000	
(490)																				*76*	*62*	50	44	38	29					150 000	
																				72	58	46	41	35	27					200 000	
																				68	*55*	44	38	33	25					250 000	
321-S51	510	235	8.1	160	157	145	133	127	121	117	113	110	107	104		103	102	102	*86*	71	57	42	36	31						100 000	
(510)																103	102	*93*	*78*	64	49	36	32	27						150 000	
																103	*97*	*89*	*74*	58	45	33	28	25						200 000	
																100	*93*	*85*	*71*	55	42	31	26	22						250 000	
347-S31	510	240	8.1	160	160	151	142	135	127	123	120	118	116																		
347-S51	510	240	8.1	160	160	151	142	135	127	123	120	118	116	115		114	113	113	*99*	82	66	53	47	41	31	23				100 000	
																	113	*112*	*93*	75	61	48	42	37	28	22				150 000	
																	113	*106*	88	72	58	45	39	34	25	19				200 000	
																	112	*102*	*85*	68	55	42	37	32	24	19				250 000	

Table K.1-6 **Design strength values (N/mm²)**

Forgings [see special Note 1b)]																							
Grade, type and method of manufacture	R_m	R_e	Material group	Ruling section	Values of f_N for design temperatures (°C) not exceeding																	Design lifetime	Notes[a]
	N/mm²	N/mm²		mm (see K.1.4.1.4)	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	h	
Carbon manganese steels to BS 1503																							
164-490	490	305	1.2	≤100*	203	187	171	156	142	128	121		115	114	*105*	90	77	65	56	48	42	100 000	15, 16, 19
		280		150*	187	173	159	145	137					*111*	95	81	68	58	50	43	38	150 000	
														104	*88*	75	63	54	46	40	34	200 000	
														98	*83*	70	59	51	43	37	32	250 000	
221-410	410	215	1.1	≤100	143	134	125	121	112	100	95		92		92	*90*	77	65	56	48	42	100 000	15, 19
		205		>100	137	126	116	113	108						92	*81*	68	58	50	43	38	150 000	
															88	75	63	54	46	40	34	200 000	
															83	70	59	51	43	37	32	250 000	
221-430	430	225	1.1	≤100	150	141	133	129	119	107	102		99	99	98	*90*	77	65	56	48	42	100 000	15, 19
		215		>100	143	133	124	121	115					99	*95*	*81*	68	58	50	43	38	150 000	
														99	88	75	63	54	46	40	34	200 000	
														98	83	70	59	41	43	37	32	250 000	
221-460	460	245	1.1	≤100	163	154	145	140	129	117	112		108	108	*105*	90	77	65	56	48	42	100 000	15, 19
		235		>100	157	146	135	131	125					108	*95*	81	68	58	50	43	38	150 000	
														104	*88*	75	63	54	46	40	34	200 000	
														98	*83*	70	59	51	43	37	32	250 000	
221-490	490	265	1.1	≤100	177	167	157	151	140	128	122	119	118	118	*105*	90	77	65	56	48	42	100 000	15, 19
		255		>100	170	158	146	141	135				118	*111*	*95*	81	68	58	50	43	38	150 000	
													118	*104*	*88*	75	63	54	46	40	34	200 000	
													115	*98*	*83*	70	59	51	43	37	32	250 000	
221-510	510	285	1.2	100	190	177	165	159	147	135	129	125	124	*121*	105	90	77	65	56	48	42	100 000	15, 19
													124	*111*	95	81	68	58	50	43	38	150 000	
													121	*104*	88	75	63	54	46	40	34	200 000	
													115	*98*	83	70	59	51	43	37	32	250 000	
221-530	530	295	1.2	100	197	185	173	167	155	141	136												14, 15, 19
221-550	550	305	1.2	100	203	192	181	175	162	149	142												14, 15, 19

Table K.1-6 **Design strength values (N/mm²)** *(continued)*

Forgings [see special Note 1b)]																							
Grade, type and method of manufacture	R_m	R_e	Material group	Ruling section	Values of f_N for design temperatures (°C) not exceeding																	Design lifetime	Notes[a]
	N/mm²	N/mm²		mm (see K.1.4.1.4)	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	h	
Carbon manganese steels to BS 1503 *(continued)*																							
223-410	410	245	1.1	≤100	163	150	138	130	120	106	99		92		91	90	77	65	56	48	42	100 000	15, 19
		230		>100	153	141	129	121	115						91	81	68	58	50	43	38	150 000	
															88	75	63	54	46	40	34	200 000	
															83	70	59	51	43	37	32	250 000	
223-430	430	260	1.1	≤100	173	160	148	139	128	114	107		100	99	98	*90*	77	65	56	48	42	100 000	15, 19
		245		>100	163	149	134	129	123						*95*	*81*	68	58	50	43	38	150 000	
															88	*75*	63	54	46	40	34	200 000	
															83	*70*	59	51	43	37	32	250 000	
223-460	460	290	1.2	≤100	193	179	164	151	140	127	119		111	110	*105*	90	77	65	56	48	42	100 000	15, 19
		270		>100	180	166	153	141	135					110	*95*	81	68	58	50	43	38	150 000	
														104	*88*	75	63	54	46	40	34	200 000	
														98	*83*	70	59	51	43	37	32	250 000	
223-490	490	320	1.2	≤100	209	194	179	165	152	139	131	125	123	*121*	105	90	77	65	56	48	42	100 000	15, 19
		295		>100	197	182	167	153	146				123	*111*	95	81	68	58	50	43	38	150 000	
													121	*104*	88	75	63	54	46	40	34	200 000	
													115	*98*	83	70	59	51	43	37	32	250 000	
223-510	510	340	1.2	≤100	217	203	190	173	159	147	138	132	131	121	105	90	77	65	56	48	42	100 000	15, 19
													128	*111*	95	81	68	58	50	43	38	150 000	
													121	*104*	88	75	63	54	46	40	34	200 000	
													115	*98*	83	70	59	51	43	37	32	250 000	
224-410	410	235	1.1	≤100	157	146	136	125	114	101	94		89	88	88	88	77	65	56	48	42	100 000	15, 19
		220		>100	147	137	127	117	109						88	81	68	58	50	43	38	150 000	
															88	75	63	54	46	40	34	200 000	
															83	70	59	51	43	37	32	250 000	
224-430	430	250	1.1	≤100	167	156	145	133	121	108	101		96	95	95	90	77	65	56	48	42	100 000	15, 19
		235		>100	157	146	135	123	117						*95*	81	68	58	50	43	38	150 000	
															88	75	63	54	46	40	34	200 000	
															83	70	59	51	43	37	32	250 000	

Table K.1-6 **Design strength values (N/mm²)** *(continued)*

Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Ruling section mm (see K.1.4.1.4)	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	Design lifetime h	Notes[a]
Forgings [see special Note 1b)]																							
Carbon manganese steels to BS 1503 *(continued)*																							
224-460	460	275	1.1	≤100	183	170	157	145	132	118	111		105	105	104	*90*	77	65	56	48	42	100 000	15, 19
		255		>100	170	158	147	135	127					105	*95*	*81*	68	58	50	43	38	150 000	
														104	*88*	*75*	63	54	46	40	34	200 000	
														98	*83*	*70*	59	51	43	37	32	250 000	
224-490	490	305	1.2	≤100	203	187	171	156	142	128	121	116	115	115	*105*	90	77	65	56	48	42	100 000	15, 19
		280		>100	187	173	159	145	137				115	*111*	*95*	81	68	58	50	43	38	150 000	
													115	104	*88*	75	63	54	46	40	34	200 000	
													115	98	*83*	70	59	51	43	37	32	250 000	
224-510	510	315	1.2	≤100	210	194	179	163	149	135	128	124	122	*121*	*105*	*90*	*77*	*65*	*56*	*48*	*42*	100 000	15, 19
													122	*111*	*95*	*81*	*68*	*58*	*50*	*43*	*38*	150 000	
													121	*104*	*88*	*75*	*63*	*54*	*46*	*40*	*34*	200 000	
													115	*98*	*83*	*70*	*59*	*51*	*43*	*37*	*32*	250 000	
225-490	490	340	1.2	≤100	209	199	189	172	160	147	137	131	130	*121*	105	90	77	65	56	48	42	100 000	15, 19
		300		>100	200	187	173	163	154	145			*128*	*111*	95	81	68	58	50	43	38	150 000	
													121	*104*	88	75	63	54	46	40	34	200 000	
													115	*98*	83	70	59	51	43	37	32	250 000	

Table K.1-6 **Design strength values (N/mm²)** *(continued)*

Forgings [see special Note 1b)]

Grade, type and method of manufacture	R_m	R_e	Material group	Ruling section	Values of f_N for design temperatures (°C) not exceeding																						Design lifetime	Notes[a]
	N/mm²	N/mm²		mm (see K.1.4.1.4)	50	100	150	200	250	300	350	400	440	450	460	480	490	500	510	520	530	540	550	560	570	580	h	
Alloy and martensitic stainless steel to BS 1503																												
243-430	430	250	1.1	≤ 200	167	161	155	145	133	111	102	99																1d, 15, 19
0.3Mo																												
620-440	440	275	5.1		183	166	149	142	131	123	113	108		105			101	101	*93*	76	62	52	42	33	27		100 000	15
1Cr½Mo																		101	83	67	55	44	35	29	24		150 000	
																		94	76	61	49	40	32	26	22		200 000	
																		88	70	57	45	37	30	25	20		250 000	
620-540	540	375	5.1	≤ 200	230	224	219	210	202	196	189	186		182	181	*162*	136	112	93	76	62	52	42	33	27		100 000	15
1Cr½Mo																*149*	124	102	83	67	55	44	35	29	24		150 000	
																138	114	94	76	61	49	40	32	26	22		200 000	
																131	107	88	70	57	45	37	30	25	20		250 000	
621-460	460	275	5.1		183	173	163	156	145	137	127	122		117		115	114	*112*	93	76	62	52	42	33	27		100 000	15
1¼Cr½Mo																	114	102	83	67	55	44	35	29	24		150 000	
																	114	*94*	76	61	49	40	32	26	22		200 000	
																	107	*88*	70	57	45	37	30	25	20		250 000	
660-460	460	300	6.1		196	190	184	178	161	150	144	139		135			134	133	*131*	115	100	87	76	66	56		100 000	15
CrMoV																		133	*122*	106	92	80	69	59	47		150 000	
																		132	*115*	100	87	75	65	55			200 000	
																		128	*111*	96	83	72	62	50			250 000	
271-560	560	370	4.2		238	229	220	215	208	204	199	192		188	186	*155*	129	107	87	69	54	41	30				100 000	19
MnCrMoV																145	118	95	76	60	44	32	23				150 000	
																132	108	87	68	51	38	26	18				200 000	
																129	105	83	65	48	35	23	14				250 000	
622-490	490	275	5.2		183	176	169	163	157	153	149	145		137	135	*131*	118	105	94	82	72	61	53	45	39	34	100 000	6, 15
2¼Cr1Mo																*122*	108	97	85	73	63	56	48	42	36	31	150 000	
																117	104	92	79	68	59	52	45	38	33	28	200 000	
																113	100	87	75	65	57	49	42	36	32	27	250 000	
622-560	560	370	5.2		238	231	224	219	213	209	201	197		*170*	157	131	118	105	94	82	72	61	53	45	39	34	100 000	
2¼Cr1Mo														*161*	148	122	108	97	85	73	63	56	48	42	36	31	150 000	
														156	143	117	104	92	79	68	59	52	45	38	33	28	200 000	
														152	139	113	100	87	75	65	57	49	42	36	32	27	250 000	

Table K.1-6 **Design strength values (N/mm²)** *(continued)*

Forgings [see special Note 1b)]																												
Grade, type and method of manufacture	R_m	R_e	Material group	Ruling section	Values of f_N for design temperatures (°C) not exceeding																						Design lifetime	Notes[a]
	N/mm²	N/mm²		mm (see K.1.4.1.4)	50	100	150	200	250	300	350	400	440	450	460	480	490	500	510	520	530	540	550	560	570	580	h	
Alloy and martensitic stainless steel to BS 1503 *(continued)*																												
622-650 2¼Cr1Mo	650	475	5.2																									1e, 15
625-520 5Cr½Mo	520	365	5.3																									1e, 15
625-590 5Cr½Mo	590	450	5.3																									1e, 15
503-490 3½Ni	490	300	9.2	≤100	200																							1d, 13, 15
509-690 9Ni	690	490	9.3	≤150	294																							1d, 13, 15
410-S21 12Cr	590	395	7.2		251	232	214	209	202	198	165	133																1d, 9, 15
420-S29 12Cr	700	515	7.2		298	295	293	287	279	272	215	158																1d, 9, 15
403-S17 12Cr	470	265	7.2		177	154	131	125	121	119	115	110																1d, 15
405-S17 12Cr	420	210	7.2		140	118	96	91	88	86	83	77																1d, 15
762-690 12Cr1Mo0.3Va	690	490	6.4	≤250	294	276	259	249	240	233	227	219		206			190	187	*173*	155	138	122	107	93	80	68	100 000	1d, 15
																		184	*168*	152	135	115	98	85	72	62	150 000	
																		180	*164*	146	128	110	94	80	68	58	200 000	
																		176	160	142	124	105	90	77	65	55	250 000	

Table K.1-6 **Design strength values (N/mm²)** *(continued)*

Forgings [see special Note 1b]

Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Ruling section mm (see K.1.4.1.4)	Values of f_N for design temperatures (°C) not exceeding: 50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	Design lifetime h	Notes[a]
Austenitic stainless steel to BS 1503																													
304-S11	480	215	8.1		143	127	111	101	95	90	86	81	80																
304-S31	490	230	8.1		153	136	119	109	103	98	93	89	87																
304-S51	490	230	8.1		153	136	119	109	103	98	93	89	87	85			83	82	82	*76*	65	55	50	45	38	33	28	100 000	
																			82	*72*	61	51	46	42	36	31		150 000	
																			81	*69*	58	48	44	40	34	30		200 000	
																			78	66	55	46	42	38	33	29		250 000	
304-S61	550	305	8.1		203	174	149	135	127	121	116	110	107	104			101												1e
310-S31	510	240	8.2																										
316-S11, S13	490	225	8.1		150	134	119	110	103	99	94	91	88																
316-S31, S33	510	240	8.1		160	143	127	118	111	106	101	99	96																
316-S51	510	240	8.1		160	143	127	118	111	106	101	99	96	93			90		89	88	*74*	58	53	46	35	28	23	100 000	6
																				85	*66*	52	46	41	32	25	19	150 000	
																				79	62	48	43	38	29	22	18	200 000	
																				75	58	45	40	35	27	22	18	250 000	
316-S61, S63	580	315	8.1		210	180	154	142	133	127	123	119	116	113			110			107									
318-S13	640	450	10.1		289	249	209	194	184	178																			
320-S33	510	245	8.1																										1e
321-S31	510	235	8.1		157	145	133	127	121	117	113	110	107																
321-S51	490	190	8.1		127	115	103	97	93	87	84	81	79	78			77		76	76	*68*	55	49	44	34	26		100 000	
(490)																				76	62	50	44	38	29			150 000	
																				72	58	46	41	35	27			200 000	
																				68	55	44	38	33	25			250 000	
321-S51	510	235	8.1		157	145	133	127	121	117	113	110	107	104		103	102	102	*86*	71	57	42	36	31				100 000	
(510)																103	101	93	78	64	49	36	32	27				150 000	
																103	97	89	74	58	45	33	28	25				200 000	
																100	93	85	71	55	42	31	26	22				250 000	
347-S31	510	240	8.1		160	151	142	135	127	123	120	118	116																
347-S51	510	240	8.1		160	151	142	135	127	123	120	118	116	115		114	113	113	*99*	82	66	53	47	41	31	24		100 000	
																	113	*112*	93	75	61	48	42	37	28	22		150 000	
																	113	106	88	72	58	45	39	34	25	19		200 000	
																	112	102	85	68	55	42	37	32	24	19		250 000	

Table K.1-7 **Design strength values (N/mm²)**

Castings																							
Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	**Material group**	**Thickness** mm	**Values of f_N for design temperatures (°C) not exceeding**																	**Design lifetime** h	**Notes**[a]
					50	**100**	**150**	**200**	**250**	**300**	**350**	**390**	**400**	**410**	**420**	**430**	**440**	**450**	**460**	**470**	**480**		
Carbon steel to BS 1504																							
161 430A	430	230	1.1		153	139	126	119	115	104	95		87										1c, 1d, 19
161 430E	430	230	1.1		153	139	126	119	115	104	95		93	93	92	*84*	71	60	51	44	37	100 000	1c, 19
														93	*89*	75	63	54	46	38	30	150 000	
														92	82	69	58	49	42	34		200 000	
														88	78	65	54	46	38			250 000	
161 480A	480	245	1.1		163	153	143	136	128	115	105	101											1c, 1d, 19
161 480E	480	245	1.1		163	153	143	136	128	115	109	107	107	106	*99*	84	71	60	51	44	37	100 000	1c, 19
													107	100	89	75	63	54	46	38	30	150 000	
													103	92	82	69	58	49	42	34		200 000	
													99	88	*78*	65	54	46	38			250 000	
161 540A	540	280	1.2		186	176	165	154	144	130	119		109										1c, 1d, 18

Table K.1-7 **Design strength values (N/mm²)** *(continued)*

Castings																												
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																						Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	440	450	460	480	490	500	510	520	530	540	550	560	570	580	h	
Low alloy steel to BS 1504																												
245A ½Mo	460	260	1.1		173	162	157	145	131	116	116	113		103														1c, 1d, 19
245E	460	260	1.1		173	167	161	155	140	121	116	113		110	109	108	95	*78*	62	51	41	32					100 000	1c, 19
½Mo																107	88	70	57	46	37	30					150 000	
																100	81	65	53	42	35	28					200 000	
																95	77	62	50	40	32	*25*					250 000	
503A 3½Ni	460	280	9.2		184																							1c
621A 1¼CrMo	480	280	5.1		187	175	161	146	141	136	123	114		108														1c, 1d
660A CrMoV	510	295	6.1		196	180	168	156	145	137	128	118		112				106										1c, 1d
622A 2¼CrMo	540	325	5.2		217	204	195	190	185	181	175	171																1c, 1d
622E	540	325	5.2		217	212	208	203	197	193	187	182	174	*170*	153	120	107	95	85	76	65	61	53	45	39	34	100 000	1c, 6
2¼CrMo													174	*161*	144	112	98	88	77	68	63	56	48	41	36	31	150 000	
													172	*156*	139	107	94	83	71	63	59	52	45	38	33	28	200 000	
													170	152	135	103	91	79	68	60	57	49	42	36	32	27	250 000	
623A 3½Cr½Mo	620	370	5.3		246	234	224	207	191	176	160	145																1c
625A, E 5Cr½Mo	620	420	5.3		264																							1c

Table K.1-7 **Design strength values (N/mm²)** *(continued)*

Castings																															
Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Thickness mm	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime h	Notes[a]
					50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	740	750		
High alloy steel to BS 1504																															
629A	620	420	5.4		264	240	216	200	184	176	157	144	123	87			55			21											1c, 1d
9Cr1Mo																															
420C 29A	620	450	7.2		264	246	228	222	219	214	176	138	100	62			26			14											1c, 1d
13Cr																															
425C 11A	770	620	7.2		328	328	328	290	255	220	192	170																			1c, 1d
13Cr 4N																															
304C 12A	430	215	8.1		143	132	120	110	103	98	95	93																			1c, 1d
304C 15A	480	240	8.1		160	143	127	115	109	104	101	99	95	92		88	88	87	*76*	65	55	45	42	38	33	28				100 000	1c, 20
																	88	*84*	*72*	61	51	42	39	36	31					150 000	
																	87	81	69	58	48	40	37	34	30					200 000	
																	85	78	66	55	46	38	35	33	29					250 000	
347C 17A	480	240	8.1		160	150	141	135	132	128	126	122	119	116	114	113	*109*	99	82	66	53	41	35	31	22					100 000	1c, 20
																112	102	93	75	61	48	37	32	28	22					150 000	
																106	98	88	72	58	45	34	29	25	19					200 000	
																102	94	85	68	55	42	32	28	24	19					250 000	
315C 16A	480	240			160	147	134	122	114	110	107	103	100	95																	1c
316C 12A	430	215	8.1		143	134	127	116	109	104	101	99																			1c
316C 16	480	240	8.1		160	147	134	122	114	110	107	103	100	95	94	93	93	92	*91*	74	58	46	41	35	28	23				100 000	1c, 20
																			85	66	52	41	36	32	25	19				150 000	
																			79	62	48	38	33	29	22	18				200 000	
																			75	58	45	35	31	27	22	18				250 000	
316C 71	510	260	8.1		173	156	140	129	122	116	110	106	102	96	94	93	93	92	91	74	58	46	41	35	28	23				100 000	1c, 20
																			85	66	52	41	36	32	25	19				150 000	
																			79	62	48	38	33	29	22	18				200 000	
																			75	58	45	35	31	27	22	18				250 000	
318C 17A	480	240	10.1		160	160	160	160	160	158	157	152	144	135	130																1c
317C 12A	430	215	8.1		143	134	127	116	109	104	101	99																			1c
317C 16A	480	240	8.1		160	147	134	122	114	110	107	103	100	95																	1c
364C 11A	430	200			134																										1c
332C 11A	430	200			134																										1c
310C 40A	450		8.2		143	137	130	124	116	112	107	104	101	97		80				69			58			46			32		1c
330C 11A	450				180	177	174	170	167	164	160	156	153	150		148	147			100			65			40			25		1c

Table K.1-8 **Design strength values (N/mm²)**

Steel boiler and superheater tubes																						
Grade, type and method of manufacture	R_m	R_0	Material group	Values of f_N for design temperatures (°C) not exceeding																	Design lifetime	Notes[a]
	N/mm²	N/mm²		50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	h	
Carbon steel to BS 3059-1																						
320	320	195	1.1	130	119	108	*97*	*86*	*77*	*71*		68										9
HFS, CFS, ERW and CEW																						
Carbon and carbon manganese steel tubes to BS 3059-2																						
360	360	235	1.1	153	139	125	117	111	*97*	*87*		*78*		77	76	*69*	60	52	44	36	100 000	9
S1, S2, ERW															*74*	65	56	48	40	32	150 000	
and CEW															71	62	53	45	37	28	200 000	
															68	59	51	42	35	26	250 000	
440	440	245	1.1	163	157	152	144	134	119	106		101	101	100	*90*	77	65	56	48	42	100 000	9
S1, S2, ERW													101	*95*	81	68	58	50	43	38	150 000	
and CEW													101	*88*	75	63	54	46	40	34	200 000	
													98	*83*	70	59	51	43	37	32	250 000	

Table K.1-8 **Design strength values (N/mm²)** *(continued)*

Steel boiler and superheater tubes																														
Grade, type and method of	R_m	R_e	Material group	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
manufacture	N/mm²	N/mm²		50	100	150	200	250	300	350	400	440	450	460	470	480	490	500	510	520	530	540	550	560	570	580	590	600	h	
Alloy steel to BS 3059-2																														
243	480	275	1.1					149	128	120	117		115		113	112	*95*	*78*	*62*	*51*	*41*	*32*							100 000	
0.3Mo																*107*	*88*	*70*	*57*	*46*	*37*	*30*							150 000	
S1, S2, ERW																*100*	*81*	*65*	*53*	*42*	*35*	*28*							200 000	
and CEW																*95*	*77*	*62*	*50*	*40*	*32*	*25*							250 000	
620-460	460	180	5.1							129	120		116			114	113	*112*	*93*	*76*	*62*	*52*	*42*	33	27				100 000	3, 9
1Cr½Mo																	113	102	*83*	*67*	*55*	*44*	*35*	29	24				150 000	
S1, S2, ERW																	113	*94*	*76*	*61*	*49*	*40*	*32*	26	22				200 000	
and CEW																	*107*	*88*	*70*	*57*	*45*	*37*	*30*	25	20				250 000	
622-490	490	275	5.2	183	176	169	163	157	153	149	145		137	135	133	*131*	118	105	*94*	*82*	*72*	*61*	*53*	45	39	34	29	26	100 000	2, 6
2¼Cr1Mo															133	122	108	*97*	*85*	*73*	*63*	*56*	*48*	42	36	31	27	23	150 000	
S1, S2															*130*	*117*	104	*92*	*79*	*68*	*59*	*52*	*45*	38	33	28	25	22	200 000	
															126	113	100	*87*	*75*	*65*	*57*	*49*	*42*	36	32	27	23	20	250 000	
629-470	470	185	5.4	123	110	*98*	*91*	*87*	*83*	*81*	*78*		*77*					*75*	*74*	*69*	*61*	*53*	*47*	42	38	34	30	26	100 000	8
9CrMo																			*74*	*65*	*56*	*49*	*44*	39	35	31	28	24	150 000	
S1, S2																			*71*	*62*	*54*	*47*	*42*	37	33	29			200 000	
																			68	*59*	*52*	*45*	*40*	35	32	28			250 000	
629-590	590	400	5.4	251	241	232	223	220	217	215	211	208	207	195	176	159	144	129	115	103	*91*	*80*	*68*	58	48	38	31	26	100 000	
9Cr1Mo													207	188	169	152	137	123	110	*97*	*85*	*74*	*62*	52	42	34	28	23	150 000	
S1, S2													*204*	*183*	165	148	132	118	105	*93*	*81*	*69*	*58*	48	38	31	25	22	200 000	
													199	*179*	161	144	129	115	102	*89*	*78*	*66*	*55*	45	35	28	24	21	250 000	
762	720	470	6.4	306	290	275	256	241	234	230	225		215			205	201	*191*	173	155	138	122	107	93	80	68	58	48	100 000	
12CrMoV																	*200*	184	168	152	135	115	*98*	85	72	62	52	44	150 000	
S1, S2																	197	180	164	146	128	110	*94*	80	68	58	49	41	200 000	
																	192	176	160	142	124	105	*90*	77	65	55	46	38	250 000	

Table K.1-8 **Design strength values (N/mm²)** *(continued)*

Steel boiler and superheater tubes																													
Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	**Material group**	**Values of f_N for design temperatures (°C) not exceeding**																								**Design lifetime** h	**Notes**[a]
				50	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**450**	**500**	**540**	**550**	**560**	**570**	**580**	**590**	**600**	**610**	**620**	**630**	**640**	**650**	**660**	**680**	**700**		
Austenitic steel to BS 3059-2																													
304-S51	490	230	8.1	157	139	121	109	101	97	94	91	88	86		83		82	82	81	76	71	65	60	55	50	45	38	33	100 000	9
CFS																		82	78	72	66	61	55	51	46	42	36	31	150 000	
																		81	75	69	63	58	53	48	44	40	34	30	200 000	
																		78	72	66	61	55	51	46	42	38	33	29	250 000	
316-S51, S52	510	240	8.1	163	146	128	118	111	106	102	98	96	93		90			89	88	88	82	74	66	58	53	46	35	28	100 000	6, 9
CFS																			88	85	75	66	58	52	46	41	32	25	150 000	
																			88	79	70	62	55	48	43	38	29	22	200 000	
																			85	75	66	58	52	45	40	35	27	22	250 000	
321-S51 (1010)	510	235	8.1	156	144	133	127	121	117	112	110	107	104	102	102	101	95	86	78	71	63	57	49	42	36				100 000	
CFS															101	93	86	78	72	64	56	49	42	36	32				150 000	
															97	89	82	74	66	58	52	45	38	33	28				200 000	
															93	85	78	71	62	55	48	42	35	31	26				250 000	
321-S51 (1105)	490	190	8.1	130	116	103	98	93	87	84	82	79	77		76				75	75	75	68	60	55	49	44	34		100 000	9
CFS																				75	68	62	56	50	44	38	29		150 000	
																				72	65	58	52	46	41	35	27		200 000	
																				68	62	55	49	44	38	33	25		250 000	
347-S51	510	240	8.1	163	155	146	138	135	130	124	122	119	119	117	117	116	109	99	91	82	74	66	59	53	47	41	31	23	100 000	9
CFS															117	112	102	93	84	75	68	61	55	48	42	37	28	22	150 000	
															116	106	98	88	79	72	64	58	51	45	39	34	25	19	200 000	
															112	102	94	85	76	68	61	55	48	42	37	32	24	19	250 000	
215-S15	540	270	8.1	180	168	156	144	141	139	136	135	133	132		130					126	125	124	123	107	87	71	52	40	100 000	
CFS																						124	114	93	75	62	46	35	150 000	
																						124	104	83	67	56	42	32	200 000	
																						118	96	76	62	52	40		250 000	

Table K.1-9 **Design strength values (N/mm²)**

Steel pipes and tubes for pressure purposes																							
Grade, type and method of manufacture	R_m	R_e	**Material group**	**Thickness**	**Values of f_N for design temperatures (°C) not exceeding**																	**Design lifetime**	**Notes**[a]
	N/mm²	N/mm²		mm	**50**	**100**	**150**	**200**	**250**	**300**	**350**	**390**	**400**	**410**	**420**	**430**	**440**	**450**	**460**	**470**	**480**	h	
Seamless and electric resistance welded carbon and carbon manganese steel to BS 3601																							
320	320	195	1.1	≤16	130	119	108	97	86	77	71		68										
ERW (BW)																							
360	360	235	1.1	≤16*	154	140	126	111	96	86	79		75										9, 16
ERW, S		225		16 ≤ 40*	150	136	122	109															
		215		40 ≤ 65*	143	131	119	107															
430	430	275	1.1	≤16*	183	166	149	132	115	103	95		89										9, 16
ERW, S, SAW		265		16 ≤ 40*	177	162	146	131															
		255		40 ≤ 65*	170	156	143	129															
Seamless and electric resistance welded including induction arc welded carbon and carbon manganese steel to BS 3602-1																							
360	360	235	1.1	≤16*	153	139	125	117	111	97	87		78		77	76	*69*	60	52	44	36	100 000	9, 16
HFS, CFS, ERW		225		16 ≤ 40*	150	137										*74*	*65*	56	48	40	32	150 000	
and CEW		215		40 ≤ 65*	143	134										*71*	*62*	53	45	37	28	200 000	
																68	*59*	51	42	35	26	250 000	
430	430	275	1.1	≤16*	183	166	149	135	120	109	101		95	95	*88*	79	69	60	52	44	36	100 000	16
HFS, CFS, ERW		265		16 ≤ 40*	177	163								93	83	74	65	56	48	40	32	150 000	
and CEW		255		40 ≤ 65*	170	161								90	80	71	62	53	45	37	28	200 000	
														87	*78*	68	59	51	42	35	26	250 000	
500-Nb	500	355	1.2	≤16*	213	204	194	178	163	148	135		125	*121*	105	90	77	65	56	48	42	100 000	16
HFS, CFS		345		16 ≤ 40*									125	*111*	95	81	68	58	50	43	38	150 000	
		355		40 ≤ 65*									*121*	*104*	88	75	63	54	46	40	34	200 000	
													115	98	83	70	59	51	43	37	32	250 000	
Longitudinally arc welded carbon and carbon manganese steel to BS 3602-2																							
430	430	250	1.1	≤16*	167	158	149	135	120	109	101												10, 16
LAW				40*	160	155																	
490	490	325	1.2	≤16	209	187	165	153	143	131	121												10
LAW		315		40																			

Table K.1-9 **Design strength values (N/mm²)** *(continued)*

Steel pipes and tubes for pressure purposes

Grade, type and method of manufacture	R_m N/mm²	R_e N/mm²	Material group	Thickness mm	Values of f_N for design temperatures (°C) not exceeding: 50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	Design lifetime h	Notes[a]
Seamless, electric resistance and induction welded carbon and alloy steel to BS 3603																							
430-LT	430	275	1.1	≤16*	183																		16
HFS, CFS, ERW		265		40*	177																		
and CEW		255		65*	170																		
503-LT	440	245	9.2	≤100	164																		
3¼Ni																							
HFS, CFS																							
509-LT	690	510	9.3	≤100	294																		
9Ni																							
HFS, CFS																							

Table K.1-10 **Design strength values (N/mm²)**

Steel pipes and tubes for pressure purposes																														
Grade, type and method of manufacture	R_m	R_e	**Material group**	**Values of f_N for design temperatures (°C) not exceeding**																									**Design lifetime**	**Notes**[a]
	N/mm²	N/mm²		**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**440**	**450**	**460**	**470**	**480**	**490**	**500**	**510**	**520**	**530**	**540**	**550**	**560**	**570**	**580**	**590**	**600**	h	
Seamless and electric resistance welded ferritic alloy steel to BS 3604-1																														
620-440	440	290	5.1	187	178	169	163	157	128	121	116		112			112	111	111	*93*	*76*	62	52	42	33	27				100 000	
1Cr½Mo																	111	*102*	*83*	*67*	55	44	35	29	24				150 000	
HFS, CFS,																	111	94	*76*	61	49	40	32	26	22				200 000	
ERW and CEW																	*107*	88	*70*	*57*	45	37	30	25	20				250 000	
621-420	420	275	5.1	179	169	158	152	145	116	110	105		101				100	100	*93*	76	62	52	42	33	27				100 000	
1¼Cr½Mo																		100	83	67	55	44	35	29	24				150 000	
HFS, CFS																		*94*	76	61	49	40	32	26	22				200 000	
																		88	70	57	45	37	30	25	20				250 000	
660-460	460	300	6.1	196	190	184	178	161	150	144	139		135				134	133	*131*	115	100	87	76	66	56				100 000	
½CrMoV																		133	122	106	92	80	69	59	47				150 000	
HFS, CFS																		132	115	100	87	75	65	55					200 000	
																		128	111	96	83	72	62	50					250 000	
622-490	490	275	5.2	183	176	169	163	157	153	149	145		137	135	133	*131*	118	105	94	82	72	61	53	45	39	34	29	26	100 000	2, 6
2¼Cr1Mo															133	*122*	108	97	85	73	63	56	48	42	36	31	27	23	150 000	
HFS, CFS															*130*	*117*	104	92	79	68	59	52	45	38	33	28	25	22	200 000	
															126	113	100	87	75	65	57	49	42	36	32	27	23	20	250 000	
625-450	450	170	5.3																											1e
5Cr½Mo																														
HFS, CFS																														
629-470	470	185	5.4	123	110	98	91	87	83	81	78		77					75	74	69	61	53	47	42	38	34	30	26	100 000	8
9Cr1Mo																			*74*	65	56	49	44	39	35	31	28	24	150 000	
HFS, CFS																			71	62	54	47	42	37	33	29			200 000	
																			68	59	42	45	40	35	32	28			250 000	

Table K.1-10 **Design strength values (N/mm²)** *(continued)*

Steel pipes and tubes for pressure purposes																														
Grade, type and method of manufacture	R_m	R_e	**Material group**	**Values of f_N for design temperatures (°C) not exceeding**																									**Design lifetime**	**Notes**[a]
	N/mm²	N/mm²		**50**	**100**	**150**	**200**	**250**	**300**	**350**	**400**	**440**	**450**	**460**	**470**	**480**	**490**	**500**	**510**	**520**	**530**	**540**	**550**	**560**	**570**	**580**	**590**	**600**	h	
Seamless and electric resistance welded ferritic alloy steel to BS 3604-1 *(continued)*																														
629-590	590	400	5.4	251	241	232	223	220	217	215	211	208	207	*195*	176	159	144	129	115	103	91	80	68	58	48	38	31	26	100 000	
9Cr1Mo													207	188	169	152	137	123	110	97	85	74	62	52	42	34	28	23	150 000	
HFS, CFS													*204*	*183*	165	148	132	118	105	93	81	69	58	48	38	31	25	22	200 000	
													199	179	161	144	129	115	102	89	78	66	55	44	35	28	24	21	250 000	
762-720	690	490	6.4	294	276	259	249	240	232	227	217		206				191	187	*173*	155	138	122	107	93	80	68	58	48	100 000	
12CrMoV																		*184*	168	152	135	115	98	85	72	62	52	44	150 000	
HFS, CFS																		180	164	146	128	110	94	80	68	58	49	41	200 000	
																		176	*160*	142	124	105	90	77	65	55	46	38	250 000	
591	610	440	2.1	260	260	260	260	260	255	249	229																			
1¼NiCuMo																														
Nb																														
Longitudinally arc welded ferritic alloy steel to BS 3604-2																														
620	480	340	5.1	204	192	180	172	165	152	143	141		137																	10
1Cr½Mo																														
LAW																														
621	515	340	5.1	219	211	203	194	187	177	170	167		163																	10
1¼Cr½Mo																														
LAW																														
622	515	310	5.2	207	198	189	183	177	173	168	163	156																		10
2¼Cr1Mo																														
LAW																														

Table K.1-11 **Design strength values (N/mm²)**

Steel pipes and tubes for pressure purposes																														
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																								Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	630	640	650	660	680	700	720	h	
Seamless and austenitic stainless steel to BS 3605-1																														
304-S11	480	215	8.1		143	127	111	101	95	90	86	81	80																	
304-S31	490	230	8.1		157	139	121	109	101	97	94	91	88																	
304-S51	490	230	8.1		157	139	121	109	101	97	94	91	88	86			83	82	82	*76*	65		55	50	45	38	33	28	100 000	
																			82	72	61		51	46	42	36	31		150 000	
																			81	69	58		48	44	40	34	30		200 000	
																			78	*66*	55		46	42	38	33	29		250 000	
316-S11, S13	490	225	8.1		150	135	119	110	103	99	94	91	88																	
316-S31, S33	510	240	8.1		163	146	128	118	111	106	102	98	96																	
316-S51, S52	510	240	8.1		163	146	128	118	111	106	102	98	96	93			90		89	88	*74*		58	53	46	35	28	23	100 000	6
																				85	*66*		*52*	46	41	32	25	19	150 000	
																				79	62		48	43	38	29	22	18	200 000	
																				75	58		45	40	35	27	22	18	250 000	
321-S31	510	235	8.1		156	144	133	127	121	117	112	110	107																	
321-S51	510	235	8.1		156	144	133	127	121	117	112	110	107	104		102	102	101	86	71	57		*42*	36	31				100 000	
(1010)																102	101	93	78	64	49		36	32	27				150 000	
																102	97	89	74	58	45		33	28	25				200 000	
																100	93	85	71	55	42		31	26	22				250 000	
321-S51	490	190	8.1		130	116	103	98	93	87	84	82	79	77			76		76	75	*68*		*55*	49	44	34	26		100 000	
(1105)																				75	62		50	44	38	29			150 000	
																				72	58		46	41	35	27			200 000	
																				68	55		44	38	33	25			250 000	
347-S31	510	240	8.1		163	155	146	138	135	130	124	122	119																	
347-S51	510	240	8.1		163	155	146	138	135	130	124	122	119	119		117	117	116	*99*	82	66	59	53	47	41	31	23		100 000	
																	117	*112*	*93*	75	61	54	48	42	37	28	22		150 000	
																	116	106	88	72	58	51	45	39	34	25	19		200 000	
																	112	102	85	68	55	48	42	37	32	24	19		250 000	
215-S15	540	270	8.1		180	168	156	144	141	139	136	135	133	132			130			126	124	123	*107*	87	71	52	40		100 000	
																					124	*114*	*93*	75	62	46	35		150 000	
																					124	104	83	67	56	42	32		200 000	
																					118	96	76	62	52	40			250 000	

Table K.1-11 **Design strength values (N/mm²)** *(continued)*

Steel pipes and tubes for pressure purposes														
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding									Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	450	
Longitudinally welded austenitic stainless steel to BS 3605-2														
304-S11	480	215	8.1		143	127	111	101	95	90	86	81	80	10
304-S31	490	230	8.1		157	139	121	109	101	97	94	91	88	10
316-S11, S13	490	225	8.1		150	135	119	110	103	99	94	91	88	10
316-S31, S33	510	240	8.1		163	146	128	118	111	106	102	98	96	10
321-S31	510	235	8.1		156	144	133	127	121	117	112	110	107	10
347-S31	510	240	8.1		163	155	146	138	135	130	124	122	119	10

Table K.1-12 **Design strength values (N/mm²)**

Steel tubes for heat exchangers																							
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																	Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	390	400	410	420	430	440	450	460	470	480	h	
Carbon steel to BS 3606																							
320	320	195	1.1		130	119	108	97	86	77	71		68										17
CFS, ERW and																							
CEW																							
400	400	230	1.1		153	142	130	123	110	96	88		86		84	*79*	69	60	52	44	36	100 000	
CFS, ERW and															83	74	65	56	48	40	32	150 000	
CEW															80	*71*	62	53	45	37	28	200 000	
															78	68	59	51	42	35	26	250 000	
440	440	265	1.1		163	157	152	144	134	119	106		101	101	100	*90*	77	65	56	48	42	100 000	
CFS, ERW and														101	*95*	*81*	68	58	50	43	38	150 000	
CEW														101	88	75	63	54	46	40	34	200 000	
														98	83	70	59	51	43	37	32	250 000	

Table K.1-12 **Design strength values (N/mm²)** *(continued)*

Steel tubes for heat exchangers																															
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
	N/mm²	N/mm²		mm	50	100	150	200	250	300	350	400	440	450	460	470	480	490	500	510	520	530	540	550	560	570	580	590	600	h	
Alloy steel to BS 3606																															
243-480	480	275	1.1		183	175	166	158	149	128	120	117		115		113	112	*95*	78	62	51	41	32							100 000	
0.3Mo																	*107*	*88*	70	57	46	37	30							150 000	
CFS, ERW																	100	81	65	53	42	35	28							200 000	
and CEW																	95	77	62	50	40	32	25							250 000	
261-540	540	400	1.3		230	230	230	230	230	230	230	227																			
½Mo																															
CFS, ERW																															
and CEW																															
620-460	460	180	5.1								129	120		116			114	113	*112*	93	76	62	52	42	33	27				100 000	3, 9
1Cr½Mo																		113	102	83	67	55	44	35	29	24				150 000	
CFS, ERW																		*113*	*94*	76	61	49	40	32	26	22				200 000	
and CEW																		107	88	70	57	45	37	30	25	20				250 000	
621-420	420	275	5.1		179	169	158	152	145	116	110	105		101			100	100	100	*93*	76	62	52	42	33	27				100 000	
1¼Cr½Mo																			100	83	67	55	44	35	29	24				150 000	
CFS, ERW																			*94*	*76*	61	49	40	32	26	22				200 000	
and CEW																			88	*70*	57	45	37	30	25	20				250 000	
622-490	490	275	5.2		183	176	169	163	157	153	149	145		137	135	133	*131*	118	105	94	82	72	61	53	45	39	34	29	26	100 000	2, 6
2¼Cr1Mo																133	122	108	97	85	73	63	56	48	42	36	31	27	23	150 000	
CFS																*130*	117	104	92	79	68	59	52	45	38	33	28	25	22	200 000	
																126	113	100	87	75	65	57	49	42	36	32	27	23	20	250 000	
625-450	450	170	5.3																												1e
5Cr½Mo																															
CFS																															
439	415	205	7.1																												1e
18Cr																															

Table K.1-12 **Design strength values (N/mm^2)** *(continued)*

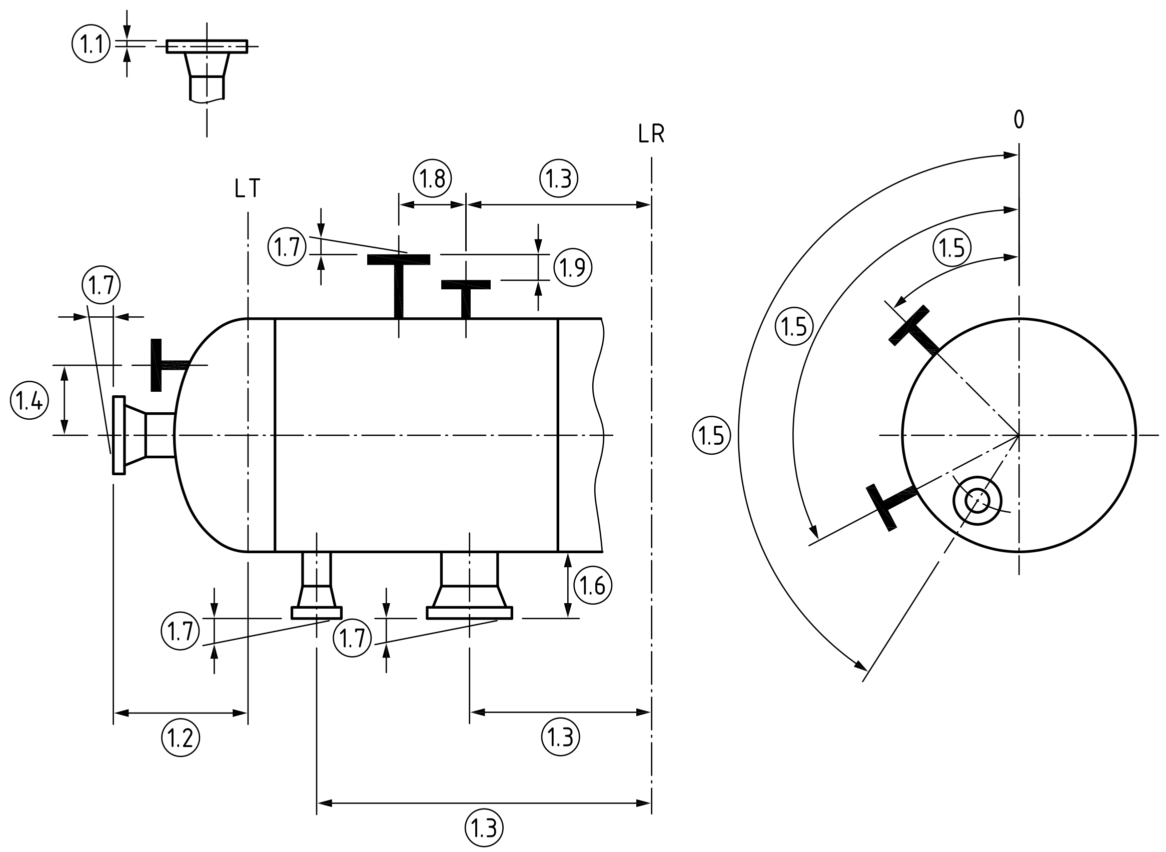
Steel tubes for heat exchangers																															
Grade, type and method of manufacture	R_m	R_e	Material group	Thickness	Values of f_N for design temperatures (°C) not exceeding																									Design lifetime	Notes[a]
	N/mm^2	N/mm^2		mm	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	740	750	h	
Austenitic stainless steel to BS 3606																															
304-S11	490	205	8.1		137	124	111	101	95	90	86	81	80																		
CFS, LWHT,																															
LWCF																															
and LWBC																															
304-S31	490	235	8.1		157	139	121	109	101	97	94	91	88	86			83	82	82	*76*	65	55	50	45	38	33	28			100 000	20
CFS, LWHT,																			82	72	61	51	46	42	36	31				150 000	
LWCF																			81	69	58	48	44	40	34	30				200 000	
and LWBC																			78	66	55	46	42	38	33	29				250 000	
316-S11, S13	490	215	8.1		143	131	119	110	103	99	94	91	88																		
CFS, LWHT,																															
LWCF																															
and LWBC																															
316-S31, S33	510	245	8.1		163	146	125	118	111	106	102	98	96	93			90		89	88	*74*	58	53	46	35	28	23			100 000	6, 20
CFS, LWHT,																				*85*	66	52	46	41	32	25	19			150 000	
LWCF																				79	62	48	43	38	29	22	18			200 000	
and LWBC																				75	58	45	40	35	27	22	18			250 000	
321-S31	510	235	8.1		156	144	133	127	121	117	112	110	107	104	103	102	102	101	*86*	71	57	42	36	31						100 000	20
CFS, LWHT																102	101	93	78	64	49	36	32	27						150 000	
LWCF																102	97	89	74	58	45	33	28	25						200 000	
and LWBC																*100*	93	85	71	55	42	31	26	22						250 000	
347-S31	510	245	8.1		163	155	146	138	135	130	124	122	119	119		117	117	*116*	99	82	66	53	47	41	31	23				100 000	20
CFS, LWHT,																	117	112	93	75	61	48	42	37	28	22				150 000	
LWCF																	*116*	106	88	72	58	45	39	34	25	19				200 000	
and LWBC																	112	102	85	68	55	42	37	32	24	19				250 000	

Annex L Guidance on structural tolerances

Figure L.1, Figure L.2, Figure L.3 and Figure L.4 of this annex give general structural tolerances. They are supplementary to the requirements of **4.2.3**, **4.2.4** and **4.2.5** and are for guidance.

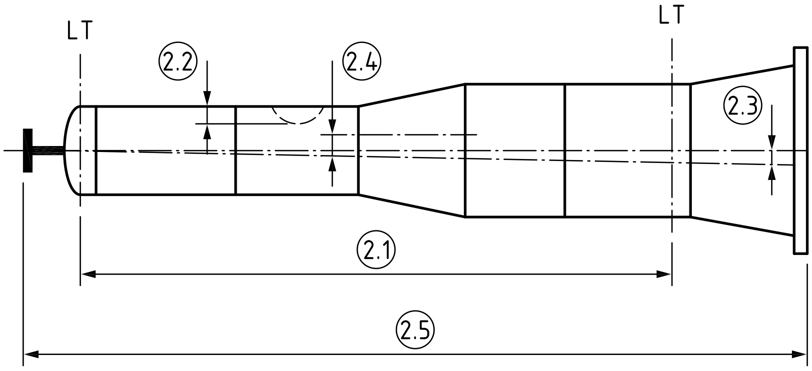
Any agreement to use these tolerances shall not remove the obligation to comply with Section 4.

Figure L.1 **Tolerances on nozzles**



Item no.	Type of deviations and elements considered		Maximum deviations authorized
1.1	Levelness of flat joint span expressed as a function of joint thickness		0.2e
1.2	Deviation between the surface of a flange and the tangential line (LT) of an end or the reference line (LR)		±5 mm
1.3	Deviation between axis of a nozzle and the reference line (LR)	Connection nozzle ≤100 mm	±5 mm
		Other nozzles and manholes	±10 mm
1.4	Deviation between the axis of a nozzle with an axis parallel to that of the vessel		±5 mm
1.5	Deviation in relation to the theoretical orientation measured by the circumferential deviation between the reference generating lines and the nozzle	Connection nozzle	±5 mm
		Manhole	±10 mm
1.6	Deviation between flange facing and vessel wall	Connection nozzle	±5 mm
		Manhole	±10 mm
1.7	Slope of the flange facing in relation to theoretical plane	Connection nozzle	±½°
		Manhole	±1°
		For measurement apparatus	±¼°
1.8	Deviation between nozzle axes for measurement apparatus		±1.5 mm
1.9	Difference in level between the two flange facings for measuring device		±1 mm

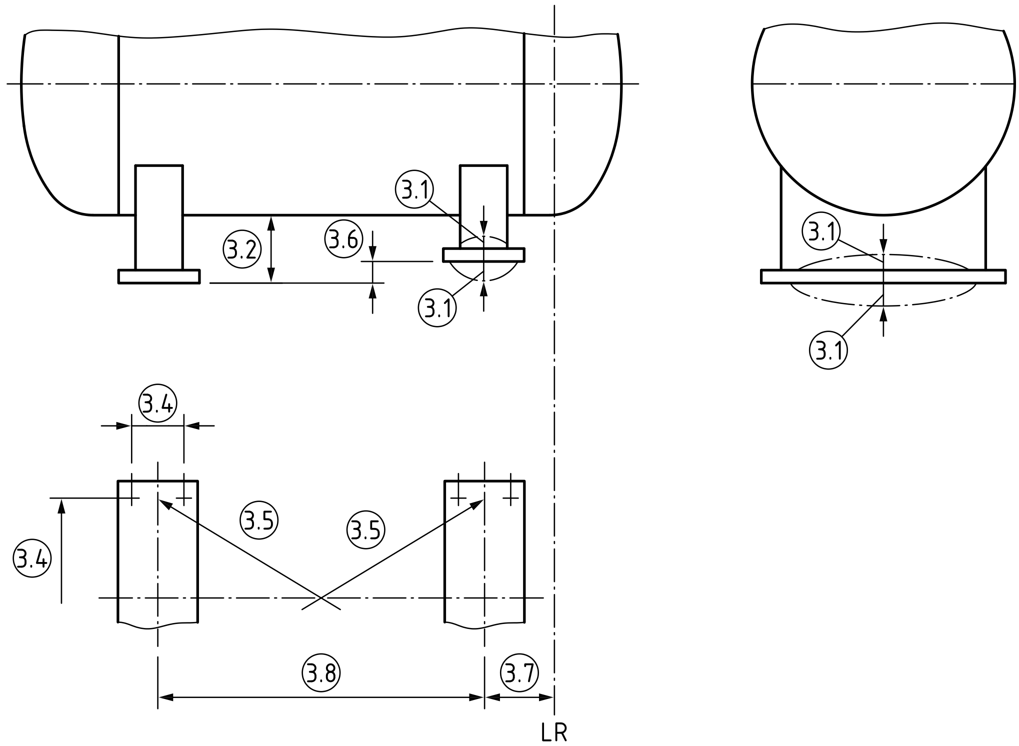
Figure L.2 **Tolerances after erection of a vertical[a) vessel**



Item no.	Type of deviations and elements considered		Maximum deviations authorized
2.1	Difference in length over distance *L* of extreme tangential lines (LT)	*L* ≤ 30 000 mm	±15 mm
		L > 30 000 mm	±20 mm
2.2	Wall straightness	Local defect measured on generating line	±6 mm
2.3	Deviation between main axis of vessel and the vertical *L* (2.1) of vessel		±min (0.001 *L*; 30 mm)
2.4	Concentricity deviation of two sections with different diameters, expressed as a function of the greater diameter *D*		±min (0.003 *D*; 20 mm)
2.5	Deviation over total height or overall length of the vessel		Cumulative tolerances

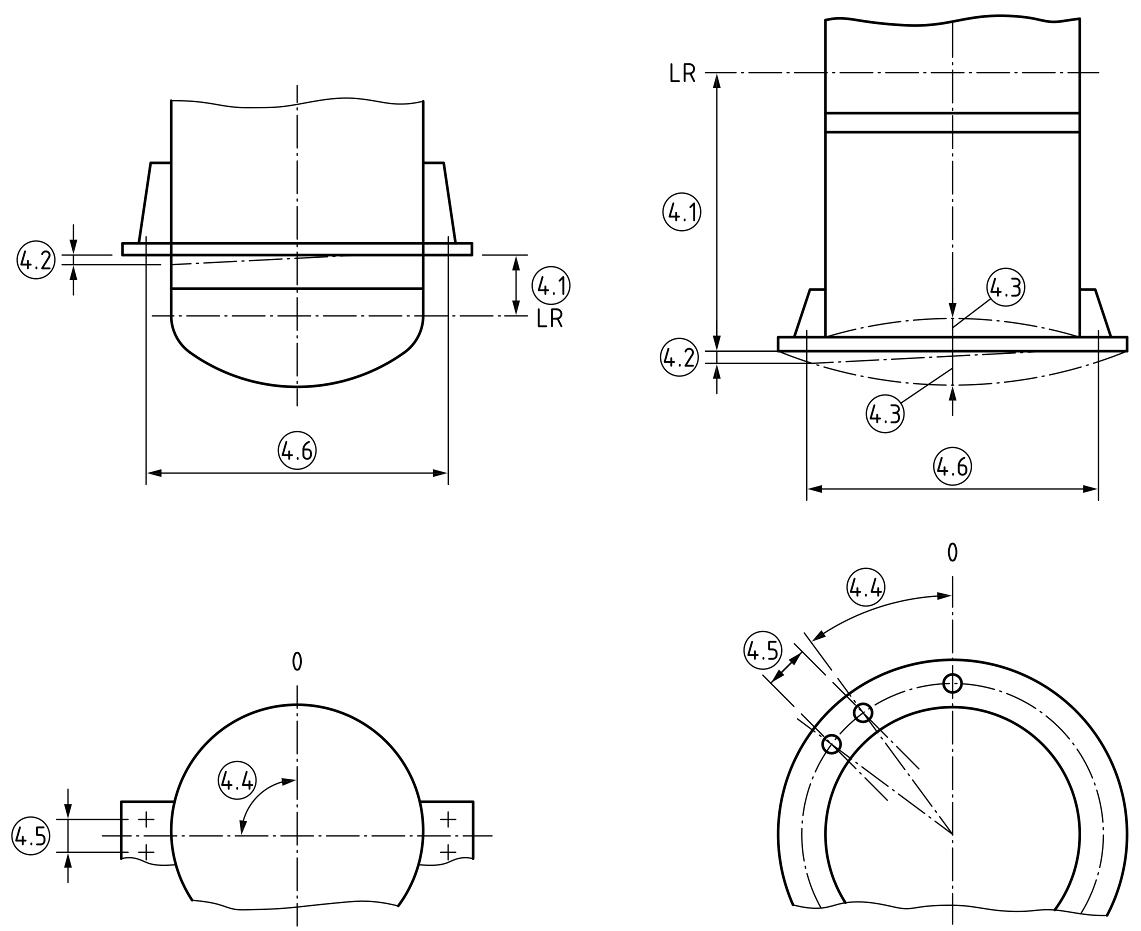
a) Items 2.1 and 2.5 also apply to horizontal vessels.

Figure L.3 **Tolerances on saddles and supports for horizontal vessels**



Item no.	Type of deviations and elements considered		Maximum deviations authorized
3.1	Flatness deviation of the bearing surface of a support	Transverse direction	±2 mm
		Longitudinal direction	±4 mm
3.2	Deviation between bearing sole plate and lower generating line of vessel		±3 mm
3.3	Deviation between axes of extreme supports		±5 mm
3.4	Deviation between axes of bolt holes		±3 mm
3.5	Deviation between diagonals of end saddles		±6 mm
3.6	Deviation between levels of end bearing sole plates		0 +5 mm
3.7	Deviation between saddle axis and vessel tangential or reference line (LR)		±5 mm

Figure L.4 **Tolerances on saddles and supports for vertical vessels**



Item no.	Type of deviations and elements considered		Maximum deviations authorized
4.1	Difference in the distance between the lower surface of supports or the base ring and the reference line (LR)		±6 mm
4.2	Perpendicularity defect of supports or base ring in relation to vessel axis or skirt		±6 mm
4.3	Flatness defect		±4 mm
4.4	Orientation deviation of axis of supports or skin reference hole	$D \leq 3\,000$ mm	±4 mm
		$3\,000 \text{ mm} < D \leq 6\,000$ mm	±8 mm
		$D > 6\,000$ mm	±12 mm
4.5	Deviation between two bolt holes		±3 mm
4.6	Anchoring diameter deviation, expressed as a function of theoretical diameter D		±min (0.002 D; 10)

Annex M Requirements for establishing the allowable external pressure for cylindrical sections outside the circularity limits specified in 3.6

M.1 General

This annex provides a procedure to determine the allowable pressure for cylinders with an out-of-roundness greater than 0.5% of radius measured from the true centre (see **3.6.2.1**).

M.2 Notation

For the purposes of this annex the symbols specified in **3.6** apply, except where modified as follows.

a_n, b_n	are Fourier series coefficients;
e	is the analysis thickness of the cylinder;
n	is the harmonic value used in Equation (M-4), and is equivalent to n_{cyl} used to evaluate ε in **3.6.2.1**;
N	is the number of measurements of radius R_r, $N \geq 24$;
p(allowable)	is the allowable external pressure as assessed by this annex, for a cylinder outside the circularity limits specified in **3.6**;
p_a	is the allowable external pressure for an otherwise similar cylinder within 0.5% tolerance (see **3.6.2.1**);
p_q	is the lower bound estimate of the collapse pressure of the cylinder;
r	is the identifier of each of the points on the shell to which the radius, R_r, is measured, r = 0 to $N - 1$;
R_r	is the radial measurement from the *assumed* centre, location to r;
W_{max}	is the maximum departure from mean circle (see **3.6.8**);
φ	is the angular increment of the measurement points, φ = 15° for N = 24.

M.3 Method

The allowable pressure shall be determined from the following equation:

$$p(\text{allowable}) = \frac{p_q}{1.5} + \left(p_a - \frac{p_q}{1.5}\right)\frac{0.005R}{|W_{max}|} \leq p_a \qquad \text{(M-1)}$$

where p_q is the lowest value of p at any location r at which:

$$\frac{pR}{e} + |\sigma_{br}| = sf \qquad \text{(M-2)}$$

$$\text{and } p_q \leq 1.5p_a \qquad \text{(M-3)}$$

where σ_{br} is given by:

$$\sigma_{br} = \frac{Ee}{2R^2(1-\nu^2)}\sum_{n=2}^{n=N/2}\left\{(n^2-1)+\nu\left(\frac{\pi R}{L}\right)^2\right\} \times \left\{p/\left(p_{m(n)}-p\right)\right\} \times$$

$$(a_n \sin nr\varphi + b_n \cos nr\varphi) \qquad \text{(M-4)}$$

in which $p_{m(n)}$ is the value of p_m from Equation (3.6.2-8) of **3.6.2.1**. In Equation (3.6.2-8) ε is determined using the same value of n as used in Equation (M-4).

$$a_n = \frac{2}{N}\sum_{r=0}^{N-1} R_r \sin(nr\varphi) \qquad \text{(M-5)}$$

$$b_n = \frac{2}{N}\sum_{r=0}^{N-1} R_r \cos(nr\varphi) \text{ when } n \neq N/2 \tag{M-6}$$

$$b_n = \frac{1}{N}\sum_{r=0}^{N-1} R_r \cos(nr\varphi) \text{ when } n = N/2 \tag{M-7}$$

If p_q is greater than $1.5p_a$ then p(allowable) = p_a.

*NOTE Calculation of p_q can be carried out by trial and error methods or a systematic iteration in order to solve Equation (M-2). A computer should be used. Example 3 of **W.1.2** is a worked example of the method.*

Annex N Requirements for vessel design and the provision of information concerning UK statutory requirements for the demonstration of the continued integrity of pressure vessels throughout their service life

N.1 General

N.1.1 This annex is mandatory for vessels to be used in Great Britain when the requirements of the Pressure Systems Safety Regulations (PSSR) [1] apply. Vessels subject to the Pressure Equipment (Safety) Regulations (PER) [4] are excepted from the requirements of the PSSR on design, construction and provisions for information and marking, as these requirements are then covered by the PER. For air receivers consideration shall be given to the requirements of the Simple Pressure Vessels (Safety) Regulations [2]. The provisions in this annex, to be specified in the purchase specification, are optional in circumstances where the aforementioned legislation does not apply. For the benefit of users in Great Britain the wording of this annex is written in the normative form, to suit the former circumstances.

NOTE The use of normative phraseology is adopted in the context of the need to conform with these requirements for the purpose of being compliant with this specification. Whilst this annex refers to legislation it cannot amend or interpret that legislation. Compliance with a Published Document does not in itself confer immunity from legal obligations.

N.1.2 Obligations on the parties under this annex are intended to be consistent with the regulations quoted in **N.1.1**. The intention is to clarify the requirements as they may apply to pressure vessels compliant with this specification. Reference should also be made to the Approved Code of Practice to the Pressure Systems Safety Regulations 2000 [3].

N.1.3 The scope of this annex is limited to:

a) the identification of the responsibilities of the manufacturer of a vessel (as defined in **1.3.2**) in respect of the design of the vessel to allow examination;

b) the identification of the responsibilities of the manufacturer to provide information relevant to the obligations of the owner/user to undertake a risk assessment, establish safe operating limits and a written scheme of examination;

c) the definition of the information to be included in the purchaser's specification so as to allow the manufacturer to fulfil the obligations defined herein.

N.1.4 It is not the intention of this annex to impose obligations on the parties involved for the consideration of issues outside their knowledge, competence and responsibilities or as established by the contractual relationship for the provision of the vessel. The manufacturer can only account for the design/operating conditions he has been made aware of. If the manufacturer does not carry out the design, the actual designer shall assume responsibilities, as are herein ascribed to the manufacturer, for those design aspects.

N.1.5 It is only the ultimate owner/user of the vessel that has the process, plant characteristics and environmental information, in respect of references [1] and [3], to decide what consideration should be given to the determination of the risk to persons and property consequent to the failure of a pressure vessel; and thus what provisions should be made for limiting that risk through the establishment of appropriate plans and a written scheme of examination.

N.2 Purchaser specification

N.2.1 The potential for failure of a vessel is increased if the operating conditions are more onerous than those used in the vessel design, thus the purchaser shall provide a sufficiently comprehensive specification appropriate to the potential consequences of vessel failure. For vessels subjected to fatigue and/or creep loading, the specification shall provide a definition of the full range of conditions likely to be encountered and the expected operating life of the vessel.

NOTE It is essential that adequate consideration is given to the pressure system design so that realistic fluid conditions can be specified for the expected operating life.

N.2.2 The purchaser shall, from knowledge of the potential consequences of vessel failure and the intended approach, both in respect of the risk assessment and the type of written scheme of examination, specify in the purchase specification the extent of information to be provided by the manufacturer, beyond that defined in **N.3**.

NOTE Annex S provides an option for the provision of potentially useful information, i.e. design calculations.

N.2.3 If the purchaser wishes the manufacturer to make recommendations for periodic examination this should be defined in the purchase specification. This specification does not recommend plans, techniques or frequencies for periodic examination.

NOTE The purchaser will need to identify conditions or physical restrictions which would limit the access to or into a vessel if they could influence the ability to undertake any periodic examination. If it is likely that the external vessel surface will have to be examined then the design of any insulation and attachments will need to be integrated with the scheme of examination.

N.3 Provision of information by the manufacturer

N.3.1 Regulation 5 of reference [1] requires that any person who designs or supplies a pressure vessel, to which those regulations apply, shall provide sufficient written information concerning its design, construction, examination, operation and maintenance as may reasonably be foreseen to be needed to enable the owner/user of the vessel and any other manufacturers and installers involved to comply with the regulations.

N.3.2 The information provided in accordance with **5.8.9**, for the nameplate of a vessel, is sufficient to satisfy the marking requirements of reference [1] but does not cover the information requirements of this annex.

N.3.3 The minimum information required from the manufacturer shall be derived from that which he has knowledge of and control over. The manufacturer shall indicate the potential susceptibility to failure of the vessel from knowledge of the margins between actual and design values of stress, the actual material properties and the existence of imperfections.

N.3.4 Based upon the manufacturer’s control and knowledge of the aforementioned margins the following types of information shall be provided for each vessel.

a) The safe operating limits of pressure, temperature (maximum and/or minimum) and where appropriate the allowable number of load cycles and the operating life of the vessel, or sufficient information to allow the user to establish the safe operating limits.

b) Corrosion allowances, as supplied, and minimum allowable metal thicknesses.

c) The nature, location and extent of any concession or accepted non conformance to this specification and the purchase order, with a definition of any special monitoring required to allow the above limits to be achieved.

d) The locations where the design and/or operating conditions give the lowest margin to the allowable stress and, where such conditions are relevant, the lowest margin against usage of the creep or fatigue life.

N.3.5 The manufacturer shall recommend (consistent with the safe operating limits) suitable operational and maintenance procedures to ensure that, if followed, the vessel will continue to be satisfactory for its specified safe operating limits. These maintenance procedures are not periodic examination procedures or written schemes, but actions to control any deterioration of the vessel from that condition it was specified to be provided in e.g., maintaining corrosion protection.

N.3.6 For vessels designed for low temperature duty the minimum metal temperature is the lowest temperature during each of the following conditions;

– normal operations;

– start up and shut down procedures;

– possible process upsets;

– when pressure or leak testing.

The safe operating limits shall be presented in the form of a permissible pressure/temperature envelope. This should be made up from the various θ_D covering the conditions assessed i.e., including those listed above (or the θ_p if one exists for that condition).

NOTE θ_D and θ_p are defined in Annex D.

N.4 Manufacturer's responsibilities for provision of certain features

N.4.1 Regulation 4 of reference [1] places responsibilities on manufacturers to properly design and construct vessels from suitable material so as to prevent danger, facilitate all necessary examinations (including providing safe access where appropriate) and provide such protective devices as may be necessary.

N.4.2 Requirements are identified in **3.12** for the provision of access and examination openings.

N.4.3 Requirements are identified in **3.13** for the provision of over pressure protection. The need for other protection, such as temperature measuring and limiting devices, would need special consideration outside this specification when such risks exist.

N.5 References

[1] GREAT BRITAIN. Pressure Systems Safety Regulations, 2000. Statutory Instrument 2000 No. 128, ISBN 0 11 085836 0, London: The Stationery Office.

[2] GREAT BRITAIN. The Simple Pressure Vessels (Safety) Regulations, 2016. Statutory Instrument 2016 No. 1092, ISBN 978 0 1111 5138 9, London: The Stationery Office.

[3] *Safety of pressure systems*, Pressure Systems Safety Regulations, Second edition 2014, L122 — Approved Code of Practice, ISBN 978 0 7176 6644 7, HSE Books.

[4] GREAT BRITAIN The Pressure Equipment (Safety) Regulations 2016. Statutory Instrument 2016 No. 1105, ISBN 978 0 1111 5149 5.

Annex Q Recommendations for preparation and testing of production control test plates

Q.1 Preparation of production control test plates (see 5.4.1)

Q.1.1 The material used for the test plates should comply with the same specification as that used in the construction of the vessel and should be manufactured by the same steel making process. The plates should be of the same nominal thickness as the shell and preferably selected from the same batch of material as that used in fabricating the vessel.

The test plates should be sufficiently large to allow for the preparation of all specimens required in **5.2.3** and Table 2 of BS EN ISO 15614-1:2017+A1. In any case the length of the plates should be not less than 350 mm.

Q.1.2 When a vessel includes one or more longitudinal seams the test plates should, wherever practicable, be attached to the shell plate on one end of one seam so that the edges to be welded in the test plate are a continuation and duplication of the corresponding edges of the longitudinal seams. The weld metal should be deposited in the test plates continuously with the welding of the corresponding longitudinal seam so that the welding process, procedure and technique are the same. When it is necessary to weld the test plates separately, the procedure used should duplicate that used in the construction of the vessel.

Where difficulties are encountered with electroslag welds in transferring from seams with different curvatures (e.g. from a cylinder to a flat coupon plate) the test plate may be welded separately, either immediately before or immediately after the welded seam, using the same welding parameters.

When test plates are required for circumferential welds they may be welded separately from the vessel providing the technique used in their preparation duplicates, as far as possible, the procedure used in the welding of the appropriate seams in the vessel.

Q.1.3 Care should be taken to minimize distortion of the test plates during welding. If excessive distortion occurs, the test plate should be straightened before post-weld heat treatment. At no time should the test plates be heated to a temperature higher than that used or to be used for the final heat treatment of the vessel.

The preheat, interpass temperature, intermediate and post-weld heat treatments of test plates should be the same as for production welding. At the option of the manufacturer the test plates may be non-destructively tested in the same manner as the production weld. If any defects in the weld of a test plate are revealed by non-destructive testing, their position should be clearly marked on the plate and test specimens should be selected from such other parts of the test plate as may be agreed upon between the manufacturer and the Inspecting Authority.

Q.2 Destructive testing of production control test plates (see 5.4.1)

Q.2.1 Test recommendations

Specimens in accordance with **5.2.3** and Table 2 of BS EN ISO 15614-1:2017+A1 should be cut from production test plates and tested and assessed in accordance with that standard except where otherwise stated in **Q.2.2** to **Q.2.7**.

Production factors result in a scatter of mechanical test results which may occasionally fall below the agreed specification level. This is recognized in the recommendations given in **Q.2.3**, **Q.2.6** and **Q.2.7**.

Q.2.2 Test temperatures

The tests should be conducted at room temperature except in the case of impact tests, which, when required for vessels operating at low temperature, should be tested at the temperature derived in accordance with Annex D.

Q.2.3 All weld tensile test

Q.2.3.1 The following additional recommendations apply.

Q.2.3.2 The tensile strength and yield stress values determined on the production test plate, are satisfactory provided they exceed 90% of the minimum specified values for the parent metal.

Q.2.3.3 The amount by which the tensile strength or yield stress may exceed the specified minimum value for the parent metal is subject to agreement between the manufacturer, purchaser and Inspecting Authority.

Q.2.3.4 The reduction in area should not be less than 35% for carbon and carbon manganese steels in material groups 1, 2 and 11 and not less than the minimum specified for the parent metal in the case of alloy steels in material groups 4 to 9.

Q.2.4 Transverse bend test (for plate less than 12 mm thick)

Face bend tests should be conducted with the surface corresponding with the outer surface of the vessel in tension. Root bend tests should be conducted with the surface corresponding with the inner surface of the vessel in tension. On completion of the test, no crack or other defect at the outer surface of the test specimen should have a dimension greater than 1.5 mm. Slight tearing at the edges of the test specimen should not constitute failure of the test.

Q.2.5 Side bend test (for plate of 12 mm thickness and greater)

On completion of the test, no crack or other defect at the outer surface of the test specimen should have a dimension greater than 3 mm. Slight tearing at the edges of the test specimen should not constitute failure of the test.

Q.2.6 Macro- and micro-examination

The specimen should be prepared for macro-examination, and for micro-examination when the necessity for the latter has been agreed between the manufacturer, purchaser and Inspecting Authority. The specimen should be located in material which has not been affected by flame cutting operations. The weld should be sound, i.e. free from cracks and substantially free from discontinuities such as slag inclusions and porosity, to an extent equivalent to that given in Table 5.7-1. The hardness survey should include the parent metal and heat affected zone on each side of the weld as well as the weld metal and the results should be recorded in the production test reports.

The results obtained from a hardness survey should be considered satisfactory provided they do not exceed 110% of the maximum specified value for the procedure test.

Q.2.7 Impact tests

Charpy V-notch impact tests, when required, should be considered satisfactory provided the average and individual results from one production test plate exceeds 90% of the minimum average and individual specified values for the procedure tests, and the average of all production test plate results for the vessel or group of vessels, exceed 110% of the minimum average and individual specified values for the procedure tests.

NOTE For example, assume a specified minimum average requirement of 40 J permitting an individual minimum requirement of 28 J (see Table D.2). Therefore one production test plate with values 36 J/26 J (90% of minimum average requirement/90% of individual minimum requirement) would be satisfactory provided the average of the results of all test plates on the vessels exceeds 44 J/31 J (110% of minimum average requirement/110% of individual minimum requirement).

Q.3 Retests

Where tests do not comply with **Q.2**, the following retests should be made:

a) *Tensile*

Two retests should be made.

b) *Bend tests*

Two retests should be made.

c) *Impact tests*

See Annex D.

Should any of the retests fail to comply with **Q.2**, the welded seams represented by these tests should be deemed not to comply with this specification.

Annex R Guidance on additional information for flat ends and flat plates

In the design of a flat plate forming a head or end of a cylinder, it is necessary to consider both the plate itself and the stresses in the cylinder. The minimum allowable value of C (0.41) in Figure 3.5-36, Figure 3.5-37 or Table 3.5-8 provides a margin of 1.5 against gross plastic deformation of the plate in the simply supported case (with a slightly higher margin if edge support is included). The sloping lines ($C > 0.41$) ensure that the maximum stress in the cylinder is less than or equal to $2.7f$. This is to be compared with the $3f$ allowed in Annex A and provides some ability to accept additional loads.

The maximum stress in the cylinder is longitudinal, and on the inside surface adjacent to the end. The following equations [taken from [1][1)]] were used in the calculation of C for Figure 3.5-36, Figure 3.5-37 or Table 3.5-8 and should be used in computer programs in preference to a curve fit (though an iterative procedure to find e is then required). The equations may also be useful in a fatigue analysis, when loads are combined, or to find the allowable pressure for a given design. For a graphical representation of the equations and further discussion see [2]. The maximum stress in the cylinder, S, is given by:

$$S = I \times \frac{pD}{2e_{\mathrm{cyl}}} \tag{R-1}$$

where

p is the pressure;

D is the mean diameter of cylinder;

e_{cyl} is the analysis thickness of cylinder (see **1.6**);

e_{cylo} is the minimum thickness of the cylindrical shell as derived from **3.5.5.2**;

e is the minimum thickness of end;

$$I = ½ + \left(\frac{C_1 a^3 - C_2 a + C_3 b^{1.5} + C_4 ab + C_5 b/a}{C_6 a^3 + C_7 a + C_8/a + C_9 b^{0.5} + C_{10} a^2/b^{0.5}} \right)$$

where

a = e/e_{cyl};

b = D/e_{cyl};

C_1 = 2.943;

C_2 = 3.74;

C_3 = 1.0;

C_4 = 0.909;

C_5 = 0.385;

C_6 = 1.907;

C_7 = 4.848;

C_8 = 1.027;

C_9 = 2.667;

C_{10} = 5.875.

All dimensions are in the corroded condition (see **3.1.5**).

References

[1] WATTS, G.W. and LANG, H.A. The stresses in a pressure vessel with a flat head closure. *Trans. Am. Soc. Mech. Engr.*, 1952, **74**, 1083–1091.

[2] ESDU. Engineering Sciences Data Item No. 66010.

[1)] The numbers in square brackets used in this annex relate to the references given at the end of the annex.

Annex S Guidance on optional documentation for supply with vessel

This annex lists some of the documentation which a manufacturer is required to generate in accordance with the provisions of this specification but which he is not required to supply for record purposes in accordance with **1.5.2**. Purchasers wishing to retain permanent copies of any such documents should define their requirements in the purchase specification order using this, or an equivalent, checklist.

	Item	If required
1.	Design calculations (**3.2**)	
2.	Material test certificates (**1.4.2**, **2.1.2**)	
3.	Records of heat treatments carried out during fabrication (**4.2.2**, **4.5.2**, **4.5.3**)	
	components	
	finished vessel	
4.	Records identifying specific location of each batch of material in finished vessel (**4.1.2**)	
5.	Records of:	
	welding procedure tests (**5.2**)	
	welder approval tests (**5.3**)	
	weld production tests (if required) (**5.4**)	
6.	NDT records:	
	parent material (**5.6.2**)	
	welds (**5.6**)	
	components prepared for welding (**5.6.3**)	
	welds (**5.6**)	
7.	Records of dimensional checks against specified tolerances (**4.2.4**)	
8.	Detailed records of pressure test[a] (**5.8**)	
9.	Records of checks made to verify any special purchaser requirements[b]	
10.	Records of qualification of NDT Personnel (**5.6**)	
11.	Records of permanent joining processes (other than welding) e.g. brazing, expansion, etc. Including personnel and procedure qualification (**4.4**, **Cu.5.2** and **Cu.5.3**)	
12.	Data related to the preparation of component parts (e.g. forming, chamfering) (**4.2**)	

[a] Essential details of pressure test are given in the Certificate of Compliance.

[b] E.g. special tolerances, “finger-printing”.

Annex T Recommendations for arc welded tube to tubeplate joints

T.1 Joint selection

T.1.1 General

The type of joint selected for welded tube to tubeplate connections may be influenced by consideration of the allowable joint load (see **3.9.6**) but is more often determined by consideration of the possibility and consequences of leaks developing in service. Because of the small cross-section of weld metal involved, minor weld defects may lead to leakage in service as a result of corrosion or propagation by in-service stresses.

The assurance of weld quality by volumetric non-destructive testing (e.g. radiography or ultrasonics) may be difficult for most types of tube to tubeplate joint, and particular attention should therefore be given to controlling the factors that can influence tube to tubeplate weld quality. These include joint geometry, method of welding, quality control in manufacture and leak testing. Examples of typical joint configurations are given in Figure T.1 to Figure T.8 showing weld details that have given satisfactory results under specific manufacturing conditions. Modifications may be required to suit particular manufacturer's techniques, but all details adopted are to be shown by the manufacturer to have produced satisfactory results using the procedures specified in Section 4.

T.1.2 Weld procedure tests

The testing of weld procedures specified in **5.2** should be in accordance with BS EN ISO 15614-8:2016.

T.1.3 Approval testing of welders

Approval testing of welders or operators involved with mechanized or automatic welding, as specified in **5.3**, should be in accordance with BS EN ISO 9606-1 or BS EN ISO 14732, as appropriate, together with the supplementary conditions in **T.6**.

T.1.4 Welding plus tube expansion

Tube expansion after welding should be considered when:

a) corrosive and scale producing media may concentrate in the crevice; or

b) severe vibration of the tubes is likely; or

c) good heat transfer is required between tube and tubeplate.

Tube expansion is not performed for strength purposes (see Table 3.9-2), but tubes expanded for the full depth of the tubeplate will have increased ligament efficiency (see **3.9.2.1**).

T.2 Tube to tubeplate joints preparation for welding

T.2.1 General

The tube holes should be free from burrs and be in accordance with the tolerances permitted in the approved drawing. Tube holes for front face welds should have their edges at the back face of the tubeplate chamfered or radiused to 2 mm approximately.

Both faces of the tubeplate, the holes and the tubes should be free from dirt, grease, scale and other foreign matter when they are assembled. To avoid possible damage during assembly, or entrapment of contaminants, baffle and support plate holes should be free from burrs and effectively cleaned prior to the commencement of tube threading. Immediately prior to assembly, tubeplates should be thoroughly cleaned and degreased using a solvent that does not leave a residue. The ends of the tubes that are to be welded should be cleaned and degreased with a suitable non-residue forming solvent both inside and out, for a length equal to the tubeplate thickness plus 25 mm or 100 mm whichever is the smaller. Chloride free solvents should be used for austenitic steels.

For welding by the TIG process, the outside ends of the tubes should be cleaned to bright metal for a minimum distance of 15 mm.

T.2.2 Positioning of tubes

Prior to welding of tube to tubeplate joints, light expansion with taper expanders may be used to locate the tubes. This expansion should be controlled to prevent the tube hole gap being completely closed beyond the weld as this can give rise to weld faults. No lubricant should be used during expansion to ensure cleanliness of the weld preparation.

Special punches may also be employed to secure tubes to the tubeplate, e.g. the punch may be designed to enable three equally spaced teeth to throw burrs from the tubeplate hole towards the tube provided the burr depths are sufficiently shallow to be fused during welding.

T.2.3 Mechanized welding

For mechanized welding processes, machine settings and meter readings should be checked at the start of each shift to ensure that they are in accordance with those detailed in the approved welding procedure.

T.2.4 Autogenous welding

Autogenous welds may be susceptible to variable penetration due to cast to cast variations in some materials. This may require a revision of the weld procedure and further weld procedure tests being carried out on representative material.

T.2.5 Expansion of welded joints

Where expansion after welding is specified for tube to tubeplate welds, it should not be carried out until after the successful completion of the low-pressure test.

Expansion is to be done with the object of sealing the back face crevice in the tube hole. Tube wall thinning should be controlled to a predetermined range that will ensure that the expansion remains tight under design conditions, but not so great as to cause cracking of the welds or tubes. The expanded region should be within 3 mm from the back of the tubeplate to 10 mm from the weld fusion line.

T.2.6 Preheat prior to welding

The preheating of tubeplates is difficult to apply, maintain and control. Sufficient temperature measurement should be made to demonstrate that preheat and interpass temperatures are not less than those specified in the approved welding procedure. Because of the difficulty in preheating, consideration should be given to eliminating preheat by use of low hydrogen processes, austenitic filler metal or austenitic clad tubeplates.

T.3 Post-weld heat treatment

Post-weld heat treatment of complex assemblies such as welded tube end connections may present difficulties and so, where applicable, consideration should be given to methods of eliminating post-weld heat treatment including the possible use of austenitic filler metals, the use of extra low carbon ferritic filler metals or the use of tubeplates clad with austenitic or extra-low carbon weld metal.

Where post-weld heat treatment is employed, the heating and cooling rates should be controlled to avoid the possibility of weld fractures and excessive tube distortion. Adequate tube support to limit tube distortion should be considered at the design stage.

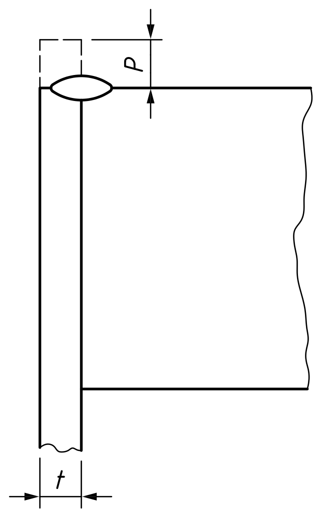
The post-weld heat treatment procedure should define charging temperature, heating and cooling rates, soak time and temperature for removal from the furnace. For fixed tubeplate heat exchangers consideration should be given to the stresses that may arise due to temperature differentials between tubes and shell or tubes and tubeplate, and sufficient temperature measurement points should be defined to monitor and control temperature differentials.

T.4 Essential testing of tube to tubeplate joints

T.4.1 General

Irrespective of vessel category, tube to tubeplate welds should be tested in accordance with **T.4.2**, **T.4.3** and **T.4.4**. The various testing options for joint designs shown in Figure T.1 to Figure T.8 are summarized in Table T.1.

Figure T.1 **Tube to tubeplate connections, tube end fusion**



NOTE 1 Typical dimensions.

P = t max.

Minimum distance between tubes = 2t.

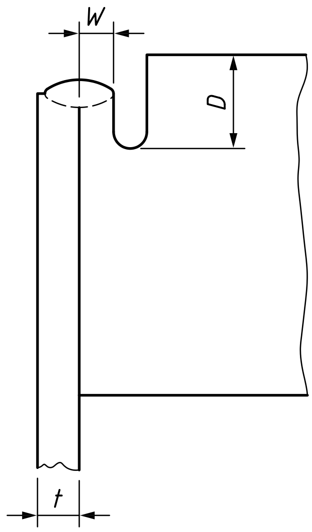
NOTE 2 Typical welding methods.

a) TIG plus filler or manual metal arc where t = 2.6 mm min.

*b) TIG without filler where t = 2.6 mm max. (See **T.2.4**).*

c) Single layer weld.

Figure T.2 **Tube to tubeplate connections, castellated weld**



NOTE 1 Typical dimensions.

W = t.

D = t min. 2t max.

Minimum distance between tubes = 3t.

NOTE 2 Typical welding methods.

a) TIG with filler or MMA where t = 2.6 mm min.

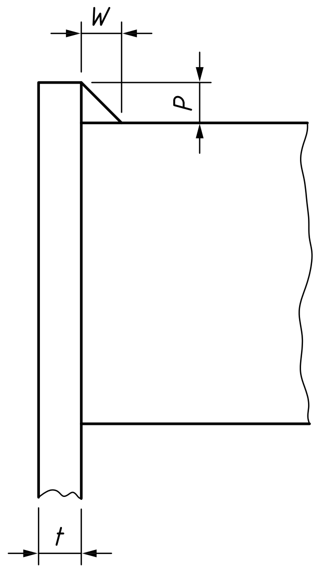
b) TIG without filler where t = 2.6 mm max. (See ***T.2.4****).*

c) Single layer weld.

NOTE 3 Special features.

Provides minimal distortion of tubeplate.

Figure T.3 **Tube to tubeplate connections, plain fillet weld**



NOTE 1 Typical dimensions.

W = t min.

P = 1.5t min.

Minimum tube wall 2.6 mm.

Minimum distance between tubes 2.5t or 8 mm whichever is least.

NOTE 2 Typical welding methods.

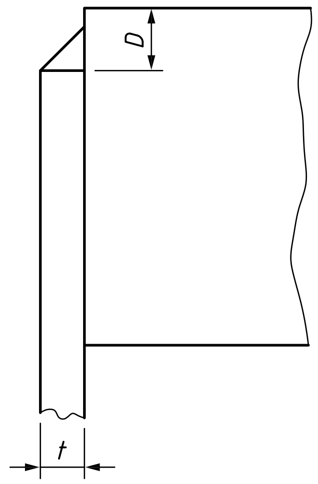
a) Manual metal arc.

b) TIG + filler.

c) MIG/MAG for multilayer welds only. Weld stop/start positions should not be coincident.

d) Single or multilayer weld.

Figure T.4 **Tube to tubeplate connections, front face bore fillet weld**



NOTE 1 Typical dimensions.

D = 0.5t to 1t for mechanized welding; t + 1.5 mm for manual welding.

t = 1.0 mm min.

Minimum distance between tubes = 2t.

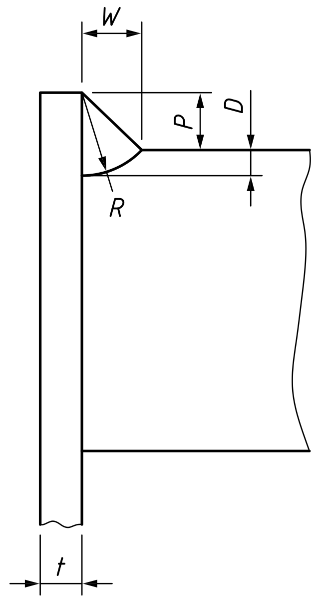
NOTE 2 Typical welding methods.

a) TIG + filler.

*b) TIG without filler (see **T.2.4**).*

c) Single layer weld.

Figure T.5 **Tube to tubeplate connections, groove plus fillet weld**



NOTE 1 Typical dimensions.

W min. = 2t or $\frac{ligament}{2}$ *mm whichever is less.*

W max. = 5 mm.

R max. = 5 mm.

D = t.

P = 2.5 for TIG welding, and 5 for MMA welding.

NOTE 2 Typical welding methods.

a) Manual metal arc throughout.

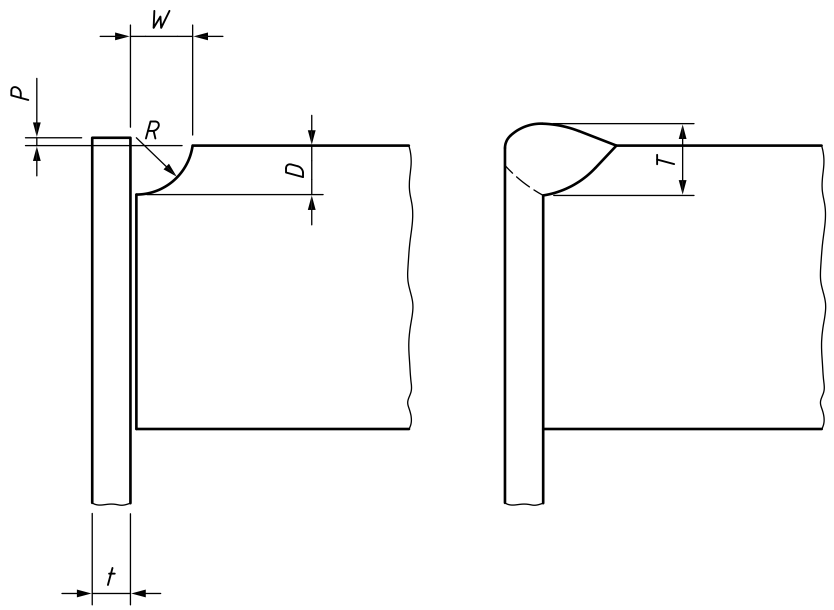
b) TIG plus filler throughout.

c) Combination of TIG and manual metal arc.

NOTE 3 Multilayer weld.

Weld stop/start positions should not be coincident.

Figure T.6 **Tube to tubeplate connections, groove weld**



NOTE 1 Typical dimensions

W min. = 2t or $\frac{ligament}{2}$ *mm whichever is less.*

W max. = 5 mm.

R max. = 1t to 1.5t.

D = 0.8t min. (Chosen to ensure design minimum throat thickness T after welding).

P = 0 to 0.8 mm.

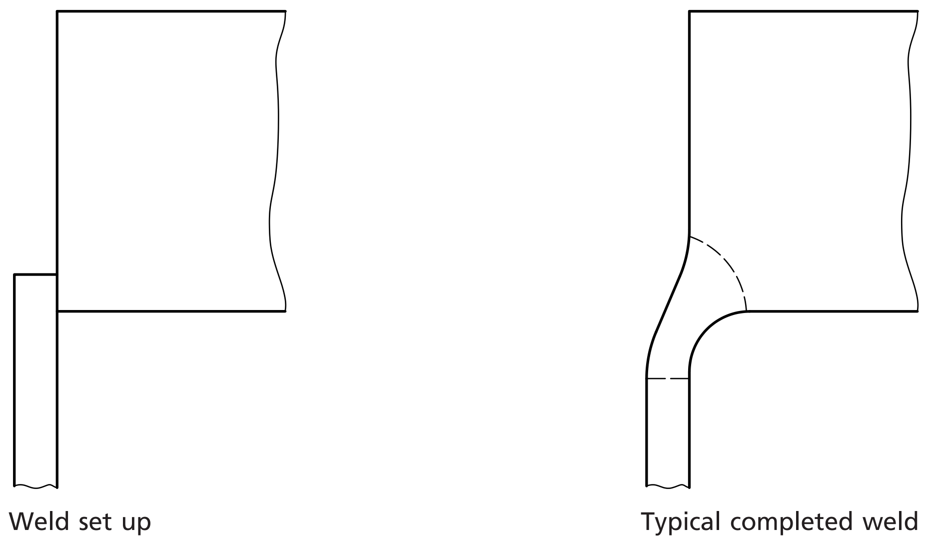
NOTE 2 Typical welding methods.

TIG: manual or mechanized with filler.

NOTE 3 Multilayer welding.

Weld stop/start positions should not be coincident.

Figure T.7 **Tube to tubeplate connections, back face inset bore weld**



NOTE 1 Dimensions.

Subject to agreement.

NOTE 2 Typical welding methods.

Only specially designed mechanized TIG equipment can be used. During welding, the back of the tubeplate in the vicinity of the joint must be protected by flux or inert gas.

Single layer (pass) weld.

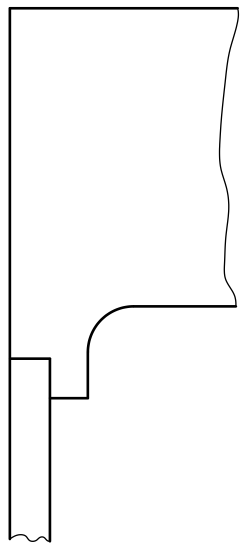
NOTE 3 Special features.

High reliability in service is possible.

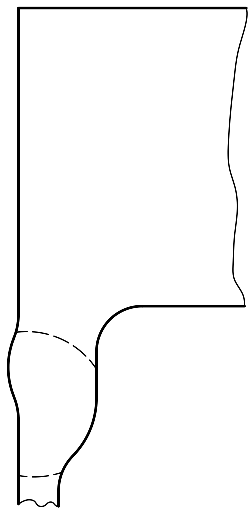
Crevice free welds are obtained.

Ultrasonic examination of the tubeplates for subsurface laminations is desirable for carbon steel.

Figure T.8 **Tube to tubeplate connections, back face stub bore weld**



Weld set up



Typical completed weld

NOTE 1 Dimensions.

Subject to agreement.

NOTE 2 Typical welding methods.

Only specially designed mechanized TIG equipment can be used. During welding, the back of the tubeplate in the vicinity of the joint must be protected by flux or inert gas.

Single layer (pass) weld.

NOTE 3 Special features.

High reliability in service is possible.

Crevice free welds are obtained.

Ultrasonic examination of the tubeplate for subsurface laminations is desirable for carbon steel.

Table T.1 **Tube to tubesheet joints: essential tests and the suitability of joint types for optional tests**

Weld detail (Figure)	**Essential tests for all joints**			**Optional tests (see T.5)**		
	Visual inspection (see T.4.2)	**Preliminary pneumatic leak test [see T.4.3a)] or dye penetrant [(see T.4.3b)]**	**Hydraulic (see T.4.4)**	**Intermediate leak test (see T.5.2)**	**Radiography (see T.5.4)**	**Final leak test (see T.4.3)**
T.1	Required	Required	Required	Not applicable	Not practicable	Where specified
T.2				Not applicable	Not practicable	
T.3				Where specified	Not practicable	
T.4				Not applicable	Not practicable	
T.5				Where specified	Not practicable	
T.6				Where specified	Not practicable	
T.7				Not applicable	Where specified	
T.8				Not applicable	Where specified	

T.4.2 Visual examination

All welds should be visually examined and should comply with the requirements of the procedure welds as defined in BS EN ISO 15614-8:2016 in respect of defects that can be revealed by visual inspection, unless otherwise agreed.

T.4.3 Dye penetrant or pneumatic testing

Before hydraulic testing welds are to be subject to either:

a) a shell side pneumatic test at 0.5 bar with the welds being examined for leaks using soapy water, or leak detection methods agreed between the manufacturer, purchaser and Inspecting Authority; or

b) a dye penetrant test in accordance with BS EN ISO 3452-1; the welds and tube wall adjacent to the weld should be free from cracks, lack of fusion or surface porosity.

All unacceptable defects revealed should be repaired and retested. This testing allows the discovery and repair of weld defects before hydraulic testing allows water into the tube to tubeplate crevice.

T.4.4 Hydraulic test

Hydraulic testing should be in accordance with **5.8.3**, and the tubeplate welds examined for leaks.

T.5 Optional tests

T.5.1 General

The purchaser may specify in the purchase specification one or more of the optional tests detailed in **T.5.2**, **T.5.3**, **T.5.4** and **T.5.5**.

T.5.2 Inter-run testing

Tests detailed in **T.4.3**a) or **T.4.3**b) may be carried out between runs for multilayer welds.

NOTE Care should be taken after any inter-run testing to ensure that the joint is adequately cleaned to prevent contamination of subsequent runs.

T.5.3 Final leak testing

After completion of the hydraulic test detailed in **T.4.4**, a sensitive tracer gas leak test may be carried out at a pressure not exceeding the design pressure, the method of test and the acceptance criteria being as agreed between the manufacturer, purchaser and Inspecting Authority.

NOTE Useful guidance on leak testing can be found in Section V of ASME Boiler and Pressure Vessel Code.

T.5.4 Radiography

If radiography is used for inspection of back-face welds (Figure T.7 and Figure T.8), unless otherwise agreed the acceptance standards are to be as specified in BS EN ISO 15614-8:2016. The radiographic technique to be employed and the extent of radiography should be agreed between the manufacturer, purchaser and Inspecting Authority.

T.5.5 Production control test pieces

Where production control test pieces are specified, the frequency of testing is to be agreed between the manufacturer, purchaser and Inspecting Authority. The production control test piece will consist of a representative tube to tubeplate weld and be designed to facilitate correct positioning of the welding head.

Certain materials are subject to cast to cast variations in autogenous welding and test piece materials should be representative of the production material.

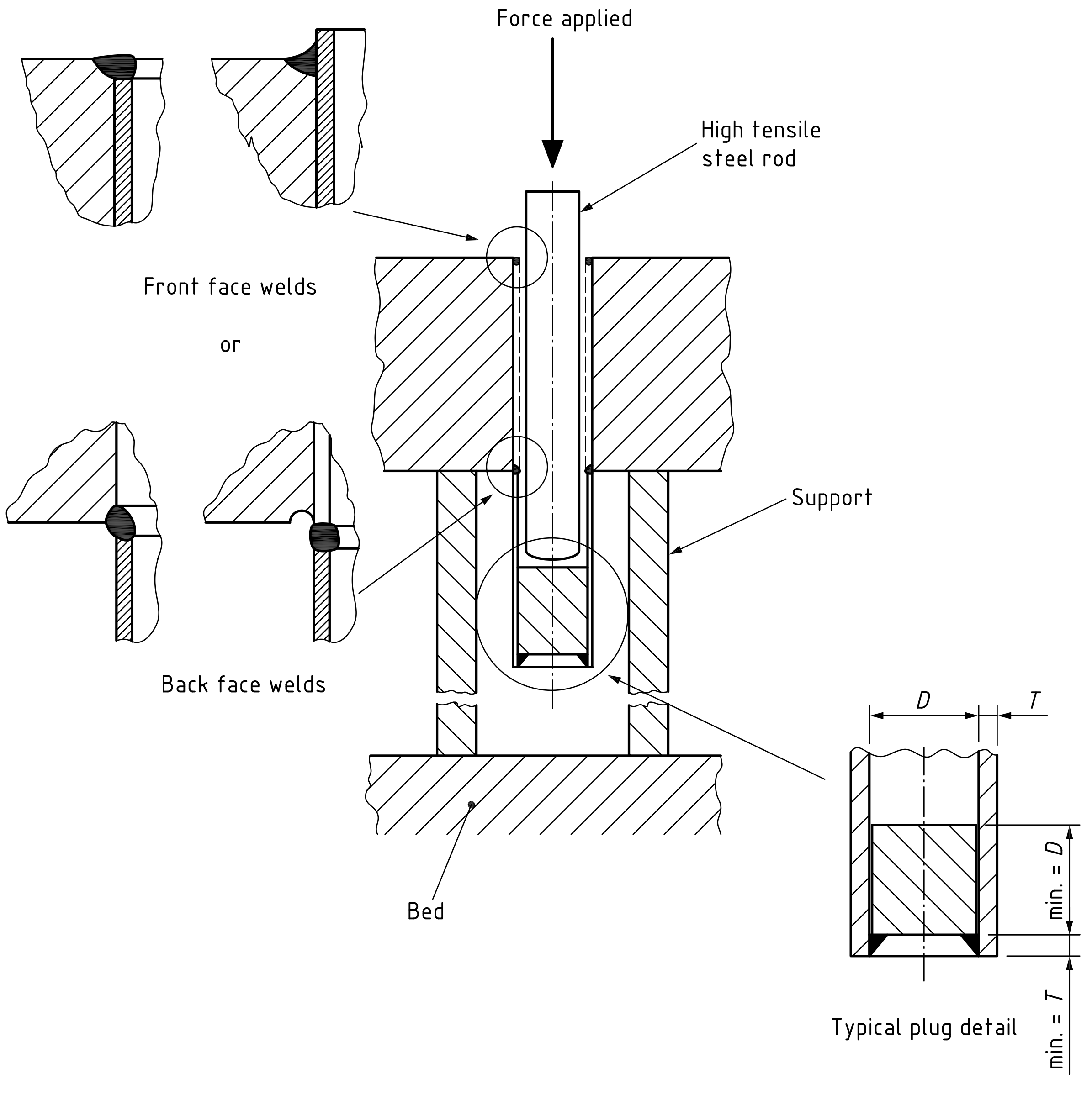
Production control welds for manual welds are not normally employed but unless otherwise agreed, they should conform to BS EN ISO 15614-8:2016.

Testing is to be in accordance with BS EN ISO 15614-8:2016 except that radiography is not required, the weld being sectioned and subject to visual assessment only. When production control test pieces are unsatisfactory, acceptance of the welds represented will be subject to agreement between the manufacturer, purchaser and Inspecting Authority.

T.5.6 Push-out test

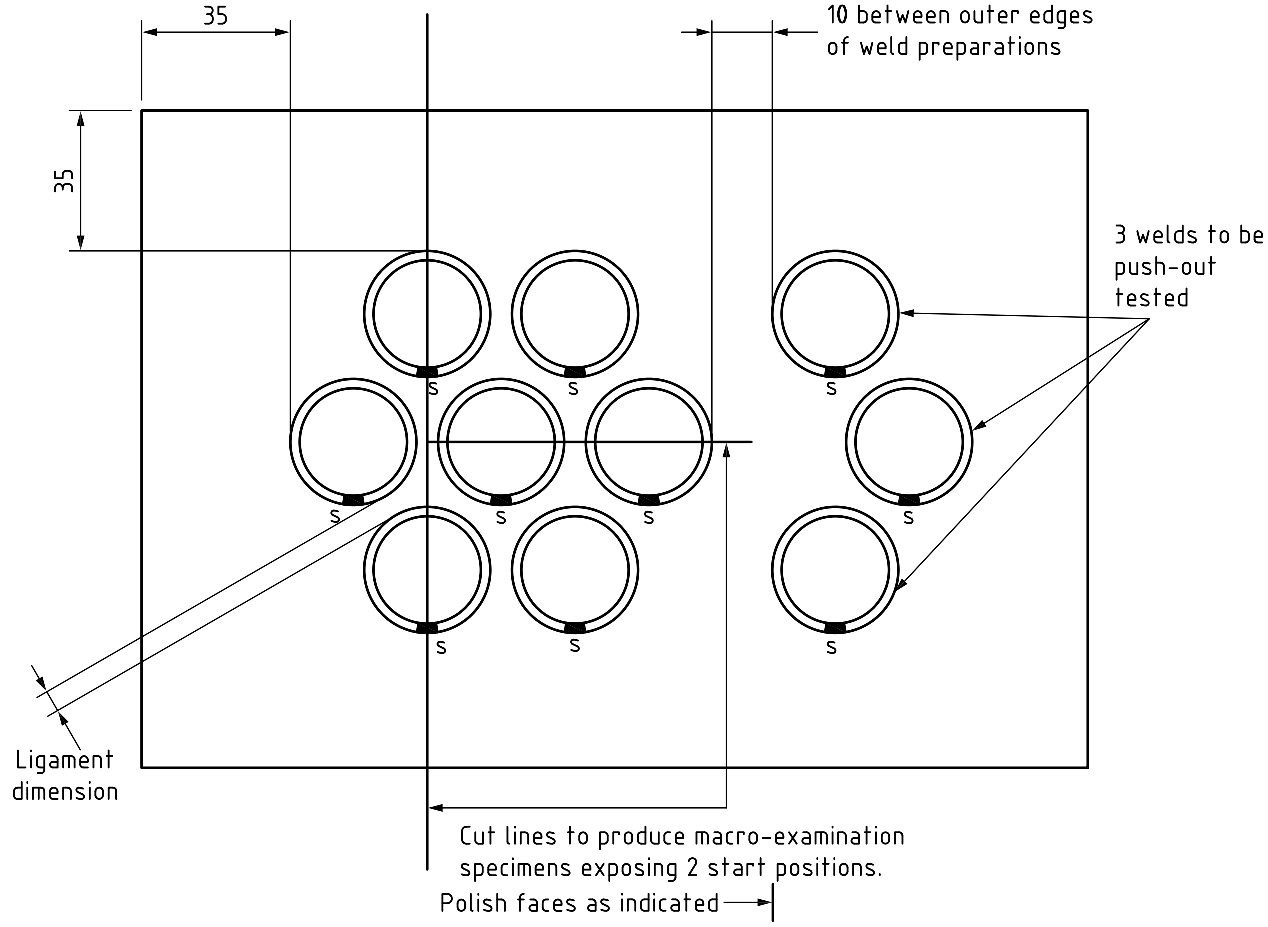
The push-out test should be arranged as shown in Figure T.9, Figure T.10 and Figure T.11. The lowest push-out load achieved in the test should be used to derive a stress in the tube which should be above the minimum tensile strength specified for the tube material. Where the tube end welds are arranged on a triangular pitch, the test piece should have a minimum of ten tube ends welded as shown in Figure T.10. Where the tube end welds are arranged on a square pitch, the test piece should have a minimum of thirteen tube ends welded as shown in Figure T.11.

Figure T.9 **General arrangement of push-out test piece for front face and back face welds**



NOTE D is the tube internal diameter and T is the tube wall thickness.

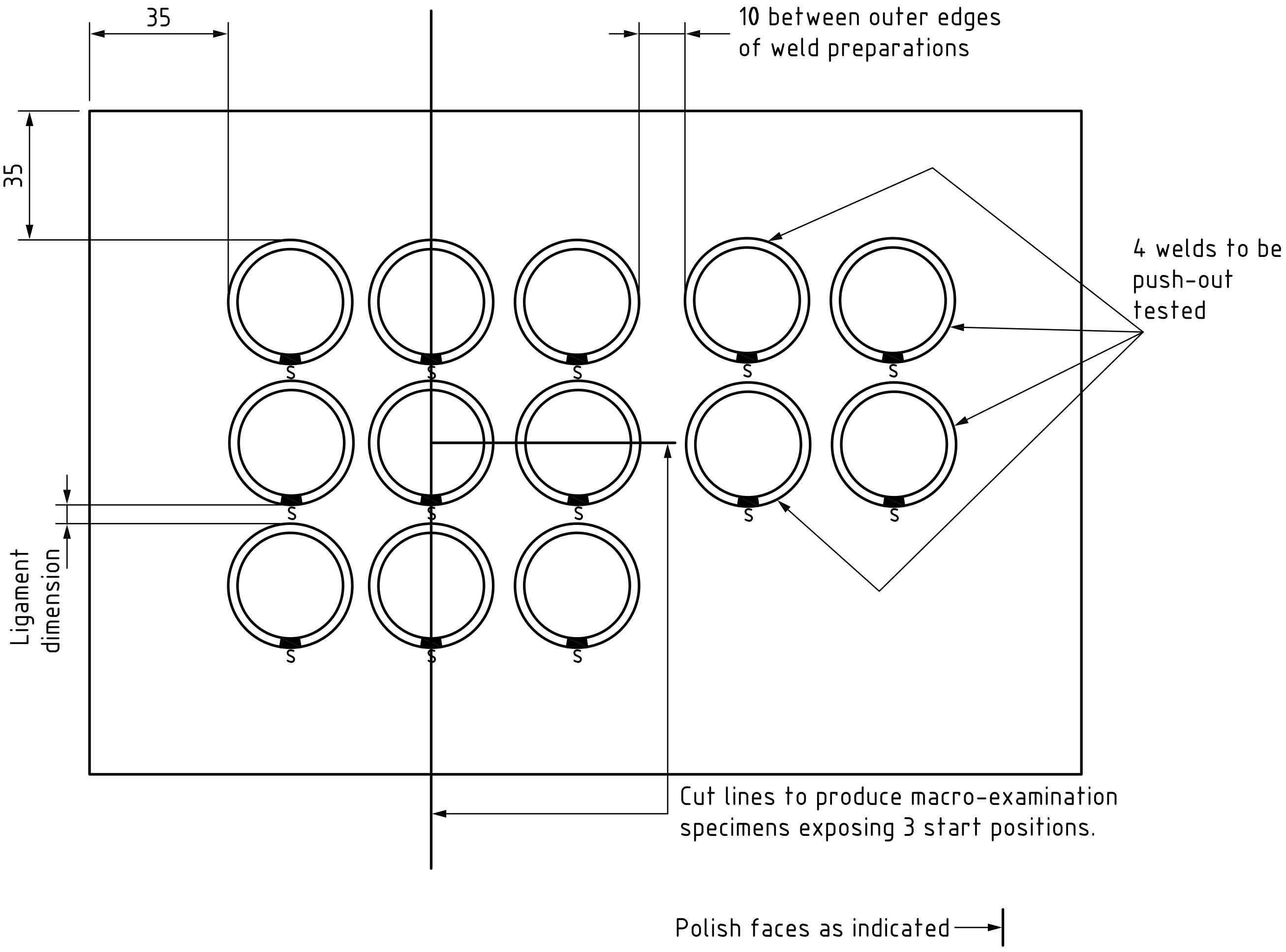
Figure T.10 **Push-out test piece for tube ends on triangular pitch**



NOTE All welds to have root run start at the position marked "S".

Dimensions in millimetres

Figure T.11 **Push-out test piece for tube ends on square pitch**



NOTE All welds to have root run start at the position marked "S".

Dimensions in millimetres

T.6 Extent of approval

T.6.1 Other than mechanized or automatic welding

T.6.1.1 General

When welder approvals are carried out in accordance with BS EN ISO 9606-1 the following essential variables should be in accordance with **T.6.1.2** to **T.6.1.8**.

T.6.1.2 Position

The approval of a welder on a test joint in one of the fundamental welding positions gives approval for welding in the following positions:

a) a test with the tubeplate in the horizontal position gives approval only for that position;

b) a test with the tubeplate in the vertical position gives approval for welding with the tubeplate in the vertical or horizontal position;

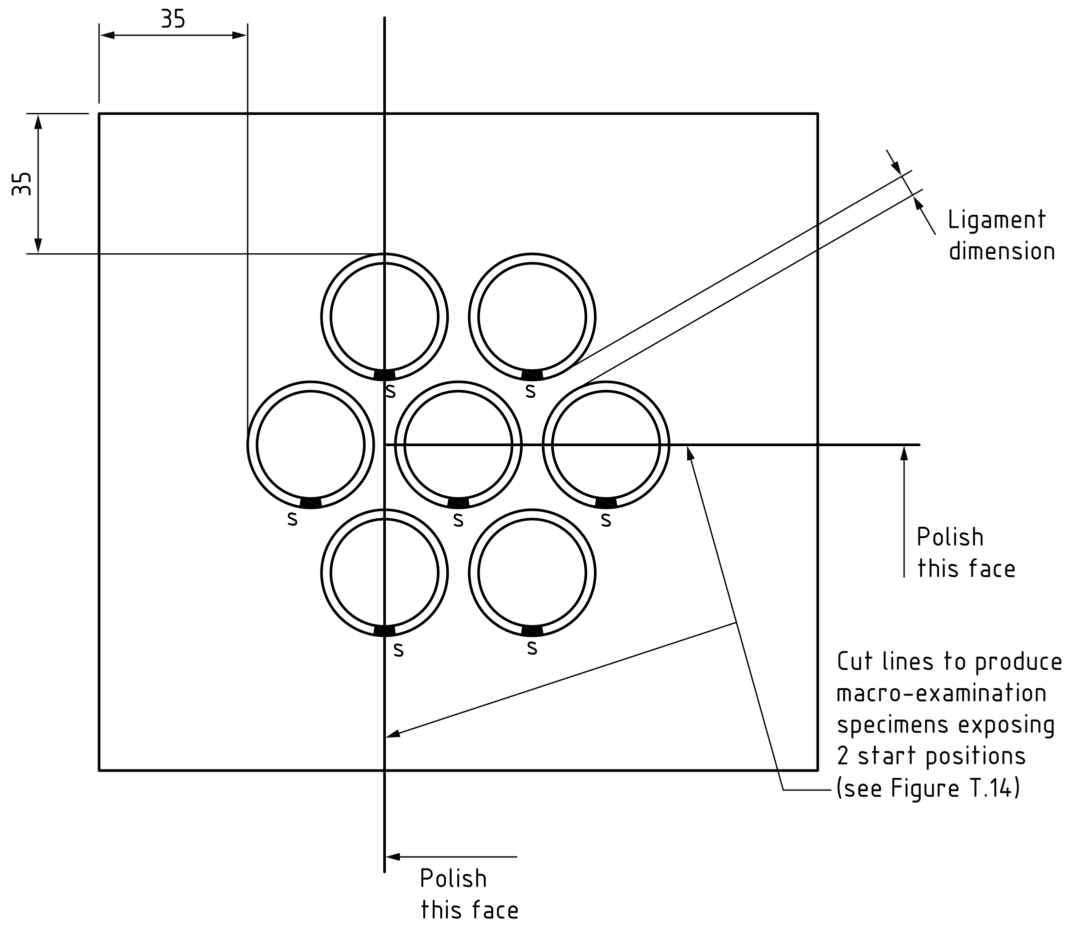
c) a test with the tubeplate in the overhead position gives approval for welding with the tube-plate in the overhead or horizontal position; and

d) welder who has carried out tests in accordance with b) and c) is approved for any position, horizontal, vertical, overhead or any intermediate position.

T.6.1.3 Tube pitch

The approval of a welder on a test joint using a particular pitch of tubes includes approval for any other type or pitch provided that the ligament dimension (see Figure T.12 and Figure T.13) is not reduced to below that given in the approved welding procedure or that used on the welder qualification test piece (see Figure T.12 and Figure T.13).

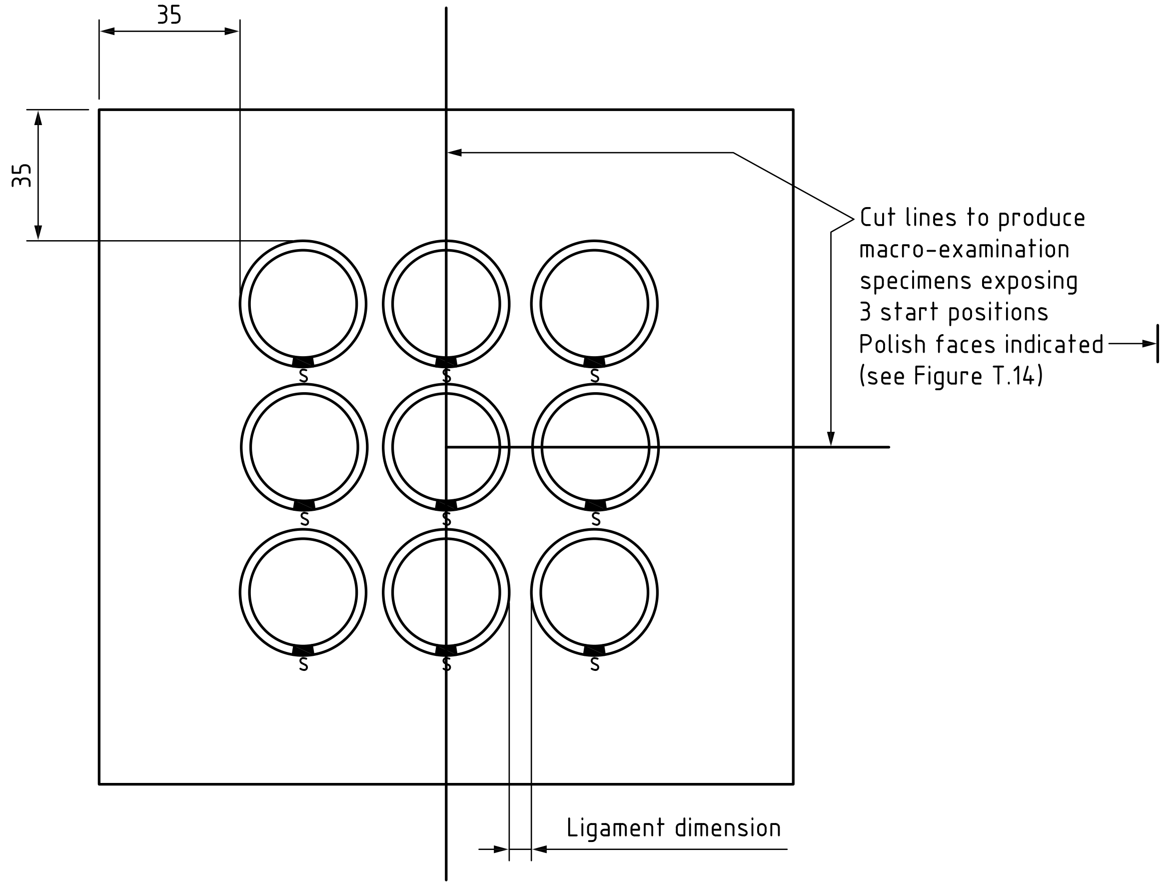
Figure T.12 **Test piece for the tube ends on triangular pitch**



NOTE All welds to have a root run start at the position marked "S".

Dimensions in millimetres

Figure T.13 **Typical examples of macro-examination specimens**



NOTE All welds to have a root run start at the position marked "S".

Dimensions in millimetres

T.6.1.4 Cladding

Irrespective of the tubeplate thickness, when welding a clad tube-plate any reduction in nominal cladding thickness entails re-approval of the welder.

T.6.1.5 Thickness

For the purposes of this subclause, T is the nominal tubeplate thickness and t is the nominal tube thickness. The approval of a welder on these thicknesses includes approval for thicknesses in the ranges given in Table T.2.

Table T.2 **Range of tubeplate and tube thickness for approval of a welder**

Tubeplate thickness T for test joint	**Range approved**
T < 10 mm	T to 2T
10 mm ≤ T < 35 mm	10 mm to 2T
T ≥ 35 mm	20 mm and above
Tube thickness t for test joint	**Range approved**
t < 1.6 mm	t to 2t
1.6 mm ≤ t ≤ 3 mm	1.6 mm to 2t
t > 3 mm	0.5t to 2t

T.6.1.6 Tube diameter

For the purposes of this subclause, D is the nominal tube outside diameter. The approval of a welder on these diameters includes approval for diameters in the ranges given in Table T.3.

Table T.3 **Range of diameters for approval of a welder**

Tube outside diameter D for test joint	Range approved
$D < 19$ mm	D to 19 mm
$D \geq 19$ mm	$0.5D$ to $2D$ with a minimum nominal diameter of 19 mm

T.6.1.7 Type of joint

The approval of a welder on a butt weld tube end joint with added filler metal also includes approval for butt and fillet weld joints.

Separate approval should be obtained for autogenous welding.

NOTE The exact joint details, such as angles and radius, do not affect welder approval, as long as the procedure has been approved in accordance with BS EN ISO 15614-1.

T.6.1.8 More than one welder

When more than one welder is employed on making a complete test weld, the satisfactory testing of the test weld results in the approval of each welder for their respective portion for that particular procedure only. When there are unacceptable defects present in a test weld and they can be attributed to one welder, this does not adversely affect the approval of the other welders.

T.6.2 Mechanized or automatic welding

T.6.2.1 When a welding operator for mechanized or automatic welding completes a welding test in accordance with BS EN ISO 14732:2013 the conditions in **T.6.2.2**, **T.6.2.3**, **T.6.2.4** and **T.6.2.5** should be included in the assessment.

T.6.2.2 For mechanized or automatic welding, most changes in the welding parameters do not affect the approval of the welder; they only affect the welding procedure approval range that is covered in BS EN ISO 15614-1. A welding operator who has successfully welded a test piece as shown in Figure T.12 or Figure T.13 is approved for all tube to tubeplate welds using the same or identical equipment for the period specified in BS EN ISO 14732:2013, Clause **5**, with the restrictions in **5.3** of PD 5500**.**

T.6.2.3 This approval covers:

a) all parent metal compositions;

b) all tube and tubeplate thicknesses and all tube outside diameters;

c) all welding positions (e.g. tubeplate vertical, flat or overhead); and

d) the essential variables listed in BS EN ISO 14732:2013, Clause **4**.

T.6.2.4 Re-approval should be obtained for major changes in joint geometry or configuration. Separate approval should be obtained for the following four main types of joint:

a) front face butt weld with filler metal addition;

b) front face butt weld without filler metal addition (autogenous welding);

c) front face fillet weld with filler metal addition; and

d) back face bore welds (autogenous welding).

T.6.2.5 The approval of a welder on a test joint of type a) (see **T.6.2.4**) should include approval for type b) (see **T.6.2.4**) but the converse should not apply.

T.6.2.6 The approval of a welding operator includes approval for the thicknesses in the ranges given in Table T.2.

T.7 Welding test pieces for welder and welding operator testing

The test pieces for welder and welding operator testing should be in accordance with Figure T.12 or Figure T.13.

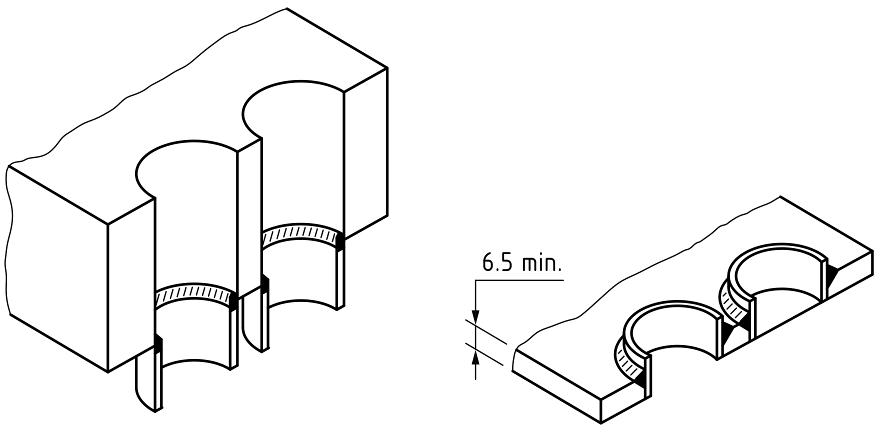
Except as permitted by **T.6.1.7**, the welder should weld a test piece as shown in Figure T.12 or Figure T.13, whichever is the most representative of the welding procedure to be used in production.

The root run start positions should be identified and positioned as shown in Figure T.12 or Figure T.13 to permit sectioning through these positions. For manual metal-arc welding the re-start should be made with a fresh electrode.

NOTE The test piece might need positioning to facilitate sectioning through start positions.

Subsurface defects that break the surface or which are revealed as the result of grinding specified in the welding procedure should not be repaired. There should be no repair welding of a completed test piece.

Figure T.14 **Typical examples of macro-examination specimens**



Dimensions in millimetres

Annex U Guidance on the use of fracture mechanics analyses

U.1 General

Annex D specifies requirements that are intended to avoid brittle fracture during operation and during pressure testing. These requirements apply only to a limited range of steels and were derived by a mixture of large-scale testing and engineering judgement (see Note to **U.2.3**). An alternative approach is given in **U.2**.

A fracture mechanics analysis following the principles of BS 7910 but observing the conditions specified in **U.2** may be used as a basis for determining the suitability of particular vessels for their intended duty when so agreed between the purchaser, manufacturer and Inspecting Authority in the following circumstances:

a) for materials not currently covered by Annex D;

b) those cases where Annex D cannot be adhered to;

c) where defects outside the requirements of **5.7** are detected;

d) when it is proposed to use group 1 materials in thicknesses greater than those permitted by Table 4.5-1 without post-weld heat treatment.

U.2 Procedure leading to requirements comparable to those in Annex D

U.2.1 The following approach is aimed at providing comparable margins to those resulting from analyses following Annex D, but for circumstances where that annex cannot be applied.

U.2.2 Fracture toughness properties should be obtained in accordance with the procedures in BS 7910 and/or BS EN ISO 15653 or BS 7448-1 on full thickness single edge notched bend specimens with fatigue cracks located through thickness on the weld centre-line and in parent material. Further tests sampling heat affected zone regions may also be specified particularly when fatigue or some other in-service crack growth mechanism might be significant.

When HAZ tests are specified special considerations are necessary with regard to the placement of the notch and metallurgical sectioning subsequent to testing, as specified in BS EN ISO 15653.

U.2.3 For materials not covered by Annex D a similar level of tolerance to fracture to that resulting from the use of Annex D can be obtained by specifying fracture toughness requirements determined from the use of an assessment procedure such as in BS 7910 with the following additional requirements.

a) Selection of a reference defect size as agreed by the interested parties (e.g. a through wall flaw of total length equal to 10 mm, or a quarter wall thickness surface flaw with length six times its depth).

b) Determination of an equivalent stress (or strain) relating to the hydraulic test condition, for a defect in a region of stress concentration.

c) For the purposes of this assessment the following residual stresses shall be assumed:

1) the room temperature yield strength of the base material for as welded components; or

2) 30% of yield for post-weld heat treated components.

NOTE The toughness requirement of Annex D for the operation of ferritic steel vessels below 0 °C was originally based on a combination of practical experience and performance of notched and welded wide plate tests and has evolved as described by Garwood S.J. and Denham B. [1].

The original criterion adopted as the critical conditions for the wide plate tests in the work reported by Woodley C.C., Burdekin F.M. and Wells A.A. [2] was the attainment of 0.5% plastic strain. For the steels employed where the yield strength averaged approximately 350 N/mm² this is equivalent to a plastic strain of three times yield, i.e. a total strain of four times yield for as welded components. This is consistent with the strain ratio used for validation purposes reported by Garwood and Denham where a PD 6493:1991 level 1 CTOD design curve based procedure was adopted.

Comparisons of the European codes procedures for the low temperature operation of pressure vessels, combined with updated fracture assessments as described by Wiesner et al. [3], led to the development of Annex B of BS EN 13445-2:, and these methods were revised further in 2009. Studies by Hadley and Garwood [4] raised some concerns for as welded applications which led to a specific test programme [5] to assess the design lines in Figure D.1 of PD 5500, Annex D. The results of this study have facilitated the latest revision to Figure D.1.

U.3 Direct application of BS 7910

U.3.1 As an alternative to the use of the procedure in **U.2**, if more specific information is available or a more rigorous approach is necessary, a more direct use of BS 7910 may be adopted in the circumstances described in **U.3.2** and **U.3.3**.

U.3.2 If non-destructive testing methods are employed which allow accurate sizing of defects, these values, together with information on the stress state of the critical regions in the vessel, can be used with BS 7910 procedures to specify more accurate toughness requirements than determined in accordance with Annex D and **U.2.3**.

U.3.3 A fitness for purpose analysis can also be employed to identify the ability to tolerate specific defects outside the requirements of Table 5.7-1. In this case the testing of surface notched fracture toughness specimens may be more appropriate to identify the toughness of specific regions in which the defects are situated.

U.3.4 Safety factors obtained for either of the above cases are likely to be different to those implicit in Annex D requirements. The difference in safety factors can be quantified by comparing results from a BS 7910 analysis using inputs defined in **U.2** with results from a case-specific BS 7910 analysis as defined in **U.3.1**.

U.3.5 It might be expected that the overall safety factors derived from the **U.3** approach would be lower than those given by the **U.2** approach, whilst still being safe.

References

[1] GARWOOD S. J. and DENHAM B. *The fracture toughness requirements of BS 5500*, ASME, 88-PVP-7, Pittsburgh, June 1988.

[2] WOODLEY C.C., BURDEKIN F.M. and WELLS A.A. *Mild steel for pressure equipment at sub zero temperature*, British Welding Journal, 11, **3**, March 1964, 123-136.

[3] WIESNER C.S., GARWOOD S.J., SANDSTROM R, STREET D.M. and COULSON K.J., *Background to requirements for the prevention of brittle fracture in the European standards for unfired pressure vessels (prEN 13445) and metallic industrial piping (prEN 13480)*, IJPVP, 78/6, 391-399.

[4] HADLEY I. and GARWOOD S, *Prevention of brittle fracture in pressure vessels: A review of the design rules of EN 13445 Annex B and BSI PD 5500 Appendix D*. International Journal of Pressure Vessels and Piping, 2019. 169: 1-15.

[5] O'CONNOR A.N., DAVIES C.M., GARWOOD S.J., and HADLEY, I. *Optimising the Safe Design of Pressurised Components*, Proceedings of the ASME 2019 Pressure Vessels & Piping Conference. Volume 1: Codes and Standards. San Antonio, Texas, USA. July 14–19, 2019. V001T01A081. ASME.

Annex V Requirements for testing and inspection of serially produced pressure vessels

V.1 Application

This annex specifies requirements for the construction, inspection, testing and certification of serially produced vessels (see **V.2.1**). It applies only to vessels made in compliance with the following.

a) The design and construction of the vessel, except where otherwise specified in this annex, is to category 2 of this specification in group 1 materials only.

b) The design pressure of the vessel does not exceed 30 bar, and the product of that pressure and the capacity of the vessel (*pV*) does not exceed 10 000 bar·L. The vessel diameter does not exceed 1.5 m, the nominal length (between tangents) does not exceed 3.5 m and the shell thickness does not exceed 15 mm. The design temperature lies in the range 300 °C to 10 °C (see **3.2.4** and **D.3.2**).

c) All type A main welds (see **5.6.4**) are welded by a mechanized or an automatic welding process.

d) Vessels identification numbers contain the suffix "V" to denote that they have been manufactured in accordance with this annex and this specification.

V.2 Definitions

V.2.1 serial production

the manufacture of identical vessels to a common approved design using the same manufacturing procedure

vessels with variations in length, nozzle position, number of nozzles and supporting locations are considered as part of the same series, subject to the confirmation of the design acceptability by the Inspecting Authority

V.2.2 batch of vessels

a part of a series where the welding of the main seams and branch welds has been essentially continuous. A stoppage in vessel production greater than five consecutive days requires the designation of a new batch

NOTE Stoppages or break-downs requiring resetting of the welding machine constitute a break in continuity. Adjustments to the welding machine within the welding procedure limitations do not qualify as resetting the welding machine.

V.3 Quality assurance

Before production commences, a detailed manufacturing and quality plan shall be prepared by the manufacturer and submitted to an Inspecting Authority for approval. This plan shall indicate the inspection or sampling points and the frequency of testing. Provision shall be made within the plan for rejected or re-worked components to be re-inspected. The quality plan shall ensure the following.

a) The materials used in the manufacture of the vessels comply with the materials standards as specified.

b) All variables in the manufacturing procedures that affect the integrity of the vessel are specified, monitored and controlled.

c) The testing and examination of the vessel is done at least at the frequency given in this specification, using appropriate test methods.

d) The inspection functions within the company are clearly prescribed.

V.4 Inspection and non-destructive testing

V.4.1 The inspection and non-destructive testing of the first vessel in a series shall be witnessed by the Inspecting Authority as follows.

a) The whole length of all welds (100%) shall be examined by the method specified in **5.6.4.2** and an assessment of any flaws shall be consistent with **5.7.2.4**.

b) The dimensions shall be checked to ensure conformity with the approved design drawings.

c) A pressure test in accordance with **5.8.2** shall be carried out.

V.4.2 At least 20% of the weld length of the first vessel of each batch shall be examined by the method specified in **5.6.4.2**. Assessment of any flaws shall be consistent with **5.7.2.4** and witnessed by the Inspecting Authority.

V.4.3 Except as permitted by **V.4.4**, each subsequent vessel produced shall be examined by the manufacturer and assessed in accordance with **5.6.4.2** and **5.7.2.4** as follows.

a) The whole length of all welds associated with nozzles shall be inspected using either a magnetic particle or a dye penetrant method.

b) Each longitudinal and circumferential weld shall be radiographed or ultrasonically examined once per vessel with a minimum sample length of 150 mm and with a minimum of 10% of the finished weld per shift. This 10% examination shall include “T” joints such that 10% of the “T” joints per shift are examined.

c) At least 10% of the length of other attachment welds shall be examined using either magnetic particle or dye penetrant inspecting methods.

V.4.4 By agreement between the manufacturer and the Inspecting Authority, the examinations in accordance with **V.4.3**b) and **V.4.3**c) may be reduced from 10% to not less than 5% if the test results are consistently satisfactory. Production shall be considered satisfactory if not more than three unacceptable defects were found in the immediately previous 100 non-destructively tested samples of the batch.

V.4.5 If not less than four and not more than ten of the same unacceptable defects are found in the immediately previous 100 non-destructively tested samples, then the inspecting frequency shall:

a) be maintained at 10% if the inspecting frequency was 10%;

b) be increased to 10% if the inspecting frequency was 5%.

Where more than 10 of the same unacceptable defects are found in the immediately previous 100 non-destructively tested samples, then the inspecting frequency shall be increased to 20% and maintained at this level until the number of unacceptable defects in the immediately previous 100 non-destructively tested samples is less than eight, at which the frequency may be reduced to the 10% specified in **V.4.3**b) and **V.4.3**c).

V.4.6 Every vessel shall be pressure tested by the manufacturer, in accordance with **5.8.5** and witnessed by the Inspecting Authority at their discretion.

V.4.7 The Inspecting Authority shall carry out surveillance during production and testing, to ensure that the manufacturer produces and inspects vessels in accordance with the manufacturing and quality plan.

V.5 Acceptance criteria

V.5.1 Isolated defects

If a defect is found during the partial non-destructive testing of a vessel, re-examination and repair shall be in accordance with **5.7.2.3**. However, defects found in “T” joint regions of the vessel examined shall be regarded as representing the seam in which they are located.

V.5.2 Multiple defects

V.5.2.1 If a recurrence of the same type of unacceptable defect is found in that vessel seam when the whole of the seam is inspected, as required by **5.7.2.3**, then the vessels produced immediately before and after it in the batch shall also have the same weld seams examined in accordance with **V.5.1**.

V.5.2.2 If no unacceptable defects are found in the appropriate seams of those two vessels, no further special examinations shall be carried out.

V.5.2.3 If unacceptable defects are found in either of the preceding or following vessels, then further vessels in sequence shall be assessed in accordance with **V.5.1** until a vessel with no unacceptable defects is found.

V.6 Marking

The Inspecting Authority shall inspect all vessels before despatch to ensure that the marking conforms to **5.8.9** of this specification. The vessels shall be marked “PD 5500V” to denote that they have been manufactured in accordance with this annex.

Where some time elapses between pressure test and despatch, e.g. stock vessels, the Inspecting Authority shall satisfy themselves that no deterioration or damage has occurred in the interim period. Where a temporary nameplate has been attached the Inspecting Authority shall ensure that the permanent plate conforms in all respects to **5.8.9**.

V.7 Documentation

A Certificate of Compliance shall be issued for each batch of vessels (see **1.4.4**). The certificate shall state clearly the serial numbers of the vessels covered. The reasons for any missing numbers in the series shall be clearly stated.

Where vessels in a batch are allocated to different purchasers, a separate copy of the certificate shall be issued to each purchaser.

If the purchaser requires records of manufacture, as described in **1.5.2**, this shall be specified in the purchase specification.

Annex W Worked examples

This annex presents a series of worked examples to indicate a method of analysis suitable for applying some of the rules of this specification. It is not possible to cover all circumstances of use but it is hoped that users will find the following examples of benefit.

List of examples

W.1 External pressure

W.1.1 General

W.1.1.1 Introduction

Clause **W.1** gives two worked examples for designs in accordance with **3.6**, one assessment to Annex M and one check in accordance with Enquiry Case 5500/33.

a) Example 1 uses simplifications giving easy hand calculations for shell thickness and light stiffener designs. The conservative assessment assumes each stiffener supports individual bays of an infinitely long cylinder.

b) Example 2 is also applicable to light stiffeners but is less conservative than Example 1 as account is taken of the increased resistance to collapse of the cylinder by using a finite length between planes of substantial support, namely the vessel ends.

c) Example 3 uses Annex M to assess the safe external pressure for a cylinder outside the circularity limits specified in **3.6**.

d) Example 4 checks the maximum departure from a cylinder mean circle with a chord gauge using the procedure given in Enquiry Case 5500/33.

The first three examples illustrate the use of hand calculations and use some simplifications which, whilst resulting in some slight increase in shell thickness and stiffener size, do greatly reduce the numerical work involved.

W.1.1.2 Notation

For the purposes of **W.1** the following symbols apply, based upon those given in **3.6**.

a_n, b_n	are Fourier series coefficients;
A	is the modified area of a stiffener;
A_c	is the cross-sectional area of a stiffener plus the effective length of shell;
A_s	is the cross-sectional area of a stiffener;
C	is a stiffener parameter;
d	is the radial height between stiffener flanges;
$\overline{d}$	is the distance to the stiffener extremity;

D_i, D_o	are inside and outside shell diameters, respectively;
e	is the shell analysis thickness;
e_f, e_w	are the analysis thicknesses of the stiffener flange and web, respectively;
E	is the modulus of elasticity from Table 3.6-3;
f, f_s	are the design stresses of shell and stiffener, respectively;
I_c	is the second moment of area of the composite stiffener;
I_s	is the second moment of area of the stiffener cross-section;
K	is an external pressure design parameter;
L_c	is the distance between heavy stiffeners;
L_e	is the effective shell length acting with the stiffener;
L_s	is the stiffener spacing;
n	is an integer used in stiffener design calculations;
N	is an inter-stiffener collapse parameter;
p	is the design external pressure;
p_a	is the calculated allowable external pressure;
p(allowable)	is the allowable external pressure for an out-of-tolerance vessel;
p_e	is the elastic instability pressure for the collapse of spherical shells;
p_m	is the elastic instability pressure for the collapse of cylindrical shells;
p_n	is the elastic instability pressure of a stiffened shell;
p_q	is the lower bound estimate of a cylindrical collapse pressure;
p_y	is the pressure at which the mean circumferential stress in a cylinder wall, midway between stiffeners, reaches yield point;
p_{ys}	is the pressure causing circumferential yield of a stiffener;
p_{yss}	is the pressure at which a spherical shell membrane stress reaches yield point;
R	is the mean radius of a shell;
R_f	is the radius of a stiffener flange;
R_r	is a measured radius;
R_s	is the radius of the stiffener centroid;
s	is a material factor;
X_c	is a an elastic instability parameter;
δ	is a chord gauge reading;
Δ	is an external design parameter;
ε	is the mean elastic circumferential strain at collapse;
ε_r	is the calculated departure from mean circle;
λ	is a stiffener parameter;
γ	is a stiffener parameter;
σ_s	is the maximum stress in a stiffener;
σ_{br}	is a calculation stress;
ν	is Poisson's ratio.

W.1.2 Worked examples

W.1.2.1 Example 1. Simple calculations for shell thickness and light stiffeners

W.1.2.1.1 Basis of analysis

This example makes use of the following simplifications for hand calculations:

a) $\gamma = 0$, $L = L_s$.

These approximations make the unsupported shell thickness required by **3.6.2.1** independent of stiffener size so that curve a) of Figure 3.6-4 may be entered using a value of $p_m/p_y = E\varepsilon/(sf)$.

b) $n = 2$, $\beta = 0$.

These approximations for stiffener assessment will lead to a conservative design (see **3.6.2.3.1**). This example illustrates the determination of shell thickness and stiffener scantlings for one section of a uniformly stiffened cylinder that is required to operate at a substantial external pressure. Table W.1-1 gives the design data for the cylindrical section.

Table W.1-1 **Design data assumed for cylindrical sections**

Design external pressure	p = 6.9 bar = 0.69 N/mm^2
Mean radius of shell	R = 2 500 mm
Stiffener spacing	L_s = 1 000 mm
Material for shell and stiffeners	Carbon steel
Design stress	f = 165 N/mm^2 s = 1.4
Effective yield strength (**3.6.1.1**)	sf = 231 N/mm^2 = sf_s
Modulus of elasticity (from Table 3.6-3)	$E = 2.07 \times 10^5$ N/mm^2
Internal stiffeners	(λ = +1)

W.1.2.1.2 Calculation of shell thickness from 3.6.2.1

Take trial values of $2R/e$ corresponding to Δ = 0.5 on Figure 3.6-4, i.e. where the effective design pressure is half that at which mean circumferential stress midway between stiffeners reaches yield; in this example, where $2R/e = sf/p = 335$. This process is summarized in Table W.1-2.

Table W.1-2 **Summary of calculation for *e***

$2R/e$ (assumed)	400	300	250	200
e	12.5	16.7	20	25
p_y [from Equation (3.6.2-7)]	1.155	1.540	1.848	2.310
ε (from Figure 3.6-2 $L/2R$ = 0.2)	0.00088	0.0013	0.0018	0.0026
$K = p_m/p_y = E\varepsilon/(sf)$ [from Equations (3.6.2-7) and (3.6.2-8)]	0.789	1.165	1.613	2.330
$\Delta = p_a/p_y$ (from Figure 3.6-4)	0.2630	0.380	0.470	0.525
p_a(N/mm^2)	0.306	0.585	0.869	1.213
Interpolating p_a versus e for the design external pressure of 0.69 N/mm^2 gives e = 17.9 mm.				

W.1.2.1.3 Calculation of effective length of shell acting with a stiffener, L_e

Calculate the effective length of shell acting with stiffener, L_e:

with e = 17.9 mm

R = 2 500 mm

L_s = 1 000 mm

$$\frac{e^2}{12R^2} = 4.27 \times 10^{-6} \frac{L_s}{2\pi R} = 0.0637$$

Using the values in Table 3.6-6 gives the values in Table W.1-3.

Table W.1-3 **Derivation of L_e/L_s**

$\frac{L_s}{2\pi R}$	$\frac{e^2}{12R^2}$		
	10^{-6}	4.27×10^{-6}	10^{-5}
0.06	0.2661		0.4483
0.0637	0.2550[a]	0.3597[b]	0.4212[a]
0.07	0.2362		0.3752

[a] Obtained by linear interpolation.

[b] Obtained by single logarithmic interpolation as follows:

given that $\log 10^{-6} = -6$, $\log 4.27 \times 10^{-6} = -5.37$ and $\log 10^{-5} = -5$;

the value of $\frac{e^2}{12R^2}$ is given by

$$0.4212 - \left[\frac{\{(-5) - (-5.37)\}}{\{(-5) - (-6)\}} \times (0.4212 - 0.2550)\right] = 0.3597;$$

giving $L_e = 360$ mm.

W.1.2.1.4 Determination of internal stiffener size

It is first necessary to decide on stiffener proportions. The proportions chosen are as shown in Figure W.1-1 and are shown later to satisfy the requirements of **3.6.2.2**. An alternative approach would be to use standard sections (possibly with cropped webs, provided that the proportions conform to the requirements of **3.6.2.2**) and find the smallest section to give $\sigma_s \leq sf_s$ for $n = 2$.

$$A_s = 0.21d^2,\ I_s = 0.0277d^4,\ R_s = 2\,491 - 0.814d,\ R_f = 2\,491 - 1.2d.$$

From **3.6.1.1** $A_c = A_s + eL_e = 0.21d^2 + 17.9 \times 360 = 6\,444 + 0.21d^2$.

From Equation (3.6.2-13),

$$A_cX_c = \left[\frac{e^2}{2}L_e + A_s\left\{\frac{e}{2} + \lambda(R - R_s)\right\}\right]$$

$$= \left[\frac{17.9^2}{2} \times 360 + 0.21d^2\left\{\frac{17.9}{2} + 1(2\,500 - 2\,491 + 0.814d)\right\}\right] \quad \text{(W.1-1)}$$

$= 57\,674 + d^2(3.759 + 0.171d)$

From Equation (3.6.2-12),

$$I_c = \left[\frac{e^3}{3}L_e + I_s + A_s\left\{\frac{e}{2} + \lambda(R - R_s)\right\}\right]^2 - \frac{(A_cX_c)^2}{A_c}$$

$$= \left[\frac{17.9^3}{2} \times 360 + 0.0277d^4 + 0.21d^2\left\{\frac{17.9}{2} + 1(2\,500 - 2\,491 + 0.814d)\right\}\right]$$

$$-\frac{(A_cX_c)^2}{A_c} \quad \text{(W.1-2)}$$

$$= 6.882 \times 105 + 0.02774d^4 + 0.21d^2[8.95 + (9 + 0.814d)]^2 - \frac{(A_c X_c)^2}{A_c}$$

From Equation (3.6.2-16), $\overline{d} = \lambda\,(R - R_f) - X_c + \frac{e}{2}$ or X_c whichever is larger

$$= 1(2\ 500 - 2\ 491 + 1.2d) - X_c + \frac{17.9}{2} \text{ or } X_c \qquad \text{(W.1-3)}$$

$= 17.9 + 12\ d - X_c$ or X_c

The stiffener size is determined by d and it is necessary to find the minimum value of d by successive approximations. A starter value of d which should reduce the amount of iteration involved can be estimated by using Equation (3.6.2-11), taking $p_n = 4p$, $n = 2$ and $\beta = 0$.

Thus

$$I_c = 1.33\frac{pR^3}{E}L_s$$

$$= 1.33 \times \frac{0.69 \times 2\ 500^3}{2.07 \times 10^5} \times 1\ 000$$

$= 6.93 \times 10^7$ mm^4

Guessing d = 160 mm and using Equations (W.1-1) and (W.1-2) gives $I_c = 7.51 \times 10^7$ mm^4.

This is close enough to 6.93×10^7 to proceed with evaluating σ_s

[Equation (3.6.2-15), for n = 2].

For d = 160 mm, R_s = 2 360 mm, A_s = 5 376 mm^2, A_c = 11 820 mm^2,

X_c = 72.28 mm

$\overline{d}$ = 138 mm

A = 6 028 mm^2 α = 0.00605 αL = 6.05

N = 1.0 (from Table 3.6-1)

p_{ys} [Equation (3.6.2-14)] = 3.53 N/mm^2

p_n [Equation (3.6.2-11)] = 2.98 N/mm^2 (with n = 2, β = 0)

σ_s [Equation (3.6.2-15)] = 238 for cold formed stiffener.

The value of σ_s just exceeds sf_s showing that the stiffener is slightly too small.

Trying d = 170 mm

p_{ys} = 3.74 N/mm^2

p_n = 3.62 N/mm^2

σ_s = 194.

This shows that the stiffener defined by d = 170 mm is adequate. Note that if the approximation A = 0 is used in Equation (3.6.2-14), the σ_s would equal 326 and a significantly larger stiffener would be required.

The criteria for stiffener proportions in **3.6.2.2** are shown to be satisfied.

Since using Figure 3.6-6 for a symmetric stiffener:

$$C = \frac{d{e_w}^3 + 8e_f{w_f}^3}{r_i\left[6d^2e_w + 12e_fw_f(2d + e_f)\right]}$$

$$C = \frac{170 \times 17^3 + 8 \times 34 \times 46.75^3}{2\,491\left[6 \times 170^2 \times 17 + 12 \times 34 \times 46.75 \times (340 + 34)\right]}$$

$C = 1.1399 \times 10^{-3} > sf_sp\ /(Ep_{ys}) = 0.206 \times 10^{-3}$

$$d/e_w = 10 \leq 1.1\sqrt{E/(sf_s)} = 32.9$$

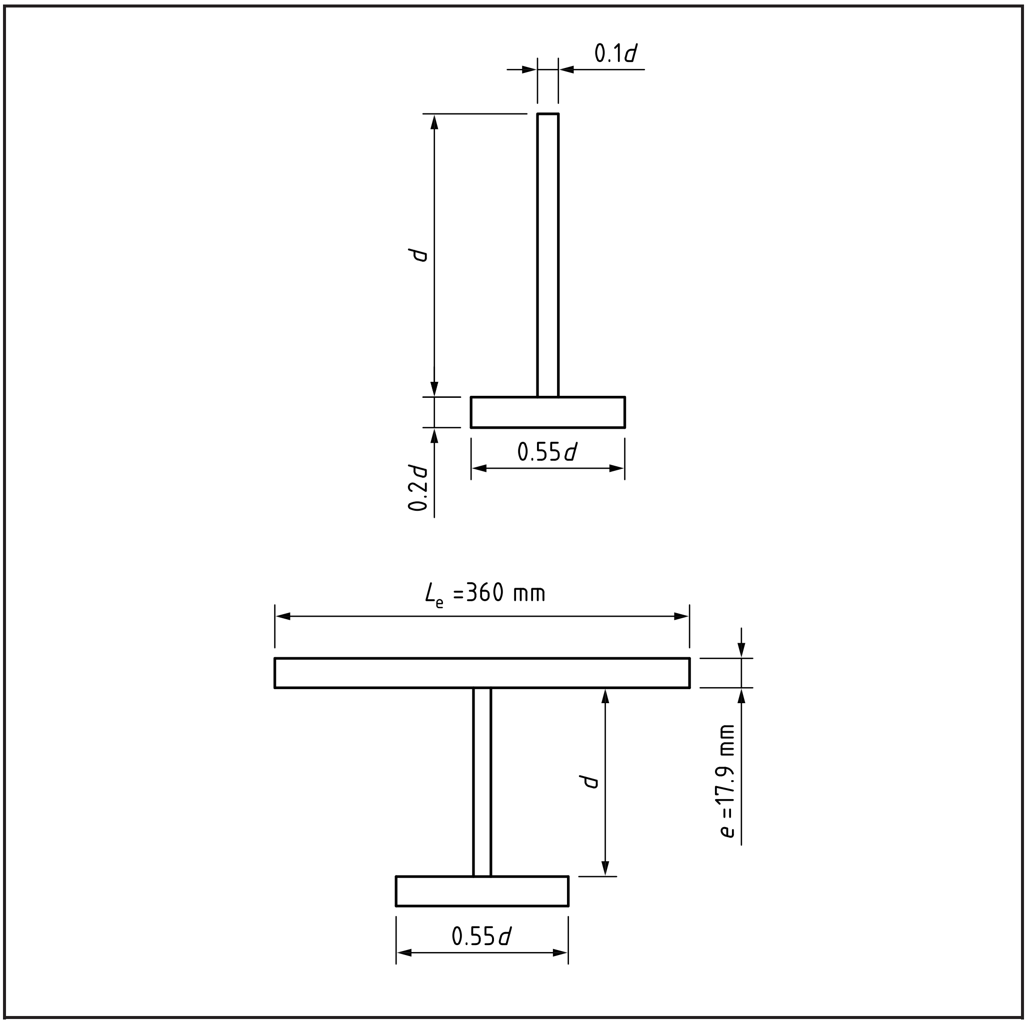
$$\leq 0.67\sqrt{Ep_{ys}/(sf_sp)} = 46.7$$

$$w_f/e_f = 1.375 \leq 0.5\sqrt{E/(sf_s)} = 15.0$$

$$\leq 0.32\sqrt{Ep_{ys}/(sf_sp)} = 22.3$$

The exact dimensions, appropriate to d = 170 mm, are unlikely to be convenient and the nearest convenient geometry can be taken with additional check calculations if thought necessary to ensure meeting the design criteria.

Figure W.1-1 **Stiffener proportions**



W.1.2.2 Example 2. Light stiffeners using finite length between ends

W.1.2.2.1 Basis of analysis

This example makes use of simplification a) in Example 1, but stiffener design is assessed using a more rigorous method and less conservative approach than used in Example 1. It illustrates the determination of scantlings for a vessel which will be subject in service to both low internal pressure and vacuum conditions. The design data is given in Table W.1-4.

Table W.1-4 **Design data assumed for complete vessel**

Inside diameter	D_i = 2 000 mm
Length along straight	L_{cyl} = 6 000 mm
Design internal pressure	p = 12.5 bar = 1.25 N/mm^2
Design external pressure	p = 1.0 bar = 0.10 N/mm^2
Modulus of elasticity (from Table 3.6-3)	$E = 2.07 \times 10^5$ N/mm^2
Poisson's ratio	$\nu = 0.3$
Design stress, cylinder	f = 165 N/mm^2
Design stress, stiffener	f_s = 165 N/mm^2
Stress factor	s = 1.4

W.1.2.2.2 Calculation of thicknesses for internal pressure

1) *Required cylinder thickness excluding corrosion allowance*

$$e = \frac{1.25 \times 2000}{(2 \times 165) - 1.25} = 7.065\text{mm}$$

(from **3.5.1.2**)

2) *Required end thickness excluding corrosion allowance*

Mean inside radius of shell R_i = 2 000 mm
Inside knuckle radius r_i = 200 mm
Inside diameter D_i = 2 000 mm

Inside dished end height $h_i = R_i - \sqrt{(R_i - D_i/2)(R_i + D_i/2 - 2r_i)}$

$$h_i = 2\,000 - \sqrt{\left(2\,000 - \frac{2000}{2}\right)\left(2\,000 + \frac{2\,000}{2} - 400\right)}$$

= 387.55 mm

Assume e = 12.5, then $h_e/D_o = \frac{400.05}{2\,025} = 0.198$

$$p/f = \frac{1.25}{165} = 0.0076$$

e/D_o = 0.0063 (from Figure 3.5-2)

$e = 0.0063 \times 2\,025$

= 12.76, say 12.75 mm

W.1.2.2.3 Check for external pressure

1) *End*

R = 2 006.375 mm

$$p_{yss} = \frac{2 \times 1.4 \times 165 \times 12.75}{2\ 006.375} = 2.9359 N/\mathrm{mm}^2$$

[from Equation (3.6.4-1)]

$$p_e = \frac{1.21 \times 2.07 \times 10^5 \times 12.75^2}{2\ 006.375} = 10.115 N/\mathrm{mm}^2$$

[from Equation (3.6.4-2)]

$p_e/p_{yss} = K = 3.445$

$p_a/p_{yss} = \Delta = 0.3175$ [from Figure 3.6-4b)]

Thus p_a(maximum allowable) = 0.932 (from **3.6.4**)

This is satisfactory as p_a(maximum allowable) > design external pressure.

2) *Cylinder*

R_i = 1 000 mm

e = 7.605 mm

R = 1 003.8 mm

L = 6 000 + 2 (0.4 × 387.55) = 6 310 mm [from Figure 3.6-1a)]

$2R/e$ = 2 007.6/7.605 = 263.98

$L/(2R)$ = 6 310/2 007.6 = 3.143

$\varepsilon = 9.6 \times 10^{-5}$ (from Figure 3.6-2)

$$p_y = \frac{1.4 \times 165 \times 7.605}{1\ 003.8}$$

[from Equation 3.6.2-7)]

= 1.750 N/mm^2

$$p_m = \frac{2.07 \times 10^{-5} \times 7.605 \times 9.6 \times 10^{-5}}{1\ 003.8}$$

[from Equation (3.6.2-8)]

= 0.11506 N/mm^2

$p_m/p_y = K = 0.0860$

$p_a/p_y = \Delta = 0.0290$ [from Figure 3.6-4a)]

Thus p_a (maximum allowable)

= 0.0290 × 1.750

= 0.05075 N/mm^2

p_a(maximum allowable) < design external pressure therefore stiffener(s) have to be provided or the shell thickness increased. The increased shell thickness that gives a value of p_a = 0.10 N/mm^2 using the above simplified calculation method is 9.8 mm.

Alternatively using method B, an internal flat bar 82 mm × 9 mm, as an intermediate cold formed light stiffener, on a cylinder thickness of 7.605 mm is found to be just adequate (see Table W.1-5, which shows key values as determined by computer using the relevant equations in **3.6**).

For each value of n, the value of F_n was less than 1.0 and greater than 0, and p_n was not less than $2p$. Thus an integral flat bar of 82 mm × 9 mm on a cylinder of 7.605 mm thickness is adequate.

Table W.1-5 **Values required for stiffener design**

Base parameters		Calculated functions				
E, N/mm^2	2.07×10^5	Functions	Values for n			
ν	0.3		2	3	4	5
e, mm	7.605	L_e, mm	138[a]	136[a]	134[a]	129[a]
R, mm	1 003.8	A_c, mm^2	1 797	1 776	1 759	1 720
L_s, mm	3 155	I_c, mm^4	1.289×10^6	1.284×10^6	1.278×10^6	1.264×10^6
d, mm	82	p_n, N/mm^2	1.984	0.807	1.268	1.974
e_w, mm	9	F_n	0.1624	0.8516	0.9013	0.8654
L_c, mm	6 310	–	–	–	–	–
p_y, N/mm^2	1.750	–	–	–	–	–
p_m, N/mm^2	0.305 (n = 5)	–	–	–	–	–
p_{ys}, N/mm^2	3.259	–	–	–	–	–

[a] L_e is determined using the approximation $L_e = Z'R$ where Z′ is taken from Table 3.6-7.

W.1.2.3 Example 3. Safe external pressure to Annex M

W.1.2.3.1 Basis of analysis

This example illustrates the use of Annex M in assessing the allowable external pressure for cylindrical sections outside the circularity limits specified in **3.6**.

A somewhat lower value of sf than would normally be applicable has been used in this example for illustrative purposes. Otherwise, with this particular geometry, the reduction in allowable pressure due to the assumed shape imperfections would have been less significant and the allowable pressure limited by the criterion $p_q \leq 1.5\ p_a$. Table W.1-6 gives the design data assumed for this example.

Table W.1-6 **Design data assumed for cylindrical sections**

Modulus of elasticity (from Table 3.6-3)	$E = 2.07 \times 10^5$ N/mm^2
Poisson’s ratio	$\nu = 0.3$
Minimum calculated thickness	e = 8.89 mm
Mean radius of shell	R = 1 000 mm
Unsupported length of shell	L = 2 313 mm
Effective yield stress	sf = 121 N/mm^2
With the approximation	$\gamma = 0$

Therefore calculating p_a the allowable external pressure of the cylinder to tolerance, from **3.6.2.1**:

p_y = 1.07 N/mm^2 [from Equation (3.6.2-7)]

ε = 0.00033954 (from Figure 3.6-2)

p_m = 0.62475 N/mm^2 [from Equation (3.6.2-8)]

p_m/p_y = 0.584

p_a/p_y = 0.195 [from Figure 3.6-4, curve a)]

Therefore p_a = 0.20825 N/mm^2, this is the value of p_a to be used in the Annex M calculations.

W.1.2.3.2 Measured shape

Values of the radii R_r measured at 24 equally spaced intervals and the calculated departures from the mean circle ε_r are given in Table W.1-7.

NOTE 1 The values of ε_r were derived by using Table 1 in Enquiry Case 5500/33.

It can be seen from Table W.1-7 that the maximum departure from the mean circle is 12.7 mm which exceeds the allowable tolerance of 0.5% of the radius. Thus Annex M has been used to derive the allowable external pressure.

In this case the procedure is as follows.

a) Initially, evaluate coefficients a_n, b_n and $p_{m(n)}$.

b) Next, evaluate for every value of r (0, 1, 2,.....,23), the pressure p at which $pR/e + |\sigma_{br}| = sf$, [Equation (M-2)], where σ_{br} is obtained from Equation (M-4).

c) The lowest value of p, obtained at any location, is p_q the lower bound estimate to the collapse pressure.

d) This value of p_q is used in Equation (M-1) to obtain p(allowable), the allowable external pressure for the out-of-tolerance cylinder.

NOTE 2 These calculations require trial and error methods or systematic iteration process, and to be practical, the use of a computer.

For the example under consideration, the calculated values of coefficients a_n, b_n and $p_{m(n)}$ are given in Table W.1-8; from the calculations for all values of r, from 0 to 23, the lowest value of p to satisfy Equation (M-2) was 0.2014 at r = 18; the corresponding terms for σ_{br} for each value of n and the results of the summation are given in Table W.1-9.

With:

σ_{br} = 98.35;

sf = 121 = 98.35 + 22.65;

therefore pR/e = 22.65;

and p = 0.2014;

this lowest value of p is thus p_q, which is less than 1.5p_a.

$$p(\text{allowable}) = \frac{p_q}{1.5} + \left(p_a - \frac{p_q}{1.5}\right)\frac{0.005R}{|W_{\max}|}$$
$$= \frac{0.2014}{1.5} + \left(0.20825 - \frac{0.2014}{1.5}\right)\frac{0.005 \times 1000}{12.7}$$

= 0.1634 N/mm^2

The allowable pressure has been reduced from 0.20825 N/mm^2 to 0.1634 N/mm^2 due to the shape being outside the 0.5% tolerance.

Table W.1-7 **Measured radii and departure from mean circle**

r	Measured radii R_r mm	Departure from mean circle ε_r mm	r	Measured radii R_r mm	Departure from mean circle ε_r mm
0	991.4	–6.0	12	993.4	–2.1
1	998.6	1.7	13	1006.2	10.2
2	996.9	0.5	14	1005.7	9.1
3	1008.6	12.7	15	994.0	–0.31
4	1006.8	11.4	16	991.3	–6.3
5	998.3	3.3	17	993.1	–4.9
6	989.8	–4.9	18	1008.4	+10.1
7	987.7	–6.8	19	1003.8	5.3
8	994.4	0	20	1003.7	5.2
9	987.7	–6.8	21	990.5	–7.9
10	989.7	–5.1	22	990.7	–7.5
11	993.8	–1.3	23	991.0	–6.9

Table W.1-8 **Values of a_n, b_n and $p_{m(n)}$**

n	a_n mm	b_n mm	$p_{m(n)}$ N/mm^2
0	—	996.48	—
1	–1.80	0.95	—
2	4.46	–1.18	46.819
3	1.79	–5.13	6.113
4	3.05	–0.10	1.473
5	–3.72	1.89	0.706
6	–0.38	–0.02	0.625
7	1.09	0.96	0.726
8	–0.15	–1.00	0.899
9	0.32	0.66	1.114
10	1.26	–2.15	1.362
11	1.23	–0.34	1.639
12	0	0.37	1.944

Table W.1-9 **Values of $\sigma_{br(n)}$**

(p = 0.2014, r = 18, $r\phi$ = 270°)	
n	$\sigma_{br(n)}$
2	0.02
3	0.53
4	–0.24
5	36.92
6	0.28
7	20.53
8	–18.64
9	–5.78
10	37.58
11	20.93
12	6.22
$\sum_{n=2}^{N/2} \sigma_{br(n)}$	98.35

W.1.2.4 Example 4. Maximum departure from mean circle using chord gauge

W.1.2.4.1 Basis of analysis

The following example shows the calculation of the maximum departure from mean circle of a cylinder using chord gauge measurements at N = 24 equally spaced positions on the circumference. This example uses the procedure of Method 3 of Enquiry Case 5500/33.

W.1.2.4.2 Calculation

For a cylinder of mean radius 2 000 mm the following chord gauge readings δ, were obtained at 15° intervals starting at the crown. The departure from mean circle at each position ε_r was calculated from the following formula using influence coefficients I_r given in Method 3 of Enquiry Case 5500/33.

$$\varepsilon_r = \sum_{i=0}^{N-1} \delta_i.17em\ I_{(i-r)}$$

θ	0	15	30	45	60	75
δ (mm)	70.2	70.6	69.1	67.0	66.2	67.1
ε_r (mm)	6.6	8.4	5.0	–0.6	–4.0	–3.4
θ	90	105	120	135	150	165
δ (mm)	68.8	69.5	68.8	67.4	67.0	67.7
ε_r (mm)	–0.5	1.1	0.1	–2.2	–2.9	–1.2
θ	180	195	210	225	240	255
δ (mm)	68.8	69.1	68.3	67.4	67.5	68.7
ε_r (mm)	1.4	2.7	2.0	0.8	1.0	2.4
θ	270	285	300	315	330	345
δ (mm)	69.6	69.1	67.4	65.9	66.1	68.1
ε (mm)	2.5	–0.3	–5.0	–8.0	–6.0	–0.2

For example, the value of ε_r at $\theta = 0°$ was obtained by summing:

$$\varepsilon_0 = (70.2)\ (1.76100) + (70.6)\ (0.85587) + (69.1)\ (0.12834) + \ldots + (68.1)\ (0.85587) = 6.6.$$

The value of ε_r at $\theta = 105°$ was obtained by summing:

$$\varepsilon_7 = (70.2)\ (-0.47097) + (70.6)\ (-0.68487) + (69.1)\ (-0.77160) + \ldots + (68.1)\ (-0.18614) = 1.1.$$

For this example it is seen that the maximum departure from the mean circle is 8.4 mm occurring at $\theta = 15°$ and is less than $0.005R = 10$ mm.

W.2 Supports and mountings for horizontal vessels

W.2.1 General

W.2.1.1 Introduction

Subclause **W.2** provides worked examples illustrating the use of **G.3.3** in the design of supports and mountings for horizontal vessels. Three different examples of possible supporting arrangements for a horizontal vessel with torispherical ends, as used for the storage of glucose syrup at low pressure, are considered.

a) Example 1 uses twin saddle supports located away from the ends.

b) Example 2 uses twin saddle supports located near the ends.

c) Example 3 utilizes a ring and leg support.

In each case the support mountings are welded to the vessel.

Starting with the minimum required vessel thickness to satisfy pressure and temperature considerations, the stresses are calculated and compared with the appropriate recommended limits in the sequence in which they are dealt with in **G.3.3.2**. When a limit is exceeded the critical calculations are repeated with an increased shell thickness or with the addition of internal or external ring stiffeners. Table W.2-1 gives the design data for these examples.

Table W.2-1 **Design data**

Design pressure (i.e. top pressure)	Zero to 2 bar gauge ($0.2\ N/mm^2$)
Design temperature	50 °C
Specific gravity of fluid contents	1.4
Specific weight of water γ	9 810 N/m^3
Construction category	3: visual inspection only
Material	BS 1501-320-S31 for all wetted parts
Inside diameter of cylindrical vessel	2 591 mm
Inside radius of cylindrical vessel, r_i	1 295.5 mm
Barrel length of cylindrical vessel, tan to tan	7 055 mm
Torispherical end:	
spherical inside radius	2 591 mm
knuckle inside radius	254 mm
depth of end, b (see Figure W.2-1)	499 mm
Distance to saddle from end, A, in Example 1 and 3	732 mm

NOTE For these examples the value of A is assumed to be fixed by site conditions.

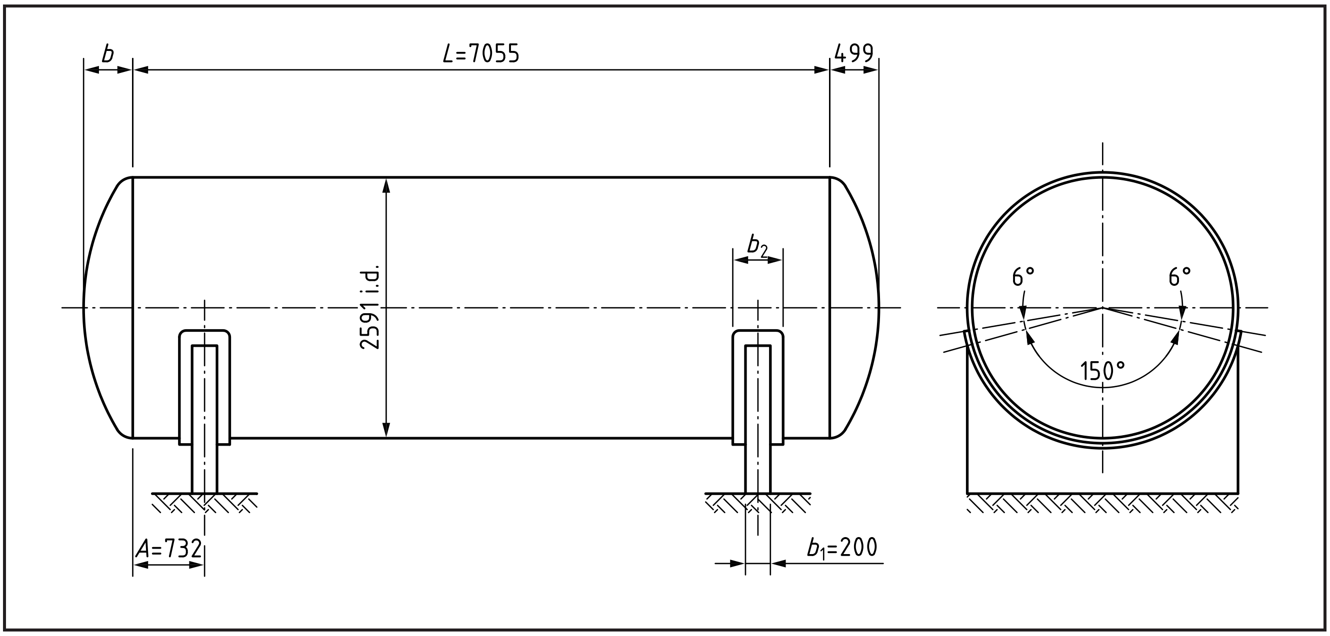
W.2.1.2 Notation

For the purposes of **W.2** the following symbols apply, based upon those given in **G.3.1.2**.

a	is the effective cross-sectional area of a stiffener;
A	is the distance from saddle support to adjacent end of cylindrical part;
b_1	is the axial width of saddle support;
b_2	is the effective axial width of saddle support, = $b_1 + 10t$;
c	is the distance from stiffener centroid to shell;
C_5	is a calculation constant;
d	is the distance from stiffener centroid to tip of stiffener;
D	is the mean diameter of the vessel;
E	is the modulus of elasticity from Table 3.6-3;
f	is the design stress;
$f_1 \ldots f_{10}$	are the resultant stresses due to mode of support;
h	is the torispherical end depth;
h_e	is the torispherical end effective depth;
I	is the second moment of area of the effective stiffener section;
K	is an external pressure design parameter;
$K_1 \ldots K_8$	are calculation constants;
M_3	is the vessel longitudinal bending moment midway between supports;
M_4	is the vessel longitudinal bending moment at supports;
p	is the design pressure (at the vessel top);
p_m	is the pressure at the vessel equator (horizontal centre-line);
q	is the shear stress in the shell;
q_e	is the shear stress in vessel end;

r_i	is the vessel inside radius;
R	is the torispherical end, spherical inside radius;
s	is a material factor;
t	is the analysis thickness of the shell;
W_1	is the maximum reaction at a support;
Δ	is an external pressure design factor;
θ	is the saddle support angle;
σ_1, σ_2	are principal stresses;
σ_θ, σ_z	are stress intensities.

Figure W.2-1 **Vessel on saddle supports**



W.2.2 Worked examples

W.2.2.1 Example 1. Twin saddle supports located remote from the ends

W.2.2.1.1 Design details

Figure W.2-1 shows the layout of the vessel on the saddles. Internal pressure and manufacturing considerations require the following thicknesses:

— cylindrical shell = 2.5 mm minimum;

— dished, torispherical ends = 5.5 mm minimum.

Assuming the vessel shell thickness t = 2.5 mm, the reaction per support, arising from the contents and self weight of the vessel, W_1 = 286 855 N.

Mean radius r = 1 295.5 + 1.25 = 1 296.75 mm.

*NOTE A value of saddle width b_1 = 200 mm was selected from previous experience, despite it being less than the suggested value of $\sqrt{30D}$ given in **G.3.3.2.7**.*

W.2.2.1.2 Longitudinal bending moments (see G.3.3.2.2)

1) *At mid-span*

$$M_3 = \frac{W_1 L}{4}\left[\frac{1+2\left(r^2-b^2\right)/L^2}{1+4b/(3L)} - \frac{4A}{L}\right] \qquad \text{(G.3.3-1)}$$

where r is the mean radius of vessel.

Thus

$$M_3 = \frac{286\ 855 \times 7\ 055}{4}\left[\frac{1+2(1\ 296.75^2-499^2)/7\ 055^2}{1+4\times 499/(3\times 7\ 055)} - \frac{4\times 732}{7\ 055}\right]$$

M_3 = +278.97 × 10^6 N·mm

2) *At the supports*

$$M_4 = -W_1 A\left[1-\frac{1-A/L+\left(r^2-b^2\right)/(2AL)}{1+4b/(3L)}\right] \qquad \text{(G.3.3-2)}$$

$$M_4 = -286\ 855 \times 732\left[1-\frac{1-732/7\ 055+(1\ 296.75^2-499^2)/(2\times 732\times 7\ 055)}{1+4\times 499/(3\times 7\ 055)}\right]$$

M_4 = −11.39 × 10^6 N·mm

W.2.2.1.3 Longitudinal stresses at mid-span (see G.3.3.2.3)

1) *The stress at the highest point of the cross-section*

$$f_1 = \frac{p_m r}{2t} - \frac{M_3}{\pi r^2 t} \qquad \text{(G.3.3-5)}$$

f_1 = (0.2 + 9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) − 278.97 × 10^6/ (π × 1 296.75^2 × 2.5)

f_1 = 35.36 N/mm^2

Check the condition when the vessel is full of product with zero top pressure, i.e. p_m = 1.4γr_1

f_1 = (9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) − 278.97 × 10^6/ (π × 1 296.75^2 × 2.5)

f_1 = −16.51 N/mm^2.

2) *The stress at the lowest point of the cross-section*

$$f_2 = \frac{p_m r}{2t} + \frac{M_3}{\pi r^2 t} \qquad \text{(G.3.3-6)}$$

f_2 = (0.2 + 9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) + 278.97 × 10^6/ (π × 1 296.75^2 × 2.5)

f_2 = 77.60 N/mm^2.

W.2.2.1.4 Longitudinal stresses at the supports (see G.3.3.2.4)

1) *The stress near the equator*

This is given by:

$$f_3 = \frac{p_m r}{2t} - \frac{M_4}{K_1 \pi r^2 t} \qquad \text{(G.3.3-7)}$$

where K_1 = 0.161 from Table G.3.3-1 for θ = 150°, since A > r/2 and the shell is not stiffened.

Thus:

$f_3 = (0.2 + 9\,810 \times 1.4 \times 1\,295.5 \times 10^{-9}) \times 1\,296.75/(2 \times 2.5) + 11.39 \times 10^6)/(0.161 \times \pi \times 1\,296.75^2 \times 2.5)$

$f_3 = 61.84$ N/mm^2.

2) *The stress at the lowest point of the cross-section*

$$f_4 = \frac{p_m r}{2t} + \frac{M_4}{K_2 \pi r^2 t} \quad \text{(G.3.3-8)}$$

where $K_2 = 0.279$ from Table G.3.3-1 for $\theta = 150°$.

Thus:

$f_4 = (0.2 + 9\,810 \times 1.4 \times 1\,295.5 \times 10^{-9}) \times 1\,296.75/(2 \times 2.5) - 11.39 \times 10^6)/(0.279 \times \pi \times 1\,296.75^2 \times 2.5)$

$f_4 = 53.39$ N/mm^2 (for zero top pressure, $f_4 = 1.52$ N/mm^2).

W.2.2.1.5 Allowable direct stresses

The calculated tensile and compressive stresses should not exceed the values permitted in **A.3.4.2.1** and **A.3.5**.

a) *Stress intensity (see **B.4**)*

The stress intensity acting at the point considered shall be taken as:

$= \max(\sigma_1 - \sigma_2; \sigma_1 + 0.5p; \sigma_2 + 0.5p)$

where σ_1 and σ_2 are the principal stresses.

For this case the stress intensity shall not exceed f. For austenitic stainless steels, category 3 construction, **3.4.2.2** indicates that the design stress f shall not exceed the values permitted for construction categories 1 and 2 vessels and the smaller of 120 N/mm^2 or 120 × 450/(400 + T) where, T is the design temperature in °C, i.e.

$f = 120 \times 450/(400 + 50) = 120$ N/mm^2

For this case the primary membrane circumferential stress:

at the highest point of the cross-section = $\sigma_\theta = pr/t$

$= 0.2 \times 1\,296.75/2.5 = 103.74$ N/mm^2;

at the lowest point of the cross-section = $\sigma_\theta = (p + 2r_i \times 1.4\,\gamma)r/t$

$= (0.2 + 2 \times 1\,295.5 \times 1.4 \times 9\,810 \times 10^{-9}) \times 1\,296.75/2.5 = 122.20$ N/mm^2.

The primary membrane stress intensities involving the longitudinal stress are $\sigma_\theta - \sigma_z$ and $\sigma_z + 0.5p$. At mid-span and the support they are as follows.

Highest point of cross-section:

$\sigma_\theta - \sigma_z = 103.74 - 35.36 = 68.38$ N/mm^2;

$\sigma_z + 0.5p = 35.36 + 0.5 \times 0.2 = 35.46$ N/mm^2.

Lowest point of cross-section:

$\sigma_\theta - \sigma_z = 122.20 - 77.60 = 44.60$ N/mm^2;

$\sigma_z + 0.5p = 77.60 + 0.5 \times 0.2 = 77.70$ N/mm^2.

At the support they are as follows:

Equator:

$\sigma_\theta - \sigma_z = 103.74 - 61.84 = 41.90$ N/mm^2;

$\sigma_z + 0.5p = 61.84 + 0.5 \times 0.2 = 61.94$ N/mm^2.

Lowest point of cross-section:

$\sigma_\theta - \sigma_z = 122.20 - 53.39 = 68.81$ N/mm^2;

$\sigma_z + 0.5p = 53.39 + 0.5 \times 0.2 = 53.49$ N/mm^2.

In each case the values are less than f and thus acceptable.

b) *Limit for the longitudinal compressive stress (see* ***A.3.5****)*

In this case the calculated compressive stress (see **A.3.5**) shall not exceed Δsf where Δ is obtained from Figure A.2 in terms of K with s and f defined in **3.6.1.1**. For this case $K = p_e/p_{yss}$.

Using Equations (3.6.4-1) and (3.6.4-2) of 3.6.4 and the nomenclature used earlier:

$$K = \frac{1.21Et^2}{r^2} \times \frac{r}{2sft} = \frac{0.605Et}{sfr}$$

where

$E = 199.7 \times 10^3$ N/mm^2 from Table 3.6-3 by interpolation;

$s = 1.1$;

$f = 163$ N/mm^2 at 50 °C for BS 1501-320-S31;

$t = 2.5$ mm.

Thus

$$K = \frac{0.605 \times 199.7 \times 10^3 \times 2.5}{1.1 \times 163 \times 1\,296.75}$$

$K = 1.30$

From Figure A.2

$\Delta = 0.5\,[1 - (1 - 0.125 \times 1.30)^2]$

$= 0.149$

Thus $\Delta sf = 0.149 \times 1.1 \times 163 = 26.76$ N/mm^2

i.e. allowable compressive stress = 26.76 N/mm^2.

It is noted that the highest longitudinal compressive stress is when the vessel is full of contents but not pressurized, i.e. $f_1 = -16.51$ N/mm^2. $f_1 < \Delta sf$ and so the stress is acceptable.

W.2.2.1.6 Tangential shearing stresses (see G.3.3.2.5) at the support

1) *Shells of thickness, 2.5 mm*

For a shell not stiffened by the end, $A > r/2$

$$q = \frac{K_3 W_1}{rt}\left[\frac{L - 2A}{L + 4b/3}\right] \qquad \text{(G.3.3-9)}$$

where $K_3 = 0.799$ from Table G.3.3-2 for $\theta = 150°$

Thus

$$q = \frac{0.799 \times 286\,855}{1\,296.75 \times 2.5}\left[\frac{7\,055 - 2 \times 732}{7\,055 + 4 \times 499/3}\right]$$

$q = 51.20$ N/mm^2

The allowable tangential shearing stress from Table G.3.3-2 is the smaller of $0.8f$ and $0.06Et/r$.

$0.8f = 0.8 \times 120 = 96$ N/mm^2

$0.06Et/r = 0.06 \times 199.7 \times 10^3 \times 2.5/1\ 296.75 = 23.10$ N/mm^2

i.e. $q > 0.06Et/r$ and therefore the design should be changed.

2) *Options for reassessment of design*

The options are:

a) increase the shell thickness;

b) move the saddles to within $r/2$ of the ends;

c) stiffen with rings in the plane of the saddle.

Options b) and c) will be used later so at this stage increase the shell thickness to 5 mm.

3) *Shell of increased thickness, 5 mm*

For this the reaction per support will be increased to W_1 = 292 511 N, and the mean radius $r = 1\ 295.5 + 2.5 = 1\ 298.0$ mm.

These changes will marginally alter M_3 and M_4 and the values of f_1, f_2, f_3 and f_4. The stresses will be lower because of the increased shell thickness. In general it is not necessary to calculate these values when the shell thickness is increased. However, for completeness, they are as follows:

$M_3 = +284.54 \times 10^6$ N·mm;

$M_4 = -11.55 \times 10^6$ N·mm;

$f_1 = 17.52$ N/mm^2 or, when the top pressure is zero, -8.44 N/mm^2;

$f_2 = 39.02$ N/mm^2;

$f_3 = 30.98$ N/mm^2;

$f_4 = 26.71$ N/mm^2.

It is noted that the stress values are approximately one half of those in the case of the thinner shell. This reduction also applies to the primary membrane circumferential stresses. The stress intensities are similar to those outlined in **W.2.2.1.5**a) and will all, therefore, be less than f and thus are acceptable.

The allowable compressive stress will be greater for this case.

$$K = \frac{0.605 \times 199.7 \times 10^3 \times 5}{1.1 \times 163 \times 1\ 298} = 2.596$$

From Figure A.2

$\Delta = 0.5\ [1 - (1 - 0.125 \times 2.596)^2] = 0.272$

Thus $\Delta sf = 0.272 \times 1.1 \times 163 = 48.76$ N/mm^2

Since the highest compressive stress $f_1 = 8.44$ N/mm^2 $f_1 < \Delta sf$ and is acceptable.

Consider again the expression for q in Equation (G.3.3-9) with the increased shell thickness and the component changes in W_1 and r:

$$q = \frac{0.799 \times 292\ 511}{1\ 298 \times 5}\left[\frac{7055 - 2 \times 732}{7\ 055 + 4 \times 499/3}\right]$$

$q = 26.08$ N/mm^2

As in **W.2.2.1.6**1), the allowable stress is the smaller of $0.8f$ and $0.06Et/r$.

$0.8f = 0.8 \times 120 = 96$ N/mm^2

$0.06Et/r = 0.06 \times 199.7 \times 10^3 \times 5/1\ 298 = 46.15$ N/mm^2

i.e. $q < 0.06Et/r$ and is thus acceptable with $t = 5$ mm.

W.2.2.1.7 Circumferential stresses (see G.3.3.2.6.2)

1) *Position of maximum stress*

The maximum values of the circumferential stress occur in the region of the saddle support. It is recommended in **G.3.3.2.6.2** that the thickness of the saddle plate, t_1, should be equal to the thickness of the shell plate, t. In the case of an extended saddle plate of width $b_2 = b_1 + 10t$, and angle not less than (θ + 12°) certain stress reductions in f_5 and f_6 can be obtained.

2) *Stress at the lowest point of the cross-section*

For saddles welded to the vessel, **G.3.3.2.6.2** should be used in conjunction with Table G.3.3-4 to determine the value of K_5. An extended saddle plate, as defined in 1) above, is to be used. If the extended saddle plate is welded to the vessel then the combined shell and saddle plate thickness may be used when deriving the stress in the *plane of the saddle*.

$$f_5 = \frac{-K_5 W_1}{(t + t_1) b_2} \qquad \text{(G.3.3-20)}$$

From Table 3.3-4 $K_5 = 0.673$ for $\theta = 150°$ (see **G.3.3.2.6.2**)

When the saddle is welded to the vessel, K_5 may be taken as one-tenth of this value, hence $K_5 = 0.0673$ (see **G.3.3.2.6.2**)

$(t + t_1) = 5 + 5 = 10$ mm;

$b_2 = b_1 + 10t = 200 + 10 \times 5 = 250$ mm;

$W_1 = 292\ 511$ N.

Thus

$f_5 = -0.0673 \times 292\ 511/(10 \times 250)$;

$f_5 = -7.87$ N/mm^2.

In the region at the edge of the saddle plate where the thickness is t:

$$f_5 = \frac{-K_5(\text{for } \theta = 162°) W_1}{t b_2}$$

where

$K_5 = 0.0651$ for $\theta = 162°$ by interpolation;

$t = 5$ mm;

$b_2 = b_1 + 10t = 200 + 10 \times 5 = 250$ mm;

$W_1 = 292\ 511$ N.

Thus

$f_5 = -0.0651 \times 292\ 511/(5 \times 250)$;

$f_5 = -15.23$ N/mm^2.

Subclause **G.3.3.2.6.1** limits f_5 to f (= 120 N/mm^2) and thus the above is acceptable.

3) *Stress at the horn of the saddles*

Since L/r = 7 055/1 298 = 5.435, use Equation (G.3.3-19)

$$f_6 = \frac{-W_1}{4tb_2} - \frac{12K_6W_1r}{Lt^2}$$

Since the saddle plate has been extended by 12° and has a width of value b_2, the stresses at both the edge of saddle (θ = 150°) and at the edge of the extended saddle plate (θ = 162°) should be determined. At the edge of the saddle, thickness = shell thickness t + saddle plate thickness t_1 = 10 mm;

b_2 = 200 + 10 × 5 = 250 mm;

θ = 150°;

K_6 = 0.0109 from Table G.3.3-3, for A/r = 732/1 298 = 0.564 and θ = 150°, by interpolation.

Thus at the edge of the saddle:

$$f_6 = \frac{-292\ 511}{4 \times 10 \times 250} - \frac{12 \times 0.0109 \times 292\ 511 \times 1\ 298}{7\ 055 \times 10^2};$$

f_6 = –99.64 N/mm^2.

At the edge of the saddle plate the thickness = 5 mm;

b_2 = 250 mm;

θ = 162°;

K_6 from Table G.3.3-3 requires double interpolation as shown in Table W.2-2.

Table W.2-2 **Interpolation of Table G.3.3-3 for K_6**

A/r	Angle θ (degrees)		
	150	162	165
≤0.5	0.0079	0.0063	0.0059
0.564	0.0109	0.0087	0.0082
≥1.0	0.0316	0.0254	0.0238

K_6 = 0.0087.

Thus at the edge of the saddle:

$$f_6 = \frac{-292\ 511}{4 \times 5 \times 250} - \frac{12 \times 0.0087 \times 292\ 511 \times 1298}{7\ 055 \times 10^2}$$

f_6 = –283.24 N/mm^2

The numerical value of circumferential stress f_6 should not exceed 1.25 times the design stress, i.e.

$f_6 \leq 1.25 \times 120 \leq 150$ N/mm^2

Thus the stress at the edge of the saddle plate is not acceptable and the design should be changed. The options are:

a) increase saddle angle;

b) increase width of saddle and saddle plate;

c) increase shell thickness (and saddle plate thickness);

d) stiffen the shell with rings;

e) move the saddle to within r/2 of ends.

Options a) and b) are unlikely to reduce the stress by the required amount.

Option e) will be considered as Example 2. Thus try option c) and d) and increase the shell to 8 mm. For this thickness of shell the reaction per support is increased to W_1 = 299 300 N.

It is only necessary to consider the stress at the edge of the saddle plate, θ = 162°.

$b_2 = b_1 + 10t = 200 + 10 \times 8 = 280$ mm;

r = 1 295.5 + 4 = 1 299.5 mm;

A/r = 732/1 299.5 = 0.563;

K_6 = 0.0087 using the double interpolation procedure given in Table W.2-2.

Thus for the shell increased to 8 mm thick:

$$f_6 = \frac{299\ 300}{4 \times 8 \times 280} - \frac{12 \times 0.0087 \times 299\ 300 \times 1\ 299.5}{7\ 055 \times 8^2};$$

f_6 = –123.33 N/mm^2.

This is now acceptable since it is less than 1.25 × f (= 150 N/mm^2).

W.2.2.1.8 Vessel stiffened with rings in the region of the saddle

For completeness, the design calculations are presented for both internal and external ring stiffening of the vessel in the region of the saddle. The internal stiffening is that of a ring in the plane of the saddle [Figure G.3.3-8a)] and the external stiffening is rings adjacent to the saddle [Figure G.3.3-8c)].

1) *Internal ring in the plane of the saddle (see Figure W.2-2)*

a) *Longitudinal stresses at mid-span and supports (see **G.3.3.2.3** and **G.3.3.2.4**).*

Values for f_1 and f_2 will correspond to those determined in **W.2.2.1.3**. The values for f_3 and f_4 are 57.35 N/mm^2 and 55.62 N/mm^2 respectively. By inspection these lead to stress intensities that are acceptable.

b) *Tangential shearing stress at supports (see **G.3.3.2.5**).*

In **W.2.2.1.6** the tangential shearing stress was determined for the saddle case, $A > r/2$ for t = 2.5 mm. Since the saddle was unstiffened the K_3 value was such that the calculated tangential shearing stress exceeded the allowable and the shell thickness was increased to 5 mm. However, when rings are used in the plane of the saddle the tangential stress is lower and the K_3 value, as obtained from Table G.3.3-2 for θ = 150°, is equal to 0.319.

Thus the tangential shearing stress for this case with t = 2.5 mm, W_1 = 286 855 N is from Equation (G.3.3-9):

$$q = \frac{K_3 W_1}{rt}\left[\frac{L - 2A}{L + 4b/3}\right];$$

$$q = \frac{0.319 \times 286\ 855}{1\ 296.75 \times 2.5}\left[\frac{7.055 - 2 \times 732}{7\ 055 + 4 \times 499/3}\right];$$

q = 20.45 N/mm^2.

This is less than 0.8f (= 96 N/mm^2) and 0.06Et/r (= 23.10 N/mm^2) and thus t = 2.5 mm is considered adequate.

c) *Circumferential stresses [see **G.3.3.2.6.3**a)]*

For this case a single rectangular cross-section ring of 150 mm radial depth and 25 mm width as shown in Figure W.2-2 is proposed.

Area of stiffener = 150 × 25 = 3 750 mm^2.

Total area a = 3 750 + (25 + 10 × 2.5) × 2.5 = 3 875 mm^2.

c = [3 750 (75 + 2.5) + 50 × 2.5 × 1.25] / (3 750 + 50 × 2.5) = 75.04 mm

d = 152.50 – 75.04 = 77.46 mm

$$I_{xx} = \frac{1}{12} \times 25 \times 150^3 + 3\ 750(77.50 - 75.04)^2 + \frac{1}{12} \times 50 \times 2.5^3 + 50 \times 2.5 \times (75.04 - 1.25)^2$$

I_{xx} = 7 734 629 mm^4

At the horn of the saddle, in the shell:

$$f_7 = \frac{C_4 K_7 W_1 rc}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-27)}$$

where K_7 = 0.0316 and K_8 = 0.303 from Table G.3.3-4 for θ = 150° and C_4 = –1 for internal rings.

Thus:

$$f_7 = \frac{-1 \times 0.0316 \times 286\ 855 \times 1\ 296.75 \times 75.04}{7\ 734\ 629} - \frac{0.303 \times 286\ 855}{3\ 875}$$

f_7 = –136.47 N/mm^2

At the horn of the saddle in the tip of the ring remote from the shell:

$$f_8 = \frac{C_5 K_7 W_1 rd}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-28)}$$

Using the previous constants and C_5 = +1 from Table G.3.3-4:

$$f_8 = \frac{1 \times 0.0316 \times 286\ 855 \times 1\ 296.75 \times 77.46}{7\ 734\ 629} - \frac{0.303 \times 286\ 855}{3\ 875}$$

f_8 = +95.29 N/mm^2

Both these values are less than 1.25f (= 150 N/mm^2) and are therefore acceptable.

2) *External rings adjacent to the saddle (see Figure W.2-3)*

In this case, the K_3 value is the same as for the saddle without rings and thus from **W.2.2.1.6** the shell thickness should be increased to t = 5 mm. This value is used for the shell thickness and leads to a use of r = 1 298 mm and W_1 = 292 511 N.

a) *Longitudinal stresses and tangential shearing stresses (see **G.3.3.2.3**, **G.3.3.2.4** and **G.3.3.2.5**).*

Values for f_1, f_2 and q will correspond to those determined in **W.2.2.1.6**. The values of f_3 and f_4 are 28.71 N/mm^2 and 27.83 N/mm^2 respectively. By inspection these lead to stress intensities that are acceptable.

b) *Circumferential stresses [see **G.3.3.2.6.3**c)].*

At the lowest point of the cross-section from Equation (G.3.3-17)

$f_5 = - K_5 W_1/(t b_2)$

If an extended saddle plate is used as in **W.2.2.1.7**, the values of f_5 at the edge of the saddle plate and in the plane of the saddle can be determined. In this case they are the same as in **W.2.2.1.7**, i.e.

in the plane of the saddle: f_5 = –7.87 N/mm^2;

at the edge of the saddle plate: f_5 = –15.23 N/mm^2.

However, if the above is not assumed:

$$f_5 = \frac{-K_5(\text{for } \theta = 150°)W_1}{tb_2}$$

where

K_5 = 0.0673 for θ = 150°, i.e. one tenth of the value from Table G.3.3-4 as the saddle is welded to the shell (see **G.3.3.2.6.2**);

t = 5 mm;

$b_2 = b_1 + 10t = 200 + 10 \times 5 = 250$ mm;

W_1 = 292 511 N.

Thus:

f_5 = –0.0673 × 292 511/(5 × 250);

f_5 = –15.75 N/mm^2.

This stress should be limited to f (= 120 N/mm^2). The above is thus acceptable.

Stresses near the equator f_7 and f_8.

For this case two rectangular rings are used adjacent to the saddle. The axial length between the stiffeners should be not less than $b_2 + 10t$ and not more than r. The proposed size of ring is 100 mm × 20 mm, as shown in Figure W.2-3.

Area of single stiffener = 100 × 20 = 2 000 mm^2.

Total area a = 2 000 + (20 + 50) × 5 = 2 350 mm^2

c = [2 000 + (50 + 5) + 70 × 5 × 2.5] / (2 000 + 70 × 5) = 47.18 mm.

d = 105 – 47.18 = 57.82 mm.

$\sum I_{xx}$ for single stiffener + plate.

$$I_{xx} = \frac{1}{12} \times 20 \times 100^3 + 2\,000\,(55.00 - 47.18)^2 + \frac{1}{12} \times 70 \times 5^3 + 70 \times 5\,(47.18 - 2.50)^2$$

I_{xx} = 2 488 407 mm^4

At a point near the equator, in the shell:

$$f_7 = \frac{C_4 K_7 W_1 rc}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-32)}$$

where K_7 = 0.0355 and K_8 = 0.219 from Table G.3.3-4 for θ = 150° and C_4 = –1.

Thus:

$$f_7 = \frac{-0.0335 \times 292\,511 \times 1\,298 \times 47.18}{2\,488\,407 \times 2} - \frac{0.219 \times 292\,511}{2\,350 \times 2}$$

f_7 = –141.41 N/mm^2

At a point near the equator, in the tip of the ring remote from the shell:

$$f_8 = \frac{C_5 K_7 W_1 rd}{I} - \frac{K_8 W_1}{a} \qquad \text{(G.3.3-33)}$$

Using the above constants and C_5 = +1 from Table G.3.3-4.

$$f_8 = \frac{0.0335 \times 292\ 511 \times 1\ 298 \times 57.82}{2\ 488\ 407 \times 2} - \frac{0.219 \times 292\ 511}{2\ 350 \times 2}$$

f_8 = +142.96 N/mm^2

Both these values are less than 1.25f (= 150 N/mm^2) and are therefore acceptable.

Further check on the f_6 *stress at the edge of the saddle plate (see* ***G.3.3.2.6.3****)*.

In view of the fact that the ring stiffeners may be located r/2 from the centre line of the saddle, it is necessary to check the f_6 saddle centre stress at the horn. The saddle region is considered to be stiffened by the rings in a manner similar to that provided by the end of the vessel, i.e. $A/r \leq 0.50$. The highest of the f_6 stresses occurs at the edge of the saddle plate θ = 162°, t = 5 mm, b_2 = 250 mm and K_6 = 0.0063 is obtained by interpolation from Table G.3.3-3.

$$f_6 = \frac{-W_1}{4tb_2} - \frac{12K_6 W_1 r}{Lt^2} \qquad \text{(G.3.3-19)}$$

$$f_6 = \frac{-292\ 511}{4 \times 5 \times 250} - \frac{12 \times 0.0063 \times 292\ 511 \times 1\ 298}{7\ 055 \times 5^2}$$

f_6 = –221.24 N/mm^2 which is greater than 1.25f.

On this basis increase the thickness to 6.5 mm.

Thus:

r = 1 295.5 + 3.25 = 1 298.75 mm;

W_1 = 296 006 N;

K_6 = 0.0063 as above;

b_2 = 200 + 10 × 6.5 = 265 mm.

Thus:

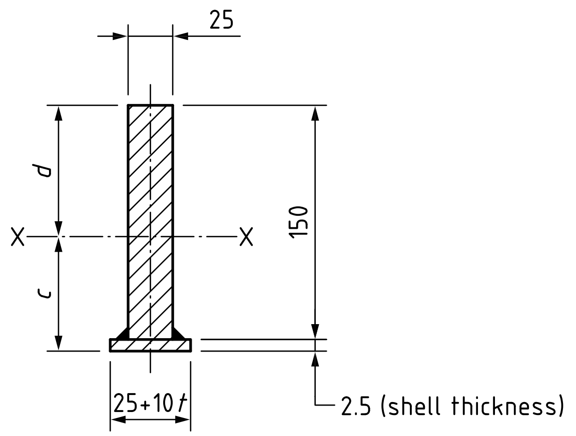
$$f_6 = \frac{-296\ 006}{4 \times 6.5 \times 265} - \frac{12 \times 0.0063 \times 296\ 006 \times 1\ 298.75}{7\ 055 \times 6.5^2}$$

f_6 = -140.47 N/mm^2;

which is satisfactory. Thus the thickness should be increased to 6.5 mm.

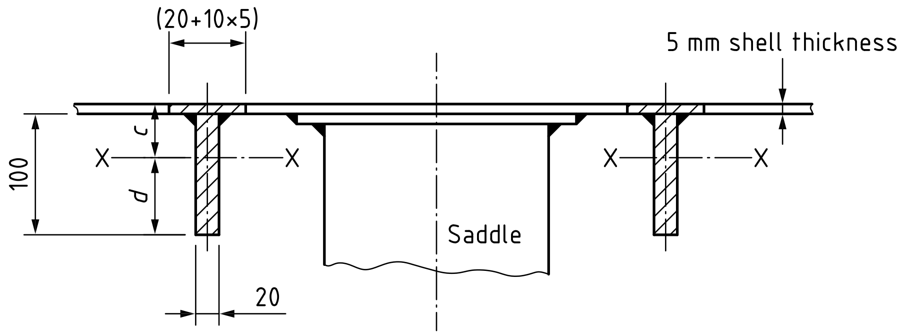
Because the thickness has been increased it would be possible to reduce marginally the size of the rings used adjacent to the saddle.

Figure W.2-2 **Internal ring stiffener in plane of saddle**



Dimensions in millimetres

Figure W.2-3 **External ring stiffeners adjacent to the saddle**



W.2.2.2 Example 2. Saddle supports near the ends

W.2.2.2.1 Design data

The design data for this example are as for Example 1 with the change that $A \leq r/2$. As in Example 1, the minimum shell thickness is assumed to be 2.5 mm and the minimum thickness of the ends is assumed to be 5.5 mm. The mean radius r = 2 591/2 + 1.25 = 1 296.75 mm.

Since $A \leq$ 1 296.75/2 $\leq$ 648.375 mm, make A = 648 mm.

The self-weight of the vessel and contents remain the same as before, i.e. W_1 = 286 855 N for t = 2.5 mm.

W.2.2.2.2 Longitudinal bending moments (see G.3.3.2.2)

1) *At mid-span*

$$M_3 = \frac{W_1 L}{4}\left[\frac{1 + 2\left(r^2 - b^2\right)/L^2}{1 + 4b/(3L)} - \frac{4A}{L}\right] \tag{G.3.3-1}$$

$$M_3 = \frac{286\ 855 \times 7\ 055}{4}\left[\frac{1 + 2(1\ 296.75^2 - 499^2)/7\ 055^2}{1 + 4 \times 499/(3 \times 7\ 055)} - \frac{4 \times 648}{7\ 055}\right]$$

M_3 = 303.07 × 10^6 N·mm

2) *At the supports*

$$M_4 = -W_1 A\left[1 - \frac{1 - A/L + \left(r^2 - b^2\right)/(2AL)}{1 + 4b/(3L)}\right] \quad \text{(G.3.3-2)}$$

$$M_4 = -286\ 855 \times 648\left[1 - \frac{1 - 648/7\ 055 + \left(1\ 296.75^2 - 499^2\right)/(2 \times 648 \times 7\ 055)}{1 + 4 \times 499/(3 \times 7\ 055)}\right]$$

M_4 = –5.01 × 10^6 N·mm

W.2.2.2.3 Longitudinal stresses at mid-span (see G.3.3.2.3)

1) *The stress at the highest point of the cross-section*

$$f_1 = \frac{p_m r}{2t} - \frac{M_3}{\pi r^2 t} \quad \text{(G.3.3-5)}$$

f_1 = (0.2 + 9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) – 303.07 × 10^6/(π × 1 296.75^2 × 2.5)

f_1 = 33.54 N/mm^2

Check the condition when the vessel is full of product with zero top pressure, i.e. p_m = 1.4 γr_i.

f_1 = (9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) – 303.07 × 10^6/(π × 1 296.75^2 × 2.5)

f_1 = –18.38 N/mm^2

2) *The stress at the lowest point of the cross-section*

$$f_2 = \frac{p_m r}{2t} + \frac{M_3}{\pi r^2 t} \quad \text{(G.3.3-6)}$$

f_2 = (0.2 + 9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/(2 × 2.5) + 303.07 × 10^6/(π × 1 296.75^2 × 2.5)

f_2 = 79.43 N/mm^2

W.2.2.2.4 Longitudinal stresses at the supports (see G.3.3.2.4)

1) *The stress at the highest point of the cross-section*

$$f_3 = \frac{p_m r}{2t} - \frac{M_4}{K_1 \pi r^2 t} \quad \text{(G.3.3-7)}$$

where K_1 = 1 from Table G.3.3-1 for θ = 150°, since $A \le r/2$.

Thus:

f_3 = (0.2 + 9 810 × 1.4 × 1 295.5 × 10^{-9}) × 1 296.75/ (2 × 2.5) + 5.01 × 10^6/(1 × π × 1 296.75^2 × 2.5);

f_3 = 56.86 N/mm^2

2) *The stress at the lowest point of the cross-section*

$$f_4 = \frac{p_m r}{2t} + \frac{M_4}{K_2 \pi r^2 t} \quad \text{(G.3.3-8)}$$

where K_2 = 1 from Table G.3.3-1 for θ = 150°, since $A \le r/2$

Thus:

$f_4 = (0.2 + 9\,810 \times 1.4 \times 1\,295.5 \times 10^{-9}) \times 1\,296.75/(2 \times 2.5) - 5.01 \times 10^6/(1 \times \pi \times 1\,296.75^2 \times 2.5)$;

$f_4 = 56.10$ N/mm^2 (For zero top pressure, $f_4 = 4.23$ N/mm^2)

W.2.2.2.5 Allowable direct stresses

The allowable direct stresses, in accordance with **A.3.4.2.1** and **A.3.5** (for $t = 2.5$ mm) are those presented in **W.2.2.1.5**, namely:

$f = 120$ N/mm^2 for maximum stress intensity;

$f = 26.76$ N/mm^2 for maximum compressive stress.

Following the procedure adopted in **W.2.2.1.5**, the value of the primary membrane circumferential stress at the highest point of the cross-section = 103.74 N/mm^2 and at the lowest point of the cross-section = 122.20 N/mm^2. The primary membrane stress intensities involving the longitudinal stress σ_z are $\sigma_\theta - \sigma_z$ and $\sigma_z + 0.5p$. From these the complete set of values can be obtained using f_1 to f_4. It is found that the maximum stress intensity values occur at the mid-span position and are as follows:

$\sigma_\theta - \sigma_z = 103.74 - 33.54 = 70.2$ N/mm^2;

$\sigma_z + 0.5p = 79.43 + 0.5 \times 0.2 = 79.53$ N/mm^2.

The maximum calculated compressive stress is f_1 and occurs when the vessel is full of contents but not pressurized, i.e.

$f_1 = -18.33$ N/mm^2

In all cases these values are less than the corresponding allowable values and therefore acceptable.

W.2.2.2.6 Tangential shearing stresses (see G.3.3.2.5)

When the shell is stiffened by the end of the vessel ($A \le r/2$) the tangential shearing stresses are given by Equations (G.3.3-10) and (G.3.3-11).

1) *Shear stresses in the shell*

$$q = \frac{K_3 W_1}{rt} \qquad \text{(G.3.3-10)}$$

where $K_3 = 0.485$ from Table G.3.3-2 for a shell stiffened by the end of the vessel, $b_1 < A \le r/2$, and $\theta = 150°$.

Thus:

$q = 0.485 \times 286\,855/(1\,296.75 \times 2.5)$

$q = 42.91$ N/mm^2

The allowable shearing stresses for $t = 2.5$ mm are given in **W.2.2.1.6**, namely the smaller of $0.8f$ (= 96 N/mm^2) and $0.06Et/r$ (= 23.10 N/mm^2). In this case the smallest value is $0.06Et/r$ and since $q > 0.06Et/r$ the design should be changed. Since this is an example without rings at the saddles the only viable alternative is to increase the shell thickness. Try $t = 4$ mm, in which case the saddle reaction is increased to $W_1 = 290\,249$ N and the mean radius $r = 1\,295.5 + 2 = 1\,297.5$ mm. Thus in Equation (G.3.3-10):

$q = 0.485 \times 290\,249/(1\,297.5 \times 4)$

$q = 27.12$ N/mm^2

The allowable tangential shear stress $0.06Et/r$ is now

$0.06 \times 1\,997 \times 10^3 \times 4/1\,297.5 = 36.93$ N/mm^2

i.e. $q < 0.06Et/r$ and the shell is acceptable with t = 4 mm.

2) *Shear stress in the vessel end*

$$q_e = \frac{K_4 W_1}{r t_e} \quad \text{(G.3.3-11)}$$

where

K_4 = 0.295 from Table G.3.3-2 for $b_1 < A \leq r/2$ and θ = 150°;

t_e = 5.5 mm (see **W.2.2.1.1**);

W = 290 249 N (t = 4 mm).

Thus:

q_e = 0.295 × 290 249/(1 297.5 × 5.5);

q_e = 12.00 N/mm^2.

The allowable tangential shearing stress for the vessel ends is $1.25f - f_{n(d)}$, as stated in Table G.3.3-2. The value of $f_{n(d)}$ can be found from Figure 3.5-2 using appropriate values of h_e/D and e/D to give p/f and hence:

$f_{n(d)} = p/(p/f)$ (see **3.5.2** and Note 2 to Table G.3.3-2);

e/D = 5.5/(2 591 + 11) = 0.00211;

$$h_e = \min\left[h; D^2/[4(r+e)]; \sqrt{D(r+e)/2}\right];$$

h = 499 + 5.5/2 = 501.8 mm;

$D^2/[4(r + e)])$ = (2 591 + 11)^2/[4(2 591 + 5.5)] = 651.9 mm;

$$\sqrt{D(r+e)/2} = \sqrt{(2\ 591 + 11)(254 + 5.5)/2} = 581.10 \text{ mm}$$

$\therefore$ h_e/D = 501.8(2 591 + 11) = 0.193.

Entering the values for e/D and h_e/D in Figure 3.5-2

p/f = 0.0018

Thus:

$f_{n(d)} = p/(p/f)$ = 0.2/0.0018 = 111.11 N/mm^2.

The allowable tangential shearing stress in the vessel end is thus:

1.25 × 120 − 111.11 = 38.89 N/mm^2.

The calculated tangential shearing stress in the end is 12.00 N/mm^2 which is less than the allowable. The end is thus acceptable.

W.2.2.2.7 Circumferential stresses (see G.3.3.2.6.2)

1) *Stress at the lowest point of the cross-section*

As in **W.2.2.1.7** for saddles welded to the vessel, **G.3.3.2.6.2** should be used in conjunction with Table G.3.3-4 to determine the value of K_5. An extended saddle plate is to be used, and if the extended saddle plate is welded to the vessel then the combined shell and saddle plate thickness may be used when deriving the stress in the plane of the saddle:

$$f_5 = \frac{-K_5 W_1}{(t + t_1) b_2} \quad \text{(G.3.3-20)}$$

where

K_5 = 0.0673 for θ = 150°;

$(t + t_1)$ = 4 + 4 = 8 mm;

$b_2 = b_1 + 10t$ = 200 + 10 × 4 = 240 mm;

W_1 = 290 249 N.

Thus:

f_5 = –0.0673 × 290 249/(8 × 240);

f_5 = –10.17 N/mm^2.

In the region of the *edge of the saddle plate* where the thickness is t:

$$f_5 = \frac{-K_5 W_1}{t b_2}$$

where

K_5 = 0.0651 for θ = 162° by interpolation;

t = 4 mm;

$b_2 = b_1 + 10t$ = 200 + 10 × 4 = 240 mm.

Thus:

f_5 = –0.0651 × 290 249/(4 × 240);

f_5 = –19.69 N/mm^2.

In **G.3.3.2.6** f_5 is limited to f (= 120 N/mm^2), therefore these stresses are acceptable.

2) *Stress at the horn of the saddles*

Since L/r = 7 055/1 297.5 = 5.437 and the extended saddle plate is welded to the vessel use Equation (G.3.3-22)

$$f_6 = \frac{-W_1}{4(t+t_1)b_2} - \frac{12 K_6 W_1 r}{L(t+t_1)^2} \qquad \text{(G.3.3-22)}$$

Using the same approach as in **W.2.2.1.7** the stresses at the edge of the saddle (θ = 150°) and the edge of the extended saddle (θ = 162°) should be checked.

a) *At the edge of the saddle*

thickness = $t_1 + t$ = 8 mm;

b_2 = 200 + 10 × 4 = 240 mm;

θ = 150°;

K_6 = 0.0079 from Table G.3.3-3 for A/r < 0.5;

W_1 = 290 249 N.

Thus:

$$f_6 = \frac{290\ 249}{4 \times 8 \times 240} - \frac{12 \times 0.0079 \times 290\ 249 \times 1\ 297.5}{7\ 055 \times 8^2}$$

f_6 = –116.86 N/mm^2.

b) *At the edge of the saddle plate*

Use Equation (G.3.3-19)

$$f_6 = \frac{-W_1}{4tb_2)b_2} - \frac{12K_6W_1r}{Lt^2} \qquad \text{(G.3.3-19)}$$

t = 4 mm;

b_2 = 200 + 10 × 4 = 240 mm;

θ = 162°;

K_6 = 0.0063 by interpolation from Table G.3.3-3 for A/r < 0.5;

W_1 = 290 249 N.

Thus:

$$f_6 = \frac{-290\ 249}{4 \times 4 \times 240} - \frac{12 \times 0.0063 \times 290\ 249 \times 1\ 297.5}{7\ 055 \times 4^2}$$

f_6 = –327.81 N/mm^2.

The allowable value for f_6 is 1.25f (= 150 N/mm^2). The stress at the edge of the saddle plate is therefore not acceptable. It is therefore necessary to increase the shell thickness. Try t = 6.5 mm. Thus r = 1 295.5 + 3.25 = 1 298.75 mm, W_1 = 296 006 N, K_6 = 0.0063 by interpolation from Table G.3.3-2, b_2 = 200 + 10 × 6.5 = 265 mm. The stress of the saddle plate is thus:

$$f_6 = \frac{-296\ 006}{4 \times 6.5 \times 265} - \frac{12 \times 0.0063 \times 296\ 006 \times 1\ 298.75}{7\ 055 \times 6.5^2}$$

f_6 = –140.47 N/mm^2;

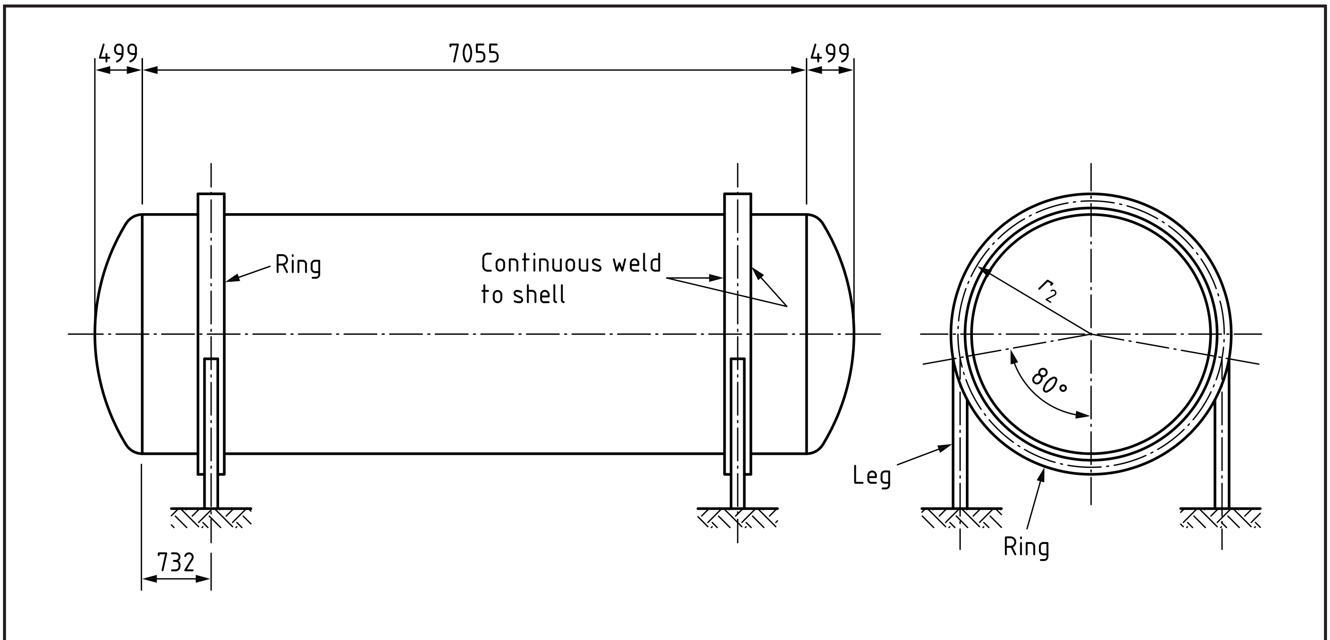
which is satisfactory. The thickness should therefore be increased to 6.5 mm.

W.2.2.3 Example 3. Ring and leg support

W.2.2.3.1 Design data

The design data used in Example 1 are relevant again for this example. The layout is as shown in Figure W.2-4. The subtended angle of the leg ϕ_1 = 80°.

Figure W.2-4 **Vessel on ring and leg support**



As in Example 1, assume the minimum shell thicknesses of the shell and ends to be 2.5 mm and 5.5 mm respectively. That is, the mean radius r = 1 296.75 mm. The self weight of the vessel and contents remain the same as in Example 1, i.e. W_1 = 286 855 N for t = 2.5 mm.

The values of the longitudinal bending moments (see **G.3.3.2.2**) M_3 and M_4 are those given in **W.2.2.1.2**, namely:

$M_3 = +278.97 \times 10^6$ N·mm;

$M_4 = -11.39 \times 10^6$ N·mm.

W.2.2.3.2 Longitudinal stresses at mid-span (see G.3.3.2.3)

The mid-span stresses f_1 and f_2 at the highest and lowest points of the cross-section respectively, are the same as in **W.2.2.1.3**, namely:

f_1 = 35.36 N/mm^2 when full and at pressure;

f_1 = −16.51 N/mm^2 when full with zero top pressure;

f_2 = 77.60 N/mm^2 when full and at pressure.

W.2.2.3.3 Longitudinal stresses at the supports (see G.3.3.2.4)

1) *The stress at the highest point of the cross-section*

$$f_3 = \frac{p_m r}{2t} - \frac{M_4}{K_1 \pi r^2 t} \tag{G.3.3-7}$$

where K_1 = 1 from Table G.3.3-1 for θ = 150° for a ring stiffened shell.

Thus:

$f_3 = (0.2 + 9\ 810 \times 1.4 \times 1\ 295.5 \times 10^{-9}) \times 1\ 296.75/(2 \times 2.5) + 11.39 \times 10^6/(1 \times \pi \times 1\ 296.75^2 \times 2.5)$;

f_3 = 57.35 N/mm^2.

2) *The stress at the lowest point of the cross-section*

$$f_4 = \frac{p_m r}{2t} + \frac{M_4}{K_2 \pi r^2 t} \tag{G.3.3-8}$$

where K_2 = 1 from Table G.3.3-1 for θ = 150° for a ring stiffened shell.

Thus:

$f_4 = (0.2 + 9\ 810 \times 1.4 \times 1\ 295.5 \times 10^{-9}) \times 1\ 296.75/(2 \times 2.5) - 11.39 \times 10^6/(1 \times \pi \times 1\ 296.75^2 \times 2.5)$;

f_4 = 55.62 N/mm^2.

W.2.2.3.4 Allowable direct stresses

The allowable direct stresses, in accordance with **A.3.4.2.1** and **A.3.5** (for t = 2.5 mm) are those presented in **W.2.2.1.5**, namely:

f = 120 N/mm^2 for maximum stress intensity;

f = 26.76 N/mm^2 for maximum compressive stress.

Following the procedure adopted in **W.2.2.1.5** the value of the primary membrane circumferential stress at the highest point of the cross-section = 103.74 N/mm^2 and at the lowest point of the cross-section = 122.20 N/mm^2. The primary membrane stress intensities involving the longitudinal stress σ_z are $\sigma_\theta - \sigma_z$ and $\sigma_z + 0.5p$. From these the complete set of values can be obtained using f_1 to f_4. It is found that the maximum stress intensity values occur at the mid-span position and are as follows:

$\sigma_\theta - \sigma_z$ = 103.74 − 35.36 = 68.38 N/mm^2;

$\sigma_z - 0.5p = 77.60 + 0.5 \times 0.2 = 77.70$ N/mm^2.

The maximum calculated compressive stress is f_1 and occurs when the vessel is full of contents but not pressurized, i.e. $f_1 = -16.51$ N/mm^2. In all cases these values are less than the corresponding allowable values and are therefore acceptable.

W.2.2.3.5 Tangential shearing stresses (see G.3.3.2.5)

The shearing stress in the shell adjacent to the ring support is given by:

$$q = \frac{0.319}{rt} W_1 [(L - 2A)/(L + 4b/3)];$$

$$q = \frac{0.319 \times 286\ 855}{1\ 296.75 \times 2.5} [(7\ 055 - 2 \times 732)/(7\ 055 + 4 \times 499/3)];$$

$q = 20.45$ N/mm^2.

The allowable tangential shearing stress from Table G.3.3-2 is the smaller of $0.8f$ and $0.06Et/r$, i.e. 96.00 N/mm^2 and 23.10 N/mm^2 (see **W.2.2.1.6**), i.e. $q < 0.06Et/r$ and is thus acceptable with $t = 2.5$ mm.

W.2.2.3.6 Circumferential stresses at the ring support (see G.3.3.3)

1) *Ring data*

The proposed ring section is a 203 mm × 89 mm mild steel channel of BS 4-1:2005 rolled "toes out" (see Figure W.2-5).

For the channel using:

Area = 3 794 mm^2;

$I_{yy} = 2\ 644\ 000$ mm^4;

$c_y = 26.5$ mm.

Total area of channel and shell:

$$= 3\ 794 + \left(203 + \sqrt{rt}\right)t = 3\ 794 + \left(203 + \sqrt{1\ 296.75 \times 2.5}\right)2.5;$$

= 3 794 + 650 = 4 444 mm^2;

$c = [3\ 794 \times (26.5 + 2.5) + (650 \times 1.25)]/4\ 444 = 24.94$ mm;

$d = 89 + 2.5 - 24.94 = 66.56$ mm.

I of the combined section = $2\ 644\ 000 + (259.9 \times 2.5^3/12) + 3\ 794(26.50 + 2.50 - 24.94)^2 + 259.9 \times 2.5(24.94 - 1.25)^2 = 3\ 071\ 527$ mm^4.

2) *Maximum circumferential stress*

$$f_{10} = \frac{K_{10} W_1 r_2}{Z} + \frac{K_{11} W_1}{a} \qquad \text{(G.3.3-38)}$$

where

K_{10}	= 0.017	from Table G.3.3-11 for $\phi_1 = 80°$;
K_{11}	= 0.29	from Table G.3.3-11 for $\phi_1 = 80°$;
r_2	= the radius through the centroid of the section = 1 295.5 + c = 1 295.5 + 24.94 = 1 320.44 mm;	
W_1	= 286 855 N;	
Z	= least section modulus = 3 071 527/66.56 = 46 147 mm^3;	
a	= effective area = 4 444 mm^2.	

Thus:

$$f_{10} = \frac{0.017 \times 286\ 855 \times 1\ 320.44}{46\ 147} + \frac{0.29 \times 286\ 855}{4\ 444};$$

f_{10} = 158.26 N/mm^2.

3) *Allowable stresses*

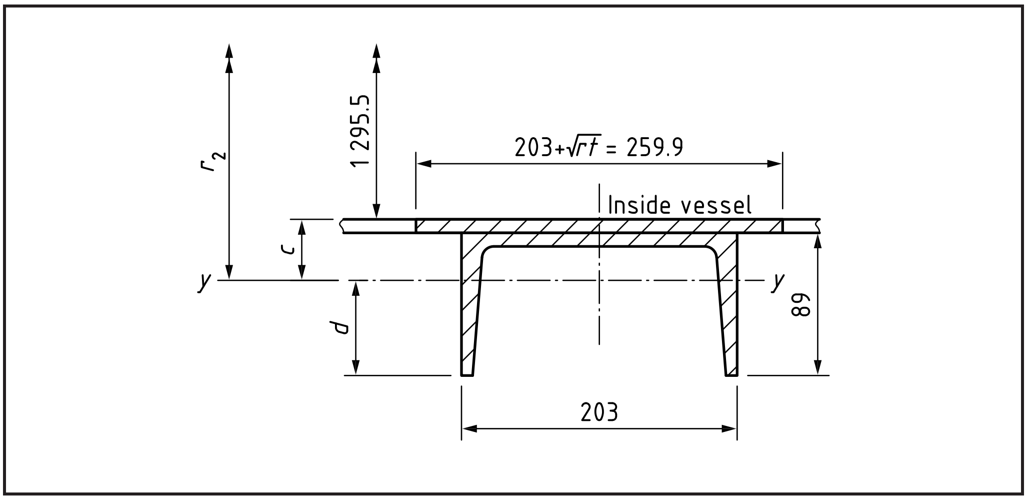
In the case of category 1 and 2 vessels the rings are in general of the same material as the vessel and constructed to the same category as the vessel with the f value obtained from **2.3.1.1**. In the case of the rings associated with category 3 vessels, it is considered acceptable to use the corresponding category 1 and 2 vessel f values as given in **2.3.1.1** providing the radial weld seams joining the segments of the rings are located in the region of low bending stress in the rings. Justification for this procedure may be found in **3.4.2.2**. The distribution of the bending moment in a typical ring support is shown in PD 6497.

Where the ring is made of a different material to that of the vessel, the f value for the weaker material should be used. In this case the value f = 163 N/mm^2 at 50 °C for BS 1501-320-S31 (see Table K.1-4) is used.

It is noted that the maximum stress < f = 163 N/mm^2 and thus the ring girder is considered acceptable.

For mild steel ring girders used on category 3 vessels and not subject to above ambient temperatures, it is acceptable to use the allowable stresses from **2.3.1.1**. In this case the ring should be designed as a separate structure without the benefit of the length of shell $\sqrt{rt}$.

Figure W.2-5 **Channel and shell as ring girder**



W.2.2.4 Conclusions

A summary of the results obtained by the above procedure for all three examples is presented in Table W.2-3. Stresses that are outside the recommended limit are underlined and the recalculated value corresponding to increased shell thickness/addition of stiffener are given in an adjacent column. When a stress has been found satisfactory it is not generally recalculated for the modified design; for such cases the letter “S” appears in the relevant column, indicating that the stress is satisfactory. Acceptable designs are indicated by means of the symbol ↑.

For this particular vessel, where external stiffening is obviously preferable, Table W.2-3 provides a basis for choosing the most economical design.

Table W.2-3 **Summary of stresses (N/mm²)**

Stress	Example 1. Saddle supports away from the ends						Example 2. Saddle supports near the ends			Example 3. Ring leg
	Plain vessel			Internal ring	External ring adjacent to saddle					
	a) t = 2.5 mm	b) t = 5 mm	c) t = 8 mm	d) t = 2.5 mm	e) t = 5 mm	f) t = 6.5 mm	g) t = 2.5 mm	h) t = 4 mm	j) t = 6.5 mm	k) t = 2.5 mm
f_1 (with top pressure)	35.36	S (17.52)	S	35.36	17.52	S	33.54	S	S	35.36
f_1 (zero top pressure)	–16.51	S (–8.44)	S	–16.51	–8.44	S	–18.33	S	S	–16.51
f_2	77.60	S (39.02)	S	77.60	39.02	S	79.43	S	S	77.60
f_3	61.84	S (30.98)	S	57.35	28.71	S	56.86	S	S	57.35
f_4	53.39	S (26.71)	S	55.62	27.83	S	56.10	S	S	55.62
Q	<u>51.20</u>	26.08	S	20.45	26.08	S	<u>42.91</u>	27.12	S	20.45
Max. f_5		–15.23	S	N/A	–15.75	S		–19.69	S	N/A
f_6 (θ = 150°)		–99.64	S	N/A				–116.86	S	N/A
f_6 (θ = 162°)		–<u>283.24</u>	–123.33	N/A	–<u>221.24</u>	–140.47		–<u>327.81</u>	–140.47	N/A
f_7	N/A	N/A	N/A	–136.47	–141.41	S	N/A	N/A	N/A	N/A
f_8	N/A	N/A	N/A	95.29	142.96	S	N/A	N/A	N/A	N/A
			↑	↑		↑			↑	↑

NOTE 1 Stresses shown underlined exceed the allowable.

NOTE 2 S indicates that the stress is satisfactory.

NOTE 3 ↑ indicates acceptable design.

W.3 Local loads

W.3.1 General

W.3.1.1 Introduction

Subclause **W.3** provides worked examples illustrating the use of Annex G to check the effects of local loads on cylinders and spheres. Five different examples are considered.

a) A circumferential moment due to a radial load applied to a square located at the mid-length of a cylinder.

b) The stresses and rotation resulting from a longitudinal moment on a branch of a cylinder.

c) The use of working forms to sum the maximum stresses due to local loads on a cylindrical shell.

d) The stresses and deflection due to a radial load on a branch of a spherical shell.

e) The stresses and deflection due to a moment on a branch of a spherical shell.

W.3.1.2 Notation

For the purposes of **W.3** the following symbols apply, based upon those given in **G.2**.

C_x	is half the longitudinal length of a rectangular load area;
C_z	is axial length of a load area subject to external longitudinal moment;
C_ϕ	is half the circumferential length of a rectangular load area;
f_p	is the circumferential or longitudinal pressure stress;
f_x	is the resultant longitudinal stress;
f_ϕ	is the resultant circumferential stress;
f_1, f_{1m}, f_2, f_{2m}	are stress intensities;
i	is a rotation;
i_b	is the slope of a branch;
L	is the length of the cylindrical portion of the shell;
M	is an external moment on a branch;
M_x, $M_{x1} \ldots M_{x3}$	are longitudinal bending moments per unit circumference;
M_ϕ, $M_{\phi 1} \ldots M_{\phi 3}$	are circumferential bending moments per unit circumference;
N_x, $N_{x1} \ldots N_{x3}$	are longitudinal membrane forces per unit circumference;
N_ϕ, $N_{\phi 1} \ldots N_{\phi 3}$	are circumferential membrane forces per unit circumference;
r	is the mean radius of the shell;
r_o	is the mean radius of a branch;
s	is the position in the shell at which forces or deflections are required;
t	is the analysis thickness of a shell;
u	defines the area over which a load is distributed;
W	is the external load distributed over the loading area;
x	is the longitudinal distance from centre of load area to point of interest;
δ	is the deflection of the shell at load point;
θ	is the polar co-ordinate of a point on a spherical vessel.

W.3.2 Worked examples

W.3.2.1 Example 1. Circumferential moment due to a radial load applied to a square at mid-length of cylinder

The cylindrical vessel is 2.5 m diameter × 6 m long × 12 mm thick. A radial load W is applied to an area 300 mm square at the mid-length of the shell. The circumferential moment at a position 600 mm from the centre of the loaded area measured along the axis of the vessel, is calculated as follows:

From Figure G.2.2-2, $C_\phi = C_x = 150$ mm:

$r = 1\,250$ mm; $r/t = 104$;

$C_x/r = 0.12$; $2C_x/L = 0.05$; $x/C_x = 4$.

For a line load, interpolating in Figure G.2.2-14:

$M_\phi/W = 0.054$ at $x/C_x = 4$.

From Figure G.2.2-6 at the ends of a line load when

$C_\phi/C_x = 0$, $0.64(r/t)(C_x/r)^2 = 90$, and $2C_x/L = 0.05$, $M_\phi/W = 0.153$

and when $C_\phi/C_x = 1.0$, $M_\phi/W = 0.072$.

Therefore when the load is distributed over an area 300 mm square:

$$M_\phi/W \text{ at } x = 0.054 \times \frac{0.072}{0.153} = 0.025.$$

Therefore the circumferential moment at a position 600 mm from the centre of the loaded area, measured along the axis of the vessel, is $0.025W$.

W.3.2.2 Example 2. Stress and rotation due to a longitudinal moment applied to a branch of a cylinder

W.3.2.2.1 General

The cylindrical vessel in this example is 2.5 m diameter × 4 m long × 12 mm thick; $E = 1.86 \times 10^5$ N/mm^2. The maximum stress due to a longitudinal moment of 1.13×10^6 N·mm applied to a branch 350 mm diameter at the mid-length, and the slope of the branch is calculated as follows.

W.3.2.2.2 Calculation of maximum stress

From **G.2.3.3**,

$$C_\phi = \frac{C_z}{2} = 0.85 \times 175 \approx 150\text{mm}$$

$$W = \pm\frac{1.5M}{C_z} = \pm\frac{1.5 \times 1.13 \times 10^6}{2 \times 150} = \pm\ 5\,650\text{ N}$$

W acts on an area $2C_\phi \times 2C_x$, where $C_x = \dfrac{C_z}{6} = 50$ mm

For this area:

$$\frac{C_\phi}{C_x} = \frac{6}{2} = 3; \frac{C_x}{r} = \frac{50}{1\,250} = 0.04; \frac{2C_x}{L} = \frac{2 \times 50}{4\,000} = 0.025$$

From Figure G.2.2-5,

$$64\frac{r}{t}\left(\frac{C_x}{r}\right)^2 \approx 10$$

The direct effect of each load W is found by interpolating for C_ϕ/C_x = 3.0 in the charts of Figure G.2.2-6, Figure G.2.2-7, Figure G.2.2-8 and Figure G.2.2-9 for $2C_x/L$ = 0.025 which gives:

$M_{\phi1}/W$ = 0.09; M_{x1}/W = 0.076; $N_{\phi1}t/W$ = –0.155;

$N_{x1}t/W$ = –0.14.

The effect of one load at the outer edge of the other is found by interpolating for $64(r/t)\ (C_x/r)^2$ = 10, x/C_x = 5.0 and $2C_x/L$ = 0.025 in the charts of Figure G.2.2-14, Figure G.2.2-15, Figure G.2.2-16 and Figure G.2.2-17 for a radial line load, and multiplying the results by a correction factor for the circumferential width of the load as in **G.2.2.3.3**.

The values interpolated from Figure G.2.2-14, Figure G.2.2-15, Figure G.2.2-16 and Figure G.2.2-17 denoted by subscript 3, are:

$M_{\phi3}/W$ = 0.065; M_{x3}/W = 0.012; $N_{\phi3}t/W$ = +0.025;

$N_{x3}t/W$ = -0.085.

Quantity	Values for C_ϕ/C_x = 0	Figure	Correction factor = $\frac{\text{value for } C_\phi/C_x = 3}{\text{value for } C_\phi/C_x = 0}$
$\frac{M_{\phi3}}{W}$	0.255	Figure G.2.2-6	$\frac{0.09}{0.255} = 0.353$
$\frac{M_{x3}}{W}$	0.16	Figure G.2.2-7	$\frac{0.076}{0.16} = 0.475$
$\frac{N_{\phi3}t}{W}$	–0.18	Figure G.2.2-8	$\frac{-0.155}{-0.18} = 0.861$
$\frac{N_{x3}t}{W}$	–0.17	Figure G.2.2-9	$\frac{-0.14}{-0.17} = 0.824$

Hence:

$$\frac{M_{\phi2}}{W} = +0.065 \times 0.353 = 0.023;$$

$$\frac{N_{\phi2}t}{W} = +0.025 \times 0.861 = 0.0215;$$

$$\frac{M_{x2}}{W} = +0.012 \times 0.475 = 0.005;$$

$$\frac{N_{x2}t}{W} = -0.085 \times 0.824 = -0.070;$$

$$M_\phi = W\left(\frac{M_{\phi1}}{W} - \frac{M_{\phi2}}{W}\right) = 5\ 650(0.09 - 0.023) = 5\ 650 \times 0.067 = 379\ \text{N} \cdot \text{mm/mm};$$

$$M_x = W\left(\frac{M_{x1}}{W} - \frac{M_{x2}}{W}\right) = 5\,650(0.076 - 0.0057) = 5\,650 \times 0.0703 = 396\ \mathrm{N \cdot mm/mm};$$

$$N_\phi = \frac{W}{t}\left(\frac{N_{\phi1}t}{W} - \frac{N_{\phi2}t}{W}\right) = \frac{5\,650}{12}(-0.155 - 0.0215) = 470 \times (-0.1765) = -83\ \mathrm{N/mm};$$

$$N_x = \frac{W}{t}\left(\frac{N_{x1}t}{W} - \frac{N_{x2}t}{W}\right) = \frac{5\,650}{12}(-0.14 + 0.07) = 470 \times (-0.07) = -33\ \mathrm{N/mm}.$$

Maximum circumferential and longitudinal stresses can then be determined in accordance with **G.2.2.2.3**. These will appear as both compressive and tensile stresses depending on which edge of the loaded area is being considered.

$$\text{Circumferential stress} = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2}$$

= –6.92 ± 15.8 (W positive)

∴ Maximum circumferential stress = -22.72 N/mm^2

$$\text{Longitudinal stress} = \frac{N_x}{t} \pm \frac{6M_x}{t^2}$$

= –2.75 ± 16.5 (W positive)

∴ Maximum longitudinal stress = -19.25 N/mm^2

W.3.2.2.3 Slope due to moment

From **W.3.2.2.2**, C_ϕ/C_x = 3, and from Figure G.2.2-20b) the half side of the equivalent square $C_1 = 2.8C_x$ = 140 mm.

In Figure G.2.2-18b):

C_1/r = 0.112;

L/r = 3.2;

r/t = 100;

whence $\delta Er/W$ = 17 000

$$\text{Therefore } \delta_1 = \frac{1.7 \times 10^4 \times 5\,650}{1.86 \times 10^5 \times 1\,250} = 0.414$$

and from **G.2.3.5**, the slope $i = \frac{3\delta_1}{C_z}$

$$= \frac{3 \times 0.414}{300}$$

= 0.00414 radians

W.3.2.3 Example 3. Summation of maximum stresses due to local loads on a cylindrical shell

In this example the recommendations of **G.2.3.6** and the suggested working form **G.3**, are used to summate the maximum stresses due to local loads on a cylindrical shell. In the first working form, these stresses are found at the nozzle outer diameter and in the second, at the edge of a reinforcing plate.

Suggested working form G3 Example 3	**Load Case:** D5 – N – 1 at nozzle o.d.		Nozzle o.d. ~~/ pad o.d. / loaded area dimensions~~[a]	219 mm
Clause **G.2.3.6** Summation of maximum stresses due to local loads on a cylindrical shell	Radial load F_R	4 410 N	~~Shell thickness /~~ Shell + pad thickness[a]	23 mm
	Shear force F_C	6 600 N		
	Shear force F_L	6 600 N	Shell i.d.	2494 mm
	Torsional moment M_T	8 900 000 N.mm	Design pressure	1.1 N/mm^2
Nozzle branch with reinforcing plate attached with full penetration weld, see **G.3.1.5**	Circumferential moment M_C	3 630 000 N.mm	Design stress (f)	151.6 N/mm^2
	Longitudinal moment M_L	3 630 000 N.mm	Yield stress	227.4 N/mm^2

	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Circumferential stresses								
	Membrane component (N_ϕ/t) due to:								
1	Radial load	1.3	1.3	1.3	1.3	1.3	1.3	1.3	1.3
2	Circumferential moment	–1.2	–1.2	–1.2	–1.2	1.2	1.2	1.2	1.2
3	Longitudinal moment	–7.8	–7.8	7.8	7.8	7.8	7.8	–7.8	–7.8
4	Sub-total due to local loads	–7.7	–7.7	7.9	7.9	10.3	10.3	–5.3	–5.3
5	Pressure (f_P from **G.2.3.6.2**)	141.5	141.5	141.5	141.5	141.5	141.5	141.5	141.5
6	Sub-total ($f_{\phi m}$)	133.8	133.8	149.4	149.4	151.8	151.8	136.2	136.2
	Bending component ($6M_\phi/t^2$) due to:								
7	Radial load	–7.9	7.9	–7.9	7.9	–7.9	7.9	–7.9	7.9
8	Circumferential moment	53.8	–53.8	53.8	–53.8	–53.8	53.8	–53.8	53.8
9	Longitudinal moment	36.7	–36.7	–36.7	36.7	–36.7	36.7	36.7	–36.7
10	Sub-total ($f_{\phi b}$)	82.6	–82.6	9.2	–9.2	–98.4	98.4	–25.0	25.0
11	Total circumferential stress (f_ϕ)	216.4	51.2	158.6	140.2	53.4	250.2	111.2	161.2
	Longitudinal stresses								
	Membrane component (N_x/t) due to:								
12	Radial load	1.4	1.4	1.4	1.4	1.4	1.4	1.4	1.4
13	Circumferential moment	–2.5	–2.5	–2.5	–2.5	2.5	2.5	2.5	2.5
14	Longitudinal moment	–2.9	–2.9	2.9	2.9	2.9	2.9	–2.9	–2.9
15	Sub-total due to local loads	–4.0	–4.0	1.8	1.8	6.8	6.8	1.0	1.0
16	Pressure (f_P from **G.2.3.6.2**)	141.5	141.5	141.5	141.5	141.5	141.5	141.5	141.5
17	Sub-total (f_{xm})	137.5	137.5	143.3	143.3	148.3	148.3	142.5	142.5
	Bending component ($6M_x/t^2$) due to:								
18	Radial load	–5.4	5.4	–5.4	5.4	–5.4	5.4	–5.4	5.4
19	Circumferential moment	25.2	–25.2	25.2	–25.2	–25.2	25.2	–25.2	25.2
20	Longitudinal moment	43.1	–43.1	–43.1	43.1	–43.1	43.1	43.1	–43.1
21	Sub-total (f_{xb})	62.9	–62.9	–23.3	23.3	–73.7	73.7	12.5	–12.5
22	Total longitudinal stress (f_x)	200.4	74.6	120.0	166.6	74.6	222.0	155.0	130.0

[a] Delete as appropriate

Suggested working form G3: Example 3 at nozzle o.d. (*continued*)									
	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Shear stresses (from **G.2.3.6.3**) due to:								
23	Torsion moment	5.1	5.1	5.1	5.1	5.1	5.1	5.1	5.1
24	Circumferential shear force	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
25	Longitudinal shear force	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
26	Total shear stress (τ)	6.7	6.7	6.7	6.7	6.7	6.7	6.7	6.7
	Check of total stress intensity (membrane + bending) to **A.3.3.1** and **A.3.3.2**)								
27	$f_1 = \left[f_\phi + f_x + \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right] / 2$	218.8	74.6	158.6	166.6	74.6	250.2	155.0	161.2
28	$f_2 = \left[f_\phi + f_x - \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right] / 2$	198.0	51.2	120.0	140.2	53.4	222.0	111.2	130.0
29	$f_2 - f_1$	–20.8	–23.4	–38.6	–26.4	–21.2	–28.2	–43.8	–31.2
	Maximum total stress intensity = maximum absolute value in rows 27, 28 and 29 = 250.2 Allowable stress at nozzle = 2.25f = 341.1 ~~, or at edge of a compensation pad, attachment or support = 2f = 303.2~~ (acceptable)								
	Check of buckling stress to **A.3.3.3**								
30	Row 4 + row 10 if row 4 is compressive (–)	74.9	–90.3					–30.3	19.7
31	Row 15 + row 21 if row 15 is compressive	63.9	–61.9						
	Maximum compressive stress in rows 30 and 31 = –90.3 Allowable compressive stress = –0.9 x yield stress = –204.7 (acceptable)								
	~~At edge of compensation pad, attachment or support, check of membrane stress intensity to **A.3.3.1**~~[a]	Not applicable							
32	$f_{1m} = \left[f_{\phi m} + f_{xm} + \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right] / 2$								
33	$f_{2m} = \left[f_{\phi m} + f_{xm} - \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right] / 2$								
34	$f_{2m} - f_{1m}$								
	Maximum membrane stress intensity = maximum absolute value in rows 32, 33 and 34 = 0.0 Allowable membrane stress = 1.2f = 181.9								
From **G.2.3.6.2**a) f_p = 141.5 circumferentially and longitudinally. From **G.2.3.6.3** shear stress due to M_T = 5.1, due to F_C = 0.8 and due to F_L = 0.8.									
[a] Delete as appropriate									

Suggested working form G3 Example 3	**Load Case:** D5 – N – 1 at pad edge		~~Nozzle o.d. /~~ pad o.d. ~~/ loaded area dimensions~~[a]	400 mm
Clause **G.2.3.6** Summation of maximum stresses due to local loads on a cylindrical shell	Radial load F_R	4 410 N	Shell thickness / ~~Shell+pad thickness~~[a]	13 mm
	Shear force F_C	6 600 N		
	Shear force F_L	6 600 N	Shell i.d.	2494 mm
	Torsional moment M_T	8 900 000 N.mm	Design pressure	1.1 N/mm^2
	Circumferential moment M_C	3 630 000 N.mm	Design stress (f)	151.6 N/mm^2
	Longitudinal moment M_L	3 630 000 N.mm	Yield stress	227.4 N/mm^2

	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Circumferential stresses								
	Membrane component (N_ϕ/t) due to:								
1	Radial load	2.1	2.1	2.1	2.1	2.1	2.1	2.1	2.1
2	Circumferential moment	–4.1	–4.1	–4.1	–4.1	4.1	4.1	4.1	4.1
3	Longitudinal moment	–15.7	–15.7	15.7	15.7	15.7	15.7	–15.7	–15.7
4	Sub-total due to local loads	–17.7	–17.7	13.7	13.7	21.9	21.9	–9.5	–9.5
5	Pressure (f_P from **G.2.3.6.2**)	106.1	106.1	106.1	106.1	106.1	106.1	106.1	106.1
6	Sub-total ($f_{\phi m}$)	88.4	88.4	119.8	119.8	128.0	128.0	96.6	96.6
	Bending component ($6M_\phi/t^2$) due to:								
7	Radial load	–11.5	11.5	–11.5	11.5	–11.5	11.5	–11.5	11.5
8	Circumferential moment	75.3	–75.3	75.3	–75.3	–75.3	75.3	–75.3	75.3
9	Longitudinal moment	35.3	–35.3	–35.3	35.3	–35.3	35.3	35.3	–35.3
10	Sub-total ($f_{\phi b}$)	99.1	–99.1	28.5	–28.5	–122.1	122.1	–51.5	51.5
11	Total circumferential stress (f_ϕ)	187.5	–10.7	148.3	91.3	5.9	250.1	45.1	148.1
	Longitudinal stresses								
	Membrane component (N_x/t) due to:								
12	Radial load	3.5	3.5	3.5	3.5	3.5	3.5	3.5	3.5
13	Circumferential moment	–8.6	–8.6	–8.6	–8.6	8.6	8.6	8.6	8.6
14	Longitudinal moment	–6.5	–6.5	6.5	6.5	6.5	6.5	–6.5	–6.5
15	Sub-total due to local loads	–11.6	–11.6	1.4	1.4	18.6	18.6	5.6	5.6
16	Pressure (f_P from **G.2.3.6.2**)	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0
17	Sub-total (f_{xm})	41.4	41.4	54.4	54.4	71.6	71.6	58.6	58.6
	Bending component ($6M_x/t^2$) due to:								
18	Radial load	–5.7	5.7	–5.7	5.7	–5.7	5.7	–5.7	5.7
19	Circumferential moment	27.7	–27.7	27.7	–27.7	–27.7	27.7	–27.7	27.7
20	Longitudinal moment	36.1	–36.1	–36.1	36.1	–36.1	36.1	36.1	–36.1
21	Sub-total (f_{xb})	58.1	–58.1	–14.4	14.4	–69.5	69.5	2.7	–2.7
22	Total longitudinal stress (f_x)	99.5	–16.7	40.3	68.5	2.1	141.1	61.3	55.9

[a] Delete as appropriate

Suggested working form G3: Example 3 at pad edge (*continued*)									
	Quadrant	**Q1**		**Q2**		**Q3**		**Q4**	
	Surface	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**	**Inside**	**Outside**
	Shear stresses (from **G.2.3.6.3**) due to:								
23	Torsion moment	2.7	2.7	2.7	2.7	2.7	2.7	2.7	2.7
24	Circumferential shear force	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
25	Longitudinal shear force	0.8	0.8	0.8	0.8	0.8	0.8	0.8	0.8
26	Total shear stress (τ)	4.3	4.3	4.3	4.3	4.3	4.3	4.3	4.3
	Check of total stress intensity (membrane + bending) to **A.3.3.1** and **A.3.3.2**)								
27	$f_1 = \left[f_\phi + f_x + \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right] / 2$	187.7	–10.7	148.3	91.3	5.9	250.1	61.3	148.1
28	$f_2 = \left[f_\phi + f_x - \sqrt{(f_\phi - f_x)^2 + 4\tau^2} \right] / 2$	99.3	–16.7	40.3	68.5	2.1	141.1	45.1	55.9
29	$f_2 - f_1$	–88.4	–6.0	–108.0	–22.8	–3.8	–109.0	–16.2	–92.2
	Maximum total stress intensity = maximum absolute value in rows 27, 28 and 29 = 250.1 Allowable stress ~~at nozzle = 2.25*f*, or~~ at edge of a compensation pad, attachment or support = 2*f* = 303.2 (acceptable)								
	Check of buckling stress to **A.3.3.3**								
30	Row 4 + row 10 if row 4 is compressive (–)	81.4	–116.8					–61.0	42.0
31	Row 15 + row 21 if row 15 is compressive	46.5	–69.7						
	Maximum compressive stress in rows 30 and 31 = –116.8 Allowable compressive stress = –0.9 x yield stress = –204.7 (acceptable)								
	At edge of compensation pad, attachment or support, check of membrane stress intensity to **A.3.3.1**[a]								
32	$f_{1m} = \left[f_{\phi m} + f_{xm} + \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right] / 2$	88.8	88.4	119.8	119.8	128.0	128.0	96.6	96.6
33	$f_{1m} = \left[f_{\phi m} + f_{xm} - \sqrt{(f_{\phi m} - f_{xm})^2 + 4\tau^2} \right] / 2$	41.0	41.4	54.4	54.4	71.6	71.6	58.6	58.6
34	$f_{2m} - f_{1m}$	–47.8	–47.0	–65.4	–65.4	–56.4	–56.4	–38.0	–38.0
	Maximum membrane stress intensity = maximum absolute value in rows 32, 33 and 34 = 128.0 Allowable membrane stress = 1.2*f* = 181.9 (acceptable)								
From **G.2.3.6.2**c) f_p = 106.1 circumferentially and f_p = 53.0 longitudinally. From **G.2.3.6.3** shear stress due to M_T = 2.7, due to F_C = 0.8 and due to F_L = 0.8.									
[a] Delete as appropriate									

W.3.2.4 Example 4. Stress and deflection due to a radial load on a branch of a spherical vessel

W.3.2.4.1 General

A load of 4 500 N is applied to a sphere 2 500 mm diameter and 12.5 mm thick through a branch 150 mm diameter. (E = 1.86 × 10^5 N/mm^2.) Stresses and deflection are calculated adjacent to the branch and 225 mm from the centre of the branch.

$$\frac{r}{t} = \frac{1\ 250}{12.5} = 100;\ \frac{r_o}{r} = \frac{75}{1\ 250} = 0.06$$

W.3.2.4.2 Next to the branch

$s = u$ = 1.09 (from Figure G.2.4-1)

Ordinate of curve for $u = s$ in Figure G.2.4-3 = –0.145.

$$\therefore \text{ Deflection} = -0.145 \times \frac{Wr}{Et^2}$$
$$= \frac{-0.145 \times 4\ 500 \times 1\ 250}{1.86 \times 10^5 \times (12.5)^2} = 0.0281 \text{ mm}$$

Ordinate of M_x curve for $u = s$ in Figure G.2.4-4 = +0.067

∴ Meridional M_x = +0.067 W = 301 N·mm/mm

Ordinate of M_ϕ curve for $u = s$ in Figure G.2.4-5 = +0.02

∴ Circumferential moment M_ϕ = +0.02 W = 90 N·mm/mm

Ordinate of N_x curve for $u = s$ in Figure G.2.4-6 = –0.11

$$\therefore \text{ Meridional membrane force } N_x = \frac{-0.11W}{t}$$

$$= \frac{-0.11 \times 4\ 500}{12.5} = 39.6 \text{ N/mm}$$

Ordinate of N_ϕ curve for $u = s$ in Figure G.2.4-7 = –0.034

$$\therefore N_\phi = \frac{-0.034\ W}{t} = \frac{-0.034 \times 4\ 500}{12.5}$$

= –12.2 N/mm^2

The resulting meridional stresses are given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} = \frac{-39.6}{12.5} \pm \frac{6 \times 301}{(12.5)^2}$$

∴ At the outside f_x = –3.17 – 11.5 = –14.67 N/mm^2 (compression)

At the inside f_x = –3.17 + 11.5 = +8.33 N/mm^2 (tension)

The resulting circumferential stresses are given by:

$$f_\phi = \frac{N_\phi}{t} \pm \frac{6M_\phi}{t^2} = \frac{-12.2}{12.5} \pm \frac{6 \times 90}{(12.5)^2}$$

∴ At the outside f_ϕ = –0.98 – 3.46 = –4.44 N/mm^2 (compression)

At the inside f_ϕ = –0.98 + 3.46 = +2.48 N/mm^2 (tension)

W.3.2.4.3 At a position 225 mm from the centre of the branch

u = 1.09 as in **W.3.2.4.2**; $\frac{x}{r} = \frac{225}{1\ 250} = 0.18$ from Figure G.2.4-1

s = 3.25

Interpolating between the curves in Figure G.2.4-3 at u = 1.09 and s = 3.25 gives:

$$\frac{\delta E t^2}{W r} = -0.030$$

$$\therefore \text{ Deflection } \delta = -0.030 \times \frac{W_r}{Et^2} = \frac{-0.030 \times 4\,500 \times 1\,250}{1.86 \times 10^5 \times (12.5)^2} = -0.0058 \text{ mm}$$

Interpolating similarly in Figure G.2.4-4, Figure G.2.4-5, Figure G.2.4-6 and Figure G.2.4-7 gives:

$$\frac{M_x}{W} = -0.008; \frac{M_\phi}{W} = +0.0007;$$

$$\frac{N_x t}{W} = -0.054; \frac{N_\phi t}{W} = +0.018$$

Whence the:

meridional moment M_x = –36.0 N·mm/mm;

circumferential moment M_ϕ = +3.2N·mm/mm;

meridional membrane force N_x = –19.4 N/mm;

circumferential membrane force N_ϕ = +6.5 N/mm.

The resulting meridional stresses are:

at the outside $f_x = \frac{-19.4}{12.5} + \frac{6 \times 36.0}{(12.5)^2} = -1.55 + 1.38 = -0.17 \text{ N/mm}^2$;

at the inside $f_x = -1.55 - 1.38 = -2.93 \text{ N/mm}^2$.

The resulting circumferential stresses are:

at the outside $f_\phi = \frac{+6.5}{12.5} - \frac{6 \times 3.2}{(12.5)^2} = +0.52 - 0.12 = +0.40 \text{ N/mm}^2$;

at the inside f_ϕ = +0.52 + 0.12 = +0.64 N/mm^2.

Hence the deflection and stresses due to the load are negligible at 225 mm from the centre of the branch, which illustrates the local nature of the stresses.

W.3.2.5 Example 5. Stress and deflection due to a moment on a branch of a spherical vessel

A moment of 1.13 × 10^5 N·mm is applied to the branch detailed in **W.3.2.4**. In this example the maximum deflection, the maximum stresses next to the branch, and the rotation of the branch due to this moment are calculated.

As before E = 1.86 × 10^5 N/mm^2, $\frac{r}{t}$ = 100 , $\frac{r_o}{r}$ = 0.06 , and, next to the branch, $s = u$ = 1.09

(from Figure G.2.4-1).

The maximum stresses and deflection are at θ = 0°;

$\therefore$ cos θ = 1

From Figure G.2.4-9

$$\delta = -0.17 \times \frac{M \cos \theta \sqrt{\frac{r}{t}}}{Et^2} = \frac{-0.17 \times 1.13 \times 10^5 \times 1 \times 10}{1.86 \times 10^5 \times (12.5)^2}$$

$\therefore$ Maximum deflection = –0.0066 mm

The deflection at θ = 180°, on the opposite side of the branch, will be +0.0066 mm. From Figure G.2.4-10, Figure G.2.4-11, Figure G.2.4-12 and Figure G.2.4-13.

Meridional moment

$$M_x = 0.182 \times \frac{\mathrm{M} \cos \theta}{\sqrt{rt}}$$

$$= \frac{0.182 \times 1.13 \times 10^5}{\sqrt{1\ 250 \times 12.5}}$$

= 164.5 N·mm/mm

Circumferential moment M_ϕ

$$= 0.055 \times \frac{\mathrm{M} \cos \theta}{\sqrt{rt}}$$

$$= 49.6 \text{ N} \cdot \text{mm/mm}$$

Meridional membrane force N_x

$$= -0.129 \times \frac{\mathrm{M} \cos \theta}{t\sqrt{rt}}$$

$$= -9.3 \text{ N/mm}$$

Circumferential membrane force N_ϕ

$$= -0.039 \times \frac{\mathrm{M} \cos \theta}{t\sqrt{rt}}$$

$$= -2.81 \text{ N/mm}$$

The maximum stresses are the resulting meridional stresses given by:

$$f_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} = \frac{-9.3}{12.5} \pm \frac{6 \times 164.5}{(12.5)^2}$$

$\therefore$ at the outside f_x = –0.74 – 6.32

= -7.06 N/mm^2 (compression)

at the inside f_x = –0.74 + 6.32

= +5.58 N/mm^2 (tension)

The slope of the branch due to this moment will be:

$$i_b = \frac{\delta_i}{r_o} = \frac{0.0066}{75} = 8.8 \times 10^{-5} \text{ radians}$$

W.4 Thermal stresses

W.4.1 General

W.4.1.1 Introduction

The example given in **W.4.2** illustrates the simplified method, as detailed in **G.4**, for assessing transient thermal stresses at a pressure vessel nozzle.

W.4.1.2 Notation

For the purposes of **W.4.2** the following symbols apply, based upon those given in **G.4**.

c	is the specific heat of material;
d	is the diffusivity of material;
$C_1 \ldots C_4$	are stress factors;
E	is the modulus of elasticity from Table 3.6-3;

k	is the conductivity of material;
k_1, k_2	are branch thermal factors;
K_1, K_2	are shell thermal factors;
K_b, K_s	are branch and shell mean temperature factors;
K_d	is the mean temperature difference factor;
m	is a thermal factor;
N	is a thermal parameter;
r_i, r_o	are the inner and outer branch radii, respectively;
S_{hi}, S_{ho}	are the inside and outside surface circumferential shell stresses, respectively;
S_{mi}, S_{mo}	are the inside and outside surface meridional shell stresses, respectively;
S'_{hi}, S'_{ho}	are the inside and outside surface circumferential branch stresses, respectively;
S'_{li}, S'_{lo}	are the inside and outside surface longitudinal branch stresses, respectively;
S_{max}	is the maximum stress intensity;
t	is the nominal branch thickness;
T	is the nominal shell thickness;
T_f	is the fluid temperature rise from start of transient;
α	is the coefficient of linear expansion;
δ	is the radial discontinuity;
θ	is the time from start of transient.

W.4.2 Worked example – Thermal stress due to transient temperatures at branch

A branch 300 mm mean diameter and 50 mm thick is welded to a steel vessel 3 m diameter and 100 mm thick. The contained fluid is subject to a ramp rise in temperature of 200 °C in 10 min. The average heat transfer coefficients to shell and branch are estimated as 570 W/(m^2·K) and 2 850 W/(m^2·K) respectively. The example calculates the thermal stress in the assembly at the end of the transient.

Take:

k = 41.5 W/(m·K)

c = 420 J/(kg·K)

ρ = 7 700 kg/m^3

$$d = \frac{k}{c\rho} = \frac{41.5}{420 \times 7\,700} = 1.28 \times 10^{-5}\ \mathrm{m^2/s}$$

Calculate thermal factors (K_1, K_2, k_1, k_2, K_d)

At end of transient θ = 600 s

For the shell:

$$N = \frac{d\theta}{T^2} = \frac{1.28 \times 10^{-5} \times 600}{(0.1)^2} = 0.77$$

$$m = \frac{k}{hT} = \frac{41.5}{570 \times 0.1} = 0.73$$

K_1 = 0.32 (from Figure G.4.3-3)

K_2 = 0.14 (from Figure G.4.3-4)

K_s = 0.30 (from Figure G.4.3-5)

For the branch:

$$N = \frac{d\theta}{t^2} = \frac{1.28 \times 10^{-5} \times 600}{(0.05)^2} = 3.1$$

$$m = \frac{k}{ht} = \frac{41.5}{2\ 850 \times 0.05} = 0.29$$

k_1 = 0.14 (from Figure G.4.3-3)

k_2 = 0.08 (from Figure G.4.3-4)

K_b = 0.82 (from Figure G.4.3-5)

Since r_o/r_i = 1.4 (i.e. r_o/r_i > 1.1)

$$k_1(\text{corrected}) = \frac{150}{125}k_1 = 0.17$$

$K_d = K_b - K_s = 0.52$

Calculate geometric factors (C_1, C_2, C_3)

R/T = 15, r/R = 0.1, $Z = T/t$ = 2.0

C_1 = 0.26 (from Table G.4.3-1)

C_2 = 0.70 (from Table G.4.3-2)

C_3 = 0.13 (from Table G.4.3-3)

Calculate total thermal stress

Rise in fluid temperature (T_f) = 200 K

$E = 21 \times 10^4$ N/mm^2

$\alpha = 12.6 \times 10^{-6}$ m/(m·K)

$E\alpha T_f = 21 \times 12.6 \times 20 \times 10^{-1} = 530$ N/mm^2

Shell:

	Total stress factor	Stress (N/mm^2)
$S_{hi} = 0.52\left(0.26 + \frac{0.3 \times 0.7}{4}\right) - 0.32 =$	–0.157	–83.2
$S_{mi} = 0.52\left(\frac{0.7}{4} - 0.13\right) - 0.32 =$	–0.296	–157.2
$S_{ho} = 0.14 + 0.52\left(0.26 - \frac{0.3 \times 0.7}{4}\right) =$	0.248	131.4
$S_{mo} = 0.14 - 0.52\left(0.13 + \frac{0.7}{4}\right) =$	–0.019	–9.9

Branch:

	Total stress factor	Stress (N/mm^2)
$S'_{hi} = 0.52[0.26 + (0.3 \times 0.7) - 1.0] - 0.17 =$	–0.446	–236.4
$S'_{li} = (0.52 \times 0.7) - 0.17 =$	0.194	102.8

	Total stress factor	Stress (N/mm^2)
$S'_{ho} = 0.08 + 0.52[0.26 - (0.3 \times 0.7) - 1.0] =$	–0.414	–219.4
$S'_{lo} = 0.08 - (0.52 \times 0.7) =$	–0.284	–150.5

From this analysis the maximum stress intensity would occur at the inner surface of the branch and would equal:

$S_{max} = 236.4 + 102.8 = 339.2\ N/mm^2$

For temperature cycling, a fatigue check would be required. The design life due to thermal cycling alone would be obtained from the appropriate fatigue design curves in Annex C using the principal stresses *S* or *S*′ from above to calculate stress ranges in accordance with **C.3.3**, considering individual locations and details in the vessel, as appropriate.

If temperature cycling coincides with pressure changes then any stresses due to pressure shall be added to those due to thermal cycling before calculating the maximum stress range in accordance with **C.3.3**.

W.5 Flat plate

W.5.1 General

W.5.1.1 Introduction

Subclause **W.5** gives an example of the stresses in a welded flat end as illustrated in Figure 3.5-34 and as calculated using the equations presented in Annex R.

W.5.1.2 Notation

a	is the ratio e/e_{cyl};
b	is the ratio D/e_{cyl};
C	is a flat plate factor from **3.5.5**;
$C_1 \dots C_{10}$	are calculation constants given in Annex R;
D	is the mean diameter of cylinder;
e	is the minimum thickness of flat end;
e_{cylo}	is the minimum thickness of the cylindrical shell (see **3.5.5.2**);
e_{cyl}	is the wall analysis thickness of cylinder (see **1.6**);
I	is a calculation constant;
p	is the design pressure;
S	is the maximum stress in the cylinder.

W.5.2 Worked example – Stress in flat plate due to pressure

1) *Design to* **3.5.5.3.1**:

$D = 1\ 200$ mm

$e_{cyl} = 10$ mm

$p = 0.15\ N/mm^2$

$f = 150\ N/mm^2$

Then:

$e_{cylo} = pD/(2f) = 0.6$ mm

$p/f = 0.001$

Initially using Figure 3.5-37 to evaluate C

$e_{cyl}/e_{cylo} = 16.67$

$C = 0.54$

$$e = 0.54 \times 1\ 200\sqrt{0.001} = 20.5 \text{ mm}$$

C can be taken as 0.41 provided that $e/e_{cyl} \leq 2.0$

$$e = 0.41 \times 1\ 200\sqrt{0.001} = 15.56 \text{ mm}$$

(say 16mm)

$e/e_{cyl} = 1.6$

2) *Stress calculation taking* e = 16 mm:

$D = 1\ 200$ mm

$e_{cyl} = 10$ mm

$e = 16$ mm

$p = 0.15$ N/mm^2

Then:

$a = e/e_{cyl} = 1.6$

$b = D/e_{cyl} = 120$

From Annex R:

$$S = I \times \frac{pD}{2e_{cyl}}$$

and $$I = ½ + \left(\frac{C_1a^3 - C_2a + C_3b^{1.5} + C_4ab + C_5b/a}{C_6a^3 + C_7a + C_8/a + C_9b^{0.5} + C_{10}a^2/b^{0.5}}\right)$$

using values for C_1 to C_{10} given in Annex R gives:

$$I = ½ + \left(\frac{1\ 524}{46.8}\right) = 33.07$$

$$S = \frac{33.07 \times 0.15 \times 1\ 200}{2 \times 10} = 298 \text{ N/mm}^2$$

Thus S, the maximum stress in the cylinder, is slightly less than $2f$ confirming that the stress in the cylinder is not controlling.

W.6 Fatigue assessment

W.6.1 General

W.6.1.1 Introduction

To illustrate the fatigue assessment methods of Annex C three examples are presented.

a) A circumferential lap joint in a cylindrical vessel subject to a given number of cycles for which the permissible cyclic pressure range is determined via four separate approaches.

b) A "simple" vessel containing a manway and nozzles subject to cycles of a constant pressure range for which the permissible number of operating cycles is determined.

c) A nozzle/end assembly subject to various loading conditions which occur at different frequencies during the life of the component, for which the fatigue damage factor is determined.

*NOTE Definitions given in Annex C are repeated only in **W.6** when confusion might otherwise occur or to illustrate the relationship to the parameters of the problem.*

W.6.1.2 Notation

For the purposes of **W.6** the following symbols apply, based upon those given in **C.1.3**.

A	is a constant in the equation of fatigue design curve;
D	is the mean shell diameter;
D_i	is the inside shell diameter;
e	is the component analysis thickness;
E	is the modulus of elasticity from Table 3.6-3;
f	is the design stress;
f_f	is a maximum stress;
f_n	is the nozzle design stress;
f_s	is the shell design stress;
K	is a stress concentration factor;
M	is a constant in the equation of fatigue design curve;
n	is the number of imposed loading cycles;
n_i	is the number of cycles experienced under stress range S_{ri};
N	is the design curve fatigue life;
p	is the static design pressure;
p_{op}	is the operating pressure;
p_r	is the pressure range;
r	is the nozzle mean radius;
S_r	is the calculated stress range;
S_{ri}	is the assumed stress range;
t	is the nozzle analysis thickness;
t_D	is the design temperature;
T	is the shell analysis thickness;
α	is the coefficient of thermal expansion;
ν	is Poisson's ratio;
σ_θ	is the circumferential stress.

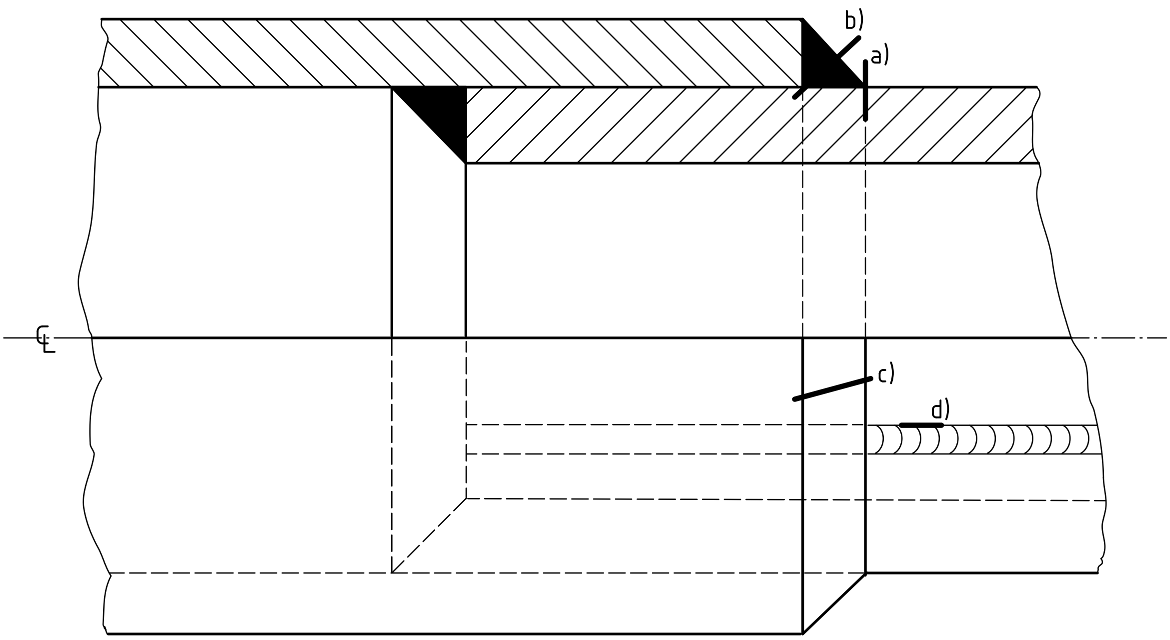
W.6.2 Worked examples

W.6.2.1 Example 1. Fatigue assessment of a lap joint

W.6.2.1.1 Problem

Consider a simple cylindrical vessel made from austenitic stainless steel containing a longitudinal butt weld and a circumferential lap joint as shown in Figure W.6-1. The lap joint is formed by two 45° fillet welds. Find the permissible cyclic pressure for the component using the design parameters given below (for the smaller diameter component).

Figure W.6-1 **Potential failure locations in circumferential lap joint**



*NOTE a) to d) represent the potential failure locations (see **W.6.2.1.4**).*

Mean diameter	D = 1 600 mm
Section thickness	e = 16 mm
Design temperature	t_D = 150 °C
Number of imposed cycles of loading	n = 100 000

NOTE Since the problem requires that the pressure be determined, the calculation will be done in the reverse order to that used in the normal design process.

W.6.2.1.2 Basic design considerations

The form of joint shown in Figure W.6-1 is permitted by Figure E.5a), but with the restrictions that it can only be used in the circumferential direction on material thickness less than 16 mm and for category 3 construction. The maximum allowable static pressure is not given for this design so it will be established from **3.5.1.2** through the use of an allowable stress derived according to **3.4.2.2b**).

Thus the design stress f (in N/mm^2) is given by:

$$\min.\left[120\left(\frac{450}{400+t_D}\right);120\right]$$

where t_D is the design temperature.

From which f is the lesser of:

$$120\left(\frac{450}{400+150}\right)=98.18,\ \text{or } 120;$$

thus f = 98.18 N/mm^2.

The maximum allowable static pressure, p, is calculated from **3.5.1.2** using Equation (3.5.1-1):

$$e=\frac{pD_i}{2f-p}$$

where

e is the thickness = 16 mm;

D_i is the inside diameter = 1 585 mm;

f is the design stress = 98.18 N/mm^2.

This gives a design pressure (maximum allowable static pressure), p, of 1.96 N/mm^2.

W.6.2.1.3 Check on criteria for establishing need for detailed fatigue analysis

1) *Use of these criteria*

If the design is based upon previous and satisfactory experience of strictly comparable service or if the conditions of either **C.2.2** or **C.2.3** are met then it is unnecessary to carry out a detailed fatigue analysis to **C.3**. In this case previous experience does not provide such a basis so the two criteria have to be tested.

2) *Criterion **C.2.2**: Limitation on number of stress fluctuations*

The maximum allowable number of stress fluctuations, N, is given by Equation (C-1):

$$N=\frac{3.8\times10^9}{f_f^3}\left(\frac{25}{e}\right)^{0.75}\left(\frac{E}{2.09\times10^5}\right)^3 \text{the greatest thickness or}$$

where

e is the maximum of “the greatest thickness or 25 mm” (see **C.2.2**) = 25 mm;

E = 1.92 × 10^5 N/mm^2 (from Table 3.6-3) at 150 °C;

N = 100 000 cycles (as the minimum number needed to be justified and thus allowed, i.e. n);

f_f is the maximum stress actually adopted in design in place of f (in N/mm^2).

Substituting in these values: f_f = 30.89 N/mm^2.

Therefore if a detailed fatigue analysis is to be avoided and the criterion of **C.2.2** is to be met this value has to be adopted as the design stress.

Using Equation (3.5.1-1) again, but taking f as 30.89 N/mm^2, the maximum allowable cyclic pressure that will not violate this criterion is calculated as 0.62 N/mm^2.

3) *Criterion **C.2.3**: Simplified fatigue analysis using design curves*

This example requires that the methodology of **C.2** is followed in reverse order so that the allowable cyclic pressure is found.

Thus from step 3 of **C.2.3**, using Equation (C-4) with a single value of i, as there is only one load:

$$\frac{n}{N} \leq 0.6\left(\frac{25}{e}\right)^{0.75}$$

where

e is the maximum of “the greatest thickness or 25 mm” (see **C.2.2**) = 25 mm;

n is the number of cycles experienced under the loading = 100 000;

N is the allowable number of cycles from the fatigue design curve, Figure C.3.

Therefore the limiting value of N = 100 000/0.6 = 166 666 cycles.

To satisfy this criterion, the curve in Figure C.3 for the lowest class of weld detail included in the vessel is used. Since the welds are loading bearing fillet welds, class W1 is used. Curve W1 in Figure C.3, at N = 167 000 cycles, gives an S_r of 82.4 N/mm^2.

Correcting S_r for differences in the elastic modulus of the materials considered in Figure C.3 and the component under consideration (see **C.3.2.2**):

$$S_r = 82.4 \times \left(\frac{1.92 \times 10^5}{2.09 \times 10^5}\right) = 75.7\ \mathrm{N/mm^2}$$

Because of the “reverse approach” used in this analysis, the value of S_r implicitly includes any stress concentration in the geometry of the lap joint, but no analysis of the concentration factor has been carried out. Therefore a conservative approach is used, taking the value of S_r calculated as the stress range due to the only loading (pressure).

Thus from step 2 of **C.2.3** the pressure range, p_r, generating the calculated stress range S_r, can be found by using Equation (C-2):

$$S_r = \left(\frac{p_r}{p}\right)3f$$

where

S_r = 75.7 N/mm^2;

f = 98.18 N/mm^2 (from **W.6.2.1.2**);

p = 1.96 N/mm^2 (from **W.6.2.1.2**).

This gives p_r = 0.50 N/mm^2 as the maximum allowable cyclic pressure that will not violate this criterion.

NOTE 1 In the calculation the value of f does not matter provided that the corresponding value of p is used (based on the design equations in Section 3). What matters is the ratio between them.

*NOTE 2 In this example no benefit has been derived from the simplified fatigue analysis of **C.2.3**. This is because the most onerous class of weld (W1) was assumed, only one source of stress fluctuation exists and the stress analysis has not been improved. It is possible to argue, from experience of the actual behaviour of welded joints, supported by the finite element analysis in **W.6.2.1.4**, that the weld throat is not where a well designed fabrication fails. That would then justify a higher weld classification, F2. Using this design curve the resultant value of p_r becomes 0.84 N/mm^2.*

W.6.2.1.4 Detailed assessment of fatigue life

1) *Potential failure locations*

   Table C.2a) indicates, for the geometry shown in Figure W.6-1, four potential locations for failure are identified. These are shown on Figure W.6-1 as a) to d) and are classified (see **C.3.4.1**) as follows:

   1) longitudinal stress in shell at weld toe: class F2;
   2) stress in weld throat: class W1;
   3) circumferential stress acting along the fillet welds: class D;
   4) circumferential stress in longitudinal seam weld: class D.

2) *A simple stress analysis approach*

   a) *Stress analysis*

      The most appropriate stress to use is the circumferential stress at the inside surface, but the complications of this problem make it acceptable to use the nominal circumferential stress.

      From the classic cylindrical vessel stress/pressure relationship:

      $$\sigma_\theta = \frac{p_r D}{2e}$$

      where

      σ_θ is the circumferential stress (in N/mm^2);

      p_r is the pressure range (in N/mm^2);

      D is the mean diameter of the shell (in mm);

      e is the wall thickness (in mm).

      For a unit pressure range, p, the circumferential stress is calculated as:

      $$\sigma_\theta = \frac{1\ 600p}{2 \times 16} = 50p$$

      The longitudinal stress is roughly half this value, i.e. 25p.

      For the location a) to d), identified in 1), the simplified stresses are as follows.

      a) The longitudinal surface in the shell at the weld toe is concentrated by the effect of the lap misalignment causing bending. Factor A_1 in **C.3.4.6.4** has the value 3 since $e_1 = e_2 = \delta$, so the stress magnification factor K_m [from Equation (C-23)] = 4. The surface stress at the weld toe is thus, 4 × 25p = 100p.

      b) The throat of each fillet weld is 0.707 × wall thickness. As there are two throats in the lap joint they share the longitudinal load. Thus the mean weld stress is 25p/1.414 = 17.7p.

      c) Although it is likely that there will be some reduction in circumferential stress due to the presence of the lap (since the circumferential load is shared between the two layers) it is assumed that the full circumferential load in the single plate adjoining the weld should be taken for this assessment, i.e. 50*p*. In addition there will be the Poisson's ratio effect from the longitudinal bending. Recalling that in an axisymmetrically loaded cylinder $M_\theta = \nu M_x$ this effect is $0.3 \times A_1 \times 25p$, where $A_1 = 3$ from a) above. The weld circumferential stress is thus $50p + (0.3 \times 3 \times 25p) = 73p$.

d) The circumferential stressing for this location is assumed to be the same as location c), i.e. 73*p*.

b) *Determination of Sr for a life of 100 000 cycles*

The allowable stress range for each class of welds given in 1), adjusted for the differences in elastic modulus value, is calculated using Equation (C-5) and **C.3.2.2**:

$N = A \times (S_r)^{-m}$

where

S_r is the stress range;

N is the allowable number of stress cycles = 100 000;

A and m are constants from Table C.1.

The results for locations a) to d) are given in Table W.6-1 below.

c) *Calculation of allowable pressure range*

The allowable pressure range, p_r, is derived by dividing the allowable stress range S_r from Table W.6-1 by the estimated stress values (as functions of *p*) given in the stress analysis a), for locations a) to d) as follows:

a) p_r = 150/100 = 1.5 N/mm^2;

b) p_r = 107/17.7 = 6.0 N/mm^2;

c) p_r = 228/73 = 3.1 N/mm^2;

d) p_r = 228/73 = 3.1 N/mm^2.

Location a), the weld toe, is shown to be the limiting feature of this design and thus the allowable cyclic pressure range for the component to avoid fatigue failure is 1.5 N/mm^2 for 100 000 cycles.

3) *Finite element analysis approach*

a) *Stress analysis*

While it is possible to provide an improved estimate of stresses by thin shell theory, compared with that in the simple analysis of 2), a finite element analysis will provide even more detail. This approach will be applied to overcome the limitations described in **W.6.2.1.3** and in the analysis 2), in respect of the lack of investigation of stress concentration effects.

NOTE 1 For the purpose of this worked example the way the figures are interpreted is of more importance than the use of finite element analysis.

Figure W.6-2 and Figure W.6-3 show, respectively, the principal stress (referring to in-plane stress only) and circumferential stress contours calculated using the finite element model of the lap joint shown in Figure W.6-1. Figure W.6-2 shows that the maximum principal stress, which in this case can be identified as the longitudinal stress, occurs in the weld region on the inside surface. Figure W.6-3 shows that the maximum circumferential stress occurs in the weld region on the outside surface.

The stress needed for this analysis is found following the guidance given in the footnote to **C.3.4.3.2**, as suggested by Niemi [1]. The locations of the most significant stresses are marked on Figure W.6-2 as W and X and on Figure W.6-3 as Y and Z. The stress values (as functions of unit pressure) for the four potential failure locations defined in 1) are as follows:

a) weld toe, longitudinal (location W, contour H of Figure W.6-2): 70*p*;

b) weld throat stress (location X, contour C/D of Figure W.6-2): 25*p*;

c) circumferential stress acting along fillet weld (location Y, contour F of Figure W.6-3): 45*p*;

d) circumferential stress in longitudinal seam weld (location Z, contour H of Figure W.6-3): 55*p*.

NOTE 2 The stresses calculated by finite element analysis are smaller than those previously estimated, except for the weld throat which is low and unimportant anyway.

b) *Calculation of allowable pressure range*

The allowable pressure range, p_r, is derived by dividing the allowable stress range S_r in Table W.6-1 by the estimated stress values (as functions of *p*) given in a) for locations a) to d).

The allowable pressure ranges (those which lead to the above stress ranges) are therefore:

a) p_r = 150/70 = 2.1 N/mm^2;

b) p_r = 107/25 = 4.3 N/mm^2;

c) p_r = 228/45 = 5.1 N/mm^2;

d) p_r = 228/55 = 4.1 N/mm^2.

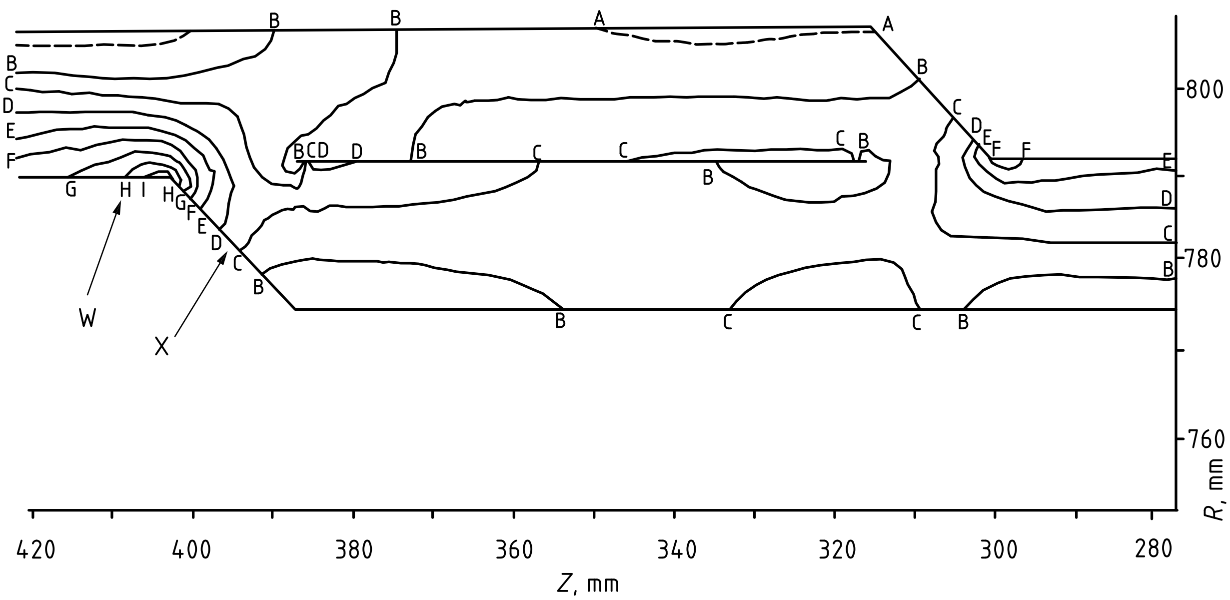
At location a), the weld toe controls the fatigue resistance of the lap joint and thus the allowable cyclic pressure range to avoid fatigue failure is 2.1 N/mm^2 for 100 000 cycles.

Since this is greater than the static design pressure (see **W.6.2.1.2**) of 1.96 N/mm^2 it is possible to cycle up to the design pressure for the requested number of cycles and above. Indeed, it is permissible to cycle between full vacuum (–0.1 N/mm^2) and the design pressure, the pressure range still being only 2.06 N/mm^2.

Table W.6-1 **Results of calculations of allowable stress range**

Location	Weld class	*A*	*m*	$S_r = 3\sqrt{\frac{A}{N}}$ N/mm²	S_r corrected for differences in *E* N/mm²
a)	F2	4.31×10^{11}	3	163	150
b)	W1	9.33×10^{10}	3	98	90
c)	D	1.52×10^{12}	3	248	228
d)	D	1.52×10^{12}	3	248	228

Figure W.6-2 **Maximum axisymmetric principal stress contours in lap joint**



Key to contours

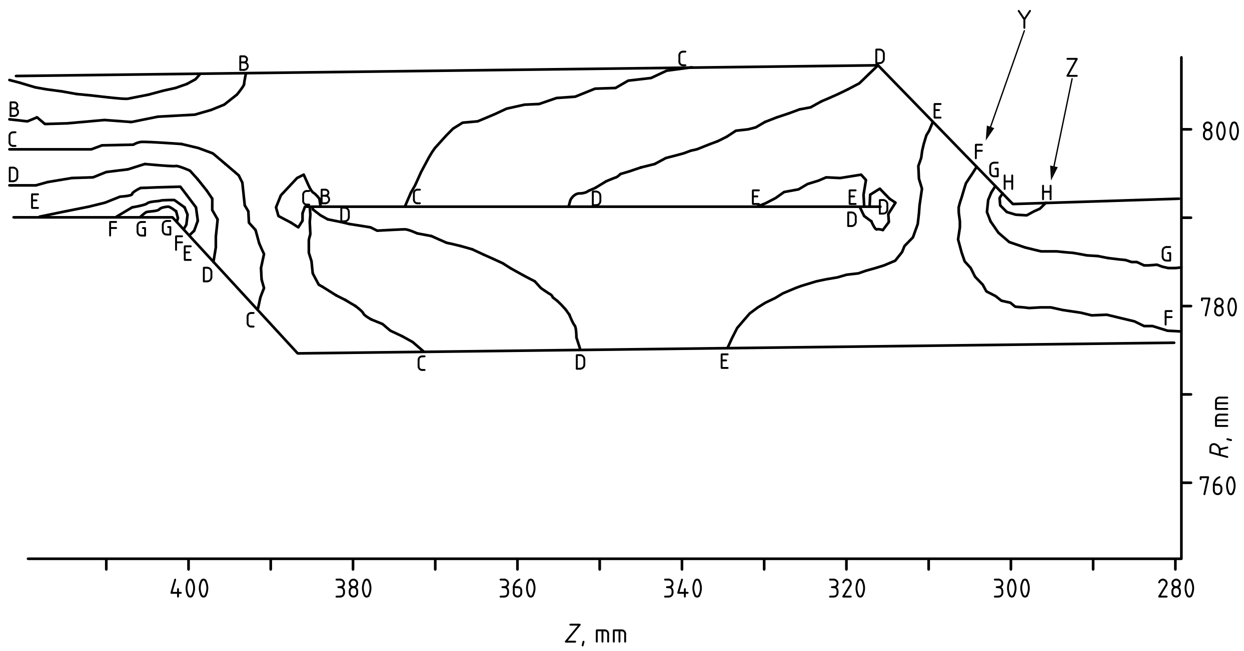
Contour	A	B	C	D	E	F	G	H	I
Stress Level (N/mm²)	0*p*	10*p*	20*p*	30*p*	40*p*	50*p*	60*p*	70*p*	80*p*

NOTE The analysis was carried out for a unit pressure loading and thus the stresses are a function of p:

maximum stress = 90p at R = 792, Z = 404 mm;

minimum stress = −2p at R = 808, Z = 410 mm.

Figure W.6-3 **Circumferential stress in lap joint**



Key to contours

Contour	A	B	C	D	E	F	G	H
Stress Level (N/mm^2)	20*p*	25*p*	30*p*	35*p*	40*p*	45*p*	50*p*	55*p*

NOTE The analysis was carried out for a unit pressure loading and thus the stresses are a function of p:

maximum stress = 58.6p at R = 792, Z = 298 mm;

minimum stress = 16.2p at R = 808, Z = 407 mm.

W.6.2.1.5 Results

The outcome of the various approaches used in this example are summarized in Table W.6-2.

The results in Table W.6-2 indicate the benefits of undertaking the detailed assessment of the fatigue resistance of the proposed design. This approach demonstrates higher levels of allowable pressure than the approach of just meeting the criteria which would allow the detailed assessment to be avoided. Additionally, the results indicate that higher pressures are demonstrated from the use of the more detailed finite element analysis. It is, however, necessary to weigh up the benefits against the greater levels of effort required for the detailed and finite element approaches. The balance will depend upon the circumstances of each individual case.

Table W.6-2 **Summary of results from Example 1**

Annex C route	Approach considered	Clause	Allowable pressure, *p* N/mm^2
Meeting criteria to avoid detailed fatigue analysis	Criterion of **C.2.2**	**W.6.2.1.3**, 2)	0.72
	Criterion of **C.2.3**	**W.6.2.1.3**, 3)	0.6
Undertake detailed fatigue analysis	Simple stress analysis analysis	**W.6.2.1.4**, 2)	1.5
	Finite element analysis	**W.6.2.1.4**, 3)	2.1

W.6.2.2 Example 2. Fatigue assessment of a simple vessel

W.6.2.2.1 Problem

Consider a vessel made of a ferritic material with a cylindrical shell closed with 2:1 ellipsoidal ends. The cylindrical shell is fitted with a 600 mm manway and a 200 mm nozzle. There is also a 150 mm nozzle in one of the dished ends. The vessel is subject to 10 000 cycles of pressure from 0 N/mm^2 to 1.35 N/mm^2. Find the fatigue life for a vessel with the design and material data as follows:

Design pressure	p	= 1.5 N/mm^2;
Pressure fluctuation range	p_r	= 1.35 N/mm^2;
Design temperature		= 150 °C;
Corrosion allowance (for inside surface only)		= 3.0 mm;
Shell design stress at 150 °C	f_s	= 148.0 N/mm^2;
Nozzle design stress at 150 °C	f_n	= 148.0 N/mm^2;
Required number of cycles	n	= 10 000;
Inside diameter of shell		= 1 500 mm;
Thickness of shell		= 15 mm;
Outside diameter of manway		= 610 mm;
Thickness of manway		= 16 mm;
Outside diameter of 200 mm nozzle		= 219.1 mm;
Thickness of 200 mm nozzle		= 12.5 mm;
Outside diameter of 150 mm nozzle		= 168.3 mm;
Thickness of 150 mm nozzle		= 11 mm.

NOTE All the component dimensions are as in the uncorroded state.

The fatigue strength of a vessel is usually governed by the fatigue strength of particular features; such as welds, nozzles, or other structural discontinuities (e.g. cone to cylinder junctions, knuckles, etc.). It is often necessary to analyse a number of these features to determine which will control the fatigue life but it is usually around the nozzles that the highest stresses occur, which is the case in this example.

W.6.2.2.2 Check on criteria for establishing need for detailed fatigue analysis

1) *Use of these criteria*

If the design is based upon previous and satisfactory experience of strictly comparable service or if the conditions of either **C.2.2** or **C.2.3** are met then it is unnecessary to carry out a detailed fatigue analysis to **C.3**. In this case previous experience does not provide such a basis so the two criteria have to be tested.

It should be noted that it is implicit in the application of these criteria that the static design of the component has been properly undertaken. However, in this case (as distinct from Example 1) it is not necessary to use the static stresses.

2) *Criterion **C.2.2**: Limitation of number of stress fluctuations*

The maximum allowable number of stress fluctuations, N, is given by Equation (C-1):

$$N = \frac{3.8 \times 10^9}{f_f^3}\left(\frac{25}{e}\right)^{0.75}\left(\frac{E}{2.09 \times 10^5}\right)^3$$

where

e is the maximum of "the greatest thickness or 25 mm" = 25 mm;

$E = 2.02 \times 10^5$ N/mm^2 (from Table 3.6-3) at 150 °C;

$f_f = f_s = 148.0$ N/mm^2 at 150 °C.

Thus:

$$N = \frac{3.8 \times 10^9}{148^3}\left(\frac{25}{25}\right)^{0.75}\left(\frac{2.02 \times 10^5}{2.09 \times 10^5}\right)^3;$$

$N = 1\ 058$ cycles.

As the required number of cycles is 10 000, the criterion of **C.2.2** has not been met.

3) *Criterion **C.2.3**: Simplified fatigue analysis using design curves*

a) *Step 1*

In this example there is a single fluctuating load (pressure) one frequency.

b) *Step 2*

A conservative estimate of the stress range S_r (in N/mm^2), due to pressure is calculated from Equation (C-2) as follows:

$$S_r = \left(\frac{p_r}{p}\right)3f$$

$$S_r = \left(\frac{1.35}{1.5}\right) \times 3 \times 148 = 399.6 \text{ N/mm}^2$$

c) *Step 3*

The number of cycles, N, is obtained from the appropriate fatigue design curve. The curve for the lowest class of weld detail to be incorporated in the vessel should be used. From Table C.2b) the toe of the weld at a nozzle attachment is class F. Using the class F fatigue curve given in Figure C.3, and correcting for differences in elastic modulus, E, of material covered by Figure C.3 and the components material (see **C.3.2.2**), the number of cycles N is given by Equation (C-5), as follows.

$N = A \times (S_r)^{-m}$

Where m and A are constants obtained from Table C.1. For class F and $N < 5 \times 10^7$ cycles, $m = 3$ and $A = 6.32 \times 10^{11}$.

Thus the allowable number of cycles, including the correction for the elastic modulus difference:

$$N = 6.32 \times 10^{11}\left(399.6 \times \frac{209\ 000}{202\ 000}\right)^{-3} = 8\ 942$$

Hence the damage factor limit, of Equation (C-4), can be tested as follows:

$$\frac{n}{N} \leq 0.6\left(\frac{25}{e}\right)^{0.75}$$

where

e is the maximum of "the greatest thickness or 25 mm" = 25 mm;

$n = 10\ 000$;

N = 8 957.Therefore

$$\frac{n}{N} = \frac{10\ 000}{8\ 957} = 1.116$$

This exceeds the limiting value of $0.6\left(\frac{25}{25}\right)^{0.75}$ therefore the criterion is not satisfied.

4) *Conclusions*

As neither of the criteria examined in 2) and 3) was satisfied, the rules of **C.2** require a detailed assessment.

In general, the procedure can be applied to individual components to determine which of them require further analysis.

W.6.2.2.3 Detailed assessment of fatigue life

1) *Basis of approach*

The fatigue life of the vessel is identified as usually being limited by the nozzle/shell detail (see **W.6.2.2.1**). For this assessment it is necessary to determine the value of the maximum stress (see Footnote to C.3.4.3.2) at the various nozzle/shell discontinuities. For the nozzles in the cylindrical shell the important stress is the circumferential stress at the junction of the vessel and the nozzles (or manway). In the case of a nozzle located in the spherical portion of a dished end, a stress concentration factor (s.c.f.) can be found from the Leckie and Penny curves given in **G.2.5.2** or, alternatively the stress may be found using the approach given in PD 6550-2:1989 [2], which makes use of curves within this specification.

When the nozzle (or manway) is located in a cylindrical shell the stresses calculated in **G.2.3.6.2** and **G.2.5.2** are suitable for assessment to Annex A and against a 2.25f criteria. For the purpose of detailed assessment for fatigue life it is proposed, for the cylindrical shell case, to use an equation suggested by Decock [3]. This formulation is based upon the fact that the maximum stress invariably occurs at the crotch corner [see 2)d)].

$$\text{s.c.f.} = \frac{2 + \frac{2d}{D}\sqrt{\frac{d}{D} \times \frac{t}{T}} + \frac{1.25d}{D}\sqrt{\frac{D}{T}}}{1 + \frac{t}{T}\sqrt{\frac{d}{D} \times \frac{t}{T}}}$$

where

d is the mean diameter of the corroded nozzle (in mm);

D is the mean diameter of the corroded shell (in mm);

t is the analysis thickness of the nozzle (in mm);

T is the analysis thickness of the shell (in mm).

2) *Shell nozzle assessment*

a) *Generally applicable shell parameters*

The following parameters are for the corroded condition.

Shell thickness	T = 12.0 mm;
Inside diameter of shell	D_i = 1 500 + 6 = 1 506 mm;
Mean diameter of shell	D = 1 506 + 12 = 1 518 mm.

b) *Nozzle stress concentration factors*

i) *For manway in the cylindrical shell*

The following parameters are for the corroded condition.

Manway thickness	t = 13.0 mm;
Inside diameter of manway	d_i = 610 – 26 = 584 mm;
Mean diameter of manway	d = 584 + 13 = 597 mm.

Using the Decock equation from 1):

$$\text{s.c.f.} = \frac{2 + \dfrac{2 \times 597}{1\ 518}\sqrt{\dfrac{597}{1\ 518} \times \dfrac{13}{12}} + \dfrac{1.25 \times 597}{1\ 518}\sqrt{\dfrac{1\ 518}{12}}}{1 + \dfrac{13}{12}\sqrt{\dfrac{597}{1\ 518} \times \dfrac{13}{12}}}$$

s.c.f. = 4.711

ii) *For 200 mm nozzle in cylindrical shell*

The following parameters are for the corroded condition and allow for a 12.5% mill tolerance.

Nozzle thickness	t = 12.5 × 0.875 – 3 = 7.94 mm
Inside diameter of nozzle	d_i = 219.1 –15.88 = 203.22 mm
Mean diameter of nozzle	d = 203.22 + 7.94 = 211.16 mm

Using the Decock equation from 1):

$$\text{s.c.f.} = \frac{2 + \dfrac{2 \times 211.16}{1\ 518}\sqrt{\dfrac{211.16}{1\ 518} \times \dfrac{7.94}{12}} + \dfrac{1.25 \times 211.16}{1\ 518}\sqrt{\dfrac{1\ 518}{12}}}{1 + \dfrac{7.94}{12}\sqrt{\dfrac{211.16}{1\ 518} \times \dfrac{7.94}{12}}}$$

s.c.f. = 3.365

c) *Stress calculations*

The highest s.c.f. calculated, i.e. 4.711, is associated with the manway. In view of the other shell details being in other respects similar, only the manway is considered further.

To obtain the peak stress at the nozzle/shell discontinuity the appropriate stress concentration factor is multiplied by circumferential membrane stress in the shell.

The circumferential membrane stress range in a cylindrical shell can be calculated from the classical stress/pressure relationship for a cylinder, as follows:

$$\sigma_\theta = \frac{p_r D}{2e}$$

where

σ_θ is the circumferential stress (in N/mm^2);

p_r is the pressure range (in N/mm^2);

D is the mean diameter of the shell (in mm);

e is the wall thickness (in mm).

Thus for the shell in the corroded condition:

$$\sigma_\theta = \frac{1.35 \times 1\ 518}{2 \times 12} = 85.39\ \mathrm{N/mm^2}$$

Therefore, the maximum stress range is 4.711 × 85.39 = 402.27 N/mm^2.

It is to be noted that the value of this maximum stress range is greater than the 3f value conservatively assumed in **W.6.2.2.2**, 3). The reason for this is that the design of the nozzle is based on centre-line shell theory and cannot predict accurately the behaviour at the crotch corner.

d) *Fatigue analysis*

For nozzles in cylindrical shells the peak stress usually occurs at the inside corner of the nozzle in the longitudinal plane (the crotch corner) in the circumferential direction. The stress on the outside surface is significantly lower (typically about half, see **2.1** of PD 6650-2:1989 [2]) and occurs on the transverse plane. Therefore, for the nozzle geometries allowed in this specification, the stress on the inside corner, calculated using the equation of Decock [3], is generally considered the worst case for fatigue assessment. Reference to Table C.2b) indicates that failure from this point may be considered as class D.

NOTE 1 It is worth noting that if the stress in the transverse plane on the outside surface of the cylindrical shell, in the region of the toe of the weld were obtained, say, by using finite element analysis then the weld category would be class F.

For this example, the higher value of the inside surface stress range, S_r = 402.27 N/mm^2 [from c)], coupled with the class D is a worst case. The membrane and bending components of the stress are not known, therefore Ω = 0. The thickness e is less than 25 mm therefore k_{tb} = 1. Using these values with Equation (C-12) the allowable number of cycles of loading can be found. From Table C.1, for class D and $N < 5 \times 10^7$ cycles;

m = 3 and $A = 1.52 \times 10^{12}$, therefore

$$N = A\left(\frac{k_{tb}E}{S_r \times 2.09 \times 10^5}\right)^m$$

$$N = 1.52 \times 10^{12}\left(\frac{1.0 \times 2.02 \times 10^5}{402.27 \times 2.09 \times 10^5}\right)^3$$

N = 21 082 cycles

NOTE 2 If the initial consideration of the nozzles crotch corner geometry (where it was classified D) it had been classified C (due, say, to the location being free of welds or flush-ground repairs) the allowable number of cycles would rise to 28 757.

3) *Dished end nozzle assessment*

a) *Dished end and nozzle parameters*

The following parameters are for the corroded condition and allow for a 12.5% mill tolerance.

Corroded nozzle thickness	t = 11 × 0.875 − 3 = 6.625 mm
Inside diameter of nozzle	d_i = 168.3 − 13.25 = 155.05 mm
Mean diameter of nozzle	d = 155.05 + 6.625 = 161.68 mm

The equivalent external diameter, D_o, of the "spherical" portion of the 2:1 ellipsoidal end, is determined from **3.5.2.4**:

$D_o = D \times$ (factor for ellipsoidal ends from Table 3.5-1)

where D = 1 530 and the factor is a function of $\frac{h_e}{D}$ which for the 2:1 end, is 0.25, giving a factor of 1.80.

Therefore $D_o = 1.8 \times 1\,530 = 2\,754$ mm.

Taking the corroded thickness of end as 12 mm, the equivalent mean diameter of end, $D = 2\,754 - 12 = 2\,742$ mm.

b) *Stress concentration factors*

The s.c.f. can be obtained from either:

— Figure G.2.5-1; or

— Equation (4) of PD 6550-2:1989 [2] through the use of Figure 3.5-10.

In both these methods the parameter ρ is used, defined as follows:

$$\rho = \frac{r}{R}\sqrt{\frac{R}{T}} \text{ which } = \frac{d}{D}\sqrt{\frac{D}{2T}}$$

where

d is the mean nozzle diameter (in mm);

D is the mean diameter of the spherical portion of the dished end (in mm);

T is the dished end thickness (in mm), which is equivalent to e_{rs} and T'.

Therefore:

$$\rho = \frac{161.68}{2\,742}\sqrt{\frac{2\,742}{2 \times 12}} = 0.630$$

In determining the ratio of the nozzle/shell wall thickness t/T' for use in Figure G.2.5-1, and e_{rb}/e_{rs} for use in Figure 3.5-10, t is the equivalent of e_{rb}.

Thus:

$$\frac{t}{T} = \frac{t}{T'} = \frac{e_{rb}}{e_{rs}}, \text{ therefore } \frac{t}{T} = \frac{6.6}{12} = 0.55.$$

Using these parameters in the two methods referred to in i) and ii):

i) plotting ρ and $\frac{t}{T'}$ on Figure G.2.5-1; gives an elastic s.c.f. = 2.5;

ii) plotting ρ and $\frac{e_{rb}}{e_{rs}}$ on Figure 3.5-10 gives

$$\frac{Ce_{rs}}{e_{ps}} = 1.21$$

which is used in Equation (4) of PD 6550-2:1989 [2],

$\frac{CT_r}{T} = \frac{1.1K}{2.25} = \frac{Ce_{rs}}{e_{ps}}$ to calculate the stress concentration K, as follows.

$$K = \frac{2.25Ce_{rs}}{1.1\ e_{ps}} = \frac{2.25}{1.1} \times 1.21 = 2.48$$

Figure 3.5-10 was developed from similar sources to Figure G.2.5-1 so it is not surprising that the results are essentially the same. However, Figure 3.5-10 and the associated Table 3.5-5 provide more readily obtained information and are therefore preferred for these calculations.

c) *Stress calculations*

To obtain the maximum stress range the concentration factor, found in i), is multiplied by the membrane stress range.

The membrane stress range in the "spherical" portion of a dished end is given by:

$$\sigma_\theta = \frac{p_r D}{4e} = \frac{1.35 \times 2\ 742}{4 \times 12} = 77.12\ \text{N/mm}^2$$

Thus the maximum stress range is:

$2.48 \times 77.12 = 191.26\ \text{N/mm}^2$

From **C.3.3.2**, for stress cycling due to the application and removal of a single load (i.e. pressure), the stress range S_r is the same as the maximum principal stress caused by the load acting alone.

d) *Fatigue analysis*

In the case of nozzles in spherical ends the maximum stress invariable occurs on the outside surface of the vessel (see **2.1** of PD 6550-2:1989 [2]) and has already been calculated, giving a value of S_r equal to 191.26 N/mm^2.

The weld category from Table C.2b) is class F. From Table C.1 for $N < 5 \times 10^7$ cycles, $m = 3$ and $A = 6.32 \times 10^{11}$, in Equation (C-12):

$$N = A\left(\frac{k_{tb}E}{S_r \times 2.09 \times 10^5}\right)^m$$

$$N = 6.33 \times 10^{11}\left(\frac{1.0 \times 2.02 \times 10^5}{191.26 \times 2.09 \times 10^5}\right)^3$$

N = 81 557 cycles

NOTE The correction for thickness is not required for thicknesses less than 25 mm.

W.6.2.2.4 Results

The analysis has given the fatigue capacity of the nozzle details in the vessel. These are given in Table W.6-3, which indicates that the main shell's manway nozzle is the fatigue life limiting feature.

Table W.6-3 **Summary of results for Example 2**

Component	Detail	Allowable number of cycles, N
Shell	Manway nozzle	21 082
	200 mm nozzle	>21 082
Dished end	150 mm nozzle	81 686

The minimum fatigue life of 21 082 cycles, for the manway, is however in excess of the design requirement of 10 000 cycles. Thus the components are all suitable for the intended duty.

W.6.2.3 Example 3. Fatigue assessment of a nozzle/end assembly subject to a range of different loads

W.6.2.3.1 Problem

Consider a hemispherical end on a vessel, containing a 90° set-in branch welded as in Figure E.8b), manufactured in material to grade 224 – 460A to BS 1501. Determine the limiting fatigue damage factor for the design.

The design and material data for the vessel are as follows:

Design temperature		= 340 °C;
Design stress of vessel and nozzle at 340 °C	f	= 112.2 N/mm^2;
Modulus of elasticity	E	= 1.87×10^5 N/mm^2 (Table 3.6-3);
Coefficient of thermal expansion	α	= 1.26×10^{-5}/°C;
Design pressure	p	= 110 bar g.;
Operating pressure	p_{op}	= 100 bar g.;
Mean radius of spherical vessel	R	= 1 479 mm;
Wall thickness of vessel in region of junction,	T'	= 92 mm;
Mean radius of set-in nozzle	r	= 216 mm;
Wall thickness of nozzle in region of junction	t	= 107 mm.

W.6.2.3.2 General considerations

Two cases (A and B) are considered in this example. Case A, illustrates the benefits of the simplified fatigue analysis, where it satisfies the criterion. Case B, shows how to carry out a detailed assessment of the fatigue life when the simplified approach is not valid.

Use is made of the stresses which have already been derived using an appropriate shell analysis, as in Annex W.A, Annex W.B and Annex W.C. In this example these stresses have been obtained using straightforward calculations based on the approach given in this specification.

It should be appreciated that a more complete stress pattern would be provided using a finite element analysis of the component. Such an analysis is able to show the location of the highest stress and correctly summate the components whereas a simplified analysis can only sum maximum stresses that may not be acting coincidentally. However, since such an approach is not always available to the designer, the approach using this specification is considered more useful in this example.

In this example the corrosion allowance or other deductions in wall thickness are not considered. These may be considered in the manner set out in **W.6.2.2**, Example 2.

W.6.2.3.3 Case A

1) *Cyclic loading*

For this case it is assumed that the nozzle/end assembly experiences stress cycles from the loads given in Table W.6-4 during its service life.

2) *Check on criteria to establishing need for detailed fatigue analysis*

a) *Use of these criteria*

If the design is based upon previous and satisfactory experience of strictly comparable service or if the conditions of either **C.2.2** or **C.2.3** are met then it is unnecessary to carry out a detailed fatigue analysis to **C.3**. In this case previous experience does not provide such a basis so the two criteria have to be tested.

b) *Criterion **C.2.2**: Limitation on number of stress fluctuations*

A detailed fatigue analysis is not necessary if the total number of stress fluctuations arising from all sources does not exceed the value calculated from Equation (C-1):

$$N = \frac{3.8 \times 10^9}{f_f^3}\left(\frac{25}{e}\right)^{0.75}\left(\frac{E}{2.09 \times 10^5}\right)^3$$

where

e is the maximum of "the greatest thickness or 25 mm";

E is the modulus of elasticity from Table 3.6-3;

N is the number of fatigue cycles allowed;

f_f is the maximum stress actually adopted in design in place of f (in N/mm^2).

A good estimate for f_f is the maximum of the various stresses, as follows:

$$\text{max.}\left[\Sigma\sigma_{(\text{primary})}; 2/3 \times \Sigma\sigma_{(\text{primary} + \text{local})}; 1/3 \times \Sigma\sigma_{(\text{primary} + \text{local} + \text{secondary})}\right]$$

or

$$\text{max.}\left[\Sigma\sigma_{(f_m)}; 2/3 \times \Sigma\sigma_{(f_m + f_L)}; 1/3 \times \Sigma\sigma_{\left(f_m + f_L + f_g\right)}\right]$$

NOTE 1 Thermal stresses are treated as secondary stresses.

NOTE 2 The factors 1, 2/3 and 1/3, arise from the stress categories and the limits of stress intensity stated in Annex W.A and given in Figure A.3.

In the static design of the assembly covered by Annex W.A, Annex W.B and Annex W.C, the following combinations of stresses have already been considered.

Primary and secondary stress due to the design pressure (S_{dp}).

Primary and secondary stress due to full external piping loads ($S_{\Delta m}$).

Transient thermal stress resulting from fluid temperature cycling ($S_{\Delta T}$), which for the purpose of fatigue analysis is treated as a secondary stress.

S_{dp} and $S_{\Delta m}$ were determined using **G.2.5** and $S_{\Delta T}$ from **G.4**. For this example the values are:

S_{dp} **W.A.2** gives a stress of 16.08p N/mm^2 for unit pressure p. Therefore for the design pressure of 11 N/mm^2 (110 bar g. = 11 N/mm^2) the resulting stress is 16.08 ×11 = 176.9 N/mm^2;

$S_{\Delta m}$ **W.B.3.5** gives 15.68 N/mm^2;

$S_{\Delta T}$ **W.C.2.4** gives 303.5 N/mm^2.

The sum of these stresses is equal to 496.1 N/mm^2.

Hence taken $f_f = 496.1/3 = 165.4$ N/mm^2.

NOTE 3 The denominator 3 comes from the factor applied to the appropriate summation of the primary, local and secondary stresses in the estimate for f_f given previously.

It follows that with $f_f = 165.4$ N/mm^2, $e = t = 107$ mm (the highest value for the nozzle and shell wall thickness should be used) and $E = 1.87 \times 10^5$ N/mm^2, that to satisfy Equation (C-1) the number of stress fluctuations shall not exceed:

$$N = \frac{3.8 \times 10^9}{165.4^3}\left(\frac{25}{107}\right)^{0.75}\left(\frac{1.87 \times 10^5}{2.09 \times 10^5}\right)^3 = 202 \text{ cycles}$$

As the actual number of cycles exceeds 202, this criterion is not satisfied.

The nozzle/end assembly has thus to be examined more carefully using a simplified fatigue analysis.

If the static calculations for the pressure and thermal loading are not available, the value of f_f can be obtained using the conservative approach (in which the stresses are assumed to exist up to the maximum possible limit) and values presented in **W.D.1** for the external loading, since no simplified method is available.

S_{dp} — $3f = 3 \times 112.2 = 336.6$ N/mm^2;

$S_{\Delta m}$ — **W.B.3.5** gives 15.68 N/mm^2;

$S_{\Delta T}$ — **W.D.1.4** gives 1 060.3 N/mm^2.

NOTE 4 In the case of $S_{\Delta T}$, a more accurate value of this stress range is available from the thermal transient analysis in Annex W.C. However, the value is not used here so that the simplified method may be illustrated.

The total of these stresses gives a value of $f_f = 1\ 412.6/3 = 470.9$ N/mm^2 and a correspondingly lower value for the number of stress fluctuations [calculated as 8 cycles using Equation (C-1)], than when using the more accurate stress values.

c) *Criterion **C.2.3**: Simplified fatigue analysis using the design curves*

Step 1

Identify the various events to be experienced by the vessel which will give rise to fluctuating stresses and the frequency at which they occur. In the simplified analysis it is assumed that the detailed stress/time history of the assembly is not available. The load combinations are therefore set down in the order of increasing frequency, as given in Table W.6-5. The number of cycles are those which occur during the life of the assembly as quoted under case A; no correction is made for cycles which have occurred in earlier events.

Step 2

For each source the maximum stress range can be calculated using the information derived in **W.D.1**.

A1	$S_{r1} = S_t$	$= 716.0$ N/mm^2;
A2	$S_{r2} = S_u + S_p + S_{\Delta;T} + S_{\Delta m}$	$= 64.3 + 306.0 + 1\ 060.3 + 15.7$ $= 1\ 446.3$ N/mm^2;
A3	$S_{r3} = S_p + S_{\Delta T} + S_{\Delta m}$	$= 306.0 + 1\ 060.3 + 15.7 = 1\ 382.0$ N/mm^2;
A4	$S_{r4} = S_{\Delta T} + S_{\Delta p} + S_{\Delta m}$	$= 1\ 060.3 + 30.6 + 15.7 = 1\ 106.6$ N/mm^2;
A5	$S_{r5} = S_{\Delta p} + S_{\Delta m}$	$= 30.6 + 15.7 = 46.3$ N/mm^2;
A6	$S_{r6} = S_{\Delta m}$	$= 15.7$ N/mm^2.

Step 3

Substituting each of the values of S_{r1} calculated in step 2 above in Equation (C-5) gives the values of N that can be used in Equation (C-4) to determine the total cumulative damage factor. In calculating the values of N the fatigue curve for the lowest class of weld detail incorporated in the vessel or part under consideration shall be used as required by step 3 in **C.2.3**. In this case, from Table C.2b), this is class F, and the value of the exponent b is 0.25.

Thus using Equation (C-5):

$$S_r^m N = A$$

Where m and A are constants obtained from Table C.1 for class F giving:

for $S > 23$ N/mm^2 (= < 5×10^7 cycles): $m = 3$; $A = 6.32 \times 10^{11}$;

for $S < 23$ N/mm^2 (= > 5×10^7 cycles): $m = 5$; $A = 3.43 \times 10^{14}$.

Taking into account the effect of the material modulus of elasticity Equation (C-5) is modified to:

$$N = A\left(S_r \times \frac{2.09 \times 10^5}{E}\right)^{-m} = A\left(S_r \times \frac{2.09 \times 10^5}{1.87 \times 10^5}\right)^{-m}$$

This gives the fatigue life as a function of each of the stress ranges S_m, as summarized in Table W.6-6:

Applying the damage rule of Equation (C-4) gives:

$$\Sigma\frac{n_i}{N_i} = 0.055, \text{ which is less than the criterion limit of :}$$

$$0.6 \times \left(\frac{25}{e}\right)^{0.75} = 0.6 \times \left(\frac{25}{107}\right)^{0.75} = 0.202$$

3) *Conclusions*

In this case the simplified fatigue analysis indicates that there is no need for a detailed fatigue analysis. The component is deemed to be satisfactory.

Table W.6-4 **Cyclic loading regime for case A**

Source	**Description**	**Number of cycles**
A1	Hydrostatic proof test at a pressure of 234 bar g. (see Note)	1
A2	Upset condition of 110% of the design pressure	1
A3	Start ups and shut downs, i.e. full pressure cycles between zero and maximum operating pressure (100 bar g.)	3
A4	Fluid temperature cycles between 10 °C and 235 °C, i.e. a heating cycle	10
A5	Pressure cycles between 90 bar g. and 100 bar g.	750
A6	External load cycle due to attached piping	1 500

*NOTE This value may be obtained using the appropriate equation in **5.8.5.1**, where f_a = 187 N/mm^2, f_t = 112.2 N/mm^2 and c = 2 mm.*

Table W.6-5 **Fatigue load cycle stress ranges**

Source	Short description	Maximum stress range	Number of cycles n	Assumed stress range S_{ri}
A1	Hydrostatic test	S_t	1	$S_{r1} = S_t$
A2	Upset pressure	S_u	1	$S_{r2} = S_u + S_p + S_{\Delta T} + S_{\Delta m}$
A3	Pressure range	S_p	3	$S_{r3} = S_p + S_{\Delta T} + S_{\Delta m}$
A4	Temperature transient cycling	$S_{\Delta T}$	10	$S_{r4} = S_{\Delta T} + S_{\Delta p} + S_{\Delta m}$
A5	Pressure cycling	$S_{\Delta p}$	750	$S_{r5} = S_{\Delta p} + S_{\Delta m}$
A6	Mechanical load	$S_{\Delta m}$	1 500	$S_{r6} = S_{\Delta m}$

Table W.6-6 **Summary of damage calculation values for case A**

Source	S_{ri}	m	A_i	N_i	n_i	n_i/N_i
A1	716	3	6.32×10^{11}	1 233	1	0.00081
A2	1 446.3	3	6.32×10^{11}	150	1	0.00668
A3	1 382.0	3	6.32×10^{11}	172	3	0.01749
A4	1 106.6	3	6.32×10^{11}	334	10	0.02993
A5	46.3	3	6.32×10^{11}	45.6×10^5	750	0.00016
A6	15.7	5	3.43×10^{14}	2.06×10^8	1 500	0.00001

W.6.2.3.4 Case B

1) *Cyclic loading*

For this case it is specified that the nozzle/end assembly experiences stress cycles from the loads given in Table W.6-7 over a 10 year period. A typical section of the stress/time history for this case is shown in Figure W.6-4. The plot shows the loading spectrum over a period of 1 week, which is repeated each week for 10 years. For ease of calculation it is assumed that the plant operates for 50 weeks per year.

This is essentially the same as case A but the number of cycles is higher in all cases, except the hydrostatic test.

Table W.6-7 **Cyclic loading regime for case B**

Source	Description	Number of cycles
B1	Hydrostatic proof test at a pressure of 234 bar g. (see Note)	1
B2	Upset condition of 110% of the design pressure	500
B3	Starts ups and shut downs, i.e. full pressure cycles between zero and maximum operating pressure (100 bar g.)	500
B4	Fluid temperature cycles between 10 °C and 235 °C, i.e. a heating cycle	500
B5	Pressure cycles between 90 bar g. and 100 bar g.	4 500
B6	External load cycles due to attached piping	16 500

NOTE This value may be obtained using the appropriate equation in ***5.8.5.1****, where f_a = 187 N/mm^2, f_t = 112.2 N/mm^2 and c = 2 mm.*

2) *Check on criteria to establish need for detailed fatigue analysis*

a) *Use of these criteria*

If the design is based upon previous and satisfactory experiences of strictly comparable service or if the conditions of either **C.2.2** or **C.2.3** are met then it is unnecessary to carry out a detailed fatigue analysis to **C.3**. In this case previous experience does not provide such a basis so the two criteria have to be tested.

b) *Criterion **C.2.3**: Limitation on the number of stress fluctuations*

The loading is similar to case A, thus the value of the design stress f_f, is the same and the allowable number of stress cycles is again 202. The actual number of stress fluctuations thus exceeds the limit imposed to satisfy the criterion of **C.2.2**, thus further analysis is required.

c) *Criterion **C.2.3**: Simplified fatigue analysis using the design curves*

The pattern of loading cycles and the maximum stress ranges S_{r1} to S_{r6} for case B are identical to those of case A. The allowable number of cycles N_1 to N_6 will therefore also be the same as in step 3 of case A. Thus applying the damage rule of Equation (C-4) to the allowable number of cycles and the imposed number (as given in Table W.6-7), the damage fraction is:

$\Sigma\frac{n_i}{N_i} = 7.76$, which very clearly exceeds the criterion limit of:

$$0.6 \times \left(\frac{25}{e}\right)^{0.75} = 0.6 \times \left(\frac{25}{107}\right)^{0.75} = 0.202$$

Thus a detailed fatigue analysis will be carried out for this case.

3) *Detailed assessment for fatigue life*

Step 1

In the present example the principal stresses are the circumferential stress, the longitudinal or meridional stress and the through thickness stress, of which the latter can be ignored because its order of magnitude is relatively small. The stress cycling is due to more than one load, but the directions of the principal stresses remain fixed. Hence S is the maximum range through which any of the principal stresses changes.

Using the reservoir cycle counting method of Annex H of BS 7608:2014+A1 [4], illustrated in Figure W.6-5, the various events which give rise to the fluctuating stresses are given in Table W.6-8. Note that, unlike the simplified analysis, the exact number of the events is closely adhered to.

Figure W.6-4 **Loading spectrum**

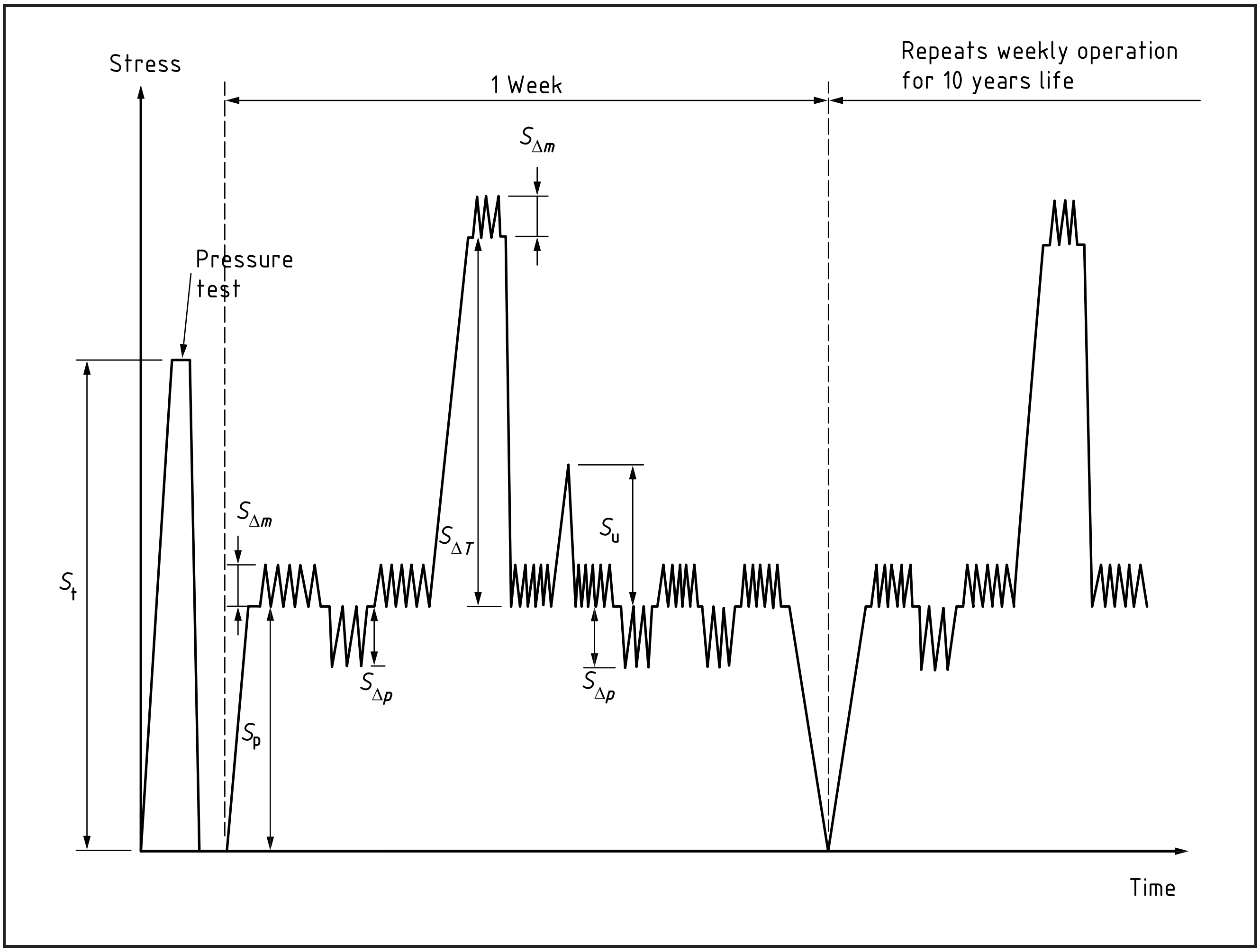


Figure W.6-5 **Reservoir cycle count**

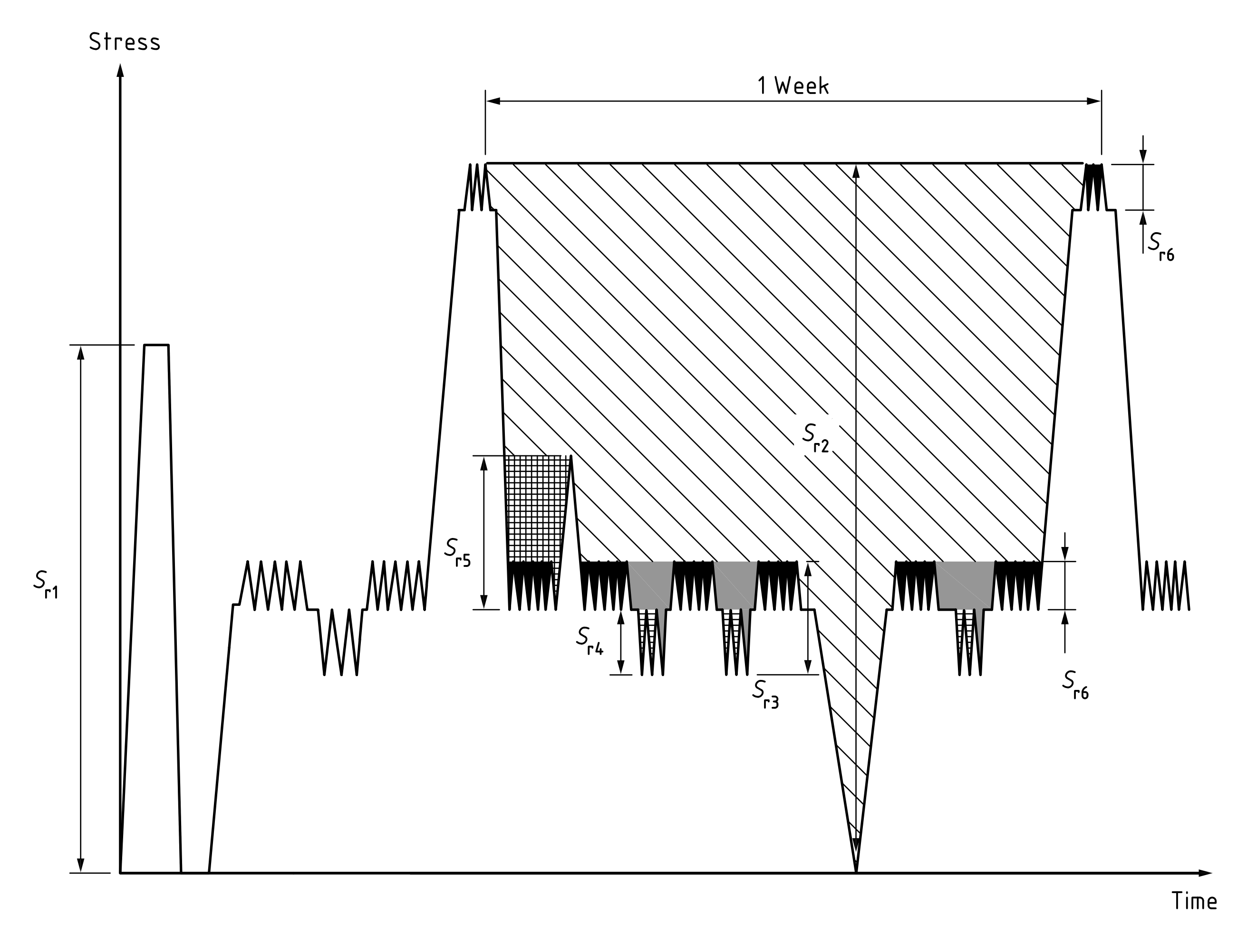


Table W.6-8 **Fatigue load cycle stress ranges**

Source	Short description	Maximum stress range	Number of cycles n	Assumed stress range S_{ri}
B1	Hydrostatic test	S_t	1	$S_{r1} = S_t$
B2	Upset pressure	S_u	500	$S_{r2} = S_p + S_{\Delta T} + S_{\Delta m}$
B3	Pressure range	S_p	1 500	$S_{r3} = S_{\Delta p} + S_{\Delta m}$
B4	Temperature transient cycling	$S_{\Delta T}$	4 500 – 1 500 = 3 000	$S_{r4} = S_{\Delta p}$
B5	Pressure cycling	$S_{\Delta p}$	500	$S_{r5} = S_u$
B6	Mechanical load	$S_{\Delta m}$	16 500 – 500 – 1 500 = 14 500	$S_{r6} = S_{\Delta m}$

Step 2

For each source the maximum stress range can be calculated using the information derived in **W.D.2**. It then follows:

B1 $S_{r1} = S_t$ $= 376.3\ \text{N/mm}^2$;

B2 $S_{r2} = S_p + S_{\Delta T} + S_{\Delta m}$ $= 160.1 + 303.5 + 15.7 = 479.3\ \text{N/mm}^2$;

B3 $S_{r3} = S_{\Delta p} + S_{\Delta m}$ $= 16.1 + 15.7 = 31.8\ \text{N/mm}^2$;

B4 $S_{r4} = S_{\Delta p}$ $= 16.1\ \text{N/mm}^2$;

B5 $S_{r5} = S_u$ = 33.8 N/mm^2;
B6 $S_{r6} = S_{\Delta m}$ = 15.7 N/mm^2.

Step 3

In this case only class F [see Table C.2b)] is considered, and the value of exponent b is 0.25. In a more general case it would be necessary to construct the stress-history profile for all possible sites that are subject to fatigue cracking.

The constants are again obtained from Table C.1 for class F:

for $S > 23$ N/mm^2 (= < 5 × 10^7 cycles): $m = 3$; $A = 6.32 \times 10^{11}$;

for $S < 23$ N/mm^2 (= > 5 × 10^7 cycles): $m = 5$; $A = 3.43 \times 10^{14}$.

Taking into account the effect of the material modulus of elasticity and the thickness of the part considered, from Equations (C-9) and (C-12):

The membrane and bending components of the stress are not known, therefore $\Omega = 0$. The value of the exponent b from Table C.2 is 0.25.

For $e > 25$ mm: $k_{tb} = \left(\frac{25}{e}\right)^b (1 + 0.18\Omega^{1.4}) = \left(\frac{25}{107}\right)^{0.25} (1 + 0.0) = 0.6952$

$$N = A\left(\frac{k_{tb}E}{S_r \times 2.09 \times 10^5}\right)^m$$

$$N = A\left(\frac{0.6952 \times 1.87 \times 10^5}{S_r \times 2.09 \times 10^5}\right)^m$$

This gives the fatigue life as a function of each of the stress ranges S_{ri}, as summarized in Table W.6-9.

Table W.6-9 **Summary of damage calculation for case B**

Source	S_{ri}	m_i	A_i	N_i	n_i	n_i/N_i
B1	376.3	3	6.32×10^{11}	2 855	1	0.00035
B2	479.3	3	6.32×10^{11}	1 382	500	0.36189
B3	31.8	3	6.32×10^{11}	4.73×10^6	1 500	0.00032
B4	16.1	5	3.43×10^{14}	29.53×10^6	3 000	0.00010
B5	33.8	3	6.32×10^{11}	3.94×10^6	500	0.00013
B6	15.7	5	3.43×10^{14}	33.49×10^6	14 500	0.00043

Applying the damage rule of Equation (C-6) leads to:

$$\Sigma\frac{n_i}{N_i} = 0.3632 < 1$$

4) *Conclusion*

The condition of Equation (C-6) has been satisfied and thus the nozzle/end assembly can be judged satisfactory from a fatigue point of view for the 10 year period considered.

W.6.2.3.5 Results

The results summarized in Table W.6-10 indicate how failure to meet the criterion of **C.2.2** can result in acceptance in case A, of a low number of 2 265 imposed cycles, that satisfy the other criterion of **C.2.3**. Whereas, in case B, its higher number of 22 501 cycles, failed both criteria, but was demonstrated through a detailed fatigue analysis, to **C.3**, to have a damage factor of 0.3632. Thus the limiting damage factor is 0.3632 for the case B loading regime.

Table W.6-10 **Summary of results**

Annex C route	Condition	Required limit	Results	
			Case A	Case B
Criterion of **C.2.2**	Number of cycles	290	2 265 (F)	22 501(F)
Criterion of **C.2.3**	Damage factor	0.183	0.055 (S)	7.73 (F)
Undertake detailed fatigue analysis of **C.3**	Damage factor	1	—	0.3632 (S)

Key
(F) indicates "fail".
(S) indicates "satisfactory".

W.6.2.3.6 Annexes to W.6.2.3

Annex W.A Calculation of stress in branch due to internal pressure

W.A.1 Method

The method proposed uses **G.2.5.2** which identifies that the maximum stress, σ_{max}, at the nozzle/sphere junction, is determined from:

$$\sigma_{max} = \text{s.c.f.} \times \frac{PR}{2T'}$$

The s.c.f. is found from Figure G.2.5-1 using the parameters:

$$\rho = \frac{r}{R}\sqrt{\frac{R}{T'}} \text{ and } \frac{t}{T'}$$

W.A.2 Calculation

Using the design and material data from **W.6.2.3**:

$$\rho = \frac{r}{R}\sqrt{\frac{R}{T'}} = \frac{216}{1\,479} \times \sqrt{\frac{1\,479}{92}} = 0.59 \text{ and } \frac{t}{T'} = \frac{107}{92} = 1.163$$

From Figure G.2.5-1 for flush nozzle (and set-in nozzle, but not protruding) this gives an s.c.f. of 2.0

$$\left(\text{at } \frac{t}{T'} = 1\right)$$

As in example 2 [see **W.6.2.2.3**l)] the alternative procedure from Figure 3.5-10 or Table 3.5-3 for flush nozzles in spherical shells with $d/D < 0.5$ could be used to find the s.c.f. However, in this case the sizes chosen are outside the scope of the graphs. One could assume:

$$\frac{CT_r}{T} = 1 \text{ and if } K = \frac{2.25CT_r}{1.1T}, \text{ then } K = 2.045.$$

From an inspection of Figure G.2.5-1 this value seems marginally too high. The value of s.c.f. = K = 2 is therefore assumed.

Therefore σ_{max} for unit pressure p is:

$$\sigma_{max} = \text{s.c.f.} \times \frac{PR}{2T'} = 2.0 \times \frac{p \times 1\,479}{2 \times 92} = 16.08p$$

Annex W.B Calculation of cyclic stress range due to external nozzle loads

W.B.1 Method

The method proposed uses **G.2.5** which identifies the equations for determining the maximum stress, σ_{max}, in a flush nozzle under the three forms of loading defined.

W.B.2 Loading conditions

The nozzle experiences piping loads alternating between zero and the following values.

Bending moment	M = 38 kN·m;
Shear force	S = 36 kN;
Thrust force	Q = 29 kN;

W.B.3 Calculation

W.B.3.1 General

The stress range due to the mechanical loads M, S and Q can be found from **G.2.5**. The maximum stress, σ_{max}, at the nozzle/sphere junction, is determined from the product of an s.c.f. and a factor, each determined for the particular type of loading from **G.2.5.3**, **G.2.5.4** and **G.2.5.5**, using the parameters:

$$\rho = \frac{r}{R}\sqrt{\frac{R}{T'}} \text{ and } \frac{t}{T'}$$

in Figure G.2.5-3, Figure G.2.5-5 and Figure G.2.5-7.

In this case the parameters are:

$$\rho = \frac{216}{1\,479} \times \sqrt{\frac{1\,479}{92}} = 0.59$$

$$\frac{t}{T'} = \frac{107}{92} = 1.163$$

W.B.3.2 Maximum stress in the shell due to the bending moment *M*

From Figure G.2.5-5 s.c.f. = 1.2

$$\sigma_{max} = \text{s.c.f.} \times \frac{M}{\pi r^2 T'}\sqrt{\frac{R}{T'}}$$

$$\sigma_{max} = 1.2 \times \frac{38 \times 10^6}{\pi \times 216^2 \times 92}\sqrt{\frac{1\,479}{92}} = 13.56\ \text{N/mm}^2$$

W.B.3.3 *Maximum stress in the shell due shear force S*

From Figure G.2.5-7 s.c.f. = 1.2

$$\sigma_{\max} = \text{s.c.f.} \times \frac{S}{2\pi r T'}$$

$$\sigma_{\max} = 1.2 \times \frac{36 \times 10^3}{2\pi \times 216 \times 92} = 0.35\ \text{N/mm}^2$$

W.B.3.4 Maximum stress in the shell due to the thrust force *Q*

From Figure G.2.5-3 s.c.f. = 1.9

$$\sigma_{\max} = \text{s.c.f.} \times \frac{Q}{2\pi r T'}\sqrt{\frac{R}{T'}}$$

$$\sigma_{\max} = 1.9 \times \frac{29 \times 10^3}{2\pi \times 216 \times 92}\sqrt{\frac{1\ 479}{92}} = 1.77\ \text{N/mm}^2$$

W.B.3.5 Summation

A conservative estimate of the stress range occurring under the action of combined mechanical loading ($S_{\Delta m}$) may be obtained by adding the stresses from each of the individual loads together.

Thus:

$$S_{\Delta m} = 13.56 + 0.35 + 1.77 = 15.68\ \text{N/mm}^2$$

Annex W.C Calculation of transient thermal stress for a flush nozzle

W.C.1 Method

The method proposed uses **G.4** which gives a simplified method for determining the maximum stress, $\sigma_{\max}$, in a flush nozzle under transient thermal loading.

W.C.2 Calculation

W.C.2.1 General

The pressurized fluid is subject to a sudden rise in temperature of 225 °C in 440 s. The average heat transfer coefficients to shell (h_s) and branch (h_b) are 446 W/m^2·K and 767 W/m^2·K respectively. The specific heat of the material (c) is 499 J/kg·K, the density (ρ) is 7 800 kg/m^3 and the thermal conductivity (k) is 41.025 W/m·K.

The materials diffusivity, d, is found from:

$$d = \frac{k}{c\rho} = \frac{41.025}{499 \times 7\ 800} = 1.054 \times 10^{-5}\text{m}^2/\text{s}.$$

W.C.2.2 Calculation of the thermal factors (K_1, K_2, k_1, k_2 and K_d)

At the end of the transient θ = 440 s.

Shell

$$N = \frac{d \times \theta}{T^2} = \frac{1.054 \times 10^{-5} \times 440}{0.092^2} = 0.55$$

$$m = \frac{k}{h_s \times T} = \frac{41.025}{446 \times 0.092} = 1.00$$

From Figure G.4.3-3: K_1 = 0.275

From Figure G.4.3-4: K_2 = 0.120

From Figure G.4.3-5: K_s = 0.165

Branch

$$N = \frac{d \times \theta}{t^2} = \frac{1.054 \times 10^{-5} \times 440}{0.107^2} = 0.41$$

$$m = \frac{k}{h_b \times t} = \frac{41.025}{767 \times 0.107} = 0.5$$

From Figure G.4.3-3: k_1 = 0.410

From Figure G.4.3-4: k_2 = 0.175

From Figure G.4.3-5: k_b = 0.215

Since

$$\frac{r_o}{r_i} = \frac{216 + \frac{107}{2}}{216 - \frac{107}{2}} = 1.658$$

is greater than the limit of applicability (i.e. 1.1) given in **G.4.5** for the method used to derive k_1 the correction prescribed is required;

$$k_1(\text{corrected}) = k_1 \times \frac{216}{216 - \frac{107}{2}} = 0.410 \times 1.329 = 0.545$$

$K_d = K_b - K_s = 0.050$

W.C.2.3 *Calculation of the geometric factors* (C_1, C_2, C_3 and C_4)

$$\frac{R}{T} = 16.0; \frac{r}{R} = 0.145;\ Z = \frac{T}{t} = \ .860$$

Via interpolation it is found that:

from Table G.4.3-1: C_1 = 0.433

from Table G.4.3-2: C_2 = 0.051

from Table G.4.3-3: C_3 = 0.204

from Table G.4.3-4: C_4 = 0.344

W.C.2.4 *Calculation of the total thermal stress*

Using the procedures of **G.4.4.1** the stresses at the various locations of interest are found from the calculated geometric and thermal factors and the following parameters:

Rise in fluid temperature, T_f = 225 K

$E = 1.87 \times 10^5$ N/mm^2

$\alpha = 1.26 \times 10^{-5}$ 1/K

and thus $E \times \alpha \times T_f$ = 567 N/mm^2. The results are given in Table W.C.1.

Table W.C.1 **Thermal stress**

<table>
<tr><th colspan="2">Position component</th><th>Surface of component</th><th>Stress direction</th><th>Stress value N/mm²</th></tr>
<tr><td rowspan="8">Junction</td><td rowspan="4">Shell</td><td rowspan="2">Inner</td><td>Circumferential</td><td>–133.8</td></tr>
<tr><td>Meridional</td><td>–149.4</td></tr>
<tr><td rowspan="2">Outer</td><td>Circumferential</td><td>74.5</td></tr>
<tr><td>Meridional</td><td>56.4</td></tr>
<tr><td rowspan="4">Branch</td><td rowspan="2">Inner</td><td>Circumferential</td><td>–303.5</td></tr>
<tr><td>Longitudinal</td><td>–287.6</td></tr>
<tr><td rowspan="2">Outer</td><td>Circumferential</td><td>77.3</td></tr>
<tr><td>Longitudinal</td><td>91.4</td></tr>
<tr><td colspan="2" rowspan="4">branch at 0.62 $\sqrt{(rt)}$</td><td rowspan="2">Inner</td><td>Circumferential</td><td>–296.9</td></tr>
<tr><td>Longitudinal</td><td>–297.2</td></tr>
<tr><td rowspan="2">Outer</td><td>Circumferential</td><td>91.1</td></tr>
<tr><td>Longitudinal</td><td>101.0</td></tr>
</table>

W.C.3 Summary

Table W.C.1 above shows that the maximum stress in the component, -303.5 N/mm^2, occurs at the junction on inner surface of the branch in the circumferential direction.

Annex W.D Calculation of stresses due to the various sources

W.D.1 Simplified method

W.D.1.1 *Source 1: hydro-test at a pressure of 234 bar g.*

A conservative estimate of the stress range due to test pressure p_t is:

$$s_t = \frac{p_t}{p} \times 3f = \frac{224}{110} \times 3 \times 112.2 = 716.0 \text{ N/mm}^2$$

This conservative estimate assumes that a stress of $3f$ may exist at a detail in the vessel not considered by the stress analysis provided in Annex W.A.

W.D.1.2 *Source 2: upset condition of 110% of design pressure*

This treatment makes the same conservative approach as that used for source 1.

$p_u = 1.1 \times p = 1.1 \times 110 = 121$ bar g.

$\Delta p_u = p_u - p_{op} = 121 - 100 = 21$ bar g.

$$s_u = \frac{\Delta p_u}{p} \times 3f = \frac{21}{110} \times 3 \times 112.2 = 64.3 \text{ N/mm}^2$$

W.D.1.3 *Source 3: start ups and shut downs*

The stress relates to full pressure cycles between zero and maximum operating pressure. This treatment makes the same conservative approach to that used for source 1.

$$s_p = \frac{p_{op}}{p} \times 3f = \frac{100}{110} \times 3 \times 112.2 = 306.0 \text{ N/mm}^2$$

W.D.1.4 ***Source 4: fluid temperature cycles between 10 °C and 235 °C***

The pressurized fluid is assumed to be subject to a sudden increase in temperature of magnitude 225 °C. A conservative estimate of the stress range due to a change of temperature difference ΔT between adjacent point is given in Equation (C-3) as:

$S_r = 2E\alpha\Delta T$

The maximum value of ΔT that can occur is 225 °C, resulting in:

$S_{\Delta T} = 2 \times 1.87 \times 10^5 \times 1.26 \times 10^{-5} \times 225 = 1\,060.3\ \text{N/mm}^2$

In this case a more accurate value of this stress range is available from the thermal transient analysis in Annex W.C. However, the value is not used here so that the simplified method may be illustrated.

W.D.1.5 ***Source 5: pressure cycles between 90 bar g. and 100 bar g.***

This treatment makes the same conservative approach to that used in source 1.

$$S_{\Delta p} = \frac{\Delta p}{p} \times 3f = \frac{10}{110} \times 3 \times 112.2 = 30.6\ \text{N/mm}^2$$

W.D.1.6 ***Source 6: external load cycles***

Since no simplified method is available for the case of external loading on the nozzle, the stress range due to these mechanical loads are determined using **G.2.5**. The calculations are given in Annex W.B.

$S_{\Delta m} = 15.7\ \text{N/mm}^2$

W.D.2 Detailed assessment

W.D.2.1 ***Source 1: hydro-test 234 bar g.***

The stress range due to the application of pressure p can be determined using **G.2.5.2**. From the calculation, which can be found in Annex W.A, it follows that:

$S_t = 16.08 \times p_t$

Hence:

$S_t = 16.08 \times 23.4 = 376.3\ \text{N/mm}^2$

W.D.2.2 ***Source 2: upset condition of 110% of design pressure***

$p_u = 1.1 \times p = 1.1 \times 110 = 121$ bar g.

$\Delta p_u = p_u - p_{op} = 121 - 100 = 21$ bar g.

$S_u = 16.08 \times \Delta p_u = 16.08 \times 2.1 = 33.8\ \text{N/mm}^2$

W.D.2.3 ***Source 3: start ups and shut downs***

The stress relates to full pressure cycles between zero and maximum operating pressure:

$S_p = 16.08 \times 10 = 160.1\ \text{N/mm}^2$

W.D.2.4 ***Source 4: fluid temperature cycles between 10 °C and 235 °C***

The simplified method for assessing transient thermal stresses given in **G.4** is used to determine the stresses in the assembly at the end of the temperature rise. From this calculation, which can be found in Annex W.C, it follows that the maximum stress is 303.5 N/mm^2, hence:

$S_{\Delta T} = 303.5\ \text{N/mm}^2$

W.D.2.5 ***Source 5: pressure cycles between 90 bar g. and 100 bar g.***

The derived pressure stress equals 16.08*p*. Therefore the stress range due to a pressure differential Δp is:

$S_{\Delta p} = 16.08 \times \Delta p = 16.08 \times 1.0 = 16.1$ N/mm^2

W.D.2.6 ***Source 6: external load cycles***

This stress has already been calculated in Annex W.B and used in the simplified fatigue analysis.

$S_{\Delta m} = 15.7$ N/mm^2

References

[1] Niemi, E — "*Recommendations concerning stress determination for fatigue analysis of welded components*" — International Institute of Welding Doc No XIII, pp 1458-1492, 1994.

[2] British Standards Institution PD 6550-2:1989 — "*Openings and branch connections*".

[3] Decock, J — "*Determination of stress concentration factors and fatigue assessment of flush and extruded nozzles in welded pressure vessels*" — 2nd Intern. Conference on Pressure Vessel Tech, Part II, ASME, San Antonio, Texas, October 1973, Paper II–59, pp 821-834.

[4] British Standards Institution BS 7608:2014+A1:2015, "*Guide to fatigue design and assessment of steel products*".

Annex X Guidance for the tensile testing of 9% nickel steel weld metal using strain-gauged tensile specimens

X.1 General

The all-weld metal tensile specimen from a thin butt-welded test plate is of such a small cross-sectional area that there is large variation in results. These results are not representative of the actual joint strength.

Since weld metal tensile properties for 9% nickel steel do not exceed the tensile properties of the parent plate, it is necessary to obtain a value of weld metal proof stress and tensile stress to carry out an effective design.

The principle of this tensile test is that a localized spot, judged to be the weakest point in a welded joint, is measured while stressed in a similar manner to the service condition, i.e. transverse stress on a vertical joint due to the hoop stress.

Since design is based on yield stress and tensile stress, the proof stress is measured by means of small gauge-length strain gauges on a joint tensile specimen from a test plate welded in accordance with the welding procedure to be used on the vessel.

X.2 Method of testing

The method of testing should be as follows.

a) The test specimens should be the full thickness of the plate at the welded joint, and in accordance with Figure X.1.

b) The weld should be ground flush with the parent plate. There should be an adequate surface finish for strain gauge attachment.

c) Misalignment should be a minimum.

d) Flattening of the specimen should be carried out where necessary to achieve an angular misalignment of less than 1°.

e) The strain gauges should be attached, using a recognized strain gauge adhesive, in accordance with Figure X.2.

f) The strain gauges on either side should have equal length and characteristics. The maximum strain gauge length consistent with being completely within the weld metal surface width should be used.

g) Initially, the gauges should be monitored independently when employing a preload, to check the gauge bonding and the extent of induced bending. If satisfactory, the gauges should be wired up with two dummy gauges mounted on similar weld metal, to eliminate the bending strains.

h) Where there are strain gauges self-compensated for temperature, precision resistors may be used in the bridge circuit to replace the dummies. This should be in accordance with Figure X.3. The output is the sum of axial strain outputs from the two gauges on the specimen.

i) The specimen should be loaded and the change in resistance of the strain gauges measured with equivalent loads, from which a stress/strain curve should be obtained.

Figure X.1 **Dimensions of tensile test specimen**

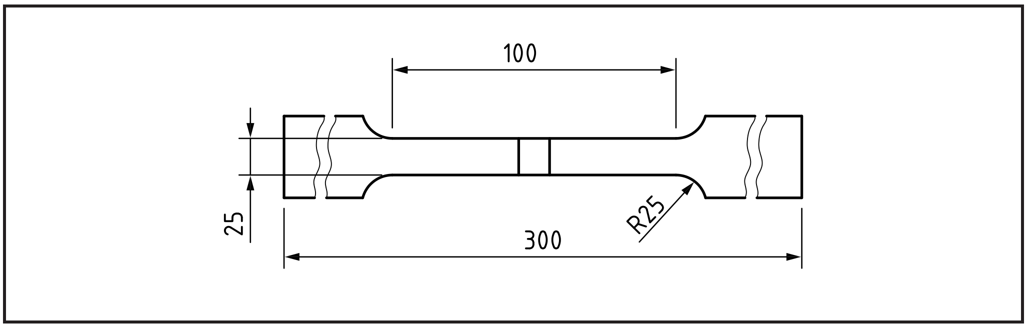


Figure X.2 **Method of attaching strain gauges**

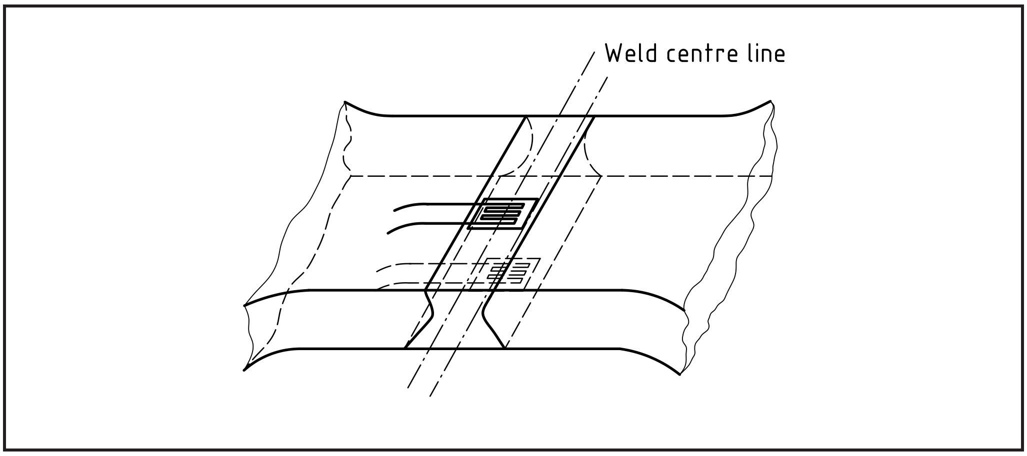
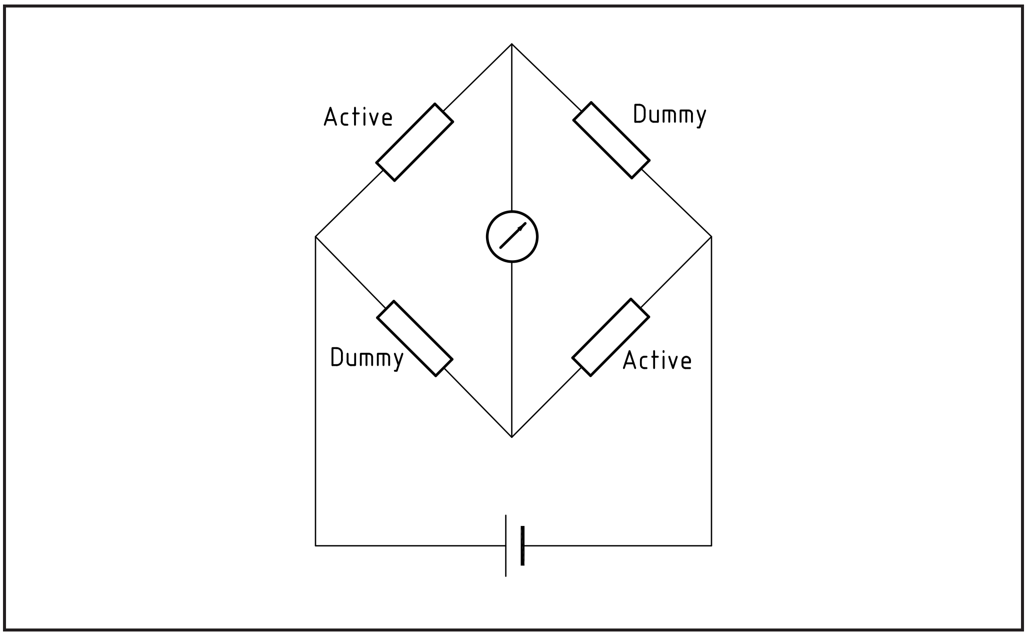


Figure X.3 **Test circuit**



Annex Z Guidance on the application of PD 5500 to pressure vessels falling within the scope of the UK Pressure Equipment (Safety) Regulations 2016 and the EU Pressure Equipment Directive

Z.1 General

The UK Pressure Equipment (Safety) Regulations 2016 (PER) [4] and the EU Pressure Equipment Directive, 2014/68/EU (PED) [1], place full responsibility upon the manufacturer of pressure equipment to comply with the provisions of the Regulations or the Directive. On completion of all the design, manufacturing, inspection and testing procedures, and after verification of conformity, the manufacturer can make a Declaration of Conformity with the PER [4] or the PED [1].

When this specification is being used to satisfy the requirements of the PER [4], as amended by Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019 [5], for pressure vessels placed on the market in Great Britain (England, Scotland and Wales), the manufacturer shall complete a Declaration of Conformity with the PER [4] and may then affix the UKCA mark when applicable. This ensures that the equipment has freedom of movement within Great Britain. For mandatory conformity assessments, the conformity assessment body shall be an approved body or user inspectorate appointed by the UK government. These requirements apply to pressure vessels manufactured in Great Britain or imported from outside the UK, but not to vessels manufactured in Northern Ireland.

For pressure vessels placed on the market in Great Britain and manufactured in Northern Ireland, special arrangements apply. The manufacturer shall complete a Declaration of Conformity with the PED [1] and may then affix the CE mark when applicable. If a notified body or user inspectorate appointed by a member state of the European Union is used for mandatory conformity assessments, the equipment has freedom of movement within Great Britain, Northern Ireland and the European Union. If an approved body or user inspectorate appointed by the UK government is used for mandatory conformity assessments, an additional UK(NI) mark is required which ensures that the equipment has freedom of movement in Great Britain and Northern Ireland, but not in the European Union.

When this specification is being used to satisfy the requirements of the PER [4], as modified by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020 [6] for pressure vessels placed on the market in Northern Ireland, the manufacturer shall complete a Declaration of Conformity with the PED and may then affix the CE mark when applicable. If a notified body or user inspectorate appointed by a member state of the European Union is used for mandatory conformity assessments, the equipment has freedom of movement within the UK and the European Union. If the conformity assessment body is an approved body or user inspectorate appointed by the UK government, an additional UK(NI) mark is required and the equipment has freedom of movement within the UK, but not the European Union. These requirements apply to pressure vessels manufactured in Northern Ireland or imported into Northern Ireland, including those manufactured in Great Britain. Equipment carrying only the UKCA mark is not valid for the Northern Ireland market.

When this specification is being used to satisfy the requirements of the PED for pressure vessels placed on the market in the EU, the manufacturer shall complete a Declaration of Conformity with the PED [1] and can then affix the CE mark when applicable. This ensures that the equipment will have freedom of movement within the European Union and Northern Ireland. For mandatory conformity assessments, the conformity assessment body shall be a notified body or user inspectorate appointed by a member state of the European Union.

The UKCA mark or CE mark can only be affixed to vessels that fall within the scope of the relevant regulations (PER [4] or PED [1]) and which conform with all the relevant requirements. The UKCA mark or CE mark shall not be affixed to vessels that are classified as being manufactured according to sound engineering practice – see Regulation 8 in Part 1 of the PER [1] and Article 4, paragraph 3 of the PED [4].

NOTE 1 CE marked pressure vessels that meet the requirements of the PED [1], while these match the requirements of the PER [4], can continue to be placed on the market in Great Britain until 31 December 2024.

NOTE 2 The UK government intends to extend recognition of the CE marking for placing pressure equipment on the market in Great Britain indefinitely, beyond December 2024.

Z.2 Compliance with essential safety requirements

An important requirement is that the manufacturer shall comply with all the Essential Safety Requirements (ESRs) in Schedule 2 of the PER [4] or Annex I of the PED [1], which are relevant to the equipment for which he makes the Declaration of Conformity.

Where this specification is used to satisfy requirements of the PER [4] or the PED [1], care should be taken to differentiate between the legal responsibility of the manufacturer under the Regulations or Directive and his contractual relationship with the purchaser.

Table Z.1 provides a summary of all the ESRs that are contained in Schedule 2 of the PER [4], as amended by Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019 [5], and Table Z.2 provides a summary of all the ESRs that are contained in Annex I of the PED [1]. These tables show, by reference to the relevant PD 5500 clause, where it is considered that PD 5500 satisfies the ESR. For materials covered by the material supplements the equivalent PD 5500 clause in the relevant material supplement should be used in place of the PD 5500 clause given in Tables Z.1 and Z.2.

This annex neither addresses the conformity assessment procedures required by the PER [4] or the PED [1] nor the involvement of the approved bodies or notified bodies.

PD 5500 is not a designated standard under the PER [1] nor a harmonized standard under the PED [1] and compliance with the clauses of this specification given in Tables Z.1 and Z.2 does not confer a presumption of conformity with the corresponding essential requirements of the PER [4] or the PED [1].

NOTE 1 The European Pressure Equipment Directive, 2014/68/EU [1], is no longer implemented within Great Britain, but is still implemented in Northern Ireland by the PER (statutory instrument SI 2016/1105)[4], as amended by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020.

NOTE 2 Attention is drawn to the existence of a European Commission Website (https://ec.europa.eu/growth/sectors/pressure-gas/pressure-equipment/directive_en) that gives the text of the PED and guidelines on its application.

NOTE 3 Attention is drawn to the existence of the following UK Government websites that give the text of the statutory instruments and guidance on their application:

a) *The Pressure Equipment (Safety) Regulations 2016: www.legislation.gov.uk/uksi/2016/1105*

b) *The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019: www.legislation.gov.uk/uksi/2019/696*

c) *The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020: www.legislation.gov.uk/uksi/2020/678/made*

Guidance:

a) *www.gov.uk/government/publications/pressure-equipment-safety-regulations-2016*

b) *www.gov.uk/guidance/placing-manufactured-goods-on-the-market-in-great-britain*

c) *www.gov.uk/guidance/using-the-ukca-marking*

d) *www.gov.uk/guidance/trading-and-moving-goods-in-and-out-of-northern-ireland*

e) *www.gov.uk/guidance/placing-manufactured-goods-on-the-eu-market*

f) *www.gov.uk/guidance/conformity-assessment-and-accreditation*

g) *www.gov.uk/government/publications/the-border-operating-model*

h) *www.gov.uk/guidance/ukca-marking-conformity-assessment-and-documentation*

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR**

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
PART 1	**GENERAL**	
1.	General requirements	
2.–(1)	Equipment to be designed, manufactured, checked, equipped and installed to ensure safety, when put into service in accordance with manufacturer’s instructions.	[a]
2.–(2)	Principles to be used by the manufacturer in assessing appropriate solutions to:	**N.3**
2.–(2)(a)	– eliminate or reduce hazards;	
2.–(2)(b)	– apply protection measures against hazards;	
2.–(2)(c)	– inform user of residual hazard.	
2.–(3)	Equipment to be designed to prevent danger from foreseeable misuse, or if not possible then adequate warning to be given against misuse.	[a]
PART 2	**DESIGN**	
	General	
3.–(1)	Equipment to be designed to ensure safety throughout intended life.	**3.1**
3.–(2)	Design to incorporate appropriate safety margins.	**3.2.2**
	Design for adequate strength	
4.–(1)	Equipment to be designed for loadings appropriate to its intended use and foreseeable operating conditions.	**3.2.1**
4.–(2)	Simultaneous loadings to be considered.	**3.2.1.4**

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
4.–(3)	Equipment to be designed for adequate strength based on a calculation method (see paragraph **5**) and to be supplemented, if necessary, by experimental methods (see paragraph **6**).	**3.2.2**
	Calculation method	
5.–(1)(a)	Allowable stresses to be limited with regard to foreseeable failure modes; safety factors to be applied.	**3.4.2**
5.–(1)(b)	Calculation methods to provide sufficient safety margins.	**3.2**
5.–(2)	Requirements to be met by applying one of the following methods:	
5.–(2)(a)	– design by formula;	**3.1**
5.–(2)(b)	– design by analysis;	Annex A[c]
5.–(2)(c)	– design by fracture mechanics.	Annex U[c]
5.–(3)(a)	Design calculations to establish the resistance of equipment, in particular:	**3.2**
5.–(3)(a)(i)	– calculation pressure to take account of maximum allowable pressure, static head, dynamic fluid forces and decomposition of unstable fluids;	**3.2.3**[c]
5.–(3)(a)(ii)	– partition walls to be designed for maximum differential pressure;	
5.–(3)(a)(iii)	– calculation temperature to allow appropriate safety margin;	**3.2.4**
5.–(3)(a)(iv)	– account to be taken of combinations of temperature and pressure;	**3.2.2**
5.–(3)(a)(v)	– maximum stresses and peak stresses to be within safe limits;	**2.3** and Annex A
5.–(3)(a)(vi)	– calculations to utilize values appropriate to material properties and to be based on documented data and have appropriate safety factors;	**2.3**, **3.4.2** and **Z.4**
5.–(3)(a)(vii)	– use of joint factors;	**3.4.1.** and **Z.3**
5.–(3)(a)(viii)	– to take account of foreseeable degradation mechanisms and attention to be drawn in instructions (see paragraph **30**) to features which are relevant to the life of equipment:	
5.–(3)(a)(viii)	(aa) creep;	**2.3.4**
5.–(3)(a)(viii)	(bb) fatigue;	Annex C
5.–(3)(a)(viii)	(cc) corrosion.	**3.3**
5.–(4)	Calculations to allow for adequate structural stability during transport and handling.	**3.2.1**

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
	Experimental design methods	
6.–(1) to (5)	Experimental design method. Design to be validated by appropriate test programme on representative sample of equipment. Test programme to include pressure strength test to check equipment does not exhibit leaks or deformation, exceeding determined threshold.	**3.2.2** and **5.8.6**[c]
	Provisions to ensure safe handling and operation	
7.–(1)	Provisions to ensure safe handling and operation. Methods to be specified for operation of equipment to preclude any foreseeable risk with attention being paid to:	[a]
7.–(1)(a)	– closures and openings;	
7.–(1)(b)	– discharge of pressure relief blow-off;	
7.–(1)(c)	– access whilst pressure or vacuum exists;	
7.–(1)(d)	– surface temperature;	
7.–(1)(e)	– decomposition of unstable fluids.	
7.–(2)	Access doors to be equipped with devices to prevent hazard from pressure.	**3.12**
	Means of examination	
8.–(1)	Pressure equipment to be designed and constructed so that examinations can be carried out.	**3.7.2.1** and **3.10.1.1**
8.–(2)	Means of determining internal condition to be available.	**3.12**
8.–(3)	Alternative means of ensuring safe condition.	[b]
	Means of draining and venting	
9.	Means to be provided for draining and venting where necessary:	[a]
9.(a)	– to avoid water hammer, vacuum collapse, corrosion and chemical reactions;	
9.(b)	– to permit cleaning, inspection and maintenance.	
	Corrosion or other chemical attack	
10.	Adequate allowance or protection provided against corrosion.	**3.3**
	Wear	
11.	Adequate measures to be taken against effects of erosion or abrasion by design, replacement parts and instructions (see paragraph **30**).	**3.3.4**[c]
	Assemblies	
12.	Assemblies to be designed so that:	[a]
12.(a)	– components to be assembled are suitable and reliable for their duty;	
12.(b)	– components are properly integrated and assembled.	

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
	Provisions for filling and discharge	
13.	Equipment to be provided with accessories, or provision made for their fitting, to ensure safe filling or discharge with respect to hazards.	[a]
13.(a)	– on filling; by overfilling or overpressurization and instability;	
13.(b)	– on discharge, by uncontrolled release of fluid;	
13.(c)	– on unsafe connection or disconnection.	
	Protection against exceeding the allowable limits of pressure equipment	
14.	Equipment to be fitted with, or provision made for, suitable protective devices, which are determined on the basis of characteristics of equipment.	**3.13**
	Safety accessories	
15.–(1)	Safety accessories to be designed and constructed to be reliable and suitable for intended duty, including maintenance; to be independent or unaffected by other functions.	**3.13.1**
	Pressure limiting devices	
16.	Pressure limiting devices to be designed so that design pressure will not be exceeded except for short duration pressure surges as specified in paragraph **39**.	**3.13.2** and Annex J
	Temperature monitoring devices	
17.	Temperature monitoring devices to have adequate response time on safety grounds.	[a]
	External fire	
18.	Where necessary pressure equipment to be designed and fitted with accessories to meet damage limitation requirements.	**1.1.5**[c]
PART 3	**MANUFACTURING**	
	Manufacturing procedures	
19.	Appropriate techniques and relevant procedures to be applied.	Section **4**
	Preparations of component parts	
20.	Preparations of component parts not to give rise to defects, cracks or changes in mechanical characteristics likely to affect safety.	**4.2**
	Permanent joining	**4.3**, **4.4**
21.–(1)	Permanent joints to be free from surface and internal defects detrimental to safety;	**5.6.4**
21.–(2)	Properties of permanent joints to meet minimum properties specified for materials to be joined or taken into account in design calculations;	**3.4.2**
21.–(3)	Permanent joining to be carried out by suitably qualified personnel according to suitable procedures;	**5.2** and **5.3**
21.–(3)(a) and (b)	Personnel and procedures to be approved by third party for Category **II**, **III** and IV equipment;	[a]

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
21.–(4)	Third party to perform examinations and tests as set out in designated standards, or have them performed.	**5.2** and **5.3**
	Non-destructive tests	
22.	Non-destructive tests of permanent joints to be carried out by suitably qualified personnel; Personnel to be approved by third party for Category **II**, **III** and IV equipment.	**5.6.1** [a]
	Heat treatment	
23.	Suitable heat treatment to be applied at appropriate stage of manufacture.	**4.5**
	Traceability	
24.	Materials making up component parts to be identified by suitable means from receipt, through production, up to final test.	**4.1.2**
	Final assessment	
25.	Equipment to be subject to final assessment in accordance with paragraphs **26** to **28**.	
	Final inspection	
26.–(1) **26.**–(2)	Final inspection to be carried out to assess visually, and by examination of documentation, compliance with the requirements of the Regulations. Tests carried out during manufacture may be taken into account. Inspection as far as possible to be carried out internally and externally on every part.	**5.8.10** **5.8.10**
	Proof test	
27.–(1) **27.**–(2) **27.**–(3)	Hydrostatic pressure tested to a pressure at least equal to value laid down in paragraph **40**. Category **I** series equipment may be tested on a statistical basis. Other tests, of equivalent validity, may be carried out where hydrostatic is harmful or impractical.	**5.8.3**[c] [a] **5.8.4**
	Inspection of safety devices	
28.	For assemblies, the safety devices to be checked to confirm compliance with paragraph **14**.	[a]
	Marking and labelling	
29.	In addition to UKCA marking, all relevant information to be provided as listed in paragraph **29**.	**1.4.4.** and **5.8.9**[c]
	Operating instructions	
30.	All relevant instructions to be provided for the user as listed in paragraph **30**.	[a]
PART 4	**MATERIALS**	
31.–(1)	Materials to be suitable for the application during the scheduled lifetime.	
31.–(2)	Welding consumables and joining materials to comply with subparagraphs **31.**–(3), (4)(a) and (5).	

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
31.–(3)	Material for pressure parts shall:	
31.–(3)(a)	– have appropriate properties for all operating and test conditions, and be sufficiently ductile and tough (see paragraph **41**);	**2.2**
31.–(3)(b)	– be chemically resistant to contained fluids and have properties not significantly affected within scheduled lifetime of equipment;	**3.3.1**
31.–(3)(c)	– not be significantly affected by ageing;	[b]
31.–(3)(d)	– be suitable for intended processing (manufacturing) procedures;	**2.1.1.1**
31.–(3)(e)	– be selected to avoid undesirable effects when put together.	**4.2.2.1**
31.–(4)(a)	Manufacturer to define material values necessary for design (see paragraph **5**) and for requirements of subparagraph **31.**–(3).	[b]
31.–(4)(b)	Manufacturer to provide technical documentation relating to compliance with material specifications of PER in one of the following forms:	**2.1.2.1**
31.–(4)(b)(i)	– by using materials which comply with designated standards;	
31.–(4)(b)(ii)	– not applicable;	
31.–(4)(b)(iii)	– by a particular material appraisal.	
31.–(5)	For equipment in Categories III or IV, the particular material appraisal to be performed by approved body in charge of conformity assessment.	[a]
31.–(6)	Equipment manufacturer to take appropriate measures to ensure that materials conform to the required specifications, and documentation by material manufacturer affirms compliance with a specification.	**4.1.2**[c]
31.–(7)	Documentation for main pressure parts in Categories II, III or IV to be a certificate of specific product control.	[a]
31.–(8)	Not applicable.	
PART 5	**SPECIFIC PRESSURE EQUIPMENT REQUIREMENTS**	
32.	Requirements of this part apply in addition to applicable requirements of Parts 1 to 4.	
33.	**Fired Or Otherwise Heated Pressure Equipment With A Risk Of Overheating** (including steam and hot-water generators, and process-heating equipment).	[a]
34.	**Piping**	[a]
PART 6	**SPECIFIC QUANTITATIVE REQUIREMENTS FOR CERTAIN PRESSURE EQUIPMENT**	
35.–(1)	Where the provisions of this part are not applied, manufacturer shall demonstrate that an equivalent overall level of safety has been achieved.	**Z.3** and **Z.4**
35.–(2)	The provisions of this part supplement the requirements of Parts 1 to 5.	

Table Z.1 **PD 5500 compliance with PER ESRs as amended by PSMR** *(continued)*

PER Schedule 2 Paragraph	PER Essential Safety Requirement	PD 5500 reference
	Allowable stresses	
36.–(1)	Definition of symbols.	
37.	Permissible general membrane stress.	**2.3** and **Z.4**
	Joint coefficients	
38.	Joint coefficients for welded joints.	**Z.3**
	Pressure limiting devices	
39.	The momentary pressure surge to be kept to 10% of design pressure.	**3.13.2** and Annex J
	Hydrostatic test pressure	
40.	Hydrostatic test pressure referred to in paragraph 27 to be not less than the greater of:	
40.(a)	– maximum loading to which equipment may be subject to in service, taking into account design pressure and design temperature, multiplied by 1.25; or	**5.8.5**
40.(b)	– design pressure multiplied by 1.43.	[b]
	Material characteristics	
41.	Unless other values are required in accordance with other criteria that shall be taken into account, a steel is considered sufficiently ductile to satisfy paragraph **31.**–(3)(a) if in a tensile test carried out by a standard procedure the following are met: – elongation after rupture is not less than 14%; – bending rupture energy measured on an ISO V test piece is not less than 27 J, at a temperature not greater than 20 °C but no higher than the lowest scheduled operating temperature.	[b]

[a] Outside the scope of the current edition of PD 5500.
[b] Requirements are not covered in the current edition of PD 5500.
[c] The PD 5500 reference clause only partially satisfies the relevant ESR.

Table Z.2 **PD 5500 compliance with PED ESRs**

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
1	**General**	
1.1	Equipment to be designed, manufactured, checked, equipped and installed to ensure safety, when put into service in accordance with manufacturer's instructions.	[a]
1.2	Principles to be used by the manufacturer in assessing appropriate solutions to: — eliminate or reduce hazards; — apply protection measures against hazards; — inform user of residual hazard.	**N.3**
1.3	Equipment to be designed to prevent danger from foreseeable misuse: — warning to be given against misuse.	[a]

Table Z.2 **PD 5500 compliance with PED ESRs** *(continued)*

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
2	**Design**	
2.1	Equipment to be designed to ensure safety throughout intended life and to incorporate appropriate safety coefficients.	**3.1**
2.2.1	Equipment to be designed for loadings appropriate to its intended use.	**3.2.1**
2.2.2	Equipment to be designed for adequate strength based on calculation method and to be supplemented, if necessary, by experimental methods.	**3.2.2**
2.2.3(a)	Allowable stresses to be limited with regard to foreseeable failure modes; safety factors to be applied. Requirements to be met by applying one of the following methods:	**3.4.2** and 3.2[c]
	— design by formula;	3.1
	— design by analysis;	Annex A[c]
	— design by fracture mechanics.	Annex U[c]
2.2.3(b)	Design calculations to establish the resistance of equipment, in particular:	**3.2**
	— calculation pressure to take account of maximum allowable pressure, static head, dynamic fluid forces and decomposition of unstable fluids;	**3.2.3**[c]
	— calculation temperature to allow appropriate safety margin;	**3.2.4**
	— account to be taken of combinations of temperature and pressure;	**3.2.2**
	— maximum stresses and peak stresses to be within safe limits;	**2.3** and Annex A
	— calculations to utilize values appropriate to material properties and to be based on documented data and have appropriate safety factors;	**2.3**, **3.4.2** and **Z.4**
	— use of joint factors;	**3.4.1** and **Z.3**
	— to take account of foreseeable degradation mechanisms and attention to be drawn in instructions to features which are relevant to the life of equipment.	
	— creep;	**2.3.4**
	— fatigue;	Annex C
	— corrosion.	**3.3**
2.2.3(c)	Calculations to allow for adequate structural stability during transport and handling.	**3.2.1**
2.2.4	Experimental design method. Design to be validated by appropriate test programme on representative sample of equipment. Test programme to include pressure strength test to check equipment does not exhibit leaks or deformation, exceeding determined threshold.	**3.2.2** and **5.8.6**[c]

Table Z.2 **PD 5500 compliance with PED ESRs** *(continued)*

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
2.3	Provisions to ensure safe handling and operation. Methods to be specified for operation of equipment to preclude any foreseeable risk with attention being paid to: — closures and openings; — discharge of pressure relief blow-off; — access whilst pressure or vacuum exists; — surface temperature; — decomposition of unstable fluids; — access doors equipped with devices to prevent hazard from pressure.	[a] **3.12**
2.4	Means of examination: — to be designed and constructed so that examinations can be carried out; — able to determine internal condition; — alternative means of ensuring safe condition.	 **3.7.2.1** and **3.10.1.1** **3.12** [b]
2.5	Means to be provided for draining and venting where necessary: — to avoid water hammer, vacuum collapse, corrosion and chemical reactions; — to permit cleaning, inspection and maintenance.	[a]
2.6	Adequate allowance or protection provided against corrosion.	**3.3**
2.7	Adequate measures to be taken against effects of erosion or abrasion by design, replacement parts and instructions.	**3.3.4**[c]
2.8	Assemblies to be designed so that: — components to be assembled are suitable and reliable for their duty; — components are properly integrated and assembled.	[a]
2.9	Equipment to be provided with accessories, or provision made for their fitting, to ensure safe filling or discharge with respect to hazards: — on filling; by overfilling or overpressurization and instability; — on discharge, by uncontrolled release of fluid; — on unsafe connection or disconnection.	[a]
2.10	Protection against exceeding the allowable limits. Equipment to be fitted with, or provision made for, suitable protective devices, which are determined on the basis of characteristics of equipment.	**3.13**
2.11.1	Safety accessories to be designed and constructed to be reliable and suitable for intended duty, including maintenance; to be independent or unaffected by other functions.	**3.13.1**
2.11.2	Pressure limiting devices to be designed so that design pressure will not be exceeded except for short duration pressure surges of 1.1 times design pressure.	**3.13.2** and Annex J

Table Z.2 **PD 5500 compliance with PED ESRs** *(continued)*

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
2.11.3	Temperature monitoring devices to have adequate response time on safety grounds.	[a]
2.12	External fire. Where necessary, pressure equipment to be designed and fitted with accessories to meet damage limitation requirements.	**1.1.5**[c]
3	**Manufacturing**	
3.1.1	Preparations of component parts not to give rise to defects, cracks or changes in mechanical characteristics likely to affect safety.	**4.2**
3.1.2	Permanent joining:	**4.3**, **4.4**
	— permanent joints to be free from surface and internal defects detrimental to safety;	**5.6.4**
	— properties of permanent joints to meet minimum properties specified for materials to be joined or taken into account in design calculations;	**3.4.2**
	— permanent joining to be carried out by suitably qualified personnel according to suitable procedures;	**5.2** and **5.3**
	— personnel and procedures to be approved by third party for Category II, III and IV equipment.	[a]
	— third party to perform examinations and tests as set out in harmonized standards, or have them performed	**5.2** and **5.3**
3.1.3	Non-destructive tests of permanent joints to be carried out by suitably qualified personnel:	**5.6.1**
	— personnel to be approved by third party for Category II, III and IV equipment.	[a]
3.1.4	Suitable heat treatment to be applied at appropriate stage of manufacture.	**4.5**
3.1.5	Traceability. Materials making up component parts to be identified by suitable means from receipt, through production, up to final test.	**4.1.2**
3.2.1	Final inspection to be carried out to assess visually, and by examination of documentation, compliance with the requirements of the Directive. Tests carried out during manufacture may be taken into account: — inspection as far as possible to be carried out internally and externally on every part.	**5.8.10**
3.2.2	Hydrostatic pressure tested to a pressure at least equal to value laid down in ESR 7.4.	**5.8.3**[c]
	— Category I series equipment may be tested on a statistical basis;	[a]
	— other tests, of equivalent validity, may be carried out where hydrostatic is harmful or impractical.	**5.8.4**
3.2.3	For assemblies, the safety devices to be checked to confirm compliance with ESR 2.10.	[a]
3.3	Marking. In addition to CE marking, all relevant information to be provided as listed in ESR 3.3.	**1.4** and **5.8.9**[c]
3.4	Operating instructions. All relevant instructions to be provided for the user as listed in ESR 3.4.	[a]

Table Z.2 **PD 5500 compliance with PED ESRs** *(continued)*

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
4	**Materials**	
4.1(a)	Materials to have appropriate properties for all operating and test conditions, to be sufficiently ductile and tough (refer to ESR 7.5).	**2.2**
4.1(b)	Materials to be chemically resistant to contained fluids: — properties not to be significantly affected within scheduled lifetime of equipment.	**3.3.1**
4.1(c)	Materials not to be significantly affected by ageing.	[b]
4.1(d)	Material to be suitable for intended processing (manufacturing) procedures.	**2.1.1.1**
4.1(e)	Materials to avoid undesirable effects when joining.	**4.2.2.1**
4.2(a)	Manufacturer to define material values necessary for design and for requirements of ESR 4.1.	[b]
4.2(b)	Manufacturer to provide technical documentation relating to compliance with material specifications of PED in one of the following forms: — by using materials which comply with harmonized standards; — by using materials covered by European approval in accordance with Article 11 of PED; — by a particular material appraisal.	**2.1.2.1**
4.2(c)	For equipment in PED Categories III or IV, the particular material appraisal to be performed by notified body in charge of conformity assessment.	[a]
4.3	Manufacturer to take appropriate measures to ensure: — materials conform to the required specifications; — documentation by material manufacturer affirms compliance with a specification; — documentation for main pressure parts in PED Categories II, III or IV to be a certificate of specific product control. — where material manufacturer has an appropriate quality-assurance system, certified by a competent body established within the Union, certificates issued by manufacturer give presumption of conformity.	 **4.1.2** [b] [a] [a]
5	**Fired Or Otherwise Heated Pressure Equipment With A Risk Of Overheating** (including steam and hot-water generators, and process-heating equipment)	[a]
6	**Piping**	[a]
7	**Specific Quantitative Requirements For Certain Pressure Equipment**	
7.1	Allowable stresses.	**Z.4**
7.2	Joint coefficients.	**Z.3**
7.3	Pressure limiting devices. The momentary pressure surge to be kept to 10 % of design pressure.	**3.13.2** and Annex J

Table Z.2 **PD 5500 compliance with PED ESRs** *(continued)*

PED Annex 1 Clause	PED Essential Safety Requirement	PD 5500 reference
7.4	Hydrostatic test pressure to be not less than the greater of: — maximum loading to which equipment may be subject to in service, taking into account design pressure and design temperature, multiplied by 1.25; or — design pressure multiplied by 1.43.	**5.8.5** b
7.5	Material characteristics. Unless other values are required in accordance with other criteria that shall be taken into account, a steel is considered sufficiently ductile to satisfy ESR **4.1**(a) if in a tensile test carried out by a standard procedure the following are met: — elongation after rupture is not less than 14%; — bending rupture energy measured on an ISO V test piece not less than 27 J, at a temperature not greater than 20 °C but no higher than the lowest scheduled operating temperature.	b

[a] Outside the scope of the current edition of PD 5500.

[b] Requirements are not covered in the current edition of PD 5500.

[c] The PD 5500 reference clause only partially satisfies the relevant ESR.

Z.3 PD 5500 construction categories and their relationship to the PER and PED joint coefficients

Z.3.1 Introduction

Pressure equipment, within a defined scope, that is to be placed on the market in the UK shall now conform to the provisions of the PER [4], modified as described in Z.1, and in the European Union it shall conform to the provisions of the PED [1]. BS EN 13445 gives a presumption of conformity to the essential safety requirements of the PER [4] and the PED [1] that are addressed by the standard, as listed in its Annex ZA. The use of PD 5500 does not provide that presumption of conformity but can be used for equipment within the scope of the PER [4] or the PED [1], subject to:

- adherence to the conformity assessment requirements of the PER [4] or the PED [1], as appropriate;
- the manufacturer satisfying himself that the pressure equipment conforms to all the relevant technical requirements of the PER [4] or the PED [1], as appropriate. Where these requirements are not covered by PD 5500 (see Table Z.1 or Z.2) they shallt be addressed by other means.

Part 1, Paragraph 6 of the PER [4], covering pressure equipment and assemblies subject to essential safety requirements, and Article 4 of the PED [1], covering technical requirements, state that equipment within its scope shall satisfy the essential safety requirements (ESRs) set out in Schedule 2 in the PER [4] or Annex I of the PED [1]. ESRs in Parts **1** to **5** of Schedule 2 in the PER [4] and Clause **1** to Clause **6** of that Annex I in the PED [1] cover general requirements, design, manufacturing, materials, fired equipment and piping. Part **6** of Schedule 2 in the PER [4] and Clause **7** of Annex I in the PED [1] supplement the other clauses and contain provisions that apply as a general rule. Where they are not applied the manufacturer shall demonstrate that the measures that have been taken achieve an equivalent overall level of safety.

Paragraph **38** of Schedule 2 in the PER [4] and Clause **7.2** in Annex I of the PED [1] deal with joint coefficients. This requires, for welded joints, the use of a coefficient of:

- 1 for equipment subject to full non-destructive testing (NDT);
- 0.85 for equipment subject to "random non-destructive testing";
- 0.7 for equipment not subject to NDT other than visual inspection.

The purpose of **Z.3** is to discuss the PD 5500 approach to Construction Categories and compare the resultant overall level of safety to that provided by joint factors in Schedule 2, Paragraph **38** of the PER [4] and Annex I, Section **7.2** of the PED [1].

Z.3.2 Background to use of joint coefficients in standards

The earliest national code for pressurized equipment was the American Society of Mechanical Engineers (ASME), Boiler and Pressure Vessel Code, ASME VIII [2]. This introduced the concept of joint coefficients that, effectively, for a given material and design condition, increased the thickness when less than full NDT was carried out. This concept was repeated in other national standards as they were developed, including the first UK national standard for pressure equipment, BS 1500, which used relatively conservative design stresses, similar to those used in ASME VIII. BS 1515 was then developed using higher design stresses, but still utilizing the joint coefficient concept.

ASME VIII, and most other national codes, use joint coefficients in an attempt to control any risk associated with partial NDT. A joint coefficient of less than 1.0 would increase the component thickness and thus reduce the membrane stress due to pressure loading in that component. PD 5500, for the first time, considered that increasing the thickness unnecessarily would increase secondary stresses caused by say thermal effects and would also increase the risk of building in a sub-surface defect that may not be picked up by partial NDT. Rather than increasing the thickness for partial NDT, PD 5500 imposed a restricted number of allowable materials having maximum component thicknesses and temperature limits.

These materials, thicknesses and temperatures were chosen to ensure good ductility so that any defects not picked up with any partial NDT would not represent a risk in normal operation. This approach was supported by UK industry.

Z.3.3 Background to PD 5500 construction categories

In the 1960s the UK pressure equipment industry faced difficulties exporting vessels, particularly to mainland Europe. The Ministry of Technology appointed a committee of enquiry on 9th December 1966, to recommend ways of improving pressure vessel technology including standards, design and manufacture. The report [3] published in 1969, recommended, amongst others things, that the British Standards codes be brought up to date and made more competitive. With the help of industry BSI developed and drafted BS 5500. This draft was reviewed and accepted by the pressure vessel industry, which included regulators, users, purchasers, consultants, designers, manufacturers and inspection authorities. The first issue of BS 5500 was in 1976 and since that date has become a de-facto International Standard. That standard, for the first time, replaced the concept of joint coefficients with the concept of construction categories.

The overall philosophy behind the construction category approach is summarized in the Note to **3.4.1**.

"Any one of the three construction categories in Table 3.4-1 will provide adequate integrity for normal purposes within the material and temperature limits specified therein."

It was accepted that the purchaser might wish to set a minimum construction category to provide additional integrity for a special reason and the note's full wording allows this option. The requirements associated with the three construction categories (i.e. full NDT, partial NDT and visual examination) are discussed in the following section.

Z.3.4 PD 5500 requirements related to construction categories

Table 3.4-1 defines the three categories of construction.

- Category 1 is non-restrictive, apart from requiring 100% NDT.
- Category 2 allows spot NDT but only permits a limited number of materials and thicknesses. It is intended that category 2 should be capable of covering a large proportion of industrial pressure vessels.
- Category 3 is the most restrictive on materials, thicknesses and temperatures; but by virtue of only requiring visual examination could produce cost effective vessels for simple applications. It should be noted that whilst design stresses would be the same for both categories 1 and 2 in a relevant material, Clause **3.4.2.2** requires a much lower design stress for a category 3 component. This clause, for example, limits the design stress of carbon steel to $\frac{R_m}{5}$ rather than the lower of $\frac{R_m}{2.35}$ or $\frac{R_e}{1.5}$ for categories 1 or 2 (see **2.3.3.2**). This reduction in design stress for category 3 components can be considered equivalent to a "joint coefficient" and by inspection would be more conservative than the joint coefficient of 0.7 prescribed by the PER or the PED for components only subject to visual inspection.

  *NOTE There is a difference between category 1 and 2 requirements in Section 4 of PD 5500 "Manufacture and workmanship". This difference, in **4.3.6.2**, relates to the use of backing strips for welds.*

For category 3 components, Table 5.1-1 of PD 5500, "Inspection and testing", lists the inspection stages where participation by the Inspecting Authority is required. The manufacturer is required to examine cut edges and heat affected zones before welding, to examine the set up of seams for welding and to inspect the second side of weld preparations after the first side weld is complete. Independent examination of these operations by the Inspecting Authority may be required for category 3 components, at the discretion of this authority.

Clause **5.6.4** of PD 5500 spells out the different requirements for NDT of welded joints, both for examination for internal flaws and surface flaws. Acceptance criteria for any weld defects revealed by visual examination or by NDT are different for the categories and are specified in **5.7** of PD 5500.

Z.3.5 Conclusions

It is considered that the PD 5500 approach to risk avoidance with partial NDT is more logical than the approach used in other national standards. PD 5500 restricts the materials, thicknesses and temperatures so that the pressure vessel component is tolerant to any defect missed by the partial NDT. The approach to risk avoidance with partial NDT used in other standards and the PER and PED is to use a joint coefficient so as to increase the thickness and thus reduce the membrane stress.

Any unnecessary increase in component thickness has been avoided in PD 5500 as it was felt that this would increase secondary stresses and increase the risk of building in defects.

Differences in specified requirements exist for the different construction categories, as indicated in **Z.3.4**, which provide protection against the occurrence or propagation of any defect.

Since 1976, PD 5500 has been increasingly used world-wide and British Standards has not been made aware of any pressure vessel component failure that could be attributed to the construction category approach.

It is concluded that PD 5500 construction categories provide a logical and safe philosophy for pressure vessel design, construction, testing and operation. The approach provides an equivalent overall level of safety to that provided by the PER [4] and PED [1] joint coefficient approach, as defined by the Essential Safety Requirements in the PER [4] in Schedule 2, Paragraph **38** and in the PED [1] in Annex **I**, Clauses **7** and **7.2**.

Z.4 Allowable stresses

Clause **2.3** of PD 5500:2024 gives factors to be applied to the specified yield strength and ultimate tensile strength of a material to enable the calculation of allowable stresses. These factors are marginally different from those given in Paragraph **37** of Schedule 2 of the PER [4] and Clause **7.1** of Annex **I** of the PED [1]. However, the PD 5500 factors have been used for many years and will provide an equivalent overall level of safety to that provided by the PER [4] and PED [1] factors.

References

Standards publications

BS 1500, *Fusion welded pressure vessels for use in the chemical, petroleum and allied industries*.

BS 1515, *Specification for fusion welded pressure vessels for use in the chemical, petroleum and allied industries*.

BS 5500:1976, *Specification for unfired fusion welded pressure vessels*.

Other documents

[1] EUROPEAN COMMUNITIES. Directive, 2014/68/EU (PED) of the European Parliament and of the Council of 15 May 2014 on the harmonisation of the laws of the Member States relating to the making available on the market of pressure equipment (Official Journal of the European Union L 189/164, 27.6.2014).

[2] UNITED STATES OF AMERICA. Boiler and Pressure Vessels Code, Section VIII. American Society of Mechanical Engineers.

[3] GREAT BRITAIN. Report of the Committee of Enquiry on Pressure Vessels, 1969. London: The Stationery Office.

[4] GREAT BRITAIN. The Pressure Equipment (Safety) Regulations 2016. London: The Stationery Office, 2016.

[5] GREAT BRITAIN. The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019. London: The Stationery Office, 2019.

[6] GREAT BRITAIN. The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020. London: The Stationery Office, 2020.

Aluminium supplement

Requirements for aluminium and aluminium alloys in the design and construction of unfired pressure vessels

This supplement shall be read in conjunction with the main body of the specification to establish the requirements for aluminium and aluminium alloy unfired pressure vessels. This supplement lists the sections of the main text applicable to and those not applicable to the design and construction of aluminium and aluminium alloy pressure vessels. In addition the supplement contains clauses specific to such vessels which replace the corresponding clauses of the main text in this context. The clause, equation, figure and table numbering of the requirements of this supplement follow those of the comparable main text requirements but are identified in the form **Al.x.y.z** to differentiate them from their main text equivalents, i.e. **x.y.z**, which might be technically different.

Al.1 Section 1: General

See main text.

Al.2 Section 2: Materials

Al.2.1 Selection of materials

Al.2.1.1 General

See main text, but subsitute.

Al.2.1.1.3 For the ease of reference in this supplement, materials are identified by a group number which has been derived from PD CEN ISO/TR 15608:2017, Table 2. This grouping is summarized in Table Al.2.1-1.

Table Al.2.1-1 **Material grouping**

Group	Sub-group	Type of aluminum and aluminium aloys
21	–	Pure aluminium ≤ 1% impurities or alloy content
22	–	Non heat treatable alloys
	22.1	Aluminium-manganese alloys
	22.2	Aluminium-manganese alloys with Mg ≤ 1.5%
	22.3	Aluminium-manganese alloys with 1.5% < Mg ≤ 3.5%
	22.4	Aluminium-manganese alloys with Mg > 3.5%
23	–	Heat treatable alloys
	23.1	Aluminium-manganese-silicon alloys
	23.2	Aluminium-zinc-manganese alloys

NOTE Materials in groups 24, 25 and 26 in PD CEN ISO/TR 15608:2017, Table 2 are not covered by this supplement.

Al.2.1.2 Materials for pressure parts

Al.2.1.2.1 General

See main text but substitute

For the construction of vessels in accordance with this supplement, materials shall satisfy the requirements of BS EN 12392:2016+A1:2022, and non-destructive testing requirements shall be as specified in **Al.5.6.2**. All the materials used in the manufacture of pressure parts shall:

a) conform to British Standards listed in Table Al.2.3-1; or

b) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2b)**] and shall conform to **Al.2.1.2.2**; or

c) be the subject of a "European approval of material" in accordance with the EU directive 2014/68/EU (see Note); or

d) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Materials for bolts and nuts shall conform to the specifications listed in Table Al.3.8-1.

Al.2.1.2.2 Materials covered by Al.2.1.2.1b)

Al.2.1.2.2.1 Other materials as specified in **2.1.2.1**b) shall comply with the general requirements of **Al.2.1.2.2.2**, **Al.2.1.2.2.3** and **Al.2.1.2.2.4**.

Al.2.1.2.2.2 The material specification shall specify the composition limits for all constituents, heat treatment and the appropriate mechanical properties for acceptance and other purposes.

Al.2.1.2.2.3 Mechanical properties at room temperature shall be specified for acceptance tests in accordance with BS EN ISO 6892-1:2019 covering:

a) the tensile strength range;

b) the minimum 0.2% proof stress ($R_{p0.2}$);

c) the specified minimum percentage elongation at fracture, referred to a gauge length of 5.65 $\sqrt{S_o}$ [1)] or 50 mm, shall be appropriate to the type of material with a lower limit of:

1) 14% for material that may be subject to further deformation/forming (i.e. shell plates, end plates, tube for bending);

2) 9% for material that is not subject to further deformation/forming (i.e. flanges, castings).

Al.2.1.2.2.4 For materials that will be used at temperatures above 50 °C, tensile data shall be provided from which the expected minimum tensile strength and minimum 0.2% proof strength at the operating temperature can be established. If the operating temperature equals or exceeds 100 °C, stress rupture data shall be available for determining the design strength and design lifetime.

Al.2.1.2.3 Additional materials for category 3 components

Main text is not applicable.

Al.2.1.2.4 Aluminium magnesium alloys

Pressure vessels in aluminium alloys containing 3.0% or more of magnesium for use at temperatures above 65 °C shall be constructed only from material supplied in the annealed (0) condition.

[1)] S_o is the original cross-sectional area of the gauge length of the tensile test specimen.

NOTE Extended service of alloys containing 3.0% or more magnesium at temperatures above 65 °C can result in grain boundary precipitation of Mg-Al intermetallic compounds which corrode in some process fluids leading to disintegration in weld areas. Alloys of this type should not be used at temperatures above 65 °C unless tests or service experience have demonstrated that they are suitable for specific duty.

Al.2.1.3 Materials for non-pressure parts

See main text.

Table Al.2.3-1 **Design strength values (N/mm²): aluminium and aluminium alloys**

Material standards, BS EN 485 BS EN 515, BS EN 573, BS EN 586, BS EN 603, BS EN 604, BS EN 754 and BS EN 755	Condition	Group	Minimum tensile strength[b] R_m N/mm²	Minimum 0.2% proof stress[b] $R_{p0.2}$ N/mm²	Values of f for design temperatures (°C) not exceeding[c]						
					50	75	100	125	150	175	200
1050A	0	21	55	—	11	11	10	10	10	9	8
3103	0	22.1	90	—	23[f]						
3003	0	22.1	96	34	23	23	23	21	17	12	10
5251	0	22.3	160	60	40	40	40	40	33	29	22
5454	0	22.3	215	80	53	53	53	52	34	28	22
5154A[e]	0	22.3	215	85	57	57[f]					
5083[e]	0	22.4	275	125	83	83[f]					
6061[g]	T6	23.1	280	225	93	93	92	86	74	54	41
6061[g]	T6 welded	23.1	165	—	55	55	55	54	51	43	32
6063[gh]	T6	23.1	185	160	62	60	58	51	38	27	15
6063A	T6	23.1	230	190	77	75	72	63	47	33	19
6063[g]	T6 welded	23.1	120	—	40	40	40	38	36	22	14
6063A	T6 welded	23.1	120	—	40	40	40	38	36	22	14
6082	T6	23.1	280	240	93	88	83	71	60	47	33
6082	T6 welded	23.1	165	—	55	55	55	54	51	43	32

[a] Not used.

[b] See **Al.2.1.2.2.4**.

[c] **Al.2.3.1.3** applies.

[d] Not used.

[e] See **Al.2.1.2.4**.

[f] See **Al.2.1.2.4**. 5154A and 5083 materials should not be used at temperatures above 65 °C, and 3103 should not be used at temperatures above 50 °C unless tests or service experience have demonstrated that they are suitable for the specific duty.

[g] These alloys are suitable only for tube type welded attachments (e.g. weld neck flanges) not subject to severe weld restraint.

[h] Extrusions and forgings up to 150 mm thick.

Al.2.2 Materials for low temperature applications

Aluminium and aluminium alloys are not susceptible to brittle fracture and no special requirements are necessary for their use at temperatures down to −196 °C.

Al.2.3 Nominal design strength

Al.2.3.1 General

Al.2.3.1.1 The design strength values given in Table Al.2.3-1 are appropriate to materials, thicknesses and form as specified in the relevant British Standards listed. It is permissible to use other thicknesses of the same form or other product forms, the minimum tensile properties being established either by reference to the material specification or by arrangement with the material supplier. If the minimum values of $R_{p0.2}$ and R_m are less than those given in Table Al.2.3-1 for the same temper or condition, the design strengths that are not time-dependent shall be reduced proportionately as follows:

a) materials 1050A, 3103, 6060/6063 and 6082 in ratio of actual minimum R_m/R_m in Table Al.2.3-1;

b) other materials in ratio of actual minimum $R_{p0.2}/R_{p0.2}$ in Table Al.2.3-1.

Al.2.3.1.2 Design strengths given in Table Al.2.3-1 were determined as follows:

a) time-independent design strength: $R_{p0.2}/1.5$ or $R_m/3$ whichever is the lower;

b) time-dependent design strength: S_{Rt} (100 000)/1.3.

These are criteria relevant to the annealed materials listed for welded construction.

NOTE 1 $R_{p0.2}$ is the expected minimum value determined by the following equation:

$$R_{p0.2}\ \text{min} = R_{p0.2}\ \text{sample} \times \frac{R_e\ \text{min}}{R_e\ \text{sample}} \qquad (\text{Al.2.3.1-1})$$

NOTE 2 S_{Rt} (100 000) was obtained by extrapolating 10 000 h test data when available. Other values have been obtained from relevant experience and other codes of practice.

Al.2.3.1.3 The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined in accordance with **Al.2.3.1.1** or **Al.2.3.1.2**. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

Al.3 Section 3: Design

Al.3.4 Construction categories and design stresses

Al.3.4.1 Construction categories

See main text with the following modified Table Al.3.4-1

Table Al.3.4-1 **Construction categories**

Construction category	Nondestructive testing (NDT)	Permitted groups and subgroups	Maximum nominal thickness of component[a] (see 1.6) mm	Temperature limit	
				Upper	Lower
1	100% (see **Al.5.6.4.1**)	All	None, except where NDT method limits	See **Al.2.3.1.3**	None
2	Limited random (spot) (see **Al.5.6.4.2**)	21, 22.1, 22.3 and 22.4	40	See **Al.2.3.1.3**	None
3	Visual only (see **Al.5.6.4.3**)	22.3 and 22.4, except alloy EN AW 5454	16	65 °C	None

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension of the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.

Al.3.4.2 Design stresses

Al.3.4.2.1 Categories 1 and 2

The design stresses for British Standard materials shall not exceed the appropriate nominal design strength value given in Table Al.2.3-1 for the material of construction at the design temperature. For welded construction, the nominal design strengths given in Table Al.2.3-1 for materials in the annealed condition, shall be used.

The design stresses for materials other than those listed in Table Al.2.3-1 shall be derived in accordance with **Al.2.3.1.2**.

Al.3.4.2.2 Category 3

The design stress shall not exceed $R_m/5$ irrespective of the orientation of the main weld seam (for definition of R_m see **2.3.2**). Main welded seams are defined as type A welds (see Figure 5.6-1).

Al.3.5 Vessels under internal pressure

See main text and additionally

Al.3.5.4.7 Nozzle and nozzle pipe minimum thickness

Add

NOTE It is recommended that nozzles of up to 80 mm nominal size in aluminium vessels should be forged or machined from wrought material, as indicated in Figure E.30, types (i), (ii) or (iii), in preference to pipe connections welded directly to the shell.

Al.3.6 Vessels under external pressure

Al.3.6.1 General

Add

In the light of current experience, materials for vessels subject to external pressure shall be restricted to materials 3103, 5154A, 5083 and 5454.

NOTE In view of the lack of appropriate data it is recommended that the use of 5154A and 5083 materials be restricted to below 65 °C and the use of 3103 materials be restricted to 50 °C.

Al.3.6.1.1 Notation

See main text but substitute

E is the modulus of elasticity of material of part under consideration at design temperature from Table Al.3.6-3;

s is the factor relating f to effective yield point of material; for the purposes of **Al.3.6**, s shall be taken to be 1.1.

Table Al.3.6-3 **E values for aluminium alloys (modulus of elasticity)**

Temperature °C	Aluminium N/mm^2
–200	76.6×10^3
–20	70.5×10^3
0	69.9×10^3
20	69.3×10^3
100	67.4×10^3
150	65.4×10^3
200	62.3×10^3

Al.3.8.1 General

See main text and additionally:

Circular bolted flanges shall either:

a) conform to the requirements of **3.8** of the main text for individually designed flanges; or

b) conform to BS EN 1092-4 or BS EN 1759-4 for aluminium alloy standard pipework flanges and be of the appropriate rating.

NOTE In selecting gasket materials for use with aluminium alloy flanges account should be taken of the relative hardness values of the gasket and flange materials.

Al.3.8.1.4 General requirements for bolting

See main text but substitute

If aluminium bolts or studs are used, special attention shall be given to the risk of fracture through over-tightening.

Table Al.3.8-1 **Recommended design stress values for flange bolting materials**

Material	BS references	Diameter	Recommended design stress (N/mm^2) for design metal temperatures (°C) not exceeding:						
		mm	50	75	100	125	150	175	200
Aluminium alloy Al Si Mg Mn	BS 1473:1972 grade HB 30 condition TF	All	57	56	54	50	43	34	24
Aluminium alloy Al Cu 4 Si Mg	BS 1473:1972 grade HB 15 condition TF	All	87	83	78	70	48	31	22
Aluminium Al Mg 5	BS 1473:1972 grade NB 6 condition H4	All	75	—	—	—	—	—	—

Al.4 Section 4: Manufacture and workmanship

Al.4.1 General aspects of construction

Al.4.1.1 General

See main text.

Al.4.1.2 Material identification

See main text.

Al.4.1.3 Order of completion of weld seams

See main text.

Al.4.1.4 Junction of more than two weld seams

Main text is not applicable.

Al.4.2 Cutting, forming and tolerances

Al.4.2.1 Cutting of material

Al.4.2.1.1 Method

All material shall be cut to size and shape preferably by machining, chipping or plasma-arc cutting. However, for plates less than 25 mm thick, it is permissible to use cold shearing provided that the cut edges are dressed back mechanically by not less than 1.5 mm to provide a suitable surface to permit a satisfactory examination of the edges prior to welding. It is permissible for plates less than 10 mm thick, which are cold sheared, not to be dressed where the cut edges are to be subsequently welded.

Surfaces that have been plasma-arc cut shall be dressed back by machining to remove severe notches and scale.

Edges that are plasma-arc cut shall be dressed back by machining for a distance of 1.5 mm unless the manufacturer can demonstrate to the satisfaction of the Inspecting Authority that the material has not been adversely affected by the cutting process.

Al.4.2.1.2 Examination of cut edges

See main text.

Al.4.2.2 Forming of shell sections and end plates

Al.4.2.2.1 General

Prior to forming, a visual examination of all plates shall be carried out, followed by measurement of the thickness.

Plates shall be formed to the required shape by any process provided that the quality of the material is not impaired. It is permissible to apply an effective heat treatment following the forming operation to restore the mechanical properties to their specified values.

The manufacturer may be required to demonstrate that the forming and heat treatment operations have not rendered the material unsuitable for the intended service.

As far as is practicable, all hot and cold forming shall be done by machine; local heating or hammering shall be used only by agreement between the manufacturer, purchaser and Inspecting Authority.

Lubricant remaining after any forming operation shall be removed by a suitable chemical cleaning process that will not impair the quality of the material.

Al.4.2.2.2 Plates welded prior to hot or cold forming

It is permissible to butt weld plates together prior to forming provided that the joint is non-destructively tested after forming by the same method as used for the final butt welds.

Al.4.2.2.3 Cold forming

If the inside radius of curvature of a pressure part is less than 10 times the thickness, an appropriate heat treatment shall be considered in order to restore properties to levels that will ensure that the material properties are not significantly different for those assumed in the design.

Al.4.2.2.4 Forming

Al.4.2.2.4.1 Aluminium hot forming

Aluminium plates to be treated or hot worked shall be heated uniformly in a neutral or oxidizing atmosphere, without flame impingement, to a temperature not exceeding 450 °C.

Deformation shall not be carried out after the temperature of the material has fallen below 300 °C. Local heating shall not be used.

Al.4.2.2.4.2 Aluminium cold working

It is permissible to soften aluminium that has been cold worked when the manufacturer, purchaser and Inspecting Authority agree that the extent of the cold working is sufficient to necessitate treatment.

The requirements for any softening treatments shall be subject to agreement between the manufacturer, purchaser and Inspecting Authority.

Al.4.2.2.5 Manufacture of shell plates and ends

See main text.

Al.4.2.2.6 Examination of formed plates

See main text.

Al.4.2.3 Assembly tolerances

See main text.

Al.4.2.4 Tolerances for vessels subject to internal pressure

See main text.

Al.4.2.5 Tolerances for vessels subject to external pressure

See main text.

Al.4.3 Welded joints

Al.4.3.1 General

See main text.

Al.4.3.2 Welding consumables

Welding consumables (e.g. wire, electrodes, flux, shielding gas) shall be the same type as those used in the welding procedure. Filler rods and wires shall comply with BS EN ISO 18273 and shall be stored in accordance with the suppliers' recommendations. The selection of filler rod or wires shall be appropriate to the parent alloy(s) and shall be in accordance with BS EN 1011-4.

In all cases where filler metals do not match parent metal compositions, or where alternative filler metals are to be used, the manufacturer shall be able to demonstrate that the combination used is suitable for the specified conditions.

Al.4.3.2.1 *Main text is not applicable.*

Al.4.3.2.2 *Main text is not applicable.*

Al.4.3.2.3 *Main text is not applicable.*

Al.4.3.2.4 *Main text is not applicable.*

Al.4.3.3 Preparation of plate edges and openings

Al.4.3.3.1 *See main text.*

Al.4.3.3.2 *See main text.*

Al.4.3.4 Assembly for welding

See main text.

Al.4.3.5 Attachments and the removal of temporary attachments

Al.4.3.5.1 Attachments

See main text.

Al.4.3.5.2 Removal of attachments

See main text.

Al.4.3.5.3 Attachments of dissimilar aluminium alloys

It is permissible to weld dissimilar aluminium alloy attachments directly to a pressure component. Compatible filler metals shall be used in an approved procedure.

Al.4.3.6 Butt joints

Al.4.3.6.1 Butt welds between plates of unequal thickness

See main text.

Al.4.3.6.2 Backing strips

Permanent backing strips shall not be used for longitudinal welds. It is permissible to weld circumferential butt joints in tubes with temporary, permanent or consumable backing rings only by agreement between the manufacturer, purchaser and Inspecting Authority.

Where a backing strip is to be used, the material shall be such that it will not adversely influence the weld. Backing strips shall be carefully removed prior to any special non-destructive tests on the joint.

Al.4.3.7 Welding: general requirements

Al.4.3.7.1 All fusion faces shall be thoroughly cleaned of oil or other foreign substances and oxide films removed to give a clean metal surface. Such cleaning shall extend for a distance of 12 mm from the edge of each fusion face.

Filler materials for TIG-welding shall be cleaned immediately before use. Filler wire for MIG-welding shall be protected from contamination during use and, in particular, between shifts.

Al.4.3.7.2 *See main text.*

Al.4.3.7.3 Each run of weld metal shall be thoroughly cleaned before the next run is deposited. All scratch brushes shall be of stainless steel and shall be used only on aluminium.

Al.4.3.7.4 *See main text.*

Al.4.3.7.5 Arcs shall be struck only where weld metal is to be applied or in the fusion path.

Al.4.3.7.6 *See main text.*

Al.4.3.7.7 Not less than two layers of weld metal shall be deposited at each weld attaching branch pipes, flanges and pads except where the particular welding procedure has been agreed between the manufacturer, purchaser and Inspecting Authority.

Al.4.3.7.8 When welding stops for any reason, care shall be taken when restarting to ensure proper fusion and penetration between the weld metal and previously deposited weld metal.

Al.4.4 Permanent joints other than welding

See main text.

Al.4.5 Heat treatment

Al.4.5.1 Preheat requirements

Al.4.5.1.1 Heating prior to welding aluminium is not normally considered necessary. Where preheating is required it shall be specified in the weld procedure. The preheat temperature depends upon the type of joint, the metal thickness, the alloy and the heat input to each weld run.

NOTE As a general guide temperatures in excess of 150 °C should not be necessary.

Al.4.5.1.2 The temperature shall be checked during the period of application using appropriate methods (e.g. thermocouples, contact pyrometers or temperature indicating coatings). Where coatings are employed they shall not be applied to fusion faces.

Al.4.5.1.3 *Main text is not applicable.*

Al.4.5.1.4 *Main text is not applicable.*

Al.4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

Main text is not applicable.

Al.4.5.3 Post-weld heat treatment

Al.4.5.3.1 For aluminium the details of any post-weld heat treatment shall be agreed between the manufacturer, purchaser and Inspecting Authority.

NOTE Stress relieving heat treatment is not normally necessary or desirable for aluminium pressure vessels.

Al.4.5.3.2 *Main text is not applicable.*

Al.4.5.3.3 *Main text is not applicable.*

Al.4.5.3.4 *Main text is not applicable.*

Al.4.5.3.5 *Main text is not applicable.*

Al.4.5.4 Methods of heat treatment

Main text is not applicable.

Al.4.5.5 Post-weld heat treatment procedure

Main text is not applicable.

Al.4.6 Surface finish

See main text.

Al.5 Section 5: Inspection and testing

Al.5.1 General

Each pressure vessel shall be inspected during construction. Sufficient inspections shall be made to ensure that the materials, construction and testing comply in all respects with this specification. Inspection by the Inspecting Authority shall not absolve the manufacturer from his responsibility to exercise such quality assurance procedures as will ensure that the requirements and intent of this specification are satisfied.

The Inspecting Authority shall have access to the works of the manufacturer at all times during which work is in progress, and shall be at liberty to inspect the manufacture at any stage and to reject any part not complying with this specification. The Inspecting Authority shall have the right to require evidence that the design complies with this specification.

The Inspecting Authority shall notify the manufacturer before construction begins regarding the stages of the construction at which special examinations of materials will be made, and the manufacturer shall give reasonable notice to the Inspecting Authority when such stages will be reached, but this shall not preclude the Inspecting Authority from making examinations at any other stages, or from rejecting material or workmanship whenever they are found defective.

NOTE Table Al.5.8-1 is included in this supplement for guidance purposes only.

Al.5.2 Approval testing of fusion welding procedures

Al.5.2.1 Approval testing of welding procedures shall be conducted, recorded and reported in accordance with BS EN ISO 15614-2 except as stated in **Al.5.5**.

For fusion welding methods other than MIG and TIG (e.g. plasma arc or electron beam) the general principles of BS EN ISO 15614-2 shall be complied with.

Al.5.2.2 The manufacturer shall supply a list of all the welding procedures required in the fabrication of the vessel, together with test pieces which are representative of the various thicknesses and materials to be used to prove each welding procedure. The production and testing of these pieces shall be witnessed by the purchaser or the Inspecting Authority except that, in cases where the manufacturer can furnish proof of previously authenticated tests and results on the same type of joint and material within the permitted variables of BS EN ISO 15614-2 and then shall be deemed exempt from any further tests.

Al.5.2.3 All welding shall be performed in accordance with a welding procedure specification, or other work instruction, conforming to BS EN ISO 15609-1.

A welding procedure test on a branch connection will only qualify a WPS for welding a branch connection to this specification when mechanical properties of the joint have been established by an equivalent butt weld. Alternatively a weld procedure approval test on a butt joint in pipe shall give approval for pipe branch connections and nozzle to shell connections, where:

a) the joint details and geometry for the branch connections have been accepted by the contracting parties; and

b) a welded branch connection using the same joint details and geometry has been previously demonstrated as sound in any steel, on the basis of volumetric and surface non-destructive examination.

A pre-existing weld procedure test performed in accordance with BS EN 288-4 or BS 4870-2, previously acceptable to an Inspecting Authority, shall remain acceptable providing it satisfies the intent of the technical requirements of BS EN ISO 15614-2. However, the range of approval of such a test shall be in accordance with the ranges in BS EN ISO 15614-2 except as modified by **Al.5.5**.

NOTE Existing procedures to BS EN 288-4 or the earlier BS 4870-2 are considered technically equivalent to BS EN ISO 15614-2 when similar types of tests have been carried out. Thus the bend tests in BS EN 288-4 or BS 4870-2 are considered equivalent to those in BS EN ISO 15614-2 even though the exact number and the bend angle differ. Similarly visual, radiographic, ultrasonic, surface crack detection, transverse tensile, hardness, macro and impact tests are considered equivalent.

Where BS EN ISO 15614-2 calls for a type of test to be performed that has not been carried out on the pre-existing BS 4870-2 procedure qualification tests, additional tests as described in the Introduction of BS EN 15614-2 should be carried out.

The alternative methods of approval of welding procedures addressed in BS EN ISO 15607 are not permitted for welding on pressure vessels made in accordance with this specification.

Al.5.2.4 *Main text is not applicable.*

Al.5.2.5 *Main text is not applicable.*

Al.5.2.6 *Main text is not applicable.*

Al.5.3 Welder and operator approval

Al.5.3.1 Approval testing of welders and operators shall be conducted, recorded and reported in accordance with BS EN ISO 9606-2.

Al.5.3.2 *See main text.*

Al.5.3.3 *See main text.*

Al.5.3.4 *See main text.*

Al.5.4 Production control test plates

Al.5.4.1 Production control test plates shall not be required unless specified by the purchaser in the purchase specification. In such cases they shall be prepared and tested in accordance with **Al.5.4** and the number of test plates shall be subject to agreement between the manufacturer, purchaser and Inspecting Authority.

Al.5.4.2 The material used for the test plates shall comply with the same specification as that used in the construction of the vessel.

The plate shall be of the same nominal thickness as the shell and should preferably be selected from the same batch of material as that used in fabricating the vessel. The test plates shall be sufficiently large to allow for the preparation of all the specimens required in BS EN ISO 15614-2 and for any additional specimens that may be required. The minimum width shall be in accordance with the following values:

Thickness of plate	Minimum width (each of two plates)
Up to and including 6 mm	250 mm
Over 6 mm up to and including 13 mm	300 mm
Over 13 mm up to and including 25 mm	450 mm
Over 25 mm up to and including 51 mm	600 mm

However, if it can be demonstrated to the satisfaction of the Inspecting Authority that the equalization temperature of the test plates has not exceeded approximately 100 °C during welding[2)] it is permissible to reduce these widths to the following values:

Thickness of plate	Minimum width (each of two plates)
Up to and including 6 mm	150 mm
Over 6 mm	250 mm

Al.5.4.3 When a vessel includes one or more longitudinal seams the test plates shall, wherever practicable, be attached to the shell plate on one end of one seam so that the edges to be welded in the test plate are a continuation and duplication of the corresponding edges of the longitudinal seams. The weld metal shall be deposited in the test plates continuously with the welding of the corresponding longitudinal seam so that the welding process, procedure and technique are the same. When it is necessary to weld the test plates separately, the procedure used shall duplicate that used in the construction of the vessel.

When the test plates are required for circumferential welds, it is permissible to weld them separately from the vessel providing the technique used in their preparation duplicates, as far as possible, the procedure used in the welding of the appropriate seams in the vessel.

Al.5.4.4 Care shall be taken to minimize distortion of the test plates during welding. If excessive distortion occurs, the test plate shall be straightened before post-weld treatment. At no time shall the test plates be heated to a temperature higher than that used or to be used for the final heat treatment of the vessel, if any (see **Al.4.5.3**).

[2)] This may be achieved by applying suitable temperature indicating paints to the outer edges (remote from the weld) of the test plates before welding. It is suggested that an 80 °C indicator and a 120 °C indicator be employed on each plate.

At the option of the manufacturer it is permissible for the test plates to be non-destructively tested in the same manner as the production weld. If any defects in the weld of a test plate are revealed by non-destructive testing, their position shall be clearly marked on the plate and test specimens shall be selected from such other parts of the test plate as may be agreed upon between the manufacturer, purchaser and the Inspecting Authority.

On completion, specimens in accordance with **Al.5.4.2** shall be cut from the production test plates and tested in accordance with **Al.5.5**.

Al.5.5 Details of destructive tests for procedure, welder and production control testing

Al.5.5.1 Test requirements

Weld procedure and production control testing shall be in accordance with BS EN ISO 15614-2, except where otherwise stated in **Al.5.5**. Approval testing of welders shall be in accordance with BS EN ISO 9606-2, except where otherwise stated in **Al.5.5**.

Al.5.5.2 Test temperature

The tests shall be conducted at room temperature.

Al.5.5.3 Transverse tensile test

For weld procedure and production control testing, transverse tensile tests shall be in accordance with BS EN ISO 15614-2. For welder approval, such testing is optional unless specified in the purchase specification when it shall be in accordance with BS EN ISO 15614-2.

Al.5.5.4 Bend test

Bend tests shall be in accordance with BS EN ISO 9606-2 as appropriate.

Al.5.5.5 Macro- and micro-examination

The specimen shall be prepared for macro-examination, and for micro-examination when the necessity for the latter has been agreed between the manufacturer, purchaser and Inspecting Authority.

The weld shall be sound, i.e. free from cracks and substantially free from discontinuities such as porosity, to an extent equivalent to that given in Table Al.5.7-1.

Table Al.5.7-1 **Acceptance levels**

Abbreviations used:

- e is the parent metal thickness. In the case of dissimilar thicknesses e applies to the thinner component;
- w is the width of defect;
- l is the length of defect;
- h is the height of defect;
- φ is the diameter of defect.

Defect type[a]		Permitted maximum	
Planar defects	Cracks and lamellar tears Lack of root fusion Lack of side fusion Lack of inter-run fusion Lack of root penetration	Not permitted	
Cavities		Category 1	Category 2
	a) Isolated pores (or individual pores in a group)	φ max. 3 mm (grade D) $\varphi \le e/4$	Such cavities may be accepted without limit provided representative mechanical specimens from production test plates comply with requirements
	b) Uniformly distributed or localized porosity	$e \le 3$ mm scattered, grade A $e > 3$ mm up to 6 mm scattered grades A and B $e > 6$ mm scattered grades A and B plus isolated, grade C not more in number than $e/2$ per 1 000 mm^2 of radiographic area[b]	
	c) Linear porosity	Linear indications parallel to the axis of the weld may indicate lack of fusion or lack of penetration and are therefore not permitted	
	d) Wormholes isolated	$l \le 3$ mm $w \le 1.5$ mm	—
	e) Wormholes aligned	As linear porosity	
	f) Crater pipes g) Surface cavities	Not permitted	
Solid inclusions	a) Oxide inclusions, linear	Not permitted	
	b) Oxide inclusions, diffuse	Isolated patches permitted provided that they do not exceed $e/4$ or 3 mm max. in average diameter and provided that they are not repetitive	
	c) Tungsten inclusions	No limit except that they shall not be accompanied by oxide inclusions and that the max. diameter of individual inclusions does not exceed $e/4$ or 3 mm max.	
	d) Copper inclusions	Not permitted	
Profile and visible surface defects	a) Insufficient weld size b) Overlap c) Shrinkage grooves and root concavity	Not permitted	
	d) Undercut	Slight intermittent undercut permitted, should not exceed approximately 0.5 mm	
	e) Excess penetration	$h \le 3$ mm. Occasional local slight excess is allowable	
	f) Reinforcement shape	The reinforcement shall blend smoothly with the parent metal and dressing is not normally required provided the shape does not interfere with the specified non-destructive testing techniques	
	g) Linear misalignment	See **4.2.3**	

Table Al.5.7-1 **Acceptance levels** *(continued)*

Grade of uniform porosity	**Approx. average diameter of pores** mm
A	0.4
B	0.8
C	1.5
D	3 or greater

NOTE The significant dimension of a defect in terms of its effect on service performance is the height or through thickness dimension. If ultrasonic flaw detection is employed, it is probable that defect indications of very minor cross-section will be obtained. In interpreting the requirements of this table, such indications having a dimension h of 1.5 mm or less should be disregarded unless otherwise agreed between the manufacturer, the purchaser and the Inspecting Authority.

[a] See below for definitions of uniform porosity.

[b] Area is the product of length and width of an envelope enclosing the affected volume of weld metal measured on a plane substantially parallel to the weld face (i.e. as seen on a radiograph). "Scattered" is defined as t to 0.25t per square centimetre (where t = metal thickness in millimetres).

Al.5.5.6 Retests

Retests shall be as specified by BS EN ISO 9606-2 for weld approvals and as specified by BS EN ISO 15614-2 for weld procedure and production control testing.

Should any production control retest specimens not comply with the requirements, the welded seams represented by these tests shall be deemed not to comply with this specification. If any retest specimens fail during weld procedure approval testing the cause of failure shall be established and the whole procedure test shall be repeated.

Al.5.6 Non-destructive testing

Al.5.6.1 General

See main text.

Al.5.6.2 Parent materials

Acceptance standards for defects revealed by non-destructive testing of unwelded parent materials shall be subject to agreement between the manufacturer and the purchaser, and/or the Inspecting Authority. Where repairs by welding are authorized, non-destructive testing techniques for the repair and subsequent acceptance standards shall also be subject to agreement between the manufacturer and the purchaser, and/or the Inspecting Authority.

Al.5.6.3 Components prepared for welding

See main text.

Al.5.6.4 Non-destructive testing of welded joints

See main text.

Al.5.6.4.1 Components to construction category 1

With the exception of materials and thicknesses permitted for construction category 2, the final non-destructive testing shall be carried out after completion of any post-weld heat treatment required (see **Al.4.5.3**).

Al.5.6.4.1.1 Examination for internal flaws

The full length of all full penetration butt welds including the welds of forged butt welded nozzles shall be examined by radiographic and/or ultrasonic methods. Unless otherwise agreed between the purchaser and the manufacturer, the full length of all other welds (e.g. nozzles and branches) in or on pressure parts shall be examined by ultrasonic and/or radiographic methods where the thickness of the thinnest part to be welded exceeds 40 mm.

Where a branch compensation plate is used, the shell and the compensation plate shall be considered as one component of total thickness equal to the combined thickness of the shell and compensation ring unless:

a) the branch to shell weld is separate from, or is completed and inspected before, the branch to compensation ring; and

b) the outer compensation ring to shell weld is not completed until the welds referred to in a) have been completed.

Al.5.6.4.1.2 Examination for surface flaws

The full length of all welds other than full penetration butt welds shall be examined in accordance with **Al.5.6.5.2**. Full penetration butt welds shall be examined by these methods when agreed between the manufacturer, the purchaser and the Inspecting Authority.

Defects revealed by non-destructive testing shall be assessed in accordance with **Al.5.7**.

Al.5.6.4.2 Components to construction category 2

Category 2 constructions shall be subject to partial non-destructive testing, as specified in **Al.5.6.4.2.1** and **Al.5.6.4.2.2**. Such non-destructive testing shall be employed at as early a stage in the fabrication process as practicable as a measure of quality control and the locations selected for testing shall be representative of all welding procedures and the work of each welder or operator employed. Results of non-destructive testing shall be assessed in accordance with **Al.5.7.3**.

Where a particular examination reveals defects in excess of the levels given in Table Al.5.7-1 all welds represented by the original examination shall be examined by the same non-destructive testing method and the results assessed in accordance with **Al.5.7**.

In cases where fabrication procedures require main seams to be welded at site, such seams shall be 100% examined by radiographic and/or ultrasonic methods generally in accordance with **Al.5.6.5.1** and the results interpreted against the acceptance levels specified in **Al.5.7.3**.

Al.5.6.4.2.1 Examination for internal flaws

See main text.

Al.5.6.4.2.2 Examination for surface flaws

See main text.

Al.5.6.4.3 Components to construction category 3

See main text.

Al.5.6.5 Choice of non-destructive test methods for welds

Al.5.6.5.1 Internal flaws

See main text.

Al.5.6.5.2 Surface flaws

The choice of method for surface crack detection depends on material. Magnetic methods are not suitable for aluminium alloys. It is permissible to employ one of the following:

a) visual examination supplemented by a ×2 or ×5 magnification glass;

b) dye penetrant examination;

c) eddy current methods;

by agreement between the manufacturer, purchaser and Inspecting Authority.

Al.5.6.6 Non-destructive testing techniques for welds

Al.5.6.6.1 Radiographic techniques

Normally radiographic examination shall be in accordance with BS EN ISO 17636-1:2013 Class B, however the maximum area for a single exposure shall conform to the requirements of Class A. Because several techniques with differing sensitivities are detailed in BS EN ISO 17636-1, it is necessary to specify for each particular application which technique is required to be used. For thicknesses up to 50 mm X-ray techniques shall normally be used. It is permissible to use other techniques provided it can be demonstrated to the satisfaction of the Inspecting Authority that adequate sensitivity can be obtained.

Al.5.6.6.1.1 Marking and identification of radiographs

See main text.

Al.5.6.6.2 Ultrasonic techniques

It is permissible to use ultrasonic examination generally in accordance with BS EN ISO 17640:2018, provided due allowance is made for different calibration tests due to the changed sound velocity.

Before carrying out ultrasonic examination of welds, the adjacent parent metal shall be ultrasonically examined to establish the thickness of the material and to locate any flaws which may prevent effective examination of the weld.

Al.5.6.6.3 Magnetic particle techniques

Main text is not applicable.

Al.5.6.6.4 Penetrant techniques

See main text.

Al.5.6.6.5 Surface condition and preparation for non-destructive testing

See main text.

Al.5.6.6.6 Marking, all non-destructive testing methods

Permanent marking of the vessel alongside welds shall be used to provide reference points for the accurate location of the seam with respect to the test report. If the purchaser requires a specific method of marking this shall be detailed in the purchase specification. Stamping shall not be used on vessels intended for low temperature service or where stamping may have a deleterious effect on the material in service.

Al.5.6.6.7 Reporting of non-destructive testing examinations

See main text.

Al.5.7 Acceptance criteria for weld defects revealed by visual examination and non-destructive testing

Al.5.7.1 General

Subject to the provisions of Annex C, the main constructional welds of pressure vessels shall comply with **Al.5.7.2**.

NOTE Guidance on the acceptance criteria for arc-welded tube to tubeplate joints is given in Annex T.

Al.5.7.2 Quality control level of acceptance

The defect acceptance levels given in Table Al.5.7-1 shall be imposed during fabrication as a means of quality control. With the exception of inclusions these are, for practical purposes, the same as those adopted for welder approval and procedure approval in BS EN ISO 15614-2 and BS EN ISO 9606-2. When inclusions are greater than those permitted in these two standards, but less than those permitted in Table Al.5.7-1, or where defect acceptance is based on **Al.5.7.3.2** or **Al.5.7.3.3**, the reasons for the occurrence of such defects shall be investigated.

Al.5.7.2.1 *Main text is not applicable.*

Al.5.7.2.2 *Main text is not applicable.*

Al.5.7.2.3 *Main text is not applicable.*

Al.5.7.2.4 *Main text is not applicable.*

Al.5.7.3 Assessment of defects

Defects shall be assessed according to one or other of the alternatives in **Al.5.7.3.1**, **Al.5.7.3.2** and **Al.5.7.3.3**. Defects that are unacceptable shall be deemed not to comply with this specification or be repaired.

Al.5.7.3.1 If defects do not exceed the levels specified in Table Al.5.7-1 the weld shall be accepted without further action.

NOTE Details for vessels intended for service in the creep range may require special consideration.

Al.5.7.3.2 When acceptance levels[3)] different from those permitted in Table Al.5.7-1 have been established for a particular application and are suitably documented, it is permissible for them to be adopted by specific agreement between the purchaser, the manufacturer and the Inspecting Authority after due consideration of material, stress and environmental factors.

Al.5.7.3.3 It is permissible to accept particular defects[3)] in excess of those permitted in Table Al.5.7-1 by specific agreement in the same way as in **Al.5.7.3.2**.

Al.5.7.4 Repair of welds

No rectification, repair or modification shall be made without the approval of the purchaser and Inspecting Authority.

Unacceptable defects shall be either repaired or deemed not to comply with this specification. Repair welds shall be carried out to an approved procedure and subjected to the same acceptance criteria as original work.

[3)] For example see Annex C or BS 7910.

Al.5.8 Pressure tests

Al.5.8.1 General

See main text.

Al.5.8.2 Basic requirements

See main text.

Al.5.8.3 Hydraulic testing

See main text.

NOTE To avoid any risk of incomplete drying out after a hydraulic test, aluminium vessels intended for cryogenic service should be pneumatically tested instead.

Al.5.8.4 Pneumatic tests

See main text.

Al.5.8.5 "Standard" test pressure

See main text.

Al.5.8.6 Proof hydraulic test

Al.5.8.6.1 A proof testing procedure to be followed for vessels (or vessel parts) of which the strength cannot be satisfactorily calculated (see **3.2.2**) shall be agreed (see **5.8.2.2**).

Al.5.8.6.2 Before the test is begun or any pressure has been applied to the vessel, strain gauges of electrical resistance or other types shall be affixed to both the inside and outside surfaces of the vessel. The number of gauges, their positions and their directions shall be chosen so that principal strains and stresses can be determined at all points of interest. The type of gauge and the cementing technique shall be chosen so that strains up to 1% can be determined.

Al.5.8.6.3 Pressure shall be applied gradually until either the "standard" test pressure for the expected design pressure is reached or significant strain of any part of the vessel occurs.

When either of these points is reached, the pressure shall not be further increased.

It is permissible to disregard indication of localized permanent set provided that there is no evidence of general distortion of the vessel.

Al.5.8.6.4 The highest pressure which is applied shall be maintained for the time sufficient to permit inspection in accordance with **5.8.2.3**.

Al.5.8.6.5 Strain readings shall be taken as the pressure is increased. The pressure shall be increased by steps of approximately 10% and unloaded between steps, until the "standard" test pressure p_t is reached or until significant general strain occurs. Strain readings shall be repeated during unloading. Should the plot of strain versus pressure during the application of pressure and unloading show evidence of non-linearity it is permissible for the pressure reached to be reapplied not more than five times until the loading and unloading curves corresponding to two successive pressure cycles substantially coincide. Should coincidence not be attained, the pressure p_y (see **5.8.6.5.2**) shall be taken as the pressure range corresponding to the linear portion of the curve obtained during the final unloading.

AI.5.8.6.5.1 *See main text.*

AI.5.8.6.5.2 *See main text.*

AI.5.8.6.6 *Main text is not applicable.*

AI.5.8.7 Combined hydraulic/pneumatic tests

See main text.

AI.5.8.8 Leak testing

See main text.

AI.5.8.9 Vessel nameplate

See main text.

AI.5.8.10 Final inspection

An internal and external examination of the completed vessel shall be carried out prior to despatch and the marking on the vessel shall be checked (see Table AI.5.8-1).

Table AI.5.8-1 **Principal stages of inspection**

Stages of inspection	Responsible party	Remarks	Clause reference
Examine materials at product maker's works, select test pieces and witness the appropriate mechanical tests	Inspecting Authority	When required by the purchaser	Section 2
Correlate the material certificates of mechanical tests and chemical analyses with the materials and check them with the specifications	Manufacturer and Inspecting Authority	The manufacturer is responsible for forwarding the certificates to the Inspecting Authority	**1.5.2**
Identify material and witness transfer of identification marks in manufacturer's works	Manufacturer and Inspecting Authority	Origin of material to be demonstrated from available records to the satisfaction of the Inspecting Authority and any transfer of identification marks in manufacturer's works to be witnessed	**4.1.2**
Visually examine material for flaws, laminations, etc. Check thickness	Manufacturer	Examination by Inspecting Authority is optional	**AI.4.2.2.1**
Examine material cut edges and heat affected zones	Manufacturer and Inspecting Authority	Examination by Inspecting Authority is optional	**AI.4.2.1.2**
Approve weld procedures to be employed	Inspecting Authority	Inspecting Authority to witness tests unless the procedures are already approved	**AI.5.2**
Approve welders and operators	Inspecting Authority	Inspecting Authority to witness tests unless the welders and operators are already approved	**AI.5.3**
Witness production weld tests	Inspecting Authority	When required by the purchaser	**AI.5.4**

Table AI.5.8-1 **Principal stages of inspection** *(continued)*

Stages of inspection	Responsible party	Remarks	Clause reference
Examine welded joints after cold forming	Manufacturer	Examination by Inspecting Authority is optional	**AI.4.2.2.2**
Examine plates after forming	Manufacturer	Examination by Inspecting Authority is optional	**AI.4.2.2.4.1** and **AI.4.2.2.4.2**
Examine set up of seams for welding, including dimensional check, examination of weld preparations, tack welds, etc.	Manufacturer and Inspecting Authority	Examination by the Inspecting Authority is optional for categories 1 and 2 components. For category 3 components the Inspecting Authority should not normally perform this examination on every joint of each component but shall exercise its discretion consequent to the results of examinations carried out	**4.3.4**
Inspect second side weld preparation after the first side weld is completed and root cleaned. This is applicable to main seams where manual or semi-automatic welding from both sides is employed	Manufacturer and Inspecting Authority	Examination by the Inspecting Authority is optional for categories 1 and 2 components. For category 3 components the Inspecting Authority should not normally perform this examination on every joint of each component but shall exercise its discretion consequent to the results of examinations carried out	**4.3.7.4**
Examine non-destructive test reports before and/or after post-weld heat treatment as required by the procedure and consider acceptability of any defects	Inspecting Authority	The manufacturer is responsible for presenting the reports to the Inspecting Authority	**5.6.6.7**
Check all main dimensions on completion of fabrications	Manufacturer and Inspecting Authority	The Inspecting Authority should witness these checks. This stage may be omitted if the vessel is to be heat treated	**4.2.4** and **AI.4.2.5**
Check post-weld heat treatment procedure	Manufacturer and Inspecting Authority	The Inspecting Authority should carry out this check when required	**AI.4.5.3**
Check all main dimensions on completion of manufacture	Manufacturer and Inspecting Authority	Inspecting Authority to witness these checks	**4.2.4** and **4.2.5**
Witness pressure test and where necessary record the amount of any permanent set	Manufacturer and Inspecting Authority		**AI.5.8**
Examine completed vessel before despatch. Check marking	Manufacturer and Inspecting Authority		**5.8.9** and **5.8.10**

Al.C Annex C: Assessment of vessels subject to fatigue

Al.C.1.3 Symbols

See main text, but substitute:

E is the modulus of elasticity at the maximum operating temperature from Table Al.3.6-3 (in N/mm^2).

Al.C.3.2.4 Environmental effects

For aluminium alloys no guidance is given, and expert advice should be sought on the effects of corrosive environments on fatigue strength.

Al.E Annex E: Recommendations for welded connections of pressure vessels

Al.E.1 Typical details for principal seams

The details indicated in this clause have given satisfactory results under specific manufacturing conditions and are included for general guidance. Modification might be required to suit particular manufacturing techniques and all details adopted have to be shown by the manufacturer to produce satisfactory results by the procedure specified in **Al.4**.

Where no root gap is shown it is intended that the joints be close butted. For requirements governing the use of backing strips see **Al.4.3.6.2**.

The following details are given:

a) typical full penetration joint preparations for one-sided welding only: aluminium and its alloys (see Figure Al.E.1);

b) typical full penetration joint preparations for two-sided welding only: aluminium and its alloys (see Figure Al.E.2);

c) typical full penetration joint preparations for one-sided welding with temporary backing or permanent backing: aluminium and its alloys (see Figure Al.E.3).

Al.E.2 Typical example of acceptable weld details

Al.E.2.1 *General*

See main text and additionally

The weld details in **E.2** are generally suitable for aluminium connections welded by an appropriate process provided that groove angles are increased to suit the welding process applied.

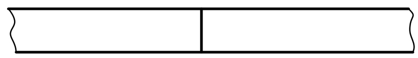
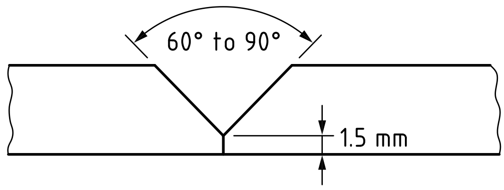
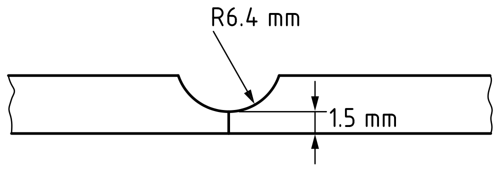
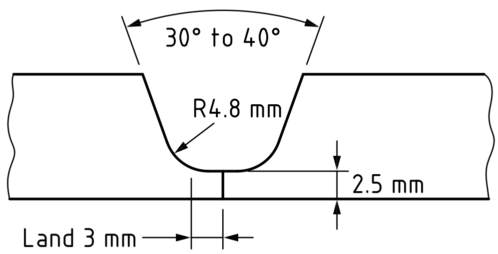
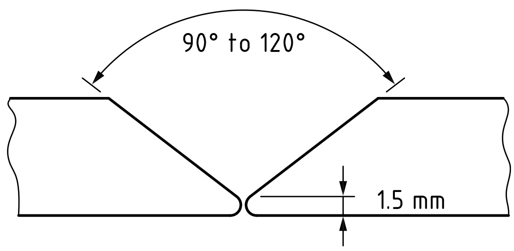
Al.E.2.5.12 These weld details are recommended only for shell thicknesses up to 16 mm for aluminium and aluminium alloys. These weld details are not recommended for corrosive or fatigue duty.

Figure E.6

See main text and additionally

NOTE The details are applicable to aluminium and aluminium alloy pressure vessels, but the groove angle α should be increased to a minimum of 45°.

Figure Al.E.1 **Typical full penetration joint preparations for one-sided welding only: aluminium and its alloys**

Material thickness	Edge preparation	Remarks
Up to 3 mm		Suitable for a.c. argon TIG, d.c. helium TIG and pulsed MIG Penetration from one side only during welding can be achieved (Manual or mechanized)
3 mm to 6.3 mm		Suitable for a.c. TIG and pulsed MIG Controlled penetration possible (Manual or mechanized)
3 mm to 4 mm		Suitable for rolled or positional fixed pipes using a.c. TIG Controlled penetration possible (Manual)
4 mm upwards		Suitable for rolled or positional fixed pipes using a.c. argon TIG Controlled penetration possible (Manual)
6.3 mm to 9.5 mm		Suitable for rolled pipes with a.c. TIG or pulsed MIG Controlled penetration possible. Root faces radiused slightly

NOTE These joint preparations are designed to permit a controlled penetration bead to be achieved on one-sided joints where accessibility to the underside is restricted.

Pipe joints preparations are also included.

Figure Al.E.2 **Typical full penetration joint preparations for two-sided welding only: aluminium and its alloys**

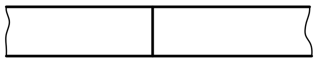
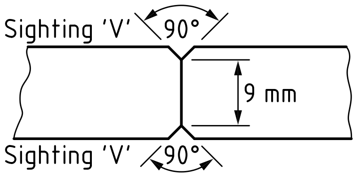
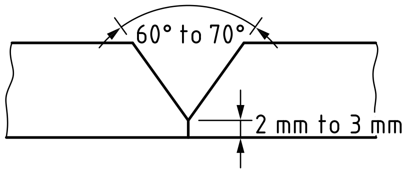
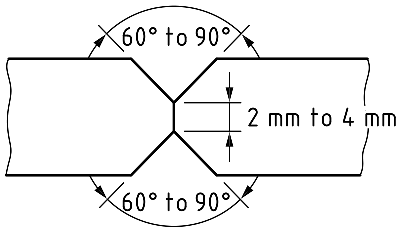
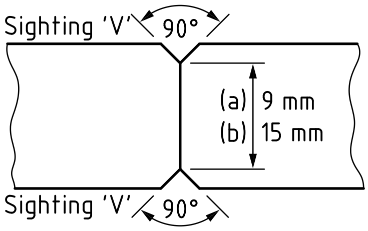
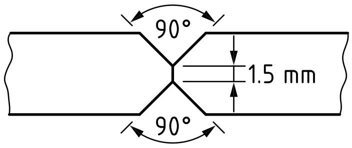
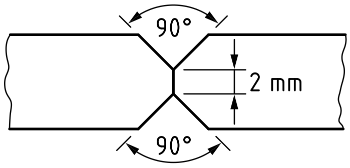
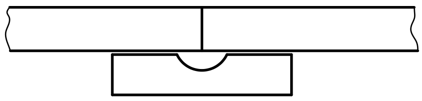
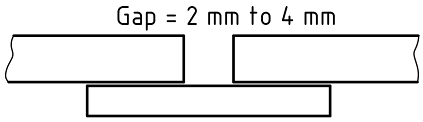
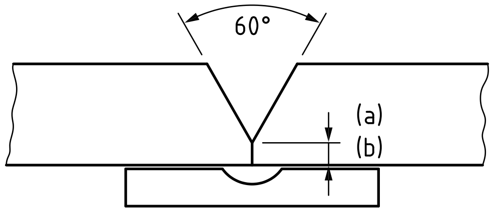
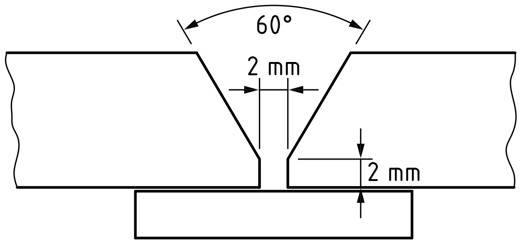
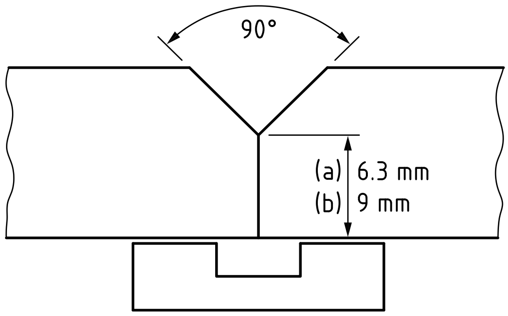
Material thickness	Edge preparation	Remarks
6.3 mm to 9.5 mm		Suitable for two run procedures (one run from each side without back cutting) Conventional MIG or d.c. helium TIG may be used (Manual or mechanized)
9.5 mm to 12.7 mm		Suitable for two run procedures (one run from each side without back cutting) Conventional MIG or d.c. helium TIG may be used (Manual welding or mechanized)
6.3 mm to 12.7 mm		Back chipped and sealed. 3 mm root face recommended when helium + argon mistures are used Conventional MIG only (Manual or mechanized)
6.3 mm to 25.4 mm		a) No back cutting required. Use 3 mm root face for argon MIG and 4 mm root faces when helium + argon mixture or helium is used with convention MIG b) Back cutting of reverse side when required using 2 mm root face only (Manual or mechanized)
12.7 mm to 25.4 mm		One run from each side Mechanized welding recommended High current applications, helium, argon, or helium + argon mix a) Root face at up to 9 mm up to 19 mm thick b) Root face at 15 mm in excess of 19 mm thick

Figure Al.E.2 **Typical full penetration joint preparations for two-sided welding only: aluminium and its alloys *(continued)***

Material thickness	Edge preparation	Remarks
4.8 mm to 6.3 mm		Double operator TIG a.c. with argon Vertical-up welding Root gap of 1.5 mm may be tolerated (Manual)
6.3 mm to 12.7 mm		Double operator TIG a.c. with argon Vertical-up welding Root gap of 1.5 mm may be tolerated (Manual)

NOTE These joint preparations are designed primarily for the use of two-sided procedures which may involve either two or more weld runs without back cutting on reverse side. Alternatively, procedures involving back cutting and a seal weld are also given.

Figure Al.E.3 **Typical full penetration joint preparations for one-sided welding with temporary backing or permanent backing: aluminium and its alloys**

Material thickness	Edge preparation	Remarks
Up to 3 mm		Temporary backed, suitable for a.c. TIG, pulsed MIG, d.c. helium TIG and conventional MIG (Manual or mechanized)
3 mm to 4.8 mm		Use 2 mm gap with conventional MIG 4 mm gap with TIG and pulsed MIG Permanent backing (Manual or mechanized)
4.8 mm to 12.7 mm		Temporary backing with conventional MIG or a.c. argon TIG a) Nil root face with 3 mm root gap for TIG b) 2 mm root face with 2 mm root gap for MIG (Manual or mechanized)
4.8 mm to 12.7 mm		Permanent backed using conventional MIG For a.c. argon TIG nil root faces will suffice (Manual or mechanized)
9.5 mm to 19 mm		High current MIG welding Argon shielded Temporary backing bar a) Root face for 9.5 mm thickness: take 6.3 mm b) Root face for thickness in excess of 9.5 mm: increse to 9 mm (Mechanized recommended)

NOTE The joint preparations are designed where temporary or permanent backing systems are required.

Copper supplement

Requirements for copper and copper alloys in the design and construction of unfired pressure vessels

This supplement shall be read in conjunction with the main body of the specification to establish the requirements for copper and copper alloy unfired pressure vessels. This supplement lists the sections of the main text applicable to, and those not applicable to, the design and construction of copper and copper alloy pressure vessels. In addition, the supplement contains clauses specific to such vessels which replace corresponding clauses of the main text in this context. The clause, equation, figure and table numbering of the requirements of this supplement follow those of the comparable main text requirements but are identified in the form **Cu.x.y.z** to differentiate them from their main text equivalents, i.e. **x.y.z**, which might be technically different.

Cu.1. Section 1: General

See main text.

Cu.2. Section 2: Materials

Cu.2.1 Selection of materials

Cu.2.1.1 General

See main text, but substitute.

Cu.2.1.1.3 For the ease of reference in this supplement, materials are identified by a group number which has been derived from PD CEN ISO/TR 15608:2017, Table 3. This grouping is summarized in Table Cu.2.1-1.

Table Cu.2.1-1 **Material grouping**

Group	Sub-group	Type of copper and copper alloys
31	–	Pure copper
32	–	Copper-zinc alloys
	32.1	Copper-zinc alloys, binary
	32.2	Copper-zinc alloys, complex
33	–	Copper-tin alloys
34	–	Copper-nickel alloys
35	–	Copper-aluminium alloys
36	–	Copper-nickel-zinc alloys
37	–	Copper alloys, low alloyed (<5% other elements) not covered by groups 31 to 36
38	–	Other copper alloys (≥5% other elements) not covered by groups 31 to 36

Cu.2.1.2 Materials for pressure parts

Cu.2.1.2.1 General

See main text but substitute

For the construction of vessels in accordance with this supplement, all the materials used in the manufacture of pressure parts shall:

a) conform to British Standards listed in Table Cu.2.3-1; or

b) conform to BS EN standards listed in Table Cu.2.3-2; or

c) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2**b)] and shall conform to **Cu.2.1.2.2**; or

d) be the subject of a "European approval of material" in accordance with the EU directive 2014/68/EU (see Note 2); or

e) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE 1 The British Standards listed in Table Cu.2.3-1 have been withdrawn and replaced by BS EN standards (see Table Cu.2.3-2 and Enquiry Case 5500/134). Enquiry Case 5500/140 gives guidance on copper and copper alloy materials covered by BS EN standards and ASTM standards.

NOTE 2 Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Cu.2.1.2.2 Materials covered by Cu.2.1.2.1c)

The material shall be covered by a written specification at least as comprehensive as the nearest equivalent British Standard listed in Table Cu.2.3-1. The specification shall, as a minimum, specify the manufacturing process, compositional limits for all constituents, heat treatment and appropriate mechanical properties for acceptance and other purposes.

Cu.2.1.2.3 Additional materials for category 3 components

Main text is not applicable.

Cu.2.1.3 Materials for non-pressure parts

See main text.

Cu.2.2 Materials for low temperature application

Materials listed in Table Cu.2.3-1 are not susceptible to brittle fracture and no special provisions are necessary for their use at temperatures down to −196 °C.

Cu.2.3 Nominal design strength

Cu.2.3.1 General

Cu.2.3.1.1 Design strength values appropriate to copper and copper alloys C106, CZ110, CZ111, CZ112, CZ123, CZ126, CN102, CN106, CN107, CA104, CA105, CA106, LG3, AB2 and SCB6 as specified in BS 1400, BS 2870, BS 2871, BS 2874 and BS 2875 are given in Table Cu.2.3-1.

The upper limiting service temperature based on time independent properties is 150 °C.

Cu.2.3.1.2 The nominal design strengths given in Table Cu.2.3-1 are related to the relevant property values given in the following British Standards.

BS 1400 Castings

BS 2870 Sheet strip and foil

BS 2871 Tube

BS 2874 Rods and sections (other than forging stock)

BS 2875 Plate.

Table Cu.2.3-1 **Design strength values (N/mm^2) of copper and copper alloys**

Material to BS 1400, BS 2870, BS 2871, BS 2874 and BS 2875	Form	Condition	Min. tensile strength R_m N/mm^2	Min. 0.2% proof stress $R_{p0.2}$ N/mm^2	Min. 1.0% proof stress $R_{p1.0}$ N/mm^2	Value for *f* for design temperature not exceeding °C				
						50	75	100	125	150
Copper C106	Plate or sheet	Annealed O	210	—	(80)	53	52	51	50	48
		Half H	(240)	(180)	—	120	116	113	110	107
		M	280	—	—	—	—	—	—	—
	Rod	Annealed O	210	—	(80)	53	52	51	50	48
		Half H	(240)	(160)	—	107	100	92	90	87
		M	—	—	—	—	—	—	—	—
	Tube	Annealed O	200	—	(80)	53	52	51	50	48
		Half H	250	(150)	—	100	95	90	88	87
		M	280	—	—	—	—	—	—	—
Special 70/30 Arsenical brass CZ126	Tube	Annealed O	280	—	(115)	77	74	72	71	71
Aluminium brass CZ110	Tube	Annealed O	300	—	(130)	87	85	83	83	83
Admiralty brass CZ111	Tube	Annealed O	(320)	—	(105)	70	69	69	68	67
60/40 Brass low lead CZ123	Plate or sheet	Annealed O	340	—	(140)	93	93	93	92	92
Naval brass CZ112	Plate or sheet	Annealed O	340	—	(175)	117	112	108	106	105
90/10 Copper-nickel-iron CN102	Plate or sheet	Annealed O	300	—	(145)	97	94	92	90	89
	Rod	Annealed O	300	—	(125)	83	81	79	77	76
	Tube	Annealed O	300	—	(115)	75	74	73	72	71
70/30 Copper-nickel CN107	Plate or sheet	Annealed O	350	—	(175)	117	113	109	108	107
	Rod or tube	Annealed O	—	—	(140)	93	90	87	85	84
60/30/2/2 Copper-nickel-iron-manganese CN108	Tube	Annealed O	(420)	(150)	—	100	96	94	93	92
15% Aluminium bronze CA104	Plate or sheet	Annealed O	(630)	(270)	—	180	176	173	170	168
10% Aluminium bronze CA105	Plate or sheet	Annealed O	(630)	(270)	—	180	177	175	174	174
7% Aluminium bronze CA106	Plate or sheet	Annealed O	460	190	—	127	127	126	126	126
Leaded gunmetal LG2	Castings	—	200	100	—	64	62	61	59	58
Aluminium bronze AB2	Castings	—	640	250	—	160	154	144	130	116
Brass for brazable castings SCB6	Castings	—	170	80	—	30	30	30	25	20

NOTE The values in brackets have been obtained from sources other than the relevant British Standards.

Table Cu.2.3-2 **BS EN material standards for copper and copper alloys**

Material to BS EN 1653, BS EN 1982, BS EN 12163, BS EN 12420, BS EN 12449, BS EN 12451, BS EN 12452 and BS EN 13348			**Nearest BS equivalent**	**Group**
EN No	**EN name**	**Form**	**BS No**	
CW024A	Cu-DHP	Plate, sheet and forgings	C106	31
CW303G	CuAl8Fe3	Plate, sheet and forgings	CA106	35
CW304G	CuAl9Ni3Fe2	Plate, sheet and forgings	CA105	35
CW307G	CuAl10Ni5Fe4	Plate, sheet, rod and forgings	CA104	35
CW352H	CuNi10Fe1Mn	Plate, sheet, tube, rod and forgings	CN102	34
CW353H	CuNi30Fe2Mn2	Tube	CN108	34
CW354H	CuNi30Mn1Fe	Plate, sheet, tube, rod and forgings	CN107	34
CW610N	CuZn39Pb0.5	Plate, sheet, rod and forgings	CZ123	32.2
CW702R	CuZn20Al2As	Plate, sheet and tube	CZ110	32.2
CW706R	CuZn28Sn1As	Tube	CZ111	32.2
CW707R	CuZn30As	Tube	CZ126	32.2
CW712R	CuZn36Sn1Pb	Rod and forgings	CZ112	32.2

Cu.2.3.1.3 Time-independent design stresses given in Table Cu.2.3-1 were determined as follows:

$$f = \frac{R_{p0.2}}{1.5} \text{ or } \frac{R_m}{2.5}$$

NOTE R_m and $R_{p0.2}$ are the minimum values at the appropriate test temperature obtained by multiplying the actual value at the test temperature by the factor:

$$\frac{\text{Minimum specified tensile strength or proof stress}}{\text{Actual room temperature tensile strength or proof stress}}$$

Cu.2.3.1.4 For materials other than those listed in Table Cu.2.3-1, the time independent design strength shall be determined in accordance with **Cu.2.3.1.3**.

Cu.2.3.1.5 Time-dependent design stresses for copper and copper alloys are not currently available for use in pressure vessel calculations.

Cu.2.3.1.6 The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined in accordance with **Cu.2.3.1.1** or **Cu.2.3.1.4**. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

Cu.3. Section 3: Design

Cu.3.1 General

See main text.

Cu.3.2 Application

Cu.3.2.1 Consideration of loads

See main text and additionally:

Fatigue loading of copper and copper alloy vessels is not covered by this supplement. Such vessels subject to this loading shall have the adequacy of the proposed design demonstrated, to the satisfaction of the purchaser and Inspection Authority, by comparison with the behaviour of similar existing vessels.

Cu.3.4 Construction categories and design stresses

Cu.3.4.1 Construction categories

See main text with the following modified Table Cu.3.4-1.

Table Cu.3.4-1 **Construction categories**

Construction category	Non-destructive testing (NDT)	Permitted materials	Maximum nominal thickness of component[a] (see 1.6) mm	Temperature limits	
				Upper	Lower
1	100% (see **Cu.5.6.4.1**)	All in Table Cu.2.3-1 and Table Cu.2.3-2	None, except where NDT method limits	150 °C	–196 °C
2	Limited random (spot) (see **Cu.5.6.4.2**)	All in Table Cu.2.3-1 and Table Cu.2.3-2	20	150 °C	–196 °C
3	Visual only (see **Cu.5.6.4.3**)	Table Cu.2.3-1 and Table Cu.2.3 in annealed (O) condition	13	150 °C	–196 °C

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension of the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.

Cu.3.4.2 Design stresses

Cu.3.4.2.1 Categories 1 and 2

The design strengths for materials not requiring subsequent heat treatment (see **Cu.4.5.3**) shall not exceed the appropriate nominal design strength given by **Cu.2.3.1.1** or **Cu.2.3.1.4** for the material at the design temperature.

Cu.3.4.2.2 Category 3

The design strengths for materials not requiring subsequent heat treatment (see **Cu.4.5.3**) shall not exceed $R_m/5$ where R_m is given in Table Cu.2.3-1, or is obtained from the relevant material standard.

Cu.3.5.4 Openings and nozzle connections

See main text with the following modified Table Cu.3.5-4.

Table Cu.3.5-4 **Thickness of nozzles**

Nozzle nominal size mm	Minimum analysis thickness mm
12.0	1.0
16.0	1.5
30.0	2.0
38.0	2.0
44.5	2.0
57.0	2.0
76.1	2.5
88.9	2.5
108.0	2.5
133.0	2.5
159.0	2.5

Cu.3.6 Vessels under external pressure

See main text and additionally:

Cu.3.6.1.1 Notation

See main text but substitute

E is the modulus of elasticity of material of part under consideration at design temperature from Table Cu.3.6-3;

s is the factor relating f to effective yield point of material; for the purposes of **Cu.3.6**, s shall be taken to be 1.1 for copper and copper alloys.

Table Cu.3.6-3 ***E* values for copper and copper alloys (modulus of elasticity)**

Temperature °C	Copper (M) N/mm^2	Copper (O) and brasses N/mm^2	Copper-nickel N/mm^2	Aluminium-bronze N/mm^2
–200	140×10^3	110×10^3	150×10^3	125×10^3
20	130×10^3	105×10^3	140×10^3	120×10^3
150	125×10^3	100×10^3	130×10^3	115×10^3

Cu.3.8 Bolted flange connections

See main text and additionally:

NOTE Full faced flanges with non-metallic soft ring type gaskets are generally most suitable for use with copper. Other types of flanges can be used with most copper alloys.

Special attention shall be given to the risk of fracture of non-ferrous bolts through over tightening.

Cu.3.8.1 General

See main text and additionally:

Circular bolted flanges shall either:

a) conform to the requirements of **3.8** of the main text for individually designed flanges; or

b) conform to BS EN 1092-3 or BS EN 1759-3 for copper alloy, or composite, standard pipework flanges and be of the appropriate rating.

Cu.4. Section 4: Manufacture and workmanship

Cu.4.1 General aspects of construction

See main text.

Cu.4.2 Cutting, forming and tolerances

Cu.4.2.1 Cutting of material

Main text is not applicable.

Cu.4.2.2 Forming of shell plates and sections

Cu.4.2.2.1 General

Prior to forming, a visual examination of all plates shall be carried out, followed by measurements of the thickness.

Cu.4.2.2.2 Plates welded prior to forming

Main text is not applicable.

Cu.4.2.2.3 Cold forming

Where brass and aluminium bronze alloys have been cold formed in the course of fabrication, a stress relieving heat treatment should be carried out. For this, the material should be subjected to a temperature in the range 350°C to 450°C for not less than 30 min, preferably in a pyrometrically controlled furnace or oven.

The stress relief of copper and copper nickel alloys after cold forming is not normally necessary. However, by agreement between the manufacturer, purchaser and Inspecting Authority, suitable heat treatments may be given in special cases.

Cu.4.2.2.4 Hot forming

Copper and copper alloy plates to be treated or hot worked shall be heated uniformly in a neutral or oxidizing atmosphere, without direct flame impingement, to a temperature within the range specified in Table Cu.4.2-1. Where hot forming is to be used, tests shall be carried out to demonstrate that the proposed heat treatment gives acceptable properties on a representative test piece.

Table Cu.4.2-1 **Hot forming temperatures**

Material	Temperature °C
Copper	750 to 950
70/30 Brass	750 to 870
Aluminium brass	575 to 725
Admiralty brass	680 to 780
60/40 Brass (low lead)	650 to 750
Naval brass	650 to 750
90/10 Copper-nickel-iron	850 to 950[a]
70/30 Copper-nickel	925 to 1 025[a]
66/30/2/2 Copper-nickel-iron-manganese	925 to 1 025[a]
Aluminium bronze	800 to 975[a]

[a] These materials should not be hot formed below this temperature range due to hot shortness.

Cu.4.2.3 Assembly tolerances

See main text.

Cu.4.2.4 Tolerances of vessels subject to internal pressure

The difference between the maximum and minimum internal diameters of vessels subject to internal pressure, measured at any one cross-section shall not exceed the following:

Up to and including 1 900 mm	1% of the internal diameter;
Over 1 900 mm up to and including 4 750 mm	19 mm;
Over 4 750 mm	0.4% of the internal diameter.

Irregularities in profile (checked by a 20° gauge) shall not exceed the lesser of 25 mm or δ where:

$$\delta = 0.05e + 0.002D$$

where

e is the plate thickness;

D is the shell outside diameter.

For copper and copper alloy vessels manufactured from tube the permissible variation in diameter (measured externally) shall be in accordance with the specification governing the manufacture of the tube.

Cu.4.2.5 Tolerances for vessels subject to external pressure

See main text.

Cu.4.2.6 Structural tolerances

See main text.

Cu.4.3 Welded joints

Cu.4.3.1 General

Brazing is only permitted for the joining of non-critical components such as bosses and lifting lugs. All brazing shall be carried out in accordance with the requirements and recommendations of BS EN 14276-1:2020, Annex B or BS EN 14324.

Cu.4.3.2 Welding and brazing consumables

Filler rods and wires shall conform to BS EN ISO 24373 for welded joints, or BS EN ISO 17672 for brazed joints, and shall be stored in accordance with the supplier's recommendations. The selection of the filler rods or wires shall be appropriate to the parent alloy.

In all cases where the filler metals do not match parent metal compositions the purchaser and Inspecting Authority shall be satisfied that the combination used is suitable for the service conditions.

Cu.4.3.3 Preparation of plate edges and openings

See **4.3.3.2**.

Cu.4.3.4 Assembly for welding

It is permissible to use tack welds for copper and copper alloys but they shall not be incorporated into the final weld except in the case of copper-nickel alloys.

Cu.4.3.5 Attachment and removal of temporary attachments

It is permissible to weld or braze dissimilar copper alloys directly to a pressure component. Compatible filler metals shall be used in an approved procedure.

Cu.4.3.6 Butt joints

Permanent backing strips shall not be used for longitudinal welds. It is permissible to weld circumferential butt joints in tubes with temporary, permanent or consumable backing rings but only by agreement between the manufacturer, purchaser and Inspecting Authority.

Where a backing strip is to be used the material shall be such that it will not adversely influence the weld.

Backing strips shall be carefully removed prior to any special non-destructive tests on the joint.

Cu.4.3.7 Welding: general requirements

Filler materials for TIG-welding shall be cleaned immediately before use. Filler wire for MIG welding shall be protected from contamination during use and in particular between shifts.

Each run of weld metal shall be thoroughly cleaned before the next run is deposited. All scratch brushes shall be of stainless steel and shall be used only on copper and copper alloys.

When welding stops for any reason, care shall be taken when restarting, to ensure proper fusion between the weld metal and previously deposited weld metal.

Cu.4.4 Permanent joints other than welding

See main text.

Cu.4.5 Heat treatment

Cu.4.5.1 Preheat requirements

The preheat requirements to achieve adequate fusion for each type of weld, including tack welds, shall be established between the purchaser and manufacturer at the time of approval of welding procedures. The preheat temperature depends upon the type of joint, the metal thickness and the heat input to each run of welding.

NOTE As a general guide temperatures in excess of 150 °C should not be necessary except in the case of copper (C106) when the preheat temperature may have to be as high as 600 °C to ensure fusion, depending upon the thickness, welding process and shielding gas used.

Cu.4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

Main text is not applicable.

Cu.4.5.3 Post weld heat treatment

Post weld heat treatment of copper and copper alloy vessels is not normally necessary or desirable.

Cu.4.5.4 Methods of heat treatment

Main text is not applicable.

Cu.4.5.5 Post-weld heat treatment procedure

Main text is not applicable.

Cu.4.6 Surface finish

The whole of the internal surface of the vessel shall be cleaned and shall be free from grit, oil, grease and carbonaceous deposits.

Cu.5. Section 5: Inspection and testing

Cu.5.1 General

See main text.

Cu.5.2 Approval testing of fusion welding (and brazing) procedures

Approval testing of welding and brazing procedures shall be conducted, recorded and reported in accordance with BS EN ISO 15614-6 or BS EN 13134 as agreed between the purchaser and manufacturer.

All the requirements for the listed inspection and testing procedures shall apply to brazing as well as welding where relevant.

A pre-existing weld procedure test performed in accordance with BS EN 288 or BS 4870, previously acceptable to an Inspecting Authority, shall remain acceptable providing it satisfies the intent of the technical requirements of BS EN ISO 15614-6. However, the range of approval of such a test shall be in accordance with the ranges in BS EN ISO 15614-6 except as modified by **Cu.5.5**.

NOTE Existing procedures to BS EN 288 or the earlier BS 4870 are considered technically equivalent to BS EN ISO 15614-6 when similar types of tests have been carried out. Thus the bend tests in BS 4870 are considered equivalent to those in BS EN ISO 15614-6 even though the exact number and the bend angle differ. Similarly visual, radiographic, ultrasonic, surface crack detection, transverse tensile, hardness, macro and impact tests are considered equivalent.

Where BS EN ISO 15614-6 calls for a type of test to be performed that has not been carried out on the pre-existing BS 4870 procedure qualification tests, additional tests as described in the Introduction of BS EN ISO 15614-6 should be carried out.

The alternative methods of approval of welding procedures addressed in BS EN ISO 15607 are not permitted for welding on pressure vessels made in accordance with this specification.

Cu.5.3 Welder and operator approval

Approval testing of welders and operators shall be conducted, recorded and reported in accordance with BS EN ISO 9606-3, in the case of those engaged in welding or with BS EN ISO 13585 in the case of those engaged in brazing.

All the approval requirements shall apply to brazing as well as welding where relevant.

Cu.5.4 Production control test plates

Cu.5.4.1 Vessels in copper (C106) sheet

Production control test pieces shall not be required unless specified by the purchaser in the purchase specification. In such cases the number of pieces to be provided and the detailed tests to be carried out on these, including acceptance criteria, shall be agreed between the manufacturer, purchaser and Inspecting Authority.

NOTE Recommendations covering the preparation and testing of production test pieces, when these are required, are given in Annex Q.

Cu.5.4.2 Vessels in other copper alloys

Production control test samples shall be provided until such time as the manufacturer has demonstrated that the production process can be relied upon to produce satisfactory joints. The number of test pieces provided and the detailed tests to be made on these shall be agreed between the manufacturer, purchaser and Inspecting Authority.

Cu.5.5 Destructive testing

Weld procedure and production control testing shall be in accordance with BS EN ISO 15614-6, except where otherwise stated in **Cu.5.5**. Approval testing of welders shall be in accordance with BS EN ISO 9606-3, except where otherwise stated in **Cu.5.5**.

Destructive testing of test pieces produced for brazing procedure and operator approval purposes shall be in accordance with BS EN 12797.

Cu.5.5.1 Test temperatures

The tests shall be conducted at room temperature.

Cu.5.5.2 Transverse tensile test

The test shall be carried out in accordance with BS EN ISO 4136 and the test results shall conform to the values given in Table Cu.5.5-1.

Table Cu.5.5-1 **Mechanical test requirements for butt weld procedure and welder approval**

Material	Maximum bend radius
Aluminium bronze	8t
Copper nickel	2t
Copper and other copper alloys	1.5t

Cu.5.5.3 Side bend test (for plate at least 10 mm thick)

On completion of this test no crack or other defect at the outer surface of the test specimen shall have a dimension greater than 3 mm. Slight tearing at the edges of the test specimen shall mean that the specimen is considered not to conform to this specification.

Cu.5.5.4 Macro- and micro-examination

The specimen shall be prepared for macro-examination, and for micro-examination, when the necessity for the latter has been agreed between the manufacturer, purchaser and Inspecting Authority. The weld shall be sound, i.e. free from cracks and substantially free from discontinuities such as porosity, to an extent equivalent to that in Table 5.7-1.

Cu.5.5.5 Retests

Retests shall be as follows.

a) *Tensile.* Where a tensile test specimen does not conform to **Cu.5.5.2** two retests shall be made.

b) *Bend tests.* Where a bend test does not conform to **Cu.5.5.3** two retests shall be made.

Should any of the retest specimens not conform to the requirements, the welded seams represented by these tests shall be deemed not to conform to this supplement. If any retest specimens fail during the weld procedure approval testing the cause of failure shall be established and the whole procedure test repeated.

Cu.5.6 Non-destructive testing

Cu.5.6.1 General

See main text.

Cu.5.6.2 Parent materials

Acceptance standards for defects revealed by non-destructive testing of unwelded parent materials shall be subject to agreement between the manufacturer and the purchaser, and/or the Inspecting Authority. Where repairs by welding are authorized, non-destructive testing techniques for the repair and subsequent acceptance standards shall also be subject to agreement between the manufacturer and the purchaser, and/or the Inspecting Authority.

Cu.5.6.3 Components prepared for welding (brazing)

Main text is not applicable.

Cu.5.6.4 Non-destructive testing of welded (and brazed) joints

Cu.5.6.4.1 Components to construction Category 1

See main text and additionally:

The full length of welds and brazed joints shall be examined by dye penetrant testing.

Cu.5.6.4.2 Components to construction Category 2

See main text and additionally:

The full length of welds and brazed joints shall be examined by dye penetrant testing.

Cu.5.6.4.3 Components to construction Category 3:

See main text.

Cu.5.6.6 Choice of non-destructive test methods for welds (and brazed joints)

Cu.5.6.6.1 Radiographic techniques

See main text and additionally:

The following modifications account for the density of copper and allow the use of the tables and graphs included in BS EN ISO 17636-1:2013:

a) the required X-radiation exposure (kV) for a given thickness of copper alloy shall be obtained by using the exposure for an equivalent steel thickness multiplied by a factor of 1.5;

b) for inspection carried out using industrial isotopes, an equivalence factor of 1.1 shall be applied.

It is permissible to use other techniques by agreement between the manufacturer and the Inspecting Authority provided that it can be demonstrated that they will achieve comparable sensitivities.

Cu.5.6.6.2 Ultrasonic techniques

Main text is not applicable.

Cu.5.6.6.3 Magnetic particle techniques

Main text is not applicable.

Cu.5.6.6.4 Penetrant techniques

See main text.

Cu.5.6.6.5 Surface condition and preparation for non-destructive surface testing

See main text.

Cu.5.6.6.6 Marking, all non-destructive testing methods

See main text.

Cu.5.6.6.7 Reporting of non-destructive testing examinations

See main text.

Cu.5.7 Acceptance criteria for welded (and brazed) joints

If any defects revealed by visual inspection and non-destructive testing in welds and brazed joints do not exceed the levels specified in Table Cu.5.7-1 then the weld or joint shall be accepted without further action.

Table Cu.5.7-1 **Acceptance levels**

Defect type*		**Permitted maximum**		
Planar defects	Cracks and lamellar tears Lack of root fusion Lack of side fusion Lack of inter-run fusion Lack of root penetration	Not permitted		
Cavities	a) Isolated pores (or individual pores in a group)	Category 1		Category 2
		ϕ max. 3 mm (grade D) $\phi \leq e/4$		Such cavities may be accepted without limit provided representative mechanical specimens from production test plates conform to requirements. See Note 2.
	b) Uniformly distributed or localized porosity	$e \leq 3$ mm scattered, grade A $e > 3$ mm up to 6 mm scattered grades A and B $e > 6$ mm scattered grades A to B plus isolated, grade C not more in number than $e/2$ per 1 000 mm^2 of radiographic area†		
	c) Linear porosity	Linear indications parallel to the axis of the weld may indicate lack of fusion or lack of penetration and are therefore not permitted.		
	d) Wormholes isolated	$l \leq 3$mm $w \leq 1.5$ mm		—
	e) Wormholes aligned	As linear porosity		
	f) Crater pipes	Not permitted		
	g) Surface cavities			
Solid inclusions	a) Oxide inclusions, linear	Not permitted		
	b) Oxide inclusions, diffuse	Isolated patches permitted provided that they do not exceed $e/4$ or 3 mm max. in average diameter and provided that they are not repetitive.		
	c) Tungsten inclusions	No limit except that they shall not be accompanied by oxide inclusions and that the max. diameter of the individual inclusion does not exceed $e/4$ or 3 mm max.		
Slag inclusions	a) Individual and parallel to major weld axis	Main butt welds	$l = e \leq 100$ mm w or $h = e/10 \leq 4$ mm	
	NOTE Inclusions to be separated on the major weld axis by a distance equal to or greater than the length of the longer and the sum of the lengths of the inclusion should not exceed the total length weld.	Nozzle and branch weld attachments	Inner half of cross-section	Outer quarters of cross-section
			w or $h = e/4 \leq 4$ mm $l = c/4 < 100$ mm	w or $h = e/8 \leq 4$ mm $l = c/8 \leq 100$ mm
	b) Individual and randomly oriented (not parallel to weld axis)	As isolated pores		
	c) Non-linear groups	As localized porosity		

Table Cu.5.7-1 **Acceptance levels** *(continued)*

Defect type*		**Permitted maximum**
Profile and visible surface defects	a) Insufficient weld size	Not permitted
	b) Overlap	
	c) Shrinkage grooves and root concavity	
	d) Undercut	Slight undercut permitted, should not exceed approximately 0.5 mm.
	e) Excess penetration	$h \leq 3$ mm. Occasional local slight excess is allowable.
	f) Reinforcement shape	The reinforcement shall blend smoothly with the parent metal and dressing is not normally required provided the shape does not interfere with the specified non-destructive testing techniques.
	g) Linear misalignment	See **4.2.3**.
Grade of porosity		**Approx. average diameter of pores** mm
A		0.4
B		0.8
C		1.5
D		3 or greater
Abbreviations used: *e* is the parent metal thickness. In the case of dissimilar thicknesses *e* applies to the thinner component; *w* is the width of defect; *l* is the length of the defect; *h* is the height of the defect; ϕ is the diameter of the defect.		
† Area is the product of length and width of an envelope enclosing the affected volume of weld metal measured on a plane substantially parallel to the weld face (i.e. as seen on a radiograph). "Scattered" is defined as t to $0.25t$ per square centimetre (where t = metal thickness in millimetres).		

* For definitions of defects, see BS 499-1. See below for definitions of uniform porosity.

NOTE 1 The significant dimension of a defect in terms of its effect on service performance is the height or through thickness dimension.

In interpreting the requirements of this table, such indications having a dimension h of 1.5 mm or less, should be disregarded unless otherwise agreed between the manufacturer, the purchaser and Inspection Authority.

NOTE 2 The tensile test for a category 2 C106 copper vessel shall be deemed satisfactory providing the tensile strength of the weld is not less than:

a) for $e < t \leq 1.1e$
   $R_t = R_m e/t$;
b) for $t > 1.1e$
   $R_t = R_m/1.1$.

where

- e is the calculated thickness;
- t is the measured thickness of plate or sheet used;
- R_m is the minimum tensile strength specified for C106 PDO copper measured at room temperature.
- R_t is the tensile strength of the test specimen calculated from the maximum load and the original cross-sectional area of the parent metal;

Where failure of the test specimen is not across weld R_t shall not be less than R_m.

Cu.E Annex E: Recommendations for welded connections of pressure vessels

Cu.E.1 Typical details for principal seams

Typical details for principal seams, applicable to copper and copper alloy pressure vessels are given, as follows:

a) full penetration joint preparations for one-sided welding only for copper and copper alloys [see Figure Cu.E.1(1)];

b) full penetration joint preparations for two-sided welding for copper and copper alloys [see Figure Cu.E.1(2)];

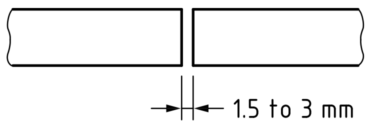
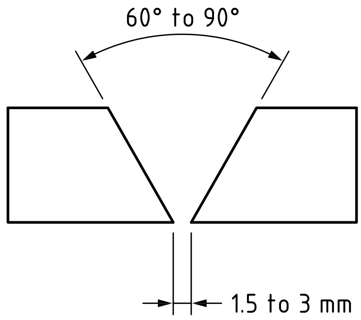
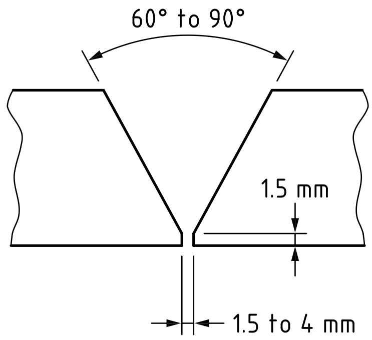
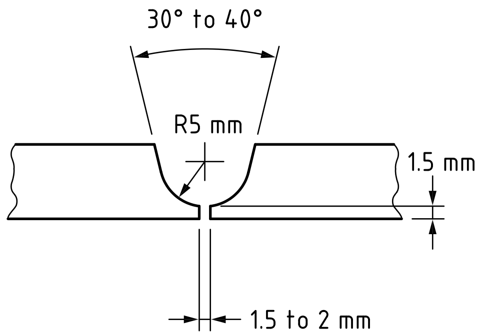
c) full penetration joint preparations for one-sided welding with temporary or permanent backing for copper and copper alloys [see Figure Cu.E.1(3)];

d) alternative joint preparations [see Figure Cu.E.1(4)].

Cu.E.2 Typical details of acceptable weld details

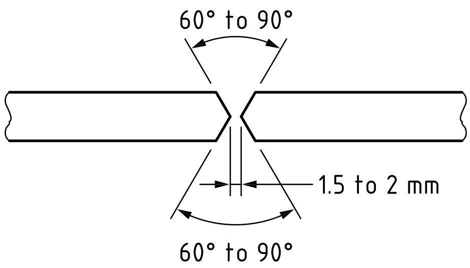
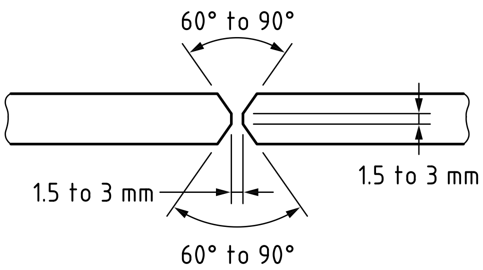
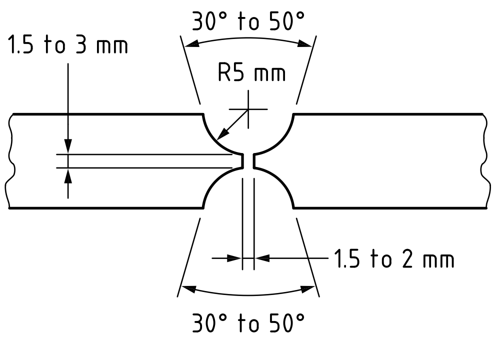
The typical details for brazing gunmetal bosses given in Figure Cu.E.2(1) are applicable for copper and copper alloy vessels.

Figure Cu.E.1(1) **Typical full penetration joint preparation for one-sided welding only: copper and its alloys**

Material thickness	Edge preparation	Remarks
Up to 7 mm		Suitable for TIG, pulsed MIG and MIG
Up to 15 mm		Suitable for TIG and MIG
Up to 20 mm		Suitable for TIG and MIG
10 mm upwards		Suitable for TIG and MIG, for pipe particularity

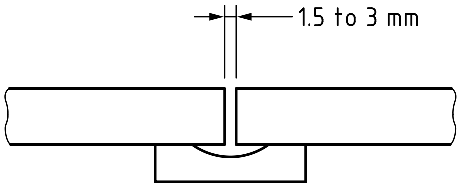
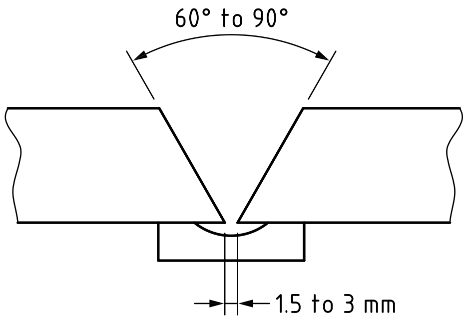
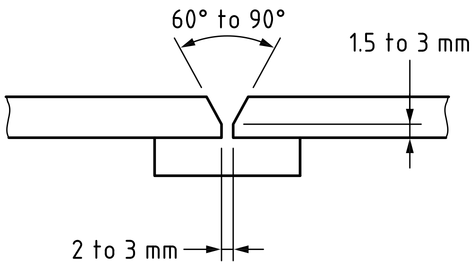
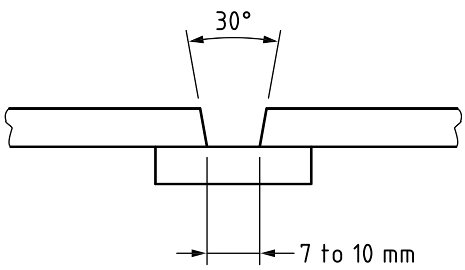
NOTE Recommended root gap of 1.5 mm only refers to the start point. Additionally gap is generally necessary during set up along the length of the weld to allow for expansion as heating proceeds.

Figure Cu.E.1(2) **Typical full penetration joint preparation for two-sided welding: copper and its alloys**

Material thickness	Edge preparation	Remarks
6 mm upwards		Suitable for TIG and MIG
6 mm upwards		Suitable for TIG and MIG
12 mm upwards		Suitable for TIG and MIG

NOTE Recommended root gap of 1.5 mm only refers to the start point. Additionally gap is generally necessary during set up along the length of the weld to allow for expansion as heating proceeds.

Figure Cu.E.1(3) **Typical full penetration joint preparation for one-sided welding with either temporary or permanent backing: copper and its alloys**

Material thickness	Edge preparation	Remarks
Up to 7 mm		Suitable for TIG, pulsed MIG and MIG
Up to 15 mm		Suitable for TIG and MIG
5 mm upwards		Permanent backed. Suitable for MIG
5 mm upwards		Permanent backed. Suitable for TIG and MIG

NOTE Recommended root gap of 1.5 mm only refers to the start point. Additional gap is generally necessary during set up along the length of the weld to allow for expansion as heating proceeds.

Figure Cu.E.1(4) **Alternative joint preparations**

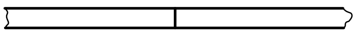
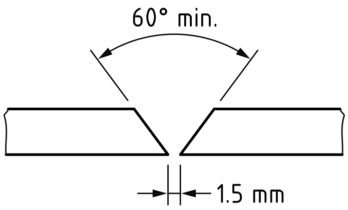
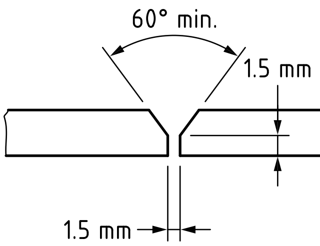
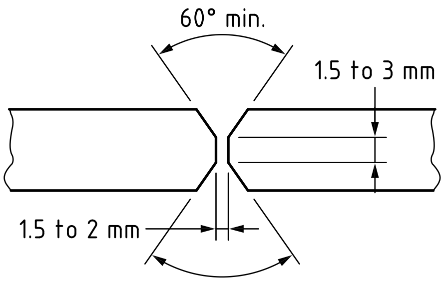
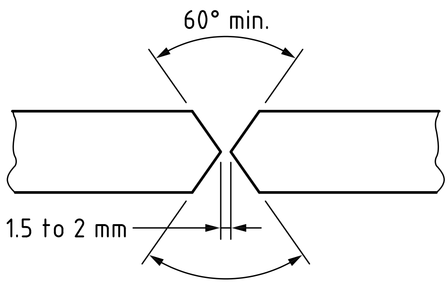
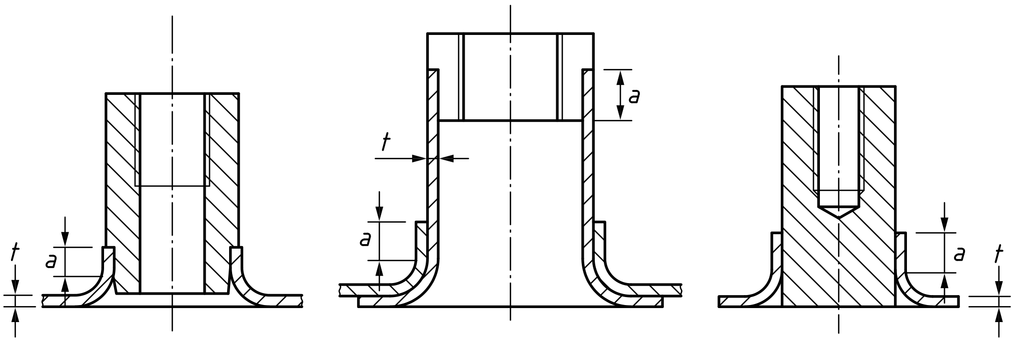
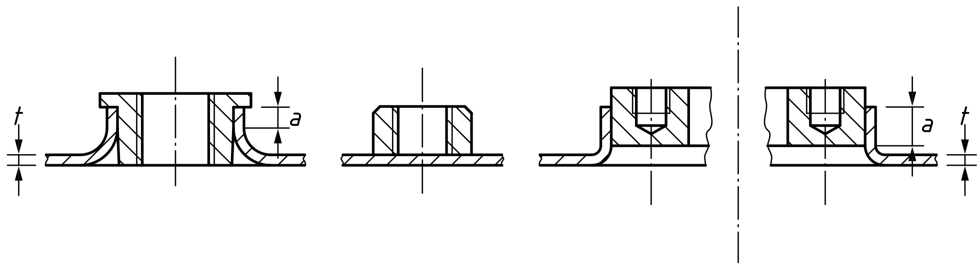
Joints	Preparation	Remarks
a) 	Close square butt	Plates having thickness not greater than 3 mm
b) 	Single-V butt	Plates having thickness not greater than 6 mm
c) 	Single-V butt with root face	
d) 	Double-V with root face	Plates over 6 mm thick
e) 	Double-V butt	

Figure Cu.E.2(1) **Typical brazed joints for gunmetal bosses to copper vessels**



a = 3*t* or 20 mm, whichever is smallest



Duplex supplement

Requirements for duplex and super duplex steels (austenitic-ferritic stainless steels) in the design and construction of unfired pressure vessels

This supplement shall be read in conjunction with the main body of the specification to establish the requirements for duplex and super duplex unfired pressure vessels. It lists the sections of the main text applicable to and not applicable to the design and construction of duplex and super duplex steel pressure vessels. In addition the supplement contains clauses specific to such vessels which replace the corresponding clauses of the main text in this context. The clause, equation, figure and table numbering of the requirements of this supplement follow those of the comparable main text requirements but are identified in the form **Du.x.y.z** to differentiate them from their main text equivalents, i.e. **x.y.z**, which might be technically different.

Du.1 Section 1: General

See main text.

Du.2 Section 2: Materials

Du.2.1 Selection of materials

Du.2.1.1 General

See main text.

Du.2.1.2 Materials for pressure parts

Du.2.1.2.1 General

See main text but substitute

For the construction of vessels in accordance with this supplement, all the materials used in the manufacture of pressure parts shall:

a) conform to BS EN standard grades given in Table Du.2.1-2; or

b) conform to BS steels given in Annex K, Table K.1-4 and Table K.1-6, for material grade 318S13; or

c) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2b)**] and shall conform to **Du.2.1.2.2**; or

d) be the subject of a "European approval of material" in accordance with the EU directive 2014/68/EU (see Note 2); or

e) conform to ASTM specifications and grades listed in Table Du.2.1-4; or

f) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE 1 The British Standards listed in Annex K, Table K.1-4 and Table K.1-6 have been withdrawn and replaced by BS EN standards (see Table Du.2.1-2 and Enquiry Case 5500/134).

NOTE 2 Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Table Du.2.1-2 **List of duplex and super duplex steels covered by BS EN material standards**

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**EN Standard**	**Material description**	**Grade**	**Material number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CEN ISO/TR 15608:2017**	**Notes**
							min	**max**		
1	Plate and strip	BS EN 10028-7	Duplex stainless steel	X2CrNiMoN22-5-3	1.4462	AT	0	75	10.1	a)
2	Plate and strip	BS EN 10028-7	Super Duplex stainless steel	X2CrNiMoCuWN25-7-4	1.4501	AT	0	75	10.2	a)
3	Bar	BS EN 10272	Duplex stainless steel	X2CrNiMoN22-5-3	1.4462	AT	0	160	10.1	a)
4	Bar	BS EN 10272	Super Duplex stainless steel	X2CrNiMoCuWN25-7-4	1.4501	AT	0	160	10.2	a)
5	Seamless tube	BS EN 10216-5	Duplex stainless steel	X2CrNiMoN22-5-3	1.4462	AT	0	30	10.1	a)
6	Seamless tube	BS EN 10216-5	Super Duplex stainless steel	X2CrNiMoCuWN25-7-4	1.4501	AT	0	30	10.2	a)
7	Welded tube	BS EN 10217-7	Duplex stainless steel	X2CrNiMoN22-5-3	1.4462	AT	0	30	10.1	a)
8	Welded tube	BS EN 10217-7	Super Duplex stainless steel	X2CrNiMoCuWN25-7-4	1.4501	AT	0	30	10.2	a)
9	Forging	BS EN 10222-5	Duplex stainless steel	X2CrNiMoN22-5-3	1.4462	AT	0	350	10.1	a)
a) Materials are considered to meet the requirements of the PER, the PED and BS EN 13445-2.										

[a] Heat treatment conditions.
AT Solution annealed

Table Du.2.1-4 **List of duplex and super duplex steels covered by ASTM material standards**

1	2	3	4	5	6	7	8		9	10
No.	**Product form**	**ASTM Standard**	**Material description**	**Nominal composition**	**UNS number**	**Heat treatment[a]**	**Thickness mm**		**Material group to PD CR ISO 15608:2000**	**Notes**
							min	**max**		
1	Plate and strip	A240	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
2	Plate and strip	A240	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	
3	Bar	A479	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
4	Bar	A479	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	
5	Tube	A789	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
6	Tube	A789	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	
7	Pipe	A790	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
8	Pipe	A790	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	
9	Forging	A182-F51	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
10	Forging	A182-F55	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	
11	Fittings	A815	Duplex stainless steel	22Cr – 5Ni – 3Mo - N	S31803	AT			10.1	
12	Fittings	A815	Super Duplex stainless steel	25Cr – 8Ni – 3Mo – W - Cu - N	S32760	AT			10.2	

[a] Heat treatment conditions.
AT solution annealed

Du.2.1.2.2 Materials covered by Du.2.1.2.1 c)

The material shall be covered by a written specification at least as comprehensive as the nearest equivalent BS EN standard listed in Table Du.2.1-2. The written specification shall as a minimum specify the manufacturing process, compositional limits for all constituents, deoxidation practice, heat treatment and appropriate mechanical properties for acceptance and other purposes.

An equivalent grouping for the material (see **2.1.1.3**) shall be chosen and agreed between manufacturer, inspection authority and purchaser. This grouping shall be used to assist in the selection of manufacturing and inspection requirements of Section 4 and Section 5.

The maximum allowable carbon, phosphorus and sulfur content in the ladle analysis shall not exceed the following values:

a) Carbon 0.03%;

b) Phosphorus 0.035%

c) Sulfur 0.015%.

Du.2.1.2.3 Additional materials for category 3 components

Main text is not applicable.

Du.2.1.2.4 Maximum service temperature

For duplex steels with Cr ≤ 24.0% (PD CEN ISO/TR 15608:2017 material group 10.1) the upper limiting service temperature shall be 300 °C.

For super duplex steels with Cr > 24.0% (PD CEN ISO/TR 15608:2017 material group 10.2) the upper limiting service temperature shall be 250 °C.

NOTE The limit of 250 °C is given for super duplex steel to reduce any risk of embrittlement that can occur with extended service above 250 °C.

Du.2.1.2.5 Chemical composition

Manufacturers of duplex and super duplex steel unfired pressure vessels shall ensure that the chemical composition of materials they use in construction have been sufficiently controlled by the steel manufacturer. The manufacturer shall ensure that an appropriate range of phase balance, tensile, impact and corrosion properties is repeatedly obtained for both the parent materials supplied and the subsequent fabrication activities e.g. forming and welding, carried out.

Du.2.1.3 Materials for non-pressure parts

See main text.

Du.2.2 Materials for low temperature applications

The impact requirements for duplex and super duplex steels used for unfired pressure vessels, designed to operate below 0 °C, shall be in accordance with Annex Du.D.

Du.2.3 Nominal design strength

Du.2.3.1 General

Du.2.3.1.1 The nominal design strength f_N shall be determined:

a) for materials to BS 1501-3 grade 318S13 or BS 1503 grade 318S13, taking values from Annex K, Table K.1-4 or Table K.1-6 respectively;

b) for materials to BS EN standards, taking values from Table Du.2.3-1;

*NOTE 1 The design strengths in Table Du.2.3-1 have been derived in accordance with **2.3.3.2** a), using tensile strength R_m and 0.2% proof stress $R_{p0.2}$ data from the relevant BS EN standard.*

c) for materials to ASTM standards, taking values from Table Du.2.3-2, which may be used without verification of $R_{e(T)}$;

*NOTE 2 The design strengths in Table Du.2.3-2 have been derived in accordance with **2.3.3.2** b), using tensile strength R_m equal to S_u from Table U and yield strength $R_{e(T)}$ equal to S_y from Table Y-1 of ASME II Part D 2015 Edition.*

d) for materials other than those listed in Table K.1-4, Table K.1-6, Table Du.2.3-1 or Table Du.2.3-2, the time-independent design strength shall be derived in accordance with **2.3.3.2** using the 0.2% proof stress $R_{p0.2}$.

The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined by one of the above methods. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

Table Du.2.3-1 **Design strength values (N/mm²) for BS EN duplex and super duplex steels**

Material standard	EN No.	Thickness mm	Min tensile strength R_m N/mm²	Min 0.2% proof stress $R_{p0.2}$ N/mm²	Elongation %	Value of *f* for design temperatures (in °C) not exceeding				
						50	100	150	200	250
BS EN 10028-7	1.4462	≤75	640	460	25	272	248	223	210	200
BS EN 10028-7	1.4501	≤50	730	530	25	311	295	280	267	253
BS EN 10272	1.4462	≤160	650	450	25	277	250	223	210	200
BS EN 10272	1.4501	≤160	730	530	25	311	295	280	267	253
BS EN 10216-5	1.4462	≤30	640	450	22	272	248	223	207	197
BS EN 10216-5	1.4501	≤30	800	550	20	340	310	280	267	253
BS EN 10217-7	1.4462	≤30	700	450	25	298	261	223	207	197
BS EN 10217-7	1.4501	≤30	730	530	25	340	310	280	267	253
BS EN 10222-5	1.4462	≤350	680	450	25	289	256	223	210	200

Table Du.2.3-2 **Design strength values (N/mm²) for ASTM A240, A789, A790, A182, A479 and A815 duplex and super duplex steels**

UNS No.	Min tensile strength R_m N/mm²	Min 0.2% proof stress $R_{p0.2}$ N/mm²	Elongation %	Value of *f* for design temperatures (in °C) not exceeding					
				50	100	150	200	250	300
S31803	621	448	25	264	248	231	221	215	209
S32205	655	448	25	279	255	231	221	215	209
S32760	750	550	25	319	292	265	256	252	—

Du.3 Section 3. Design

Du.3.1 General

See main text, except:

Du.3.1.7 Requirements to safeguard against brittle fracture of pressure vessels manufactured in duplex and super duplex steels are given in Annex Du.D (see also **Du.2.2**). Requirements to safeguard against fatigue failure of pressure vessels manufactured in duplex and super duplex steels are given in Annex Du.C.

Du.3.4 Construction categories and design stresses

Du.3.4.1 Construction categories

See main text with the following modified Table Du.3.4-1.

Table Du.3.4-1 **Construction categories**

Construction category	Non-destructive testing (NDT)	Permitted material groups	Maximum nominal thickness of component[a] (see 1.6) (mm)	Temperature limits	
				Upper	Lower
1	100% (see **5.6.4.1**)	10	None, except where NDT method limits apply	See **Du.2.1.2.4** and **Du.2.3.1.1**	See Annex Du.D
2	Limited random (spot) (see **5.6.4.2**)	10	16	See **Du.2.1.2.4** and **Du.2.3.1.1**	See Annex Du.D
3	Not permitted				

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension of the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.

Du.3.5 Vessels under internal pressure

See main text.

Du.3.6 Vessels under external pressure

See main text and additionally

Du.3.6.1.1 Notation

See main text but substitute

E is the modulus of elasticity of material of part under consideration at design temperature from Table Du.3.6-3;

s is the factor relating f to effective yield point of material; for the purposes of **Du.3.6**, s shall be taken to be 1.1 for duplex and super duplex steels.

Table Du.3.6-3 *E* values for duplex steels (modulus of elasticity)

Temperature °C	Duplex steel N/mm^2	Super duplex steel N/mm^2
−50	205 000	210 000
0	200 000	205 000
50	195 000	200 000
100	190 000	195 000
150	185 000	190 000
200	180 000	185 000
250	175 000	180 000
300	170 000	

NOTE Linear interpolation is permitted in the above table.

Du.3.7 to Du.3.13 *See main text*.

Du.4 Section 4. Manufacturing and workmanship

Du.4.1 General aspects of construction

See main text.

Du.4.2 Cutting, forming and tolerances

Du.4.2.1 Cutting of material

Du.4.2.1.1 Method

All materials shall be cut to size and shape by cold shearing, plasma cutting, laser cutting, or by machining; the use of thermal cutting processes is not permitted.

NOTE There is no particular requirement for material cut or shaped by any of the above processes to be dressed before welding, unless specific contractual requirements dictate otherwise.

Where machining is used, the chloride content of the machining oil shall be no greater than 1.0 ppm.

Du.4.2.1.2 Examination of cut edges

Before carrying out further work, cut surfaces and heat affected zones shall be examined for defects, including laminations, cracks and slag inclusions.

The purchaser shall specify in the purchase specification where supplementary non-destructive testing of cut edges is required in addition to the usual visual methods.

The manufacturer shall notify any major defects, and provide a proposed method of rectification, to the purchaser and Inspecting Authority for their agreement.

Any material damaged in the process of cutting to size and preparation of edges shall be removed by machining or grinding back to undamaged metal.

Du.4.2.2 Forming of shell sections and plates

Du.4.2.2.1 General

See main text, additionally

An appropriate post forming heat treatment is considered to be a solution treatment followed by rapid cooling to restore the properties to those of the as-received material. The manufacturer shall provide evidence that the heat treatment has restored the properties.

Du.4.2.2.2 Plates welded prior to hot or cold forming

The welding of plates prior to forming is not permitted.

Du.4.2.2.3 Cold forming

For cold formed items, an appropriate post forming heat treatment (see **Du.4.2.2.1**) shall be applied in the following instances, to restore the properties to those of the as-received material.

a) For items designed to operate above 0 °C, when the inside radius of curvature of a formed cylindrical pressure part is less than 10 times the nominal thickness.

b) For items designed to operate at or below 0 °C, when the inside radius of curvature of a formed cylindrical pressure part is less than 18 times the nominal thickness.

c) For all dished ends, including conical and toriconical ends.

Du.4.2.2.4 Hot forming

For hot formed items, an appropriate post forming heat treatment (see **Du.4.2.2.1**) shall be applied.

Du.4.2.2.4.1 Warm forming

Warm forming is defined by a maximum forming temperature of 300 °C, above this temperature items shall be treated as having been hot formed.

For warm formed items the requirements of **Du.4.2.2.3** shall be applied.

Du.4.2.2.5 Manufacture of shell plates and ends

See main text.

Du.4.2.2.6 Examination of formed plates

See main text.

Du.4.2.3 to Du.4.2.6

See main text.

Du.4.3 Welded joints

Du.4.3.1 General

Production welding shall not commence until:

a) welding procedures proposed have been approved in accordance with **5.2**;

b) welder/operators have been approved in accordance with **5.3**;

c) where stipulated by the purchaser in the purchase specification, production control test plate requirements have been agreed.

Du.4.3.2 Welding consumables

Welding consumables (e.g. wire, electrodes, flux, shielding gas) shall be the same type as those used in the welding procedure and shall be within the limits specified in BS EN ISO 15614-1. To ensure that no unacceptable deterioration occurs, the storing and handling of welding consumables shall be controlled in accordance with procedures written on the basis of the makers' information.

The manufacturer of the vessel shall provide evidence that the deposited weld metal is suitable in all respects for the intended duty and has tensile properties derived from the weld procedure tests not less than those specified for the parent material.

Du.4.3.3 Preparation of plate edges and openings

Weld preparations and openings of the required shape shall be formed in accordance with **Du.4.2.1**.

The profile of the weld preparation shall be as specified in the approved welding procedure (see **5.2**).

Du.4.3.4 Assembly for welding

See main text.

Du.4.3.5 Attachments and the removal of temporary attachments

See main text, additionally

Du.4.3.5.3 Attachments of dissimilar metals

Dissimilar metal welds shall only be made where agreement has been obtained from the purchaser and Inspecting Authority, and where an approved Welding Procedure Specification is present.

Du.4.3.6 Butt joints

See main text.

Du.4.3.7 Welding: general requirements

See main text, additionally

Du.4.3.7.7 Autogenous welding shall not be used.

Du.4.4 Permanent joints other than welding

See main text, except that the requirements for brazing are not applicable to duplex and super duplex steels.

Du.4.5 Heat treatment

Du.4.5.1 Preheat requirements

Preheating of duplex and super duplex steels is not recommended. It may be used to remove condensation, provided that the temperature does not exceed 100 °C, and weld preparations have already been cleaned.

Du.4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

Main text is not applicable.

Du.4.5.3 Post-weld heat treatment

Post weld heat treatment of duplex and super duplex steels is not normally needed but may be applied by agreement between the purchaser, manufacturer and Inspection Authority. In these circumstances it shall be noted that significant degradation of toughness might result and appropriate allowances shall be made for this effect.

Local post-weld heat treatment is not permitted.

Appropriate welding procedures shall be suitably qualified in accordance with BS EN ISO 15614-1.

Du.4.5.4 Methods of heat treatment

See main text.

Du.4.5.5 Post-weld heat treatment procedure

The post-weld heat treatment procedure shall be by agreement between the manufacturer, purchaser and Inspecting Authority.

Du.4.5.6 Mechanical properties after heat treatment

See **Du.4.5.3**.

Du.4.6 Surface finish

See main text.

Du.5 Section 5. Inspection and testing

Du.5.1 General

See main text.

Du.5.2 Approval testing of fusion welding procedures

See main text, additionally

Welding procedures shall be qualified with a maximum variation on heat input of ±10% per weld run, calculated in accordance with BS EN 1011-1.

Testing shall include microstructure examination and ferrite determination in accordance with ASTM E562 at the following locations:

a) Weld cap (including, where applicable, both weld caps of double sided welds).

b) Weld root.

c) HAZ adjacent to the weld cap (including, where applicable, the HAZ adjacent to both weld caps of double sided welds).

d) HAZ adjacent to the weld root.

Both weld and HAZ shall demonstrate a phase balance between 30% and 70% ferrite, and shall demonstrate freedom from deleterious phases (e.g. Sigma, Laves and Chi), however these need not necessarily be a cause for rejection if the corrosion and mechanical properties can be justified in a particular application.

Charpy V impact testing shall be carried out in the weld metal, and at the fusion line (FL), FL + 2 mm and HAZ (FL + 5 mm). The number of sets of impact test specimens shall be in accordance with Annex D.7.2.3 b) irrespective of the heat treatment condition. The tests shall demonstrate an average impact energy value of 50 J (average of results from three test pieces), with no individual value below 40 J. For material less than 12 mm thick testing is still required though sub size specimens may be used, in which case the required energy values shall be in accordance with Table D.2.

All test specimens shall be prepared after the test pieces have been given a heat treatment that is the same as that applied to the vessel. In the case of production test plates, the purchaser might specify that the plates be heat treated with the vessel.

Fillet welds shall be separately qualified.

Du.5.3 Welder and operator approval

See main text, additionally

Welder and welding operator approvals shall be only valid for the range of base metal thickness and weld deposit thickness covered by the weld procedure qualification in accordance with BS EN ISO 15614-1, that backs up the welding procedure specification used in the qualification test. In addition, the welder approval is limited to the testing positions (as specified in BS EN ISO 6947) used in the qualification test piece.

Welder approval tests shall be carried out every 12 months, irrespective of the level of production welding being carried out, or when the welder has not carried out manual welding of duplex and/or super duplex steels for a period of more than 3 months.

Welding operator approval tests shall be carried out every 2 years.

Welder approval tests associated with branch, fitting or fillet welding procedure specifications may be simulated by suitable weld joint set ups using plate and/or pipe, or by a BS EN ISO 15613 pre-production test weld.

In addition to the specified test methods, all welder and welding operator approval tests shall be subject to 2-off micro sections; where the test piece represents a branch weld these shall be taken from the crotch and flank positions.

Both macro sections shall be subject to hardness testing and micro structural analysis; both weld and HAZ shall demonstrate a phase balance between 30% and 70% ferrite, and shall demonstrate freedom from deleterious phases (e.g. Sigma, Laves and Chi), unless otherwise agreed with the purchaser and Inspecting Authority.

Du.5.4 Production control test plates

Main text is not applicable.

Du.5.4.1 Vessels in duplex and super duplex steels

The testing requirements shall be as specified in **Du.5.2**, unless specifically stated otherwise below.

For each vessel, production control test plates shall be carried out as follows:

a) **Longitudinal welds**

   1-off production control test plate shall be provided, representative of all longitudinal welds, per welding process, covered by a single approved welding procedure qualification record.

When the manufacturer can demonstrate completion of a minimum of 10-off previous satisfactory production control test plates, involving more than one cast and/or grade of material, the number of production control test plates may be reduced to one test plate per 5-off vessels of the same material grade. Welding of the production control test plate shall be witnessed by an independent Inspecting Authority.

When welding is carried out by a mechanised, semi-automatic or automatic process, the reduction to one test plate per 5-off vessels may be carried out when the results from a minimum of 5-off previous satisfactory production control test plates are available.

b) **Circumferential welds**

Production control test plates are only required where the welding process is different to that used for longitudinal welds. In such cases the requirements of **Du.5.4.1a)** shall apply.

Du.5.5 Destructive testing

See main text.

Du.5.6 Non-destructive testing

See main text.

Du.5.7 Acceptance criteria for weld defects revealed by visual examination and non-destructive testing

See main text, except for ***5.7.3****, replaced by.*

Du.5.7.3 Repair of welds

Du.5.7.3.1 General

All repair welds shall be subject to a welding procedure qualification test which, as a minimum, repeats the testing requirements of the original weld as described by **Du.5.2**.

Phase balance assessments of the repair weld should be carried out in accordance with ASTM E562, supplemented by a visual examination of the micro sections for precipitated phases.

Multiple repairs, where permitted, should ensure that the secondary repair site is excavated with a larger profile than the original repair, and that the HAZ resulting from the first repair is fully removed.

Du.5.7.3.2 Duplex weld repairs

The maximum number of permitted weld repairs in any one area shall be two, unless a specific repair weld simulation test has been carried out in accordance with BS EN ISO 15613.

Du.5.7.3.3 Super duplex weld repairs

Multiple weld repairs are prohibited, unless specifically agreed between the manufacturer, purchaser and Inspecting Authority.

Du.5.8 Pressure tests

See main text.

Du.C Annex C: Assessment of vessels subject to fatigue

See main text, additionally

Du.C.1.3 Symbols

See main text but substitute

E is the modulus of elasticity at the maximum operating temperature from Table Du.3.6-3 (in N/mm^2);

Du.C.3.2.4 Environmental effects

For duplex steels no guidance is given, and expert advice should be sought on the effect of corrosive environments on fatigue strength.

Du.D Annex D: Requirements for vessels designed to operate below 0 °C

See main text but substitute

Du.D.2 Scope

The requirements of this annex apply to the design, materials and manufacturing processes of duplex and super duplex vessels which have a minimum design temperature, θ_D, less than 0 °C.

These requirements apply to all pressure parts and attachments welded thereto but not to non-pressure parts such as internal baffles, etc. that are not welded to a pressure part and are not otherwise an integral part of a pressure part.

Where there is a requirement for a minimum design temperature below -50 °C, and/or a nominal thickness of greater than 50 mm, specific material properties shall be agreed between the manufacturer and the purchaser.

Annex U provides recommendations for the application of fracture mechanics for cases outside the scope of this annex.

Du.D.6 Required impact parameters

Du.D.6.1 Required Charpy impact test temperature

The impact test temperature shall be determined graphically from Figure Du.D.1a), Figure Du.D.1b) or Figure D.1c). The figures relate the design reference temperature, the design reference thickness and the required impact test temperature.

If the design reference temperature and the design reference thickness are known Figure Du.D.1a), Figure Du.D1b) and Figure D.1c) can be used to determine the required material impact test temperature.

Alternatively, if the actual impact test temperature for the material is known the figures can be used to determine the minimum acceptable design reference temperature for any given design reference thickness, or the maximum acceptable design reference thickness for any given design reference temperature.

When the reference thickness exceeds 50 mm an alternative method of assessment shall be used (e.g. fracture mechanics as outlined in Annex U).

For all thicknesses below 10 mm, the 10 mm lines are to be used. When the 10 mm lines are used, the energy requirements at 10 mm shall be scaled down in proportion to specimen thickness.

Figure Du.D.1a) **Permissible design reference temperature/design reference thickness/required impact test temperature relationships for duplex and super duplex steels with $R_e \leq 385$ N/mm²**

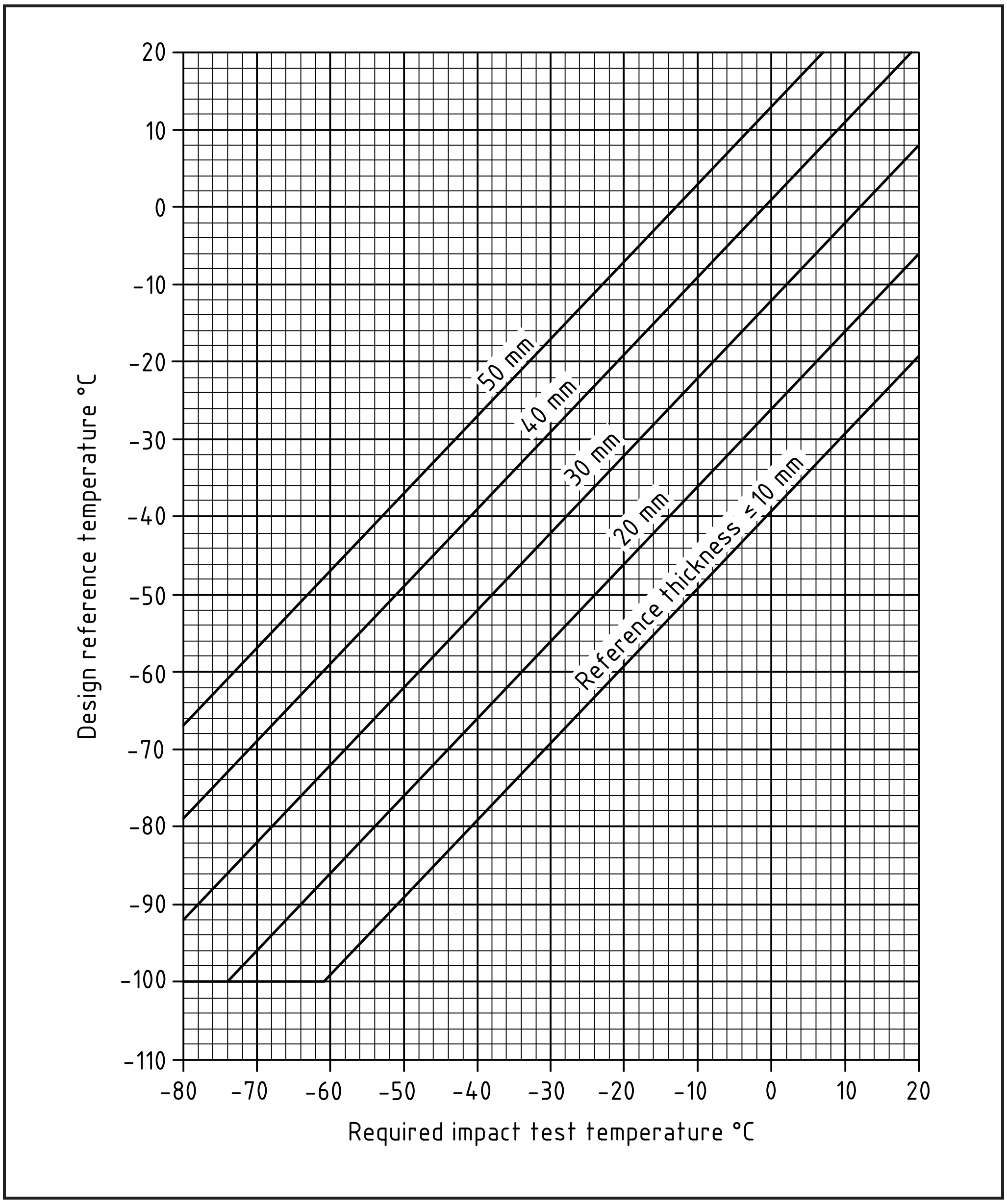


Figure Du.D.1b) **Permissible design reference temperature/design reference thickness/required impact test temperature relationships for duplex and super duplex steels with $R_e \leq 465$ N/mm^2**

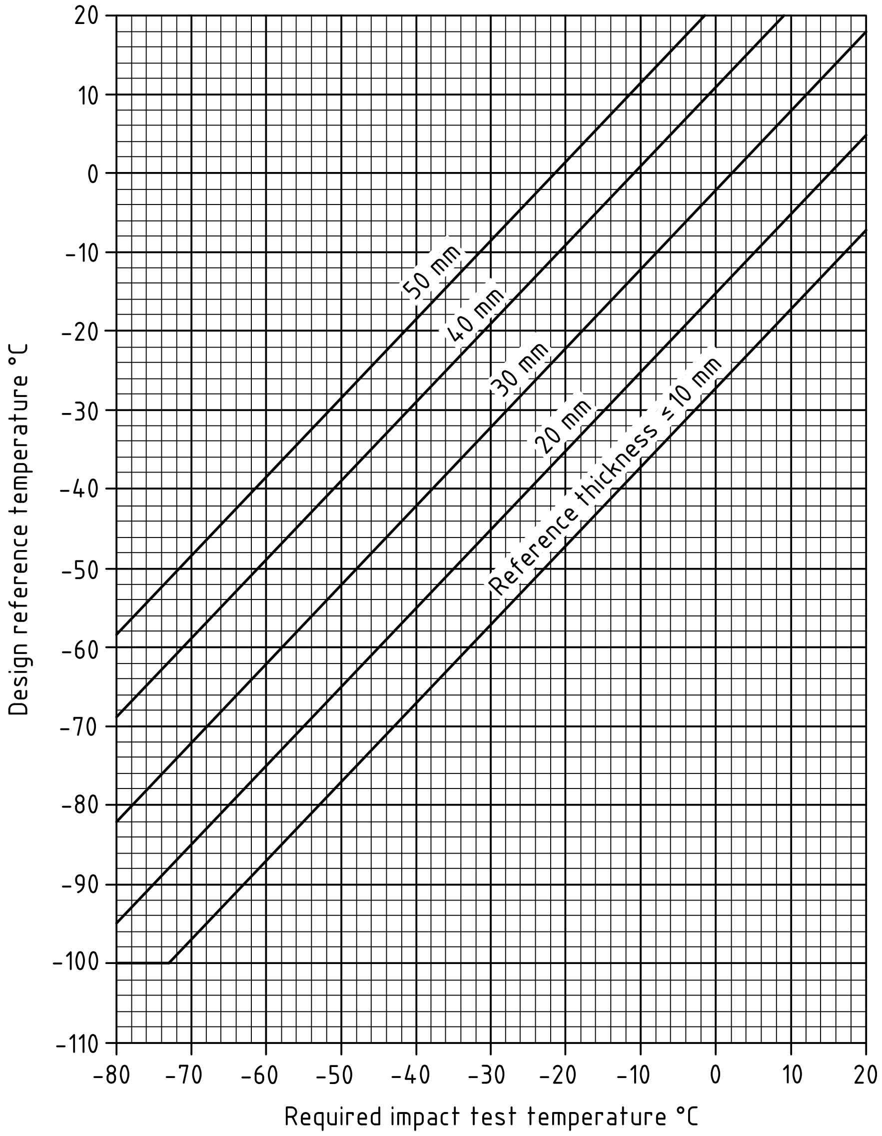
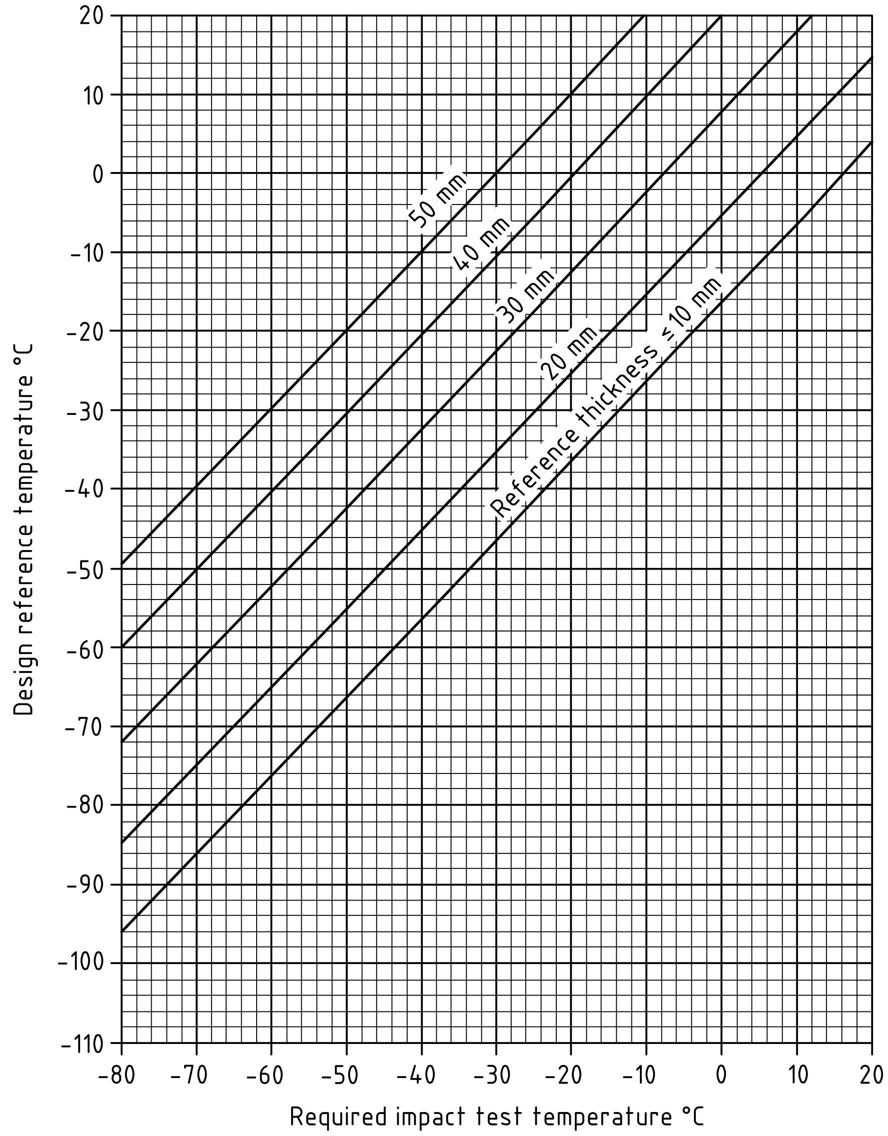


Figure Du.D.1c) **Permissible design reference temperature/design reference thickness/required impact test temperature relationships for duplex and super duplex steels with $R_e \leq 550$ N/mm²**



Du.D.6.2 Required impact energy values

See main text.

Du.D.9 Start-up and shut down procedure

To avoid brittle fracture occurrence of pressure vessels made of duplex or super duplex steels during start-up and shut-down procedures, the pressure shall not exceed 50 % of the design pressure at temperatures lower than 20 °C.

This start-up and shut-down procedure need not be considered if the evaluation of the required impact test temperature using the methods given in **D.5.1.1** and **Du.D.6.1** allows the full design pressure to be used at temperatures lower than 20 °C.

Du.D.10 Pressure test

The hydrostatic test of pressure vessels made of duplex or super duplex steels shall not be carried out at material temperatures lower than 10 °C.

This temperature limitation need not be considered if the evaluation of the required impact test temperature using the methods given in **D.5.1.1** and **Du.D.6.1** allows the test pressure to be used at temperatures lower than 10 °C.

Nickel supplement

Requirements for nickel and nickel alloys in the design and construction of unfired pressure vessels

This supplement shall be read in conjunction with the main body of the specification to establish the requirements for nickel and nickel alloy unfired pressure vessels. It lists the sections of the main text applicable to, and those not applicable to, the design and construction of nickel and nickel alloy pressure vessels. In addition the supplement contains clauses specific to such vessels which replace the corresponding clauses of the main text in this context. The clause, equation, figure and table numbering of the requirements of this supplement follow those of the comparable main text requirements but are identified in the form **Ni.x.y.z** to differentiate them from their main text equivalents, i.e. **x.y.z**, which might be technically different.

Ni.1 Section 1: General

See main text.

Ni.2 Section 2: Materials

Ni.2.1 Selection of materials

Ni.2.1.1 General

See main text, but substitute.

Ni.2.1.1.3 For the ease of reference in this supplement, materials are identified by a group number which has been derived from PD CEN ISO/TR 15608:2017, Table 4. This grouping is summerized in Table Ni.2.1-1.

Table Ni.2.1-1 **Material grouping**

Group	Type of nickel and nickel alloys
41	Pure nickel
42	Nickel-copper alloys (Ni-Cu) Ni ≥ 45%, Cu ≥ 10%
43	Nickel-chromium alloys (Ni-Cr-Fe-Mo) Ni ≥ 40%
44	Nickel molybdenum alloys (Ni-Mo) Ni ≥ 45%, Mo ≤ 32%
45	Nickel-iron-chromium alloys (Ni-Fe-Cr) Ni ≥ 31%
46	Nickel-chromium-cobalt alloys (Ni-Cr-Co) Ni ≥ 45%, Co ≥ 10%
47	Nickel-iron-chromium-copper alloys (Ni-Fe-Cr-Cu) Ni ≥ 45%
48	Nickel-iron-cobalt alloys (Ni-Fe-Co-Cr-Mo-Cu) 31% ≤ Ni ≤ 45% and Fe ≥ 20%

Ni.2.1.2 Materials for pressure parts

Ni.2.1.2.1 General

See main text but substitute

For the construction of vessels in accordance with this supplement, all the materials used in the manufacture of pressure parts shall:

a) conform to British Standards listed in Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5; or

b) conform to BS EN standards listed in Table Ni.2.3-6 (see **Ni.2.1.2.2**); or

c) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2b)**] and shall conform to **Ni.2.1.2.3**; or

d) be the subject of a "European approval of material" in accordance with the EU directive 2014/68/EU (see Note 2); or

e) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE 1 The British Standards listed in Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5 have been withdrawn and have not been replaced. Enquiry Case 5500/140 gives guidance on nickel and nickel alloy materials covered by EAMs (European approval of materials) and ASME/ASTM standards.

NOTE 2 Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Ni.2.1.2.2 Materials covered by Ni.2.1.2.1b)

Material numbers 1.4529, 1.4539, 1.4558, 1.4563, 1.4958, 1.4958 + RA and 1.4959 are classified as group 8.2 austenitic stainless steels in PD CEN ISO/TR 20172:2021, Table 1, but for this specification they shall be treated as group 45 nickel-iron-chromium alloys and the requirements of this supplement shall be applied for vessels or components designed and manufactured from these materials.

Ni.2.1.2.3 Materials covered by Ni.2.1.2.1c)

Ni.2.1.2.3.1 The material shall be covered by a written specification at least as comprehensive as the nearest equivalent British Standard listed in Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5. The specification shall, as a minimum, specify the manufacturing process, compositional limits for all constituents, heat treatment and appropriate mechanical properties for acceptance and other purposes.

Ni.2.1.2.3.2 The specified minimum percentage elongation at fracture, referred to a gauge length of $5.65\sqrt{S_o}$ [1)] or 50 mm, shall be appropriate to the type of material with a lower limit of 25%, unless the material is subject to special agreement (see **2.1.1.2**).

Ni.2.1.2.3.3 Materials which have mechanical properties enhanced by precipitation hardening are excluded from this supplement, unless they are to be used for bolting applications.

Ni.2.1.2.3.4 If the design temperature exceeds 450 °C, stress rupture data shall be available for determining the time-dependent design strength.

Ni.2.1.2.4 Additional materials for category 3 components

Main text is not applicable.

Ni.2.1.3 Materials for non-pressure parts

See main text.

Ni.2.2 Materials for low temperature application

The alloys specified in **Ni.2.1.1** are not susceptible to brittle fracture and no special provisions are necessary for their use at temperatures down to −196 °C.

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1) S_o is the original cross-sectional area of the gauge length of the tensile test specimen.

Ni.2.3 Nominal design strength

Ni.2.3.1 General

Ni.2.3.1.1 Design strength values, appropriate to specified British Standard materials, shall be in accordance with Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5.

*NOTE These design strengths have either been calculated in accordance with the rules given in **2.3.3.3**b) using the typical mechanical properties given in BS 3072, BS 3074 and BS 3076 or, for alloy NA 21, have been assumed to be equal to the design stresses in ASME II Part D, for the equivalent ASTM material. Higher design values, calculated in accordance with **2.3.3.3**a) may be used provided elevated temperature tests on representative material are carried out to confirm or determine the properties at the design temperature.*

Ni.2.3.1.2 For materials other than those listed in Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5, the time-independent design strength shall be derived in accordance with **2.3.3.3** using the 1% proof stress $R_{p1.0}$. The time-dependent design strength shall be derived in accordance with **2.3.4**.

Ni.2.3.1.3 The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined in accordance with **Ni.2.3.1.1** or **Ni.2.3.1.2**. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

Table Ni.2.3-1 **Design strength values (N/mm^2) for nickel and nickel alloy plate conforming to BS 3072**

Type	Group	R_m N/mm^2	R_e N/mm^2	Values of f for design temperatures (in °C) not exceeding								
				50	100	150	200	250	300	350	400	450
NA 11	41	380	130	87	87	87	86	84	82	79	—	—
NA 12	41	350	110	73	69	64	62	60	59	58	55	52
NA 13	42	480	220	147	134	122	107	107	107	107	103	103
NA 14	43	550	265	176	168	160	148	143	138	134	131	128
NA 15	45	520	235	157	149	142	134	129	124	121	117	114
NA 16	45	590	270	180	171	161	152	147	141	138	134	131
NA 21	43	830	415	236	236	236	232	228	224	221	217	213

NOTE 1 The above design strength values are for hot and cold rolled material in the annealed condition.

*NOTE 2 The design strength is derived from 1% proof stress ($R_{p1.0}$) using **2.3.3.3**b); except for NA 21 for which values from ASME II Part D 2015 Edition, Table 1B, specification SB-443 Grade 1, UNS*

No. N06625, in the annealed condition have been used.

Table Ni.2.3-2 **Design strength values (N/mm²) for nickel and nickel alloy seamless tube conforming to BS 3074**

Type	Group	R_m N/mm²	R_e N/mm²	Values of *f* for design temperatures (in °C) not exceeding								
				50	100	150	200	250	300	350	400	450
NA 11	41	380	130	87	87	87	86	84	82	79	—	—
NA 12	41	350	110	73	72	70	69	67	65	62	59	55
NA 13	42	480	220	147	137	127	117	117	117	117	114	110
NA 14	43	550	230	153	145	136	128	124	121	117	114	110
NA 15	45	450	200	133	124	116	107	103	100	97	93	90
NA 16	45	520	200	133	127	120	114	110	107	103	100	97
NA 21[a]	43	—	—	—	—	—	—	—	—	—	—	—

NOTE 1 The above design strength values are for hot worked and annealed pipe and tube up to and including 125 mm outside diameter.

*NOTE 2 The design strength is derived from 1% proof stress ($R_{p1.0}$) using **2.3.3.3**b).*

[a] No data.

Table Ni.2.3-3 **Design strength values (N/mm²) for nickel and nickel alloy seamless tube conforming to BS 3074**

Type	Group	R_m N/mm²	R_e N/mm²	Values of *f* for design temperatures (in °C) not exceeding								
				50	100	150	200	250	300	350	400	450
NA 11	41	380	130	87	87	86	84	82	79	—	—	—
NA 12	41	350	110	73	72	70	69	67	65	62	59	55
NA 13	42	480	220	147	137	127	117	117	117	117	114	114
NA 14	43	550	265	176	167	157	148	143	138	134	131	127
NA 15	45	520	235	157	143	128	114	108	103	100	97	97
NA 16	45	590	270	180	167	147	131	128	124	121	117	114
NA 21	43	830	415	236	236	236	232	228	224	221	217	213

NOTE 1 The above design strength values are for cold worked and annealed tube (up to and including 115 mm outside diameter only for NA 11, NA 12, NA 13, NA 14, NA 15 and NA 16).

*NOTE 2 The design strength is derived from 1% proof stress ($R_{p1.0}$) using **2.3.3.3**b), except for NA 21 for which values from ASME II Part D 2015 Edition, Table 1B, specification SB-444 Grade 1, UNS No. N06625, in the annealed condition have been used.*

Table Ni.2.3-4 **Design strength values (N/mm^2) for nickel and nickel alloy seamless tube conforming to BS 3074**

Type	Group	R_m	R_e	Values of *f* for design temperatures (in °C) not exceeding								
		N/mm^2	N/mm^2	50	100	150	200	250	300	350	400	450
NA 11	41	380	110	73	73	72	72	71	69	65	—	—
NA 12	41	350	95	63	63	62	62	60	58	58	55	50
NA 13	42	480	195	130	122	115	107	107	107	107	103	103
NA 14	43	520	195	130	122	115	107	103	100	100	97	96
NA 15	45	450	200	133	124	116	107	103	100	100	93	90
NA 16	45	520	200	133	127	120	114	110	107	107	100	97
NA 21[a]	43	—	—	—	—	—	—	—	—	—	—	—

NOTE 1 The above design strength values are for hot worked and annealed pipe and tube above 125 mm outside diameter.

*NOTE 2 The design strength is derived from 1% proof stress ($R_{p1.0}$) using **2.3.3.3**b).*

[a] No data.

Table Ni.2.3-5 **Design strength values (N/mm^2) for nickel and nickel alloy forgings conforming to BS 3076**

Type	Group	R_m	R_e	Values of *f* for design temperatures (in °C) not exceeding								
		N/mm^2	N/mm^2	50	100	150	200	250	300	350	400	450
NA 11	41	380	130	87	87	86	86	84	83	79	—	—
NA 12	41	340	95	63	63	62	62	60	59	59	55	50
NA 13	42	480	195	130	122	115	107	107	107	107	103	103
NA 14	43	550	265	176	168	160	148	143	138	134	131	128
NA 15	45	520	235	157	149	142	134	129	124	121	117	114
NA 16	45	590	270	180	171	161	152	147	141	138	134	131
NA 21	43	830	415	217	217	215	209	204	199	194	191	187

NOTE 1 The design strength values are for cold worked and annealed and hot worked and annealed forgings.

*NOTE 2 The design strength is derived from 1% proof stress ($R_{p1.0}$) using **2.3.3.3**b), except for NA 21 for which values from ASME II Part D 2015 Edition, Table 1B, specification SB-564, UNS No. N06625, in the annealed condition, thickness ≤ 250 mm have been used.*

Table Ni.2.3-6 **BS EN material standards for nickel alloys**

Standard	Material number	Group	Material name	Product form
BS EN 10028-7	1.4529	45[a]	X1NiCrMoCuN25-20-7	Plate and strip
BS EN 10028-7	1.4539	45[a]	X1NiCrMoCu25-20-5	Plate and strip
BS EN 10028-7	1.4563	45[a]	X1NiCrMoCu31-27-4	Plate and strip
BS EN 10028-7	1.4958	45[a]	X5NiCrAlTi31-20	Plate and strip
BS EN 10028-7	1.4958 + RA	45[a]	X5NiCrAlTi31-20 + RA	Plate and strip
BS EN 10028-7	1.4959	45[a]	X8NiCrAlTi32-21	Plate and strip
BS EN 10216-5	1.4529	45[a]	X1NiCrMoCuN25-20-7	Seamless tube
BS EN 10216-5	1.4539	45[a]	X1NiCrMoCu25-20-5	Seamless tube
BS EN 10216-5	1.4558	45[a]	X2NiCrAlTi32-20	Seamless tube
BS EN 10216-5	1.4563	45[a]	X1NiCrMoCu31-27-4	Seamless tube
BS EN 10216-5	1.4958	45[a]	X8NiCrAlTi31-20	Seamless tube

Table Ni.2.3-6 **BS EN material standards for nickel alloys** *(continued)*

Standard	Material number	Group	Material name	Product form
BS EN 10216-5	1.4958 + RA	45[a]	X8NiCrAlTi31-20 + RA	Seamless tube
BS EN 10216-5	1.4959	45[a]	X8NiCrAlTi32-21	Seamless tube
BS EN 10217-7	1.4529	45[a]	X1NiCrMoCuN25-20-7	Welded tube
BS EN 10217-7	1.4539	45[a]	X1NiCrMoCu25-20-5	Welded tube
BS EN 10217-7	1.4563	45[a]	X1NiCrMoCu31-27-4	Welded tube
BS EN 10269	2.4952	43	NiCr20TiAl	Fastener
BS EN 10269	2.4668	43	NiCr19Fe19Nb5Mo3	Fastener
BS EN 10269	2.4669	43	NiCr15Fe7TiAl	Fastener
BS EN 10272	1.4529	45[a]	X1NiCrMoCuN25-20-7	Bar
BS EN 10272	1.4539	45[a]	X1NiCrMoCu25-20-5	Bar
BS EN 10272	1.4563	45[a]	X1NiCrMoCu31-27-4	Bar

[a] See **Ni.2.1.2.2.**

Ni.3 Section 3: Design

Ni.3.1 General

See main text.

Ni.3.2 Application

See main text.

Ni.3.3 Corrosion, erosion and protection

See main text.

Ni.3.4 Construction categories and design stresses

Ni.3.4.1 Construction categories

See main text with the following modified Table Ni.3.4-1.

Table Ni.3.4-1 **Construction categories**

Construction category	Non-destructive testing (NDT)	Permitted material grades	Maximum nominal thickness of component[a] (see 1.6) mm	Temperature limits	
				Upper	Lower
1	100% (see **5.6.4.1**)	All	None, except where NDT method limits apply	NA 11[b]: 350 °C All other nickel alloys: 450 °C	−196 °C
2	Limited random (spot) (see **5.6.4.2**)	All	40	NA 11[b]: 350 °C All other nickel alloys: 450 °C	−196 °C
3	Not permitted				

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension on the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.
[b] Or equivalent – EN 2.4066 or UNS N02200.

Ni.3.4.2 Design stresses

The design strengths for materials entering service without any subsequent heat treatment following the removal of test coupons at the material manufacturer's works shall not exceed the appropriate nominal design strength given by **Ni.2.3.1.1** or **Ni.2.3.1.2** for the material at the design temperature.

If any material is subjected to subsequent heat treatment (for example, in the manufacture of dished and flanged ends) representative material test coupons shall be heat treated with the components and subjected to the same mechanical tests as used to certify the material at the material manufacturer's works. The nominal design strengths shall then be calculated in accordance with **2.3.3.3**b).

If this design strength is lower than that used in the original calculations, the design of that component and any other related components shall be repeated using the actual material thicknesses and the newly derived nominal design stress. The actual thickness shall be equal to or greater than the thickness determined from Section 3 of this specification.

NOTE In using the data in this supplement for designs with nickel and nickel alloys, attention is drawn to the effect of heat treatment on the materials, and care should therefore be taken when determining the thickness of materials that will receive subsequent heat treatments during manufacture.

Ni.3.5 Vessels under internal pressure

See main text.

Ni.3.6 Vessels under external pressure

See main text and additionally.

Ni.3.6.1.1 Notation

See main text but substitute

E is the modulus of elasticity of material of part under consideration at design temperature from Table Ni.3.6-3;

s is the factor relating f to effective yield point of material; for the purposes of **Ni.3.6**, s shall be taken to be 1.1 for nickel and nickel alloys.

Table Ni.3.6-3 ***E* values for nickel alloys (modulus of elasticity)**

Temperature °C	NA 11 N/mm^2	NA 12 N/mm^2	NA 13 N/mm^2	NA 14 N/mm^2	NA 15 N/mm^2	NA 16 N/mm^2	NA 21 N/mm^2
−200	221 400	221 400	191 800	229 000	210 400	206 900	221 400
−150	218 400	218 400	189 300	226 000	207 400	204 000	218 400
−100	215 000	215 000	186 400	222 300	204 400	200 600	215 000
−50	211 500	211 500	183 400	218 400	201 200	197 200	211 500
0	208 200	208 200	180 500	215 100	197 900	194 300	208 200
20	206 900	206 900	179 300	213 800	196 600	193 100	206 900
50	204 900	204 900	177 600	211 500	194 600	191 100	204 900
100	201 600	201 600	174 800	207 900	191 300	187 900	201 600
150	198 500	198 500	172 300	205 400	188 900	185 400	198 500
200	196 700	196 700	170 500	203 600	187 000	183 600	196 700
250	194 200	194 200	168 000	200 600	184 000	181 100	194 200
300	192 300	192 300	166 500	198 500	182 400	179 200	192 300
350	189 500	189 500	164 500	195 700	179 900	176 900	189 500

Table Ni.3.6-3 ***E* values for nickel alloys (modulus of elasticity)** *(continued)*

Temperature °C	NA 11 N/mm²	NA 12 N/mm²	NA 13 N/mm²	NA 14 N/mm²	NA 15 N/mm²	NA 16 N/mm²	NA 21 N/mm²
400	186 100	186 100	161 300	192 300	176 800	173 700	186 100
450	182 400	182 400	157 800	188 600	173 400	170 200	182 400

NOTE Values have been taken from ASME II Part D, Table TM-4.

Ni.3.7 to Ni.3.13 *See main text.*

Ni.4 Section 4: Manufacturing and workmanship

Ni.4.1 General aspects of construction

See main text.

Ni.4.2 Cutting, forming and tolerances

See main text and additionally:

Ni.4.2.1 Cutting of material

Material shall be cut to size and shape, by machining or by a thermal cutting technique, (include plasma-arc cutting). Plates less than 20 mm thick may be cold sheared provided that the cut edges are dressed back mechanically by not less than 1.5 mm to provide a suitable surface to permit a satisfactory examination of the edges prior to welding.

Plates less than 10 mm thick, which are cold sheared, need not be dressed when the cut edges are to be subsequently welded. Surfaces that have been thermally cut shall be dressed back by machining or grinding for a minimum distance of 1.5 mm to remove metal dross and fused layer.

Table Ni.4.2-1 **Maximum temperature for heating nickel and nickel alloys**

Material group	Maximum temperature °C
41	930
42	980
43	1 040
44	1 120
45	980
46	1 140

Ni.4.2.2 Forming of shell sections and plates

Ni.4.2.2.1 Hot forming

Nickel and nickel alloys to be heated or hot worked shall be heated uniformly, without flame impingement, in accordance with the material manufacturer's recommendations. Suggested maximum temperatures are given in Table Ni.4.2-1. The hot forming temperature shall not exceed the final annealing temperature.

Nickel and nickel alloys shall be cleaned before heating as they can be embrittled by sulfur, phosphorus, lead, zinc and other low melting point metals and alloys which can be present in marking materials, die lubricants, pickling liquids, dirt accumulated in storage, furnace slag and cinder. Any foreign substance, even those which are not embrittling, can burn into the surface of the metal at high temperatures.

Most fuels may be used provided that detrimental impurities, such as sulfur, are kept at low levels. In view of the above, however, it is preferable that nickel and nickel alloys are cold worked whenever possible.

*NOTE Manufacturers constructing vessels in accordance with the provisions of this supplement, who carry out mechanical testing in accordance with **Ni.3.1** on materials which have been heat treated during manufacture, are requested to forward details of the mechanical properties resulting to: The Committee Manager of PVE/1, British Standards Institution, 389 Chiswick High Road, London W4 4AL.*

Ni.4.2.2.2 Cold forming

If the inside radius of curvature of a cold formed cylindrical pressure part is less than 10 times the thickness, an appropriate post forming heat treatment, as described in **Ni.4.5**, shall be given to remove the effects of cold work hardening.

Dished and flanged ends that have been cold formed shall be subsequently softened, as described in **Ni.4.5**, when the inside radius of curvature of the minimum radius is less than 15 times the thickness, when the thickness exceeds 5 mm.

Ni.4.3 Welded joints

Ni.4.3.1 General

See main text.

Ni.4.3.2 Welding consumables

Filler rods and wires for welded joints shall conform to BS EN ISO 18274 and shall be stored in accordance with the supplier's recommendations. The selection of the filler rods or wires shall be appropriate to the parent alloy.

Ni.4.3.3 Preparation of plate edges and openings

See main text.

Ni.4.3.4 Assembly for welding

See main text.

Ni.4.3.5 Attachments and the removal of temporary attachments

See main text.

Ni.4.3.6 Butt joints

See main text.

Ni.4.3.7 Welding: General requirements

See main text, additionally:

Ni.4.3.7.1 All surfaces to be welded shall be cleaned of oxide scale, grease, dirt, cutting fluids, paints and films arising from atmospheric contamination. Clean metal surfaces shall be exposed, if necessary by abrasive means, to a distance of 20 mm from each welding edge. Degreasing shall be undertaken immediately prior to welding.

Ni.4.4 Permanent joints other than welding

See main text, except that the requirements for brazing are not applicable to nickel and nickel alloys.

Ni.4.5 Heat treatment

Ni.4.5.1 Preheat requirements

Preheating is not normally necessary for nickel and nickel alloys.

Ni.4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

Main heading and main text are not applicable, replace with:

Annealing: Nickel and nickel alloys

After completion of the hot forming operation, or if required after cold forming, the material shall be given a final annealing in accordance with the material manufacturer's recommendations. Care should be taken not to exceed the maximum annealing temperature given on the material test certificate. Suggested temperature ranges are given in Table Ni.4.5-1.

Precautions shall be taken to avoid contamination and embrittlement (as described in **Ni.4.2**). After annealing the surfaces shall be descaled.

Ni.4.5.3 Post-weld heat treatment

Post weld heat treatment is not normally necessary for nickel and nickel alloys. If vessels are required for service in contact with caustic soda, fluorosilicates or some mercury salts, a stress relieving procedure can be desirable.

Ni.4.5.4 Methods of heat treatment

See main text.

Ni.4.5.5 Post-weld heat treatment procedure

If post-weld heat treatment is required then a heat treatment procedure shall be agreed between the purchaser, manufacturer and Inspecting Authority, with precautions taken as for hot forming.

Ni.4.5.6 Mechanical properties after heat treatment

Main text is not applicable.

Table Ni.4.5-1 **Annealing temperature for nickel and nickel alloys**

Material group	Annealing temperature °C
41	815 to 930
42	870 to 980
43	930 to 1 040
44	1 070 to 1 120
45	870 to 980
46	1 110 to 1 140

Ni.4.6 Surface finish

See main text.

Ni.5 Section 5: Inspection and testing

See main text, additionally:

Ni.5.2 Approval testing of fusion welded procedures

BS EN ISO 15614-1 for approval of welding procedures shall be used for nickel and nickel alloys.

Ni.5.3 Welder and operator approval

Ni.5.3.1 BS EN ISO 9606-4 for approval testing of welders and operators shall be used for nickel and nickel alloys.

For welder qualification, all materials covered by this supplement shall be considered as one group.

Ni.C Annex C: Assessment of vessels subject to fatigue

Ni.C.1.3 Symbols

See main text, but substitute:

E is the modulus of elasticity at the maximum operating temperature from Table Ni.3.6-3 (in N/mm^2), but not exceeding the value for ferritic steels at the maximum operating temperature from Table 3.6-3;

Ni.C.3.2.4 Environmental effects

For nickel alloys no guidance is given, and expert advice should be sought on the effects of corrosive environments on fatigue strength.

Titanium supplement

Requirements for titanium and titanium alloys in the design and construction of unfired pressure vessels

This supplement shall be read in conjunction with the main body of the specification to establish the requirements for commercially pure titanium and titanium alloy unfired pressure vessels. It lists the sections of the main text applicable to and not applicable to the design and construction of commercially pure titanium and titanium alloy pressure vessels. In addition the supplement contains clauses specific to such vessels which replace the corresponding clauses of the main text in this context. The clause, equation, figure and table numbering of the requirements of this supplement follow those of the comparable main text requirements but are identified in the form **Ti.x.y.z** to differentiate them from their main text equivalents, i.e. **x.y.z**, which might be technically different.

Ti.1 Section 1: General

See main text.

Ti.2 Section 2: Materials

Ti.2.1 Selection of materials

Ti.2.1.1 General

See main text, but substitute.

Ti.2.1.1.3 For the ease of reference in this supplement, materials are identified by a group number which has been derived from PD CEN ISO/TR 15608:2017, Table 5. This grouping is summarized in Table Ti.2.1-1.

Table Ti.2.1-1 **Material grouping**

Group	Sub-group	Type of titanium and titanium alloys
51	–	Pure titanium
	51.1	Titanium with $O_2 \leq 0.20\%$
	51.2	Titanium with $0.20\% < O_2 \leq 0.25\%$
	51.3	Titanium with $0.25\% < O_2 \leq 0.35\%$
	51.4	Titanium with $0.35\% < O_2 \leq 0.40\%$
52	–	Alpha alloys
53	–	Alpha-beta alloys
54	–	Near beta and beta alloys

Ti.2.1.2 Materials for pressure parts

Ti.2.1.2.1 General

See main text but substitute

For the construction of vessels in accordance with this supplement, all the materials used in the manufacture of pressure parts shall:

a) conform to ASTM specifications and grades listed in Table Ti.2.3-1; or

b) be agreed between purchaser, Inspecting Authority and manufacturer, be documented [see **1.5.2.2**b)] and shall conform to **Ti.2.1.2.2**; or

c) be the subject of a “European approval of material” in accordance with the EU directive 2014/68/EU (see Note 3); or

d) be detailed in supplements, annexes or enquiry cases to this specification.

NOTE 1 There are currently no British or European Standards specifically for titanium or titanium alloys for pressure purposes.

NOTE 2 Material grades are grouped into types of titanium and titanium alloy in Table 5 of PD CEN ISO/TR 15608:2017.

NOTE 3 Use of materials covered by European approval for material does not provide compliance with the Pressure Equipment (Safety) Regulations 2016, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

Ti.2.1.2.2 Materials covered by Ti.2.1.2.1b)

Ti.2.1.2.2.1 The material shall be covered by a written specification at least as comprehensive as the nearest equivalent ASTM specification listed in Table Ti.2.3-1. The specification shall, as a minimum, specify the manufacturing process, compositional limits for all constituents, heat treatment and appropriate mechanical properties for acceptance and other purposes.

Ti.2.1.2.2.2 Mechanical properties at room temperature shall be specified for acceptance tests in accordance with BS EN ISO 6892-1:2019 covering:

a) the tensile strength range;

b) the minimum 0.2% proof stress (R_p);

c) the specified minimum percentage elongation at fracture, referred to a gauge length of either 5.65 $\sqrt{S_o}$[1)] or 50 mm, shall be appropriate to the type of material. Minimum values of percentage elongation at fracture are given in the relevant ASTM material specification as listed in Table Ti.2.3-1.

Ti.2.1.2.3 Additional materials for category 3 components

Main text is not applicable.

Ti.2.1.3 Materials for non-pressure parts

See main text.

Ti.2.2 Materials for low temperature application

The alloys specified in **Ti.2.1.1** are not susceptible to brittle fracture and no special provisions are necessary for their use at temperatures down to at least −100 °C. Tensile and other strength values at room temperature may be used for operational service down to −30 °C.

Ti.2.3 Nominal design strength

Ti.2.3.1 General

Ti.2.3.1.1 Design strength values for commercially pure titanium and titanium alloy grades, appropriate to specified ASTM materials, shall be in accordance with Table Ti.2.3-1. These design strength values may be used without verification of $R_{e(T)}$ values.

NOTE These design strengths have been derived in accordance with ***Ti.2.3.2****, using tensile strength $R_{m(T)}$ equal to S_u from Table U and yield strength $R_{p0.2(T)}$ equal to S_y from Table Y-1 of ASME II Part D 2015 Edition.*

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1) S_o is the original cross-sectional area of the gauge length of the tensile test specimen.

Table Ti.2.3-1 **Design strength values (N/mm^2): commercially pure titanium and titanium alloys of material specifications ASTM B265, B338, B348, B363, B381, B861, and B862**

Material grades	Material group	Minimum tensile strength R_m N/mm^2	Minimum 0.2% proof stress $R_{p0.2}$ N/mm^2	Values of *f* for design temperatures not exceeding						
				20	**50**	**100**	**150**	**200**	**250**	**300**
				°C						
1	51	240	138	80	77	62	49	38	30	25
2	51	345	275	115	112	98	83	72	63	55
2H[a]	51	400	275	133	130	113	96	83	71	58
3	52	450	380	150	146	124	102	85	71	62
7	51	345	275	115	112	98	83	72	63	55
7H[a]	51	400	275	133	130	113	96	83	71	58
9	53	620	485	207	205	197	181	164	148	140
11	51	240	138	80	77	62	49	38	30	25
12	52	485	345	162	159	148	130	116	106	101
16	51	345	275	115	112	98	82	72	62	55
16H[a]	51	400	275	133	130	113	96	83	71	58
17	51	240	138	80	78	62	49	38	30	25
26	51	345	275	115	112	98	82	72	62	55
26H[a]	51	400	275	133	130	113	96	83	71	58
27	51	240	138	80	77	62	49	38	30	25
28	53	620	485	207	205	197	181	164	148	140

[a] Material is identical to the corresponding numeric grade (for example, Grade 2H = Grade 2) except for the higher guaranteed minimum tensile strength, and may always be certified as meeting the requirements of its corresponding numeric grade. In general over 99% of materials in these grades will meet the 400 MPa minimum tensile strength value.

Ti.2.3.1.2 For materials other than those listed in Table Ti.2.3-1, the time-independent design strength shall be determined in accordance with **Ti.2.3.2**.

Ti.2.3.1.3 The maximum design temperature as defined in **3.2.4** shall not exceed the upper temperature for which data are available to enable the design strength f_N to be determined in accordance with **Ti.2.3.1.1** or **Ti.2.3.1.2**. Where extrapolation of the data is required, this shall be on a basis agreed between the manufacturer, purchaser and Inspecting Authority.

Ti.2.3.2 Time-independent design strength

The time-independent design strength values given in Table Ti.2.3-1 were determined by $R_{p0.2(T)}/1.5$ or $R_{m(T)}$, whichever was the lower, where $R_{p0.2(T)}$ is the 0.2% proof stress at the design temperature and $R_{m(T)}$ is the tensile strength at the design temperature.

Ti.2.3.3 Time-dependent design strength

Commercially pure titanium and titanium alloys can exhibit time dependent deformation when loads are sustained for long periods near the proof stress value.

Time-dependent deformation is particularly relevant where the design conditions require $R_{m(T)}/3$ to exceed 70% of the 0.2% proof strength value. In such circumstances the designer shall consider the effect of those properties which influence time dependent deformation, and take specialist metallurgical advice as appropriate.

Ti.3 Section 3: Design

Ti.3.1 General

See main text.

Ti.3.2 Application

See main text.

Ti.3.3 Corrosion, erosion and protection

See main text and additionally:

Commercially pure titanium and titanium alloys have outstanding resistance to a wide range of mildly reducing, neutral and oxidizing corrosive media. As a general rule no allowance is required for pitting or general corrosion.

Caution is required in the design of joints and the selection of gasket materials where crevice corrosion could occur.

Commercially pure titanium and titanium alloys are highly resistant to erosion.

Titanium alloy selection for use in "sour service" conditions may be restricted by the requirements of NACE Standard MR0175 [1].

NOTE Further information on selection is available from the "Titanium Information Group Designers and Users Handbook, Titanium for Offshore and Marine Applications" [2]. See www.titaniuminfogroup.co.uk for information on Titanium Information Group members and documents.

Ti.3.4 Construction categories and design stresses

Ti.3.4.1 Construction categories

See main text with the following modified Table Ti.3.4-1.

Table Ti.3.4-1 **Construction categories**

Construction category	Non-destructive testing (NDT)	Permitted material grades	Maximum nominal thickness of component[a] (see 1.6) mm	Temperature limits	
				Upper	Lower
1	100% (see **5.6.4.1**)	All in Table Ti.2.3-1	None, except where NDT method limits apply	Grades 1, 11, 17, 27: 250 °C Grades 2, 3, 7, 9, 12, 16, 18, 26, 28: 300 °C	−30 °C
2	Limited random (spot) (see **5.6.4.2**)	All in Table Ti.2.3-1	None, except where NDT method limits apply	Grades 1, 11, 17, 27: 250 °C Grades 2, 3, 7, 9, 12, 16, 18, 26, 28: 300 °C	−30 °C
3	Not permitted				

[a] In the case of welded flat ends, tubesheets and flanges, the limitation on thickness applies to the governing dimension on the attachment weld and not to the thickness of the flat end, tubesheet or flange itself. If the flat end, tubesheet or flange is made from more than one piece of material butt welded together then the limitation on thickness applies to this butt weld.

Ti.3.4.2 Design stresses

The design strengths for materials entering service in the annealed condition, following the removal of test coupons at the material manufacturer's works, shall not exceed the appropriate nominal design strength given by **Ti.2.3.1.1** or **Ti.2.3.1.2** for the material at the design temperature.

NOTE In using the data in this supplement, attention is drawn to the effect of post-weld heat treatment on these materials, and in particular the formation of the brittle alpha case when an oxidizing atmosphere is used. Care with heat treatment atmospheres should therefore be taken when carrying out any stress relieving treatment. Titanium and titanium alloys should only be heat treated in argon or helium, or in a vacuum. A stress relieving post-weld heat treatment is not normally required for commercially pure titanium.

Ti.3.5 Vessels under internal pressure

See main text, additionally

Ti.3.5.4.7 Nozzle and nozzle pipe minimum thickness

Add

NOTE It is recommended that nozzles of up to 80 mm nominal size in titanium vessels should be forged or machined from wrought material, as indicated in Figure E.30, types (i), (ii) or (iii), in preference to pipe connections welded directly to the shell.

Ti.3.6 Vessels under external pressure

See main text and additionally

Ti.3.6.1.1 Notation

See main text but substitute

E is the modulus of elasticity of material of part under consideration at design temperature from Table Ti.3.6-3;

s is the factor relating *f* to effective yield point of material; for the purposes of **Ti.3.6**, *s* shall be taken to be 1.1.

Table Ti.3.6-3 ***E* values for titanium alloys (modulus of elasticity)**

Temperature °C	*E* N/mm^2
0	107 900
20	106 900
50	105 500
100	103 100
150	100 600
200	96 900
250	92 600
300	88 200

NOTE Values have been taken from ASME II Part D, Table TM-5.

Ti.3.7 Supports, attachments and internal structures

See main text.

Ti.3.8 Bolted flange connections

See main text and additionally:

In addition to titanium and titanium alloy bolts, nuts and washers, flanging arrangements might also include the use of stainless steel and mild steel epoxy coated components.

Gaskets made from or containing polymers which could release fluoride on thermal or acid decomposition shall not be used.

Ti.3.9 Flat heat exchanger tubesheets

See main text.

Ti.3.10 Design of welds

See main text.

Ti.3.11 to Ti.3.13

See main text.

Ti.4 Section 4: Manufacturing and workmanship

Ti.4.1 General aspects of construction

See main text.

Ti.4.2 Cutting, forming and tolerances

Ti.4.2.1 Cutting of material

Ti.4.2.1.1 Method

All material shall be cut to size and shape preferably by machining, plasma-arc or laser cutting, or water-jet cutting. Surfaces cut by other thermal processes shall be mechanically dressed or machined back for a distance of 6 mm.

Edges to be welded, that have been plasma-arc or laser cut shall be dressed back by grinding or machining for a distance of 1.5 mm, unless the manufacturer can demonstrate to the satisfaction of the Inspecting Authority that the material has not been adversely affected by the cutting process.

As-guillotined edges shall not be welded.

NOTE Fire safety procedures should be applied for the handling and control of titanium fines and turnings.

Ti.4.2.1.2 Examination of cut edges

See main text.

Ti.4.2.2 Forming of shell sections and plates

Ti.4.2.2.1 General

See main text.

Ti.4.2.2.2 Plates welded prior to hot or cold forming

Main text is not applicable.

Ti.4.2.2.3 Cold forming

Cleanliness of tooling and selection of correct lubrication are of particular importance. Suitable interface material shall be used between the forming equipment and the workpiece.

All cold forming methods may be used. The minimum bend radius on cold forming, without subsequent heat treatment, shall be that shown in Table Ti.4.2.2-1.

Table Ti.4.2.2-1 **Minimum bend radii**

Material grade	Minimum bend radius
Grades 1, 2, 7, 11, 16, 17, 26 and 27	4t
All other grades	5t

NOTE t = material thickness.

Where the minimum bend radii is less than that shown in Table Ti.4.2.2-1, the cold formed material shall be heat treated in accordance with Table Ti.4.5-1.

NOTE Compensation for springback (15° to 25°) is made by over forming.

Ti.4.2.2.4 Hot forming

Hot forming shall be carried out in accordance with written manufacturing procedures. These procedures shall include information such as material preparation for heating, heating times and temperatures, inspection and quality controls and any subsequent heat treatment and cleaning procedures.

Suitable interface material shall be used between the forming equipment and the workpiece.

The material shall be heated uniformly in a furnace to a maximum temperature of 600 °C, using a slightly oxidizing or inert atmosphere. Soaking times shall not exceed one hour per 50 mm of section thickness, and shall be kept to a minimum.

Ti.4.2.3 Assembly tolerances

See main text.

Ti.4.2.4 Tolerances for vessels subject to internal pressure

See main text.

Ti.4.2.5 Tolerances for vessels subject to external pressure

See main text.

Ti.4.3 Welded joints

Ti.4.3.1 General

See main text.

Ti.4.3.2 Welding consumables

Filler rods and wires for welded or brazed joints shall conform to AWS A5.16 or ASTM B 863, and shall be stored in accordance with the manufacturer's instructions.

In all cases where the filler metals do not match parent metal combinations the purchaser and Inspecting Authority shall be satisfied that the combination used is suitable for the service conditions.

Ti.4.3.3 Preparation of plate edges and openings

See main text.

Ti.4.3.4 Assembly for welding

See main text.

Ti.4.3.5 Attachments and the removal of temporary attachments

See main text and additionally:

Ti.4.3.5.3 Attachment of dissimilar metal

Dissimilar metal attachments are generally incompatible by fusion welding with most commercially available materials. Dissimilar metal welds shall only be made where agreement has been obtained from the purchaser and Inspecting Authority, and where an approved Welding Procedure Specification is present.

Ti.4.3.6 Butt welds

See main text and additionally:

Ti.4.3.6.2 Backing strips

Permanent backing strips shall not be used.

Ti.4.3.7 Welding: general requirements

Ti.4.3.7.1 All surfaces to be welded shall be thoroughly cleaned, on both sides of the joint, for a distance of 50 mm from each welding edge. Cleaning shall be by degreasing using a suitable solvent such as acetone on a lint-free cloth, before and after stainless steel or titanium wire brushing. Surfaces shall be dry before welding commences.

NOTE Methyl alcohol or sulfur-containing cleaning fluids should not be used.

Ti.4.3.7.3 Each run of weld metal shall be thoroughly cleaned before the next run is deposited. Brushes shall be of either stainless steel or titanium, and shall only be used on titanium and titanium alloys.

Ti.4.3.7.4 Before welding the second side of double sided joints the metal at the bottom of the first side shall be cut back to sound metal by machining or filing.

Ti.4.3.7.5 *See main text and additionally:*

Where arc strikes show a rejectable oxide discolouration the offending area shall be cut out and repaired.

Ti.4.3.7.7 To avoid contamination of heated surfaces by oxygen, hydrogen or nitrogen, welding shall be carried out either:

a) in a suitable chamber containing argon; or

b) by using trailing and purging argon gas shields.

This is in order to protect the weld bead and surrounding area until it has cooled below 250 °C.

Ti.4.3.7.8 Filler wire shall be thoroughly degreased prior to welding. When the filler wire is removed from the gas shield during or after welding, the first 20 mm of wire shall be discarded before welding re-commences.

Ti.4.3.7.9 After welding has been stopped for any reason, care shall be taken on re-starting to ensure satisfactory gas coverage of the welding zone together with satisfactory fusion and penetration with the parent material.

Ti.4.4. Permanent joints other than welding

See main text, except that the requirements for brazing are not applicable to titanium and titanium alloys.

Ti.4.5 Heat treatment

Ti.4.5.1 Preheat requirements

Heating prior to welding commercially pure titanium and its alloys is not normally considered necessary.

Ti.4.5.2 Normalizing: ferritic steels (material groups 1 to 6, 9 and 11)

Main heading and main text are not applicable, replace with:

Annealing

Following any hot forming operation, or when specified after cold forming, the material shall be given an annealing treatment in accordance with Table Ti.4.5-1. Precautions shall be taken to avoid contamination and embrittlement. After annealing the surfaces might require a descaling treatment.

Table Ti.4.5-1 **Heat treatment temperatures for commercially pure titanium and titanium alloys**

Grade	Stress relief		Anneal	
	Temperature °C	Time hr	Temperature °C	Time hr
1, 2, 3, 7, 11, 16, 17, 26, 27	400 – 550	½ – 2	650 – 700	½ – 4
9, 12, 18 ,28	500 – 600	½ – 4	700 – 750	½ – 4

NOTE 1 Attention is drawn to the effect of heat treatment on these materials, and in particular the formation of the brittle alpha case when an oxidizing atmosphere is used.

NOTE 2 Heat treatment should only be carried out in argon or helium, or in a vacuum.

NOTE 3 Heat treatment in a reducing atmosphere results in hydrogen absorption and causes embrittlement.

Ti.4.5.3 Post-weld heat treatment

For commercially pure titanium and titanium alloys, the requirement to carry out any post-weld heat treatment shall be agreed with the manufacturer, purchaser and Inspecting Authority.

Ti.4.5.4 Methods of heat treatment

Heat treatment shall be carried out in a furnace in a manner that ensures that the fabrication receives the full heat treatment process. Partial insertion into the furnace, or local post-weld heat treatment shall not be permitted.

Ti.4.5.5 Post-weld treatment procedure

Post-weld heat treatment temperature and time at temperature shall be as given in Table Ti.4.5-1, unless otherwise agreed with the manufacturer, purchaser and Inspecting Authority.

Ti.4.5.6 Mechanical properties after heat treatment

Main text is not applicable.

Ti.4.6 Surface finish

Ti.4.6.1 *See main text.*

Ti.4.6.2 *See main text and additionally:*

The method used to remove the alpha case surface layer shall be specified by the purchaser in the purchase specification.

Ti.5 Section 5: Inspection and testing

Ti.5.1 General

See main text.

Ti.5.2 Approval testing of fusion welding procedures

Ti.5.2.1 Approval testing of fusion welding procedures shall be conducted, recorded and reported in accordance with BS EN ISO 15614-5 or BS EN ISO 15614-8 as appropriate.

Ti.5.3 Welder and operator approval

Ti.5.3.1 Approval testing of welders and operators shall be conducted, recorded and reported in accordance with BS EN ISO 9606-5.

Ti.5.4 Production control test plates

Production control test plates are not required.

Ti.5.5 Destructive testing

Destructive testing is not required.

Ti.5.6 Non-destructive testing

See main text and additionally:

Ti.5.6.6.1 Radiographic techniques

Allowance shall be made for the lower absorption of X-rays by titanium and titanium alloys, when compared with ferritic materials.

An aluminium Image Quality Indicator shall be used to demonstrate radiographic sensitivity.

Ti.5.7 Acceptance criteria for weld defects revealed by visual examination and non-destructive testing

See main text and additionally:

Ti.5.7.2 Assessment of defects

Visual examination of titanium and titanium alloy welds is used to assess the adequacy or otherwise of the gas shielding methods applied, making use of the interference colours generated by thin layers of surface oxide in the weld zone. Further information is available from the "Titanium Information Group Designers and Users Handbook, Welding Titanium" [3].

Ti.C Annex C: Assessment of vessels subject to fatigue

Ti.C.1.3 Symbols

See main text, but substitute:

E is the modulus of elasticity at the maximum operating temperature from Table Ti.3.6-3 (in N/mm^2);

Ti.C.3.2.4 Environmental effects

The *S*-*N* curves in Figure C.3 and Figure C.4, adjusted for elastic modulus in accordance with **C.3.2.2**, are applicable in air and unprotected seawater up to 150 °C. For titanium alloys in other environments no guidance is given, and expert advice should be sought on the effect of corrosive environments on fatigue strength.

References

[1] NACE. International Standard MR0175 Parts 1 to 3, *Petroleum and natural gas industries — Materials for use in H_2S-containing environments in oil and gas production*.

[2] TITANIUM INFORMATION GROUP. *Titanium for offshore and marine applications: A designers and users handbook*, 1999.

[3] TITANIUM INFORMATION GROUP. *Welding titanium: A designers and users handbook*, 1999.

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List of references

See also references given in Annex A, Annex C, Annex G, Annex N, Annex R, Annex U, Annex W, Annex Z and the Titanium Supplement.

NOTE Where standards are identified as withdrawn, reference should be made to the Foreword.

Normative references

The following referenced documents are indispensable for the application of this specification. For dated references, only the edition cited applies. Subsequent amendments and revisions to dated references only apply when incorporated into this specification. For undated references, the latest edition of the referenced document (including any amendments) applies.

BS 21, *Specification for pipe threads for tubes and fittings where pressure-tight joints are made on the threads (metric dimensions).* (withdrawn)

BS 470, *Specification for inspection, access and entry openings for pressure vessels.*

BS 1123, *Fusible plugs for steam boilers and compressed air installations. Specification.*

BS 1400, *Specification for copper alloy ingots and copper alloy and high conductivity copper castings.* (withdrawn)

BS 1449-1.1, *Steel plate, sheet and strip — Part 1: Carbon and carbon-manganese plate, sheet and strip — Section 1.1: General specification.*

BS 1473:1972, *Specification for wrought aluminium and aluminium alloys for general engineering purposes — Rivet, bolt and screw stock.*

BS 1501, *Steels for pressure purposes.* (withdrawn)

BS 1502, *Specification for steels for fired and unfired pressure vessels: Sections and bars.* (withdrawn)

BS 1503, *Specification for steel forgings for pressure purposes.* (withdrawn)

BS 1504, *Specification for steel castings for pressure purposes.* (withdrawn)

BS 1580-1:2007, *Unified screw threads — Part 1: Screw threads with diameter ¼ in and larger — Requirements*.

BS 2870, *Specification for rolled copper and copper alloys: Sheet, strip and foil.* (withdrawn)

BS 2871 (all parts), *Specification for copper and copper alloys.* (withdrawn)

BS 2874, *Specification for copper and copper alloy rods and sections (other than forging stock).* (withdrawn)

BS 2875, *Specification for copper and copper alloys — Plate.* (withdrawn)

BS 2915:1990, *Specification for bursting discs and bursting disc devices.* (withdrawn)

BS 3059-1, *Steel boiler and superheater tubes — Part 1: Specification for low tensile carbon steel tubes with-out specified elevated temperature properties.* (withdrawn)

BS 3059-2, *Steel boiler and superheater tubes — Part 2: Specification for carbon, alloy and austenitic stainless steel tubes with specified elevated temperature properties.* (withdrawn)

BS 3072, *Specification for nickel and nickel alloys: Sheet and plate.* (withdrawn)

BS 3074, *Specification for nickel and nickel alloys: Seamless tube.* (withdrawn)

BS 3076, *Specification for nickel and nickel alloys: Bar.* (withdrawn)

BS 3601, *Specification for carbon steel pipes and tubes with specified room temperature properties for pressure purposes.* (withdrawn)

BS 3602-1, *Steel pipes and tubes for pressure purposes: Carbon and carbon manganese steel with specified elevated temperature properties — Part 1: Specification for seamless and electric resistance welded including induction welded tubes.* (withdrawn)

BS 3602-2, *Steel pipes and tubes for pressure purposes: Carbon and carbon manganese steel with specified elevated temperature properties — Part 2: Specification for longitudinally arc welded tubes.* (withdrawn)

BS 3603, *Specification for carbon and alloy steel pipes and tubes with specified low temperature properties for pressure purposes.* (withdrawn)

BS 3604-1, *Steel pipes and tubes for pressure purposes: Ferritic alloy steel with specified elevated temperature properties — Part 1: Specification for seamless and electric resistance welded tubes.* (withdrawn)

BS 3604-2, *Steel pipes and tubes for pressure purposes: Ferritic alloy steel with specified elevated temperature properties — Part 2: Specification for longitudinally arc welded tubes.*

BS 3605-1, *Austenitic stainless steel pipes and tubes for pressure purposes — Part 1: Specification for seamless tubes.* (withdrawn)

BS 3605-2, *Austenitic stainless steel pipes and tubes for pressure purposes — Part 2: Specification for longitudinally welded tubes.* (withdrawn)

BS 3606, *Specification for steel tubes for heat exchangers.* (withdrawn)

BS 3643-1:2007, *ISO metric screw threads — Part 1: Principles and basic data.*

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

BS 3799:1974, *Specification for steel pipe fittings, screwed and socket-welding for the petroleum industry.*

BS 3920-1:1973, *Derivation and verification of elevated temperature properties for steel products for pressure purposes.* (withdrawn)

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

BS 4870-1, *Specification for approval testing of welding procedures — Part 1: Fusion welding of steel.* (withdrawn)

BS 4870-2, *Specification for approval testing of welding procedures — Part 2: TIG or MIG welding of aluminium and its alloys.* (withdrawn)

BS 4871-3, *Specification for approval testing of welders working to approved welding procedures — Part 3: Arc welding of tube to tube-plate joints in metallic materials.* (withdrawn)

BS 4882:1990, *Specification for bolting for flanges and pressure containing purposes.*

BS 5046, *Method for the estimation of equivalent diameters in the heat treatment of steel.*

BS 7608:2014+A1:2015, *Guide to design and assessment of steel products.*

BS 7910:2019, *Guide to methods for assessing the acceptability of flaws in metallic structures.*

BS EN 287-1, *Qualification test of welders — Fusion welding — Part 1: Steels.* (Withdrawn)

BS EN 288-3, *Specification and approval of welding procedures for metallic materials — Part 3: Welding procedure tests for the arc welding of steels.* (withdrawn)

BS EN 288-4, *Specification and approval of welding procedures for metallic materials — Part 4: Welding procedure tests for the arc welding of aluminium and its alloys.* (withdrawn)

BS EN 485, *Aluminium and aluminium alloys — Sheet, strip and plate.*

BS EN 515, *Aluminium and aluminium alloys — Wrought products — Temper designations.*

BS EN 573, *Aluminium and aluminium alloys — Chemical composition and form of wrought products.*

BS EN 586, *Aluminium and aluminium alloys — Forgings.*

BS EN 603, *Aluminium and aluminium alloys — Wrought forging stock.*

BS EN 604, *Aluminium and aluminium alloys — Cast forging stock.*

BS EN 754, *Aluminium and aluminium alloys — Cold drawn rod/bar and tube.*

BS EN 755, *Aluminium and aluminium alloys — Extruded rod/bar, tube and profiles.*

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

BS EN 1011-4, *Welding — Recommendations for welding of metallic materials — Part 4: Arc welding of aluminium and aluminium alloys.*

BS EN 1092, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated.*

BS EN 1369, *Founding — Magnetic particle inspection.*

BS EN 1371-1, *Founding — Liquid penetrant inspection — Part 1: Sand, gravity die and low pressure die castings.*

BS EN 1653, *Copper and copper alloys — Plate, sheet and circles for boilers, pressure vessels and hot water storage units.*

BS EN 1759, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, class designated.*

BS EN 1759-3, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, class designated — Part 3: Copper alloy flanges.*

BS EN 1759-4, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, class designated — Part 4: Aluminium alloy flanges*.

BS EN 1982, *Copper and copper alloys. Ingots and castings.*

BS EN 1991-1-4, *Eurocode 1: Actions on structures — Part 1-4: General actions – Wind actions.*

BS EN 1998-1, *Eurocode 8: Design of structures for earthquake resistance — Part 1: General rules, seismic actions and rules for buildings.*

BS EN 1998-6, *Eurocode 8: Design of structures for earthquake resistance — Part 6: Towers, masts and chimneys.*

BS EN 10002-1:2001, *Tensile testing of metallic materials — Part 1: Method of test at ambient temperature.* (withdrawn)

BS EN 10025-2, *Hot rolled products of structural steels — Part 2: Technical delivery conditions for non-alloy structural steels.*

BS EN 10028-2, *Specification for flat products made of steels for pressure purposes — Part 2: Non-alloy and alloy steels with specified elevated temperature properties.*

BS EN 10028-3, *Specification for flat products made of steels for pressure purposes — Part 3: Weldable fine grain steels, normalized.*

BS EN 10028-4, *Specification for flat products made of steels for pressure purposes — Part 4: Nickel alloy steels with specified low temperature properties.*

BS EN 10028-6, *Specification for flat products made of steels for pressure purposes — Part 6: Weldable fine grain steels, quenched and tempered.*

BS EN 10028-7, *Specification for flat products made of steels for pressure purposes — Part 7: Stainless steels.*

BS EN 10088, *Stainless steels.*

BS EN 10160, *Ultrasonic testing of steel flat product of thickness equal or greater than 6 mm (reflection method).*

BS EN 10213, *Steel castings for pressure purposes*.

BS EN 10216-1, *Seamless steel tubes for pressure purposes — Technical delivery conditions — Part 1: Non-alloy steel tubes with specified room temperature properties.*

BS EN 10216-2, *Seamless steel tubes for pressure purposes — Technical delivery conditions — Part 2: Non-alloy and alloy steel tubes with specified elevated temperature properties.*

BS EN 10216-3, *Seamless steel tubes for pressure purposes — Technical delivery conditions — Part 3: Alloy fine grain steel tubes.*

BS EN 10216-4, *Seamless steel tubes for pressure purposes — Technical delivery conditions — Part 4: Non-alloy and alloy steel tubes with specified low temperature properties.*

BS EN 10216-5, *Seamless steel tubes for pressure purposes — Technical delivery conditions — Part 5: Stainless steel tubes.*

BS EN 10217-1, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 1: Non-alloy steel tubes with specified room temperature properties.*

BS EN 10217-2, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 2: Electric welded non-alloy and alloy steel tubes with specified elevated temperature properties.*

BS EN 10217-3, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 3: Alloy fine grain steel tubes.*

BS EN 10217-4, *Welded steel tubes for pressure purposes —Technical delivery conditions — Part 4: Electric welded non-alloy steel tubes with specified low temperature properties.*

BS EN 10217-5, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 5: Submerged arc welded non-alloy and alloy steel tubes with specified elevated temperature properties.*

BS EN 10217-6, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 6: Submerged arc welded non-alloy steel tubes with specified low temperature properties.*

BS EN 10217-7, *Welded steel tubes for pressure purposes — Technical delivery conditions — Part 7: Stainless steel tubes.*

BS EN 10222-2, *Steel forgings for pressure purposes — Part 2: Ferritic and martensitic steels with specified elevated temperature properties.*

BS EN 10222-3, *Steel forgings for pressure purposes — Part 3: Nickel steels with specified low-temperature properties.*

BS EN 10222-4, *Steel forgings for pressure purposes — Part 4: Weldable fine-grain steels with high proof strength.*

BS EN 10222-5, *Steel forgings for pressure purposes — Part 5: Martensitic, austenitic and austenitic-ferritic stainless steels.*

BS EN 10269:2013, *Steels and nickel alloys for fasteners with specified elevated and/or low temperature properties*.

BS EN 10272, *Stainless steel bars for pressure purposes.*

BS EN 10273, *Hot rolled weldable steel bars for pressure purposes with specified elevated temperature properties.*

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BS EN ISO 15614-2, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 2: Arc welding of aluminium and its alloys.*

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PD 5500:2024

Specification for unfired pressure vessels

Enquiry Cases: Introduction

Enquiry cases are published for information and guidance and thus the application of any particular enquiry cases to a vessel is by agreement of the relevant parties.

Some of the following Enquiry Cases refer to various editions of BS 5500 and were most recently published as part of BS 5500:1997, inclusive of amendments 1 to 6. As this Published Document is technically identical to that standard, these Enquiry Cases are equally relevant to the content of this specification.

Enquiry Cases published after 15 January 2000 relate directly to the edition of PD 5500 existing at the time of their publication.

In accordance with the provisions of Clause **1.2**, the publication of Enquiry Cases will be notified in *BSI Update Standards* and will be made available for inclusion in the ring-binder in this separate section. The table below is for recording cases as they are published and included in the binder, and for noting their subsequent routeing. The relevant pages of this table will be replaced when an existing Enquiry Case is withdrawn or a new Enquiry Case is issued.

In general, cases will be extant, as adjuncts to the specification and open to public comment, until the text of the specification is amended to incorporate the substance of particular cases. This will be done in the course of the normal amendment procedure and each case so dealt with will be recorded on the final page of the relevant amendment.

When an Enquiry Case is superseded by amended text to the specification, this fact will be recorded in the "routeing" column at the time of the amendment. Replacement pages for this summary will be included in the amendment. In a subsequent edition of the specification the references to such withdrawn or superseded cases will be deleted. Consequently, the numerical sequence of Enquiry Case numbers in the first column will not be continuous because of these omissions. Where a number within the published range has been allocated to an enquiry but its issue has been delayed, this will be indicated.

Enquiry Case No.	Date of publication	Subject of the Enquiry Case	Subsequent Case routeing (e.g. incorporated into PD 5500)
5500/33	October 2016	Verification of shape of vessels subject to external pressure	This is an update and revision to the August 2002 issue
5500/73	September 2020	Use of enhanced 2¼ Cr1Mo steel in welded pressure vessels complying with PD 5500	This is an update and revision to the September 2019 issue
5500/82	September 2009	Design stresses for ASTM A193 bolting material	This is an update and revision to the January 1989 issue
5500/87	September 2014	Use of duplex steels for the construction of pressure vessels to PD 5500	Withdrawn September 2016, incorporated as Duplex supplement
5500/91	September 2023	Use of ASTM and API materials for the construction of pressure vessels in accordance with PD 5500	This is an update and revision to the September 2020 issue
5500/107	May 1994	UT sensitivity setting for sizing and sentencing of indications against Table 5.7(2)	
5500/123	October 2008	External loads on flanges	
5500/127	September 2019	Wind loading	This is an update and revision to the September 2018 issue
5500/128	September 2019	Preliminary rules for a less conservative design of jacket blocking rings	This is an update and revision to the September 2011 issue
5500/133	September 2023	Flat unstayed ends of non-circular shape and associated flanges	This is an update and revision to the September 2021 issue
5500/134	September 2023	Superseded referenced standards	This is an update and revision to the September 2022 issue
5500/135	November 2005	Stress concentration factors for flush nozzles in cylinders	
5500/136	November 2005	Use of titanium for the construction of pressure vessel components in accordance with PD 5500	Withdrawn September 2010, incorporated as Titanium supplement
5500/137	September 2019	Flexibility of nozzle-vessel interface	This is an update and revision to the December 2016 issue
5500/138	October 2008	Guidance on alternative rules for nozzle reinforcement	This is an update and revision to the October 2007 issue
5500/139	September 2021	ISO 16528-2:2007 Conformance Tables	This is an update and revision to the September 2020 issue
5500/140	September 2022	Use of copper, copper alloy, nickel and nickel alloy materials covered by European approval of materials and ASME or ASTM standards	This is an update and revision to the September 2019 issue
5500/141	September 2020	Alternative limits for longitudinal compressive general membrane stress in a vessel	This is an update and revision to the January 2018 issue

Enquiry Case No.	Date of publication	Subject of the Enquiry Case	Subsequent Case routeing (e.g. incorporated into PD 5500)
5500/142	September 2018	Cylindrical shells of varying thickness under external pressure	
5500/143	September 2019	Permanent joints other than welding	Withdrawn September 2020, incorporated into **4.4**
5500/144	September 2023	Post Brexit UK pressure equipment legislation	This is an update and revision to the September 2022 issue

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Enquiry Case 5500/33

Verification of shape of vessels subject to external pressure

Enquiry

Can the committee provide further guidance on methods which can be used to verify that the shapes of vessels subject to external pressure are satisfactory.

Reply

The following methods can be used to establish the out of circularity of cylinders and cones and the deviations in the radius of curvature of spheres, as required by **3.6**. These methods are based on physical measurement techniques, however, surveying techniques such as optical, infrared or laser measurements can be used as an alternative for cylindrical and conical sections. With translation, the results of these surveyed measurements can be used for these sections in methods 2 and 3, as detailed below.

a) Cylindrical and conical sections

Any of the following three methods may be used, particular attention being given to sections lying in the plane of stiffeners.

The first method, using templates, is relatively simple compared with the others but has relatively tight limits. Vessels, the shape of which fall outside these limits, may be shown to be acceptable by one of the other methods.

Method 1. The shape should be checked against a template of chord length approximately equal to the vessel radius. The template should have a nominal radius approximately 1% greater or smaller than the nominal radius depending upon whether external or internal measurements are made. Stand off distance pieces should be provided as shown in Figure 1 and the departure from the true shape should not exceed 0.2% of the nominal radius. This criterion has been derived to ensure that the departure from the mean circle does not exceed $0.005R$ and can be over-conservative.

Method 2. Radii or differences from a constant radius should be measured at an even number of equally spaced intervals around the circumference sufficient to define the profile of the section being considered, but not fewer than 24. This can be done either by swinging an arm internally or by rotating the vessel about its longitudinal axis and making external measurements. The axis of rotation of the internal swinging arm or of the vessel, as the case may be, should approximate to the true centre of circularity of the section under consideration (see Figure 2 and Figure 3 and the suggested working form given in Table 1).

The radial measurements should be made to a precision of about $0.000\,1R$ and should be corrected for the mean and for the error in positioning the centre. This should be done by finding the coefficients b_0, a_1 and b_1 in the expression (see **3.6.8**):

$$R_\mathrm{r} = b_0 + a_1 \sin r\varphi + b_1 \cos r\varphi + \varepsilon_\mathrm{r}$$

where

$$b_0 = \frac{1}{N}\sum_{r=0}^{r=N-1} R_\mathrm{r}$$

$$a_1 = \frac{2}{N}\sum_{r=0}^{r=N-1} R_\mathrm{r} \sin r\varphi$$

$$b_1 = \frac{2}{N}\sum_{r=0}^{r=N-1} R_\mathrm{r} \cos r\varphi$$

where

N is the number of measuring points;

r is the identifying number of the points, equally spaced around the shell circumference, to which the radius R_r, is measured;

R_r is the radial measurement to the shell surface at the location r, from the assumed centre;

φ is the angular increment of the measuring points r;

ε_r is the deviation from mean circle at any measuring station r. Its value is unaffected by the radii being measured as a difference from a constant radius.

Then the departure, ε_r from the mean circle at any point r is given by:

$$\varepsilon_r = R_r - b_0 - a_1 \sin r\varphi - b_1 \cos r\varphi$$

The vessel is of adequate circularity provided that the maximum value of ε_r does not exceed $0.005R$.

Method 3. Chord (or bridge if preferred) gauge measurements at no fewer than 24 equally spaced positions on the circumference should be made to give values of the "rise" δ_1, termed chord (bridge) gauge readings, or differences from a constant rise (see Figure 4). The readings should be measured to a precision of 0.1 mm. The departures from the mean circle can be calculated from the readings using the following influence coefficients[1)].

$$\varepsilon_r = \sum_{i=0}^{N-1} \delta_i I_{(i-r)}$$

Values for I_r for two values of N are given in Table 2 where N is the number of equally spaced measuring points.

NOTE $I_r = I_{(N-r)}$, *e.g.* $I_{10} = I_{14}$ *with* $N = 24$.

The vessel is of adequate circularity if the maximum value of ε_r does not exceed $0.005R$.

b) Spheres and spherical sections

1) In the following guidelines on the verification of shape:

R is the design mean radius;

R_{max} is the maximum local radius measured at an inward local deviation $R_{max} > R$;

e is the analysis thickness of the section under consideration.

1) A check should be made of the spherical surface to demonstrate that the shape is within the limits specified in **3.6.4**. The degree of checking should be agreed with the designer and, typically, consist of checking three planes, one horizontal plane at the equator and two vertical planes at 90° to each other. The requirement for R_{max} not greater than $1.3R$ will be demonstrated using a template of design radius (or bridge gauge) of arc length $2.74\sqrt{Re}$, and checking that the inward deviation from design shape is no greater than $0.216e$.

[1)] Alternatively the departures from the mean circle can be calculated using the method described in KENDRICK Shape imperfections in cylinders and spheres – Their importance in design and methods of measurement *J. Strain Analysis for Eng Design*, **12**, No. 2, April 1977.

2) An alternative to measuring inward deviations is to use a template of radius 1.3R, i.e. R_{max}, as follows.

Internal template. If the template fits on the plate without rocking, the local radius will be equal to or less than R_{max} and therefore acceptable.

External template. If the template fits on the plate without rocking and there is clearance at the centre of the template, this will indicate that the local radius is greater than R_{max} and therefore unacceptable.

3) In the case of large or site erected vessels, the checks may be made on plates after pressing and before welding. Care should be taken, however, in the support of plates which would otherwise distort if supported incorrectly, whilst these checks are made. Additionally, after fabrication a check should be made throughout the length of all seams, using a template of arc length $2.74\sqrt{Re}$ and spanning the welded seam equally on either side. Where doubt arises concerning the local form away from, or along the welded seam, this should be subject to further verification.

4) Any weld overfill or misalignment of abutting plates, within the limits permitted by **4.2.3** may be disregarded. There should be no discernable flats, changes in curvature should be gradual and peaking at welded seams limited to:

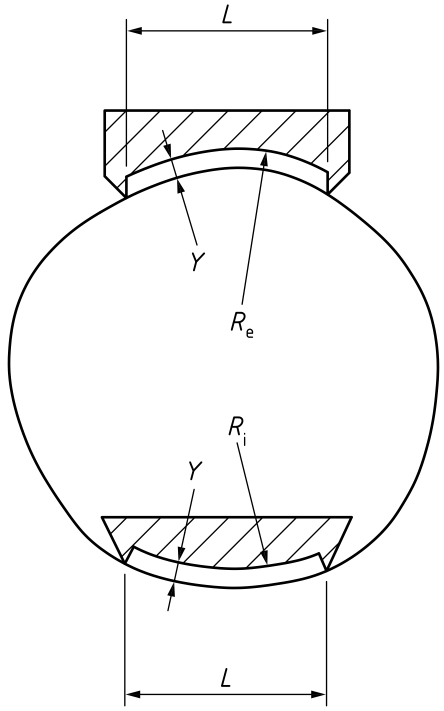
– the lesser of $e/4$ or 6 mm for external peaking;

– the lesser of $e/4$ or 3 mm for internal peaking;

referred to a gauge length extending five times the plate thickness from the edge of the weld on either side.

5) The designer may specify on the drawing other template dimensions, different R_{max} and different allowable deviations. Table 3 shows allowable local deviations and associated design pressure reduction factors where it is agreed that R_{max} can exceed 1.3R. The template gauge to be used to check deviations has a length of $2.74\sqrt{Re}$.

Figure 1 **Sketch showing use of templates**



$0.9R \le L \le 1.1R$

For an external template:

R_e is approximately equal to $1.01R$

$R_e - 1.002R \le Y \le R_e - 0.998R$

For an internal template:

R_i is approximately equal to $0.99R$

$0.998R - R_i \le Y \le 1.002R - R_i$

Figure 2 **Clarification of symbols**

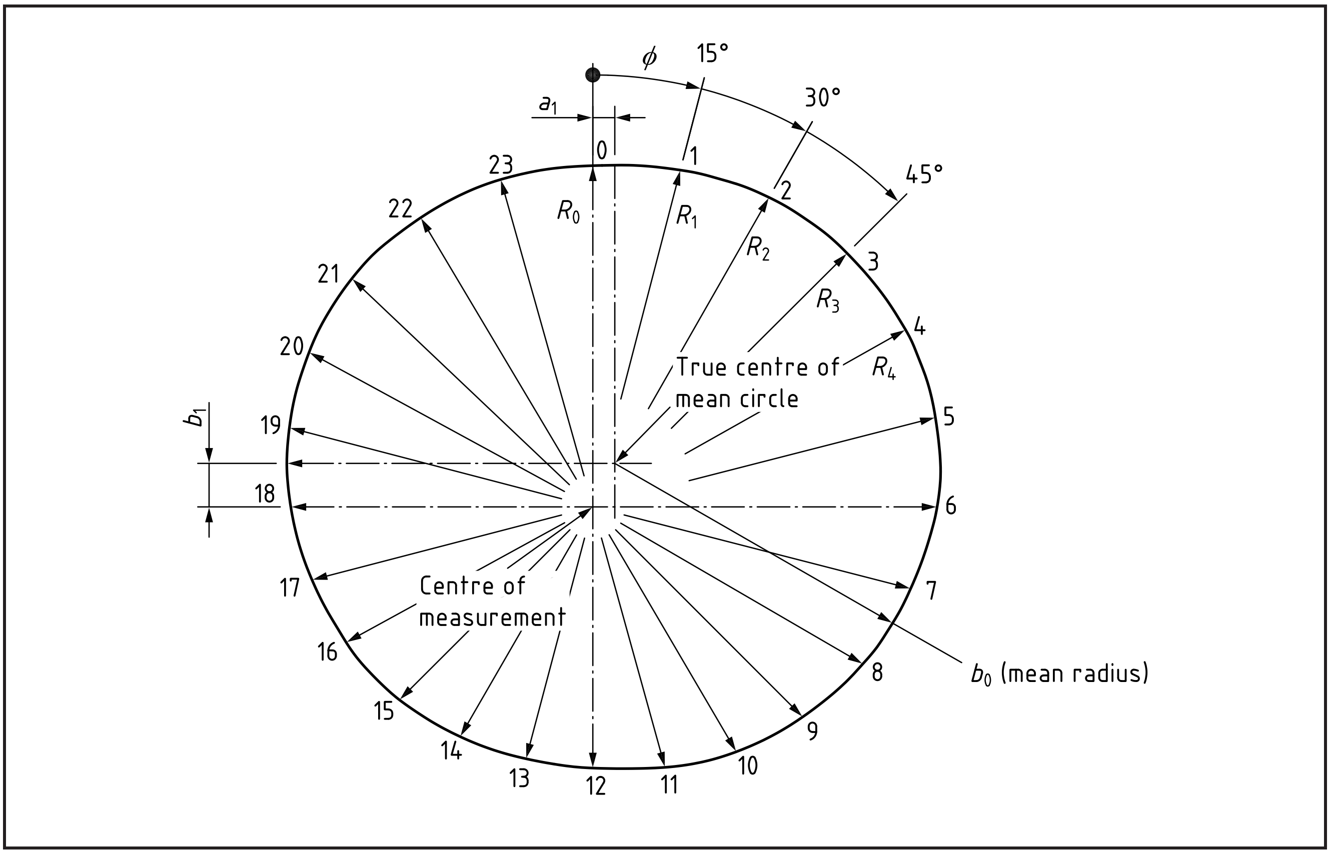
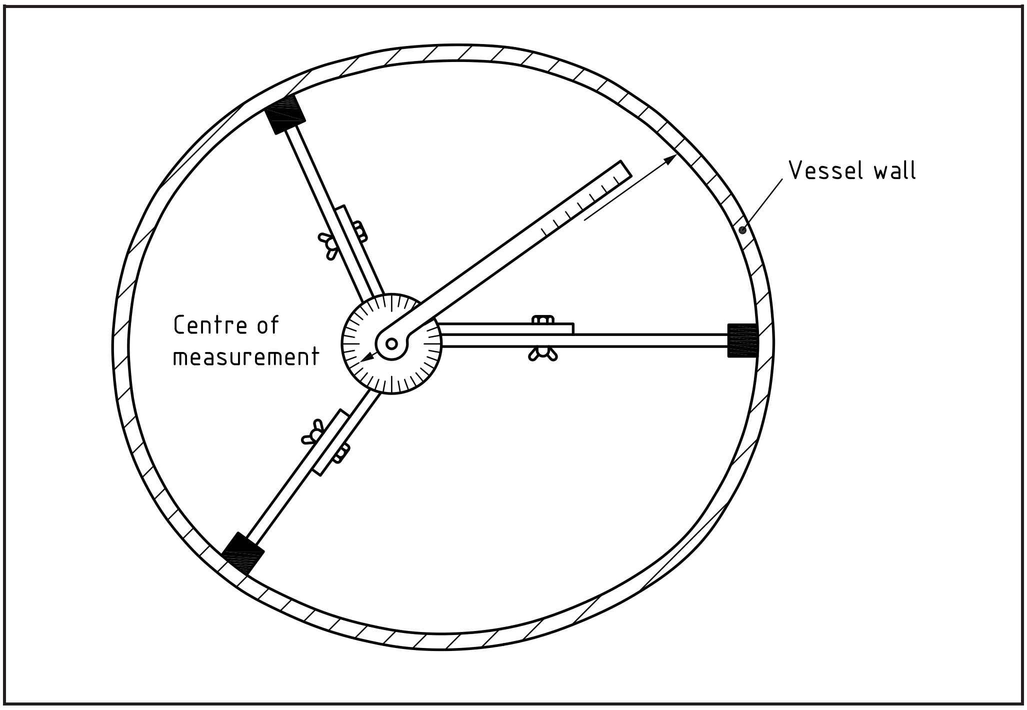


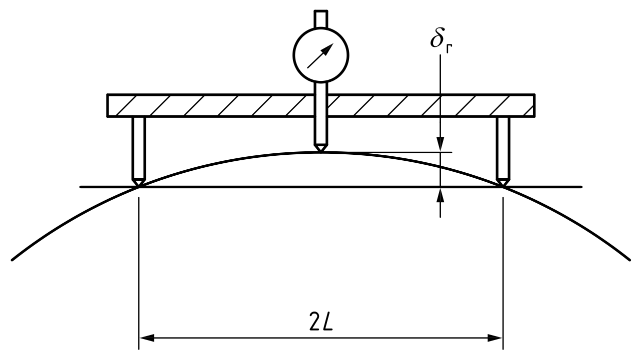
Figure 3 **Example of measuring gauge**



(1) Point no. r	(2) Reference angle $r\varphi$	(3) $\sin r\varphi$	(4) $\cos r\varphi$	(5) Measured radius R_r	(6) $R_r \sin r\varphi$	(7) $R_r \cos r\varphi$	(8) $a_1 \sin r\varphi$	(9) $b_1 \cos r\varphi$	(10) $a_1 \sin r\varphi + b_1 \cos r\varphi$	(11) $b_0 + a_1 \sin r\varphi + b_1 \cos r\varphi$	(12) Deviation ε_r $R_r - (b_0 + a_1 \sin r\varphi + b_1 \cos r\varphi)$
	Degree			mm	column (3) × column (5)	column (4) × column (5)	column (3) × a_1	column (4) × b_1	column (8) + column (9)	column (10) + b_0	column (5) – column (11)
0	0	0.0000	1.0000								
1	15	0.2588	0.9659								
2	30	0.5000	0.8660								
3	45	0.7071	0.7071								
4	60	0.8660	0.5000								
5	75	0.9659	0.2588								
6	90	1.0000	0.0000								
7	105	0.9659	–0.2588								
8	120	0.8660	–0.5000								
9	135	0.7071	–0.7071								
10	150	0.5000	–0.8660								
11	165	0.2588	–0.9659								
12	180	0.0000	–1.0000								
13	195	–0.2588	–0.9659								
14	210	–0.5000	–0.8660								
15	225	–0.7071	–0.7071								
16	240	–0.8660	–0.5000								
17	255	–0.9659	–0.2588								
18	270	–1.0000	0.0000								
19	285	–0.9659	0.2588								
20	300	–0.8660	0.5000								
21	315	–0.7071	0.7071								
22	330	–0.5000	0.8660								
23	345	–0.2588	0.9659								
				$\Sigma_1 =$	$\Sigma_2 =$	$\Sigma_3 =$					
$b_0 = \frac{1}{24}\Sigma_1 =$			$a_1 = \frac{1}{12}\Sigma_2 =$			$b_1 = \frac{1}{12}\Sigma_3 =$					
NOTE Shaded area indicates negative values.											

Table 1 **Determination of departure from mean circle (see method 2): Example of working form for inspectors**

Figure 4 **Example of use of chord or bridge gauge**



$$L = \frac{2\pi R}{N}$$

Table 2 **Influence coefficient I_r, for N = 24 and 48**

r	*N* = 24	*N* = 48	*r*	*N* = 24	*N* = 48	*r*	*N* = 48	*r*	*N* = 48
0	1.76100	3.6185	12	0.60124	–1.3835	24	1.2101	36	–1.3835
1	0.85587	2.6580	13	0.54051	–1.1944	25	1.1791	37	–1.5076
2	0.12834	1.7753	14	0.36793	–0.9544	26	1.0873	38	–1.5538
3	–0.38800	0.9834	15	0.11136	–0.6780	27	0.9385	39	–1.5107
4	–0.68359	0.2923	16	–0.18614	–0.3804	28	0.7385	40	–1.3689
5	–0.77160	–0.2910	17	–0.47097	–0.0763	29	0.4957	41	–1.1210
6	–0.68487	–0.7624	18	–0.68487	0.2201	30	0.2201	42	–0.7624
7	–0.47097	–1.1210	19	–0.77160	0.4957	31	–0.0763	43	–0.2910
8	–0.18614	–1.3689	20	–0.68359	0.7385	32	–0.3804	44	0.2923
9	0.11136	–1.5107	21	–0.38800	0.9385	33	–0.6780	45	0.9834
10	0.36793	–1.5538	22	0.12834	1.0873	34	–0.9544	46	1.7753
11	0.54051	–1.5076	23	0.85587	1.1791	35	–1.1944	47	2.6580

Table 3 **Maximum permissible local deviations from design shape**

$\dfrac{R_{max}}{R}$	**Maximum permissible inwards deviation from design shape** $\left[0.72\left(\dfrac{R_{max}}{R}-1\right)\right]e$	**Design pressure reduction factor =** $\left[\dfrac{R_{max}}{1.3R}\right]^2$
1.30	0.216e	1.00
1.40	0.288e	1.16
1.50	0.360e	1.33
1.60	0.432e	1.51
1.70	0.504e	1.71
1.80	0.576e	1.92
1.90	0.648e	2.14
2.00	0.720e	2.37
2.10	0.792e	2.61
2.20	0.864e	2.86
2.30	0.936e	3.13
2.40	1.008e	3.41
2.50	1.080e	3.70

Intermediate values may be obtained by linear interpolation.

Enquiry Case 5500/73

Use of enhanced 2¼Cr1Mo steel in welded pressure vessels complying with PD 5500

Enquiry

An extensive materials testing programme has recently been carried out under the aegis of API[1)]/MPC[2)] to investigate the feasibility of improving the utilization of 2¼Cr1Mo steel for the construction of welded pressure vessels with peak operating temperatures up to 450 °C. In the light of the materials data obtained in the course of this programme [ref 1[3)]] the use of grade 622 material complying with BS 1501 and BS 1503, but with the tensile and toughness requirements set out in Table 1 of this Enquiry Case would appear to be feasible for thicknesses up to 300 mm. The design strength of such materials would be as indicated in Table 2 of this Enquiry Case.

Table 1 **Tensile and toughness requirements**

Property		Value N/mm^2
Tensile strength R_m at room temperature		600 to 770
Yield strength, minimum		
R_e	at room temperature, °C	385
$R_{e(T)}$	at 350 °C	310
	at 450 °C	280
Elongation measured on a gauge length of 5.65 $\sqrt{S_o}$ as per BS 1501 and BS 1503 as relevant		
Toughness, min. Charpy V-notch, full size transverse, at −20 °C		50 J average of three values, no individual value less than 45 J

Table 2 **Design strength**

Value of design strength (f) for design temperature (T) not exceeding											
T	50	100	150	200	250	300	350	400	440	450	°C
f	255	240	224	221	216	212	207	202	190	187	N/mm^2

Will the committee please confirm that it is permissible to use such a material with these design strengths in the construction of welded pressure vessels complying with PD 5500?

Reply

It is the opinion of the committee that, in the light of the data presented in ref 1, grade 622 material complying with BS 1501 or BS 1503 but with the mechanical properties as set out in Table 1 of this Enquiry Case may be used for welded pressure vessels in thicknesses up to 300 mm with the design strength values set out in Table 2 of this Enquiry Case.

Additionally, the committee considers that the following additional conditions should be met.

a) The maximum design metal temperature not to exceed 450 °C.

b) The final post-weld heat treatment to be at a metal temperature not lower than 620 °C, and the minimum holding time to be in accordance with Table 4.5-1 of PD 5500 for sub-group 5.2 material.

1) American Petroleum Institute.
2) The Materials Properties Council, Inc, New York.
3) Ref 1: An unedited compilation of recent mechanical property test results on quenched and tempered 2¼Cr1Mo steel in the 85 to 110 ksi strength range: MPC report HPV-31.

c) Separate welding procedure approvals to be made for this material, welded to itself or to other materials. The approvals to conform to BS EN ISO 15614-1, the maximum tensile strength at room temperature being 770 N/mm^2.

d) Tensile tests at room temperature, 350 °C and 450 °C to be made to represent post-weld heat treatment at both the minimum and maximum times at maximum temperature, and impact tests to be made to represent the minimum time at minimum temperature. Hot tests to be in accordance with BS EN ISO 6892-2, the results of all the tests being in accordance with the values given in Table 1 of this Enquiry Case.

e) Each heat or lot of consumable welding electrodes and each heat or lot of filler wire and flux combination to be tested to meet the requirements of d) above.

f) Vessels designed for a temperature above 440 °C to be subject to periodic inspection such that any evidence of creep damage bearing on the remanent life can be detected.

g) Ultrasonic testing to be used to conform to **5.6.5** of PD 5500.

Enquiry Case 5500/82

Design stresses for ASTM A193 and EN 10269 bolting material

Enquiry

Can the design stress values recommended in Table 3.8-1 be applied to those ASTM 193 and EN 10269 material grades which correspond with the BS 4882 material grades specified in column 2 of this table?

Reply

Yes.

ASTM and EN bolting materials which correspond to BS material grades are given in Table 1 below.

Table 1 **BS, ASTM and EN bolting materials**

BS	ASTM	EN Name	EN No.
BS 4190 Gr 4.6	A-307-GrB	EN ISO 898-1 Gr 4.6	
BS 4882-B7	A-193-B7	EN 10269 42CrMo4	1.7225
BS 4882-L7	A-320-L7	EN 10269 42CrMo4	1.7225
BS 4882-B16	A-193-B16	EN 10269 40CrMoV4-6	1.7711
BS 4882-B6	A-193-B6		
BS 4882-B8M	A-193-B8M Class 1	EN 10269 X5CrNiMo17-12-2	1.4401
BS 4882-B8MX	A-193-B8M Class 2	EN 10269 X5CrNiMo17-12-2 [a]	1.4401
BS 4882-B8	A-193-B8 Class 1	EN 10269 X5CrNi 18-10	1.4301
BS 4882-B8X	A-193-B8 Class 2	EN 10269 X5CrNi 18-10 [a]	1.4301
BS 4882-B8T	A-193-B8T Class 1	EN 10269 X6CrNiTiB 18-10	1.4941
BS 4882-B8TX	A-193-B8T Class 2		
BS 4882-B8C	A-193-B8C Class 1		
BS 4882-B8CX	A-193-B8C Class 2		

[a] These materials are cold work hardened.

Enquiry Case 5500/91

Use of ASTM and API materials for the construction of pressure vessels in accordance with PD 5500

NOTE This revised Enquiry Case incorporates Enquiry Case 5500/112 which is withdrawn.

Enquiry

For availability reasons it might be necessary to consider the application of ASTM or API materials for the construction of pressure vessels in accordance with PD 5500. Can the committee confirm what design strength values may be used for the following common materials without values of $R_{e(T)}$ being verified?

The following ASTM and API materials should be considered:

ASTM A53	Standard specification for pipe, steel, black and hot-dipped, zinc-coated, welded and seamless;
ASTM A105	Specification for forgings, carbon steel, for piping components;
ASTM A106	Specification for seamless carbon steel pipe for high-temperature service;
ASTM A182	Specification for forged or rolled alloy-steel pipe flanges, forged fittings, and valves and parts for high-temperature service;
ASTM A213	Specification for seamless ferritic and austenitic alloy-steel boiler, superheater, and heat-exchanger tubes;
ASTM A234	Specification for piping fittings of wrought carbon steel and alloy steel for moderate and elevated temperatures;
ASTM A240	Specification for heat-resisting chromium and chromium-nickel stainless steel plate, sheet and strip for pressure vessels;
ASTM A285	Standard specification for pressure vessel plates, carbon steel, low- and intermediate-tensile strength;
ASTM A312	Specification for seamless and welded austenitic stainless steel pipe;
ASTM A333	Specification for seamless and welded steel pipe for low-temperature service;
ASTM A350	Specification for forgings, carbon and low-alloy steel, requiring notch toughness testing for piping components;
ASTM A403	Specification for wrought austenitic stainless steel piping fittings;
ASTM A420	Specification for piping fittings of wrought carbon steel and alloy steel for low-temperature service;
ASTM A515	Standard specification for pressure vessel plates, carbon steel, for intermediate- and higher-temperature service;
ASTM A516	Standard specification for pressure vessel plates, carbon steel, for moderate- and lower-temperature service;
API 5L	Specification for line pipe.

Reply

Yes. The committee confirms that it is acceptable to use the materials listed in the enquiry, subject to conformity to **2.1.2.1**c). Based on requirements current in PD 5500:2003, design strength values given in Table 1 and Table 2 of this Enquiry Case may be used without verification of $R_{e(T)}$ values for construction categories 1 and 2. Design stress values for construction category 3 should be determined in accordance with **3.4.2.2** except that for austenitic steels the 0.8 factor only need be applied to type 304L materials. Carbon steels and carbon manganese steels in material group 11 or with a specified minimum tensile strength greater than 432 N/mm^2 are not permitted for construction category 3.

The time-independent design strengths values given in Table 1 have been calculated in accordance with **2.3.3.2**b). The values of tensile strength and yield strength for the materials have been taken from the 2001 issue of ASME Section II, Part D[1)], Table Y-1, including 2002 updates. Time-dependent values in Table 1 have been assumed to be the same as for equivalent BS materials, as quoted in Table K.1-2 to Table K.1-12.

The design strength values given in Table 2 have been taken from Table 1b) of the 1991 issue of this Enquiry Case 5500/91.

The use of ASTM A53 and API 5L pipe materials is permitted with the following additional restrictions.

a) Grade B only is permitted.

b) Spirally welded pipe is not permitted.

c) Longitudinal seams to be 100% volumetrically inspected to the requirements given in Section 5.

The design strength values for API 5L can be assumed as being the same as given for ASTM A53 in Table 1.

[1)] ASME Boiler and pressure vessel code. Section II, *Materials*. Part D, *Properties*, 2001.

Table 1 **Design strength values (N/mm²) for carbon steels**

Specification	Grade	Material Group	R_m		R_e		Values of *f* for design temperatures (°C) not exceeding																		Design lifetime
			p.s.i	N/ mm²	p.s.i	N/ mm²	50	100	150	200	250	300	350	380	390	400	410	420	430	440	450	460	470	480	hours
A-285[b]	C	11.1	55 000	379	30 000	207	138	126	114	111	106	101	95	92	91	90	88	*87*	*79*	*69*	*60*	*52*	*44*	*36*	100 000
A-516	55	1.1															88	*83*	*74*	*65*	*56*	*48*	*40*	*32*	150 000
																	88	*80*	*71*	*62*	*53*	*45*	*37*	*28*	200 000
																	87	*78*	*68*	*59*	*51*	*42*	*35*	*26*	250 000
A-350[b]	LF1	11.1	60 000	414	30 000	207	138	126	114	111	106	101	95	92	91	90	90	90	*90*	*77*	*65*	*56*	*48*	*42*	100 000
																			81	*68*	*58*	*50*	*43*	*38*	150 000
																			75	*63*	*54*	*46*	*40*	*34*	200 000
																			70	*59*	*51*	*43*	*37*	*32*	250 000
A-515	60	1.1	60 000	414	32 000	221	147	134	122	118	113	108	101	98	97	96	94	*88*	*79*	*69*	*60*	*52*	*44*	*36*	100 000
A-516	60	1.1															*93*	*83*	*74*	*65*	*56*	*48*	*40*	*32*	150 000
																	90	*80*	*71*	*62*	*53*	*45*	*37*	*28*	200 000
																	87	*78*	*68*	*59*	*51*	*42*	*35*	*26*	250 000
A-53[b]	E/B	11.1	60 000	414	35 000	241	161	147	133	129	124	118	111	107	106	104	98	*88*	*79*	*69*	*60*	*52*	*44*	*36*	100 000
A-53[b]	S/B	11.1															*93*	*83*	*74*	*65*	*56*	*48*	*40*	*32*	150 000
A-106[b]	B	11.1															*90*	*80*	*71*	*62*	*53*	*45*	*37*	*28*	200 000
A-234[b]	WPB	11.1															*87*	*78*	*68*	*59*	*51*	*42*	*35*	*26*	250 000
A-333[b]	6	11.1																							
A-420[b]	WPL6	11.1																							

Table 1 **Design strength values (N/mm^2) for carbon steels *(continued)***

Specification	Grade	Material Group	R_m		R_e		Values of *f* for design temperatures (°C) not exceeding																		Design lifetime
			p.s.i	N/mm^2	p.s.i	N/mm^2	50	100	150	200	250	300	350	380	390	400	410	420	430	440	450	460	470	480	hours
A-515[b]	65	11.1	65 000	448	35 000	241	161	147	133	129	124	118	111	107	106	104	98	*88*	*79*	*69*	*60*	*52*	*44*	*36*	100 000
A-516	65	1.1															*93*	*83*	*74*	*65*	*56*	*48*	*40*	*32*	150 000
																	90	*80*	*71*	*62*	*53*	*45*	*37*	*28*	200 000
																	87	*78*	*68*	*59*	*51*	*42*	*35*	*26*	250 000
A-105[a],[b]		11.1	70 000	483	36 000	248	165	151	136	133	127	121	114	110	109	107	106	*105*	*90*	*77*	*65*	*56*	*48*	*42*	100 000
A-350[b]	LF2	11.1																*98*	*81*	*68*	*58*	*50*	*43*	*38*	150 000
																		88	*75*	*63*	*54*	*46*	*40*	*34*	200 000
																		83	*70*	*59*	*51*	*43*	*37*	*32*	250 000
A-515[b]	70	11.1	70 000	483	38 000	262	175	160	145	140	135	128	120	116	115	113	98	*88*	*79*	*69*	*60*	*52*	*44*	*36*	100 000
A-516[b]	70	11.1															*93*	*83*	*74*	*65*	*56*	*48*	*40*	*32*	150 000
																	90	*80*	*71*	*62*	*53*	*45*	*37*	*28*	200 000
																	87	*78*	*68*	*59*	*51*	*42*	*35*	*26*	250 000

NOTE Italicized values are based upon stress to rupture criteria.

[a] The use of A105 on duties below 0 °C without impact tests requires agreement between the purchaser, Inspecting Authority and manufacturer. Reference may be made to EEMUA publication 153/92, "EEMUA Supplement to ANSI B31.3, *Process piping*, American Society of Mechanical Engineers", for requirements to ensure that the material conforms to the requirements specified in Annex D.

[b] Where the carbon content of these materials, as specified in the design documentation, does not exceed 0.25% (see **2.1.2.2.2.1**) the material group may be taken as 1.1. Where the carbon content is not specified the material group should be taken as 11.1.

Table 2 **Design strength values (N/mm²) for austenitic steels to specifications A182, A213, A240, A312 and A403**

Type	Material Group	R_m		R_e		Values of *f* for design temperatures (°C) not exceeding																							Design lifetime
		p.s.i.	N/mm²	p.s.i.	N/mm²	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	hours
304L	8.1	70 000	485	25 000	170	137	122	107	97	91	86	83	78	77															
304	8.1	75 000	515	30 000	205	153	136	119	109	103	98	93	89	87															
304H[a]	8.1	75 000	515	30 000	205	153	136	119	109	103	98	93	89	87	85		84	83	*81*	*68*	*57*	*47*	*38*	*35*	*31*	*23*	*18*		100 000
	8.1																84	*83*	*76*	*62*	*52*	*42*	*33*	*30*	*26*	*20*	*15*		150 000
	8.1																84	*78*	*72*	*58*	*48*	*38*	*31*	*27*	*24*	*18*			200 000
	8.1																*82*	*75*	*68*	*56*	*45*	*36*	*28*	*25*	*22*	*17*			250 000
316L	8.1	70 000	485	25 000	170	135	121	107	99	93	89	85	82	79															
316	8.1	75 000	515	30 000	205	160	143	127	118	111	106	101	99	96															
316H[a]	8.1	75 000	515	30 000	205	160	143	127	118	111	106	101	99	96	93			90		89	*88*	*72*	*58*	*53*	*46*	*35*	*28*	*23*	100 000
	8.1																				*85*	*66*	*52*	*46*	*41*	*32*	*25*	*19*	150 000
	8.1																				*79*	*62*	*48*	*43*	*38*	*29*	*22*	*18*	200 000
	8.1																				*75*	*58*	*45*	*40*	*35*	*27*	*22*	*18*	250 000

Table 2 **Design strength values (N/mm²) for austenitic steels to specifications A182, A213, A240, A312 and A403 *(continued)***

Type	Material Group	R_m		R_e		Values of *f* for design temperatures (°C) not exceeding																							Design lifetime
		p.s.i.	N/mm²	p.s.i.	N/mm²	50	100	150	200	250	300	350	400	450	500	520	540	550	560	580	600	620	640	650	660	680	700	720	hours
321	8.1	75 000	515	30 000	205	157	145	133	127	121	117	113	110	107															
321H	8.1	75 000	515	30 000	205	157	145	133	127	121	117	113	110	107	104		102	102	*101*	*86*	*71*	*57*	*42*	*36*	*31*				100 000
	8.1																102	102	*93*	*78*	*64*	*49*	*36*	*32*	*27*				150 000
	8.1																102	*97*	*89*	*74*	*58*	*45*	*33*	*28*	*25*				200 000
	8.1																*100*	*93*	*85*	*71*	*55*	*42*	*31*	*26*	*22*				250 000
347	8.1	75 000	515	30 000	205	160	151	142	135	127	123	120	118	116															
347H	8.1	75 000	515	30 000	205	160	151	142	135	127	123	120	118	116	115		113	113	*113*	*99*	*82*	*66*	*53*	41	*35*	*31*	*23*		100 000
	8.1																	113	*112*	*93*	*75*	*61*	*48*	*37*	*32*	*28*	*22*		150 000
	8.1																	113	*106*	*88*	*72*	*58*	*45*	*34*	*29*	*25*	*19*		200 000
	8.1																	*112*	*102*	*85*	*68*	*55*	*42*	*32*	*28*	*24*	*19*		250 000

NOTE Italicized values are based upon stress to rupture criteria.

[a] Subject to continued fitness for service reviews (see **3.2.4**) being instituted at 2/3 of the design lifetime indicated, time-dependent values for 304H and 316H materials may be increased by up to 10% providing the resulting value does not exceed the lowest time-independent value given.

Enquiry Case 5500/107

UT sensitivity setting for sizing and sentencing of indications against Table 5.7(2)

Enquiry

When applying Table 5.7(2) of BS 5500:1994 in the sentencing of indications found by the application of BS 3923-1:1986, should this be carried out with the 14 db gain enhancement applied as specified in Table 2 of BS 3923-1:1986?

Reply

The gain enhancement specified in Table 2 of BS 3923-1:1986 is only required to be applied for the scanning and preliminary evaluation steps (see **17.2** of BS 3923-1:1986). When sizing and sentencing indications in accordance with Table 5.7(2) of BS 5500:1994, that gain enhancement should have been removed.

This will result in echo heights being judged against the DAC curve as originally established and corrected for differences between the material and surface condition of the test block and the part to be examined. In this context users are reminded of the interpretation provided in *BSI News*, January 1992, of **10.2** of BS 3923-1:1986, which was as follows.

"Interpretation of Clause 10.2 of BS 3923-1:1986

Due to the significant number of enquiries about the interpretation of the above clause, WEE/34, the Technical Committee responsible for the standard has agreed the following explanation of the procedure.

a) Plot the DAC curve using a reference block containing 3 mm side-drilled holes.

b) Increase the gain to allow for the transfer loss and material attenuation between the calibration block and the specimen.

   NOTE This may increase the first signal from the DAC reference block above the full screen height but this should be ignored as the gain will be correct for the material to be tested.

c) Increase the gain further by the appropriate decibel value shown in Table 2. This is the scanning sensitivity.

d) Scan for discontinuity indications.

e) At the scanning sensitivity defined in c) record any discontinuity indications that have an amplitude greater than the DAC curve drawn on the screen from item a). This will be cause for evaluation (see Clause **17**)."

Enquiry Case 5500/123

External loads on flanges

Enquiry

In the design of flanges, Note 4 of **3.8.1** mentions several loading conditions not covered in the rules of section 3.8 but which may need to be taken into account in the assessment of the flanges. The note suggests that special consideration should be given where the flange is subject to significant additional loading. Can the committee give some guidance where this additional loading takes the form of an external axial load or moment?

Reply

Yes the committee feels that the following points can give guidance on the assessment of external loads on flanges. The committee understands that the so-called "Kellogg Method" [1] of assessment of external loads on flanges has been used by industry for many years, particularly for pipework flanges. This method calculates an equivalent additional pressure for the axial load and moment, adds this to the flange design pressure and calculates the flange using this new total pressure. More recently BS EN 1591-1 has provided a method which calculates an additional load from the external loads which can then be incorporated into a flange assessment. It can be shown that these two methods have the same basis however neither method gives detailed guidance on how resulting calculated flange and bolt stresses and gasket loads should be assessed.

The following rules, which are closely related to the above methods, can be used to assess external axial loads and moments on flanges. Clauses a) and b) relate to flanges designed to section 3.8 and clause c) to standard pipework flanges as permitted by **3.8.1**. Unless stated otherwise the notation is as given in **3.8.2** and the loads are applied as given in Figure 1 below.

a) To check the flange and bolting for additional tensile loading, calculate an additional equivalent pressure p_a where:

$$p_a = \frac{4F}{\pi G^2} + \frac{16M}{\pi G^3}$$

F = the total external axial load (positive direction shown in Figure 1)

M = the total external moment

Using the combined pressure ($p + p_a$) recalculate A_{m1}, the bolt area for the operating condition, and check that it is less than the actual bolt area. With the combined pressure recheck the flange stresses for the operating condition.

Where the external moment is the more significant external load it is recognized that the additional equivalent pressure has a localized maximum at the point furthest from the neutral axis. Therefore where the moment contributes 80%, or more, of this additional equivalent pressure, the committee feels that the allowable flange and bolt stresses can be increased by 20% as long as the additional loading results from thermal effects or short term operating conditions such as wind or earthquake loading.

b) To check the gasket for additional loading, calculate an additional compressive gasket load H_a where:

$$H_a = \frac{4M}{G} - F$$

A check for local gasket crushing under operating conditions can be made using a combined total load of (W_{m1} + H_a). The calculated local gasket stress can be taken as y_1 where:

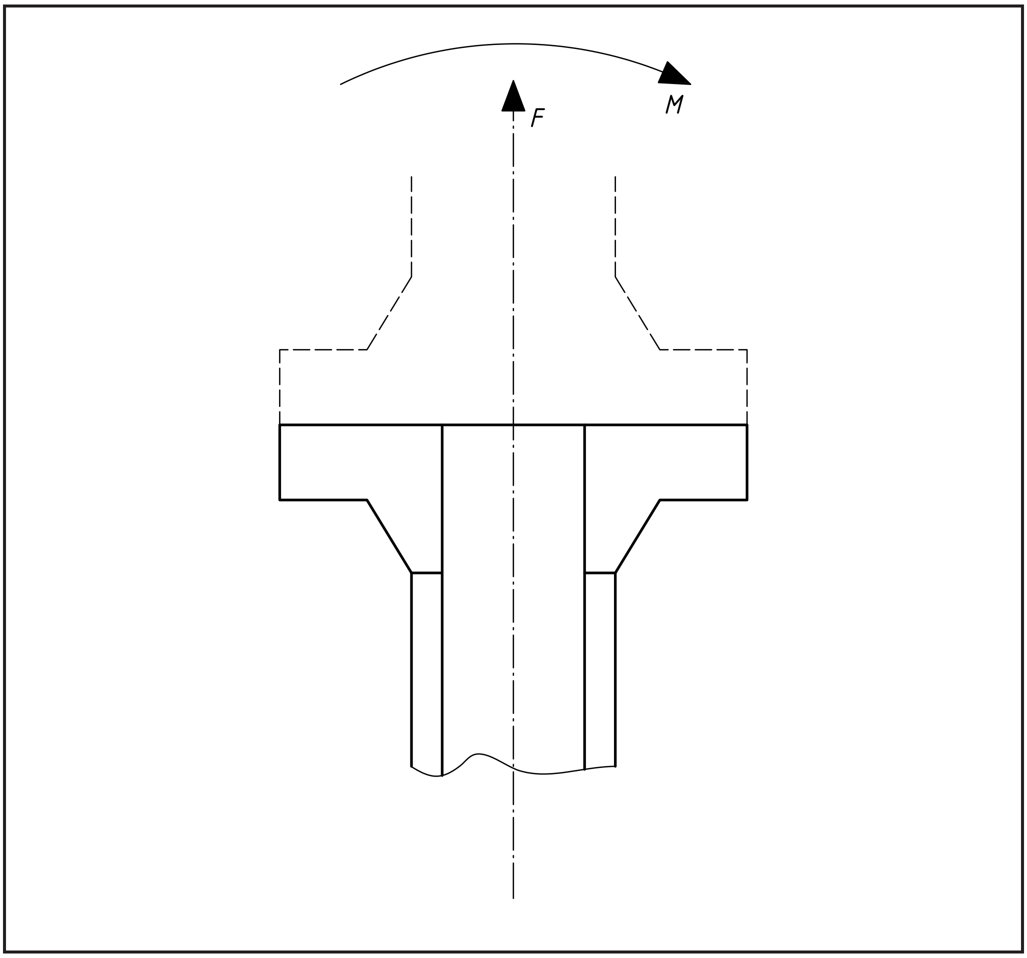
$$y_1 = \frac{W_{m1} + H_a}{\pi G b}$$

The value of y_1 should not exceed the yield strength of the flange material at the design temperature, and the gasket supplier may need to be involved in the assessment if the calculated local gasket stress is significantly higher than $2y$, where y is the gasket design seating stress.

c) A basis for assessment of standard pipework flanges could be to calculate the combined pressure (p + p_a) as in a) above. The relevant pipework flange standard should then be checked to ensure that the pressure/temperature rating for the flange is not exceeded by this combined pressure. Again, where the moment contributes 80%, or more, of the additional equivalent pressure p_a the committee suggests the relevant pressure/temperature rating could be increased by 20% to check this combined pressure. This assumes that the additional loading results from thermal effects or short term operating conditions such as wind or earthquake loading. If it is not possible to satisfy these conditions then the geometry of the standard flange can be checked using the design rules of section 3.8 of the main text and a) above.

[1] Design of piping systems, M W Kellogg Company, John Wiley & Sons Inc., 1956

Figure 1 **Check of flange and bolting for additional loading**



Enquiry Case 5500/127

Wind loading

Enquiry

In Annex B, paragraph B.5, reference is made to BS EN 1991-1-4 for information on wind loading. Can the committee provide further guidance on the application of BS EN 1991-1-4 to pressure vessels?

Reply

In order to calculate the wind loading on pressure vessels the following procedure may be used. The procedure given in sections 1 to 4 is based on the methods given in BS EN 1991-1-4:2005+A1, including the UK National Annex published in January 2011, and the guidance document PD 6688-1-4:2015.

BS EN 1991-1-4 (Eurocode 1) was first published in April 2005, and following the original publication of the National Annex (NA) to BS EN 1991-1-4 in September 2008 there was a three year coexistence period during which Member States were encouraged to withdraw conflicting national rules before the end of the coexistence period. BS 6399-2 was withdrawn in March 2010 and the procedure based on the methods in BS 6399-2 is no longer included in this Enquiry Case.

BS EN 1991-1-4 Eurocode 1: Actions on Structures — Part 1-4: General actions — Wind actions

References, where quoted, are to BS EN 1991-1-4 unless stated otherwise. References prefixed by the letters NA are to the UK National Annex to BS EN 1991-1-4.

NOTE Other National Annexes apply for vessels installed in other European countries.

1 Notation

A	altitude of site (m)
A_{ref}	reference area (m^2)
a	distance between centres of adjacent vertical cylinders (m)
a_L	normalized limiting amplitude
b	diameter (m)
C_c	aerodynamic constant
c_{alt}	altitude factor
c_d	dynamic factor
c_{dir}	directional factor
$c_e(z)$	exposure factor at height z
c_{eT}	exposure correction factor
c_f	force coefficient
$c_{f,x}$	along wind force (horizontal) coefficient
$c_{f,z}$	vertical force coefficient
$c_{f,0}$	force coefficient of cylinders without free-end flow
c_{lat}	lateral force coefficient
$c_{lat,0}$	basic value of lateral force coefficient
$c_o(z)$	orography factor
c_{prob}	probability factor
$c_r(z)$	roughness factor at height z
$c_{r,T}$	roughness correction factor
c_s	size factor
c_{season}	seasonal factor
c_1	constant
c_2	constant

$F_j(z)$	inertia force acting on a section j of vessel (N)
$F_L(z)$)	inertia force per unit length (N/m)
F_w	resultant wind force (N)
h	total height of vessel above ground (m)
h_{dis}	displacement height (m)
$I_v(z)$	turbulence intensity at height z
$I_v(z)_{flat}$	turbulence intensity
K	shape parameter
K_a	aerodynamic excitation parameter
K_w	effective correlation length factor
k	equivalent roughness (m)
k_I	turbulence factor
$k_{I,T}$	turbulence correction factor
k_p	peak factor
L_j	effective correlation length (m)
l	vessel length (m)
$M_j(z)$	bending moment acting on a section j of the vessel at a height z (N·m)
$M_j(z_b)$	bending moment acting on the vessel at the bottom of a section j (N·m)
$m(z)$	mass per unit length (kg/m)
m_e	equivalent mass per unit length as defined in EN 1991-1-4:2005+A1, **F.4**(1) (kg/m)
N	number of load cycles caused by vortex excited oscillation
n	exponent
n_1	fundamental natural frequency of vibration of vessel (Hz)
p	annual probability of exceedence
$q_p(z)$	peak velocity pressure at height z (N/m^2)
q_b	reference mean (basic) velocity pressure (N/m^2)
Re	Reynolds number
$Re(v_{crit,1})$	Reynolds number at critical wind velocity
Sc	Scruton number
St	Strouhal number
$v(z_e)$	peak wind velocity at height z_e (m/s)
v_b	basic wind velocity (m/s)
$v_{b,0}$	fundamental value of the basic wind velocity (m/s)
$v_{b,map}$	fundamental value of the basic wind velocity before altitude correction is applied (m/s)
$v_{crit,i}$	critical wind velocity (m/s)
$v_m(z)$	mean wind velocity at height z (m/s)
W	weight of vessel (kg)
W_j	weight of vessel section j (kg)
x	deflection at centre of gravity of vessel due to its self-weight applied in the vibration direction (m)
x_j	deflection at centre of gravity of vessel section j due to its self-weight applied in the vibration direction (m)
z	height above ground (m)
z_e	reference height (m)
z_g	distance from ground to considered component (m)
δ_s	logarithmic decrement of structural damping
$\Phi_1(z)$	fundamental flexural mode shape at elevation z
φ	solidity ratio
κ	factor for vertical cylinders in a row arrangement

λ	effective slenderness ratio
ν	kinematic viscosity of air (m^2/s)
ρ	density of air (kg/m^3)
σ_y	standard deviation of the displacement (m)
ψ_λ	end effect factor

2 Calculation method

The method given in BS EN 1991-1-4 employs equivalent static loads to represent the effect of fluctuating loads. The wind velocity and the velocity pressure are composed of a mean and a fluctuating component.

The mean wind velocity is evaluated from the basic wind velocity v_b and the height variation of the wind determined from the terrain roughness and orography (e.g. hills, cliffs, etc.).

The fluctuating component of the wind is represented by the turbulence intensity I_v.

For tall vertical vessels the wind velocity will vary with the effective height above ground z. It is permissible to perform the calculations for a single value of z equal to the height at the top of the vessel and to use the calculated wind velocity and velocity pressure to determine the wind force acting on the whole vessel. For tall vessels it is advantageous to divide the vessel into a number of parts and to evaluate the velocity pressure for each part using the appropriate values of the relevant factors corresponding to the effective height at the top of the part.

No specific guidance is given in BS EN 1991-1-4 regarding the division of the height into parts for vertical structures of circular cross-section. Based on **7.2.2** and Figure 7.4 it is recommended that the following three cases should be considered:

– A vertical vessel, whose height h is less than its diameter b should be considered to be one part.

– A vertical vessel, whose height h is greater than its diameter b, but less than $2b$, may be considered to be two parts, comprising: a lower part extending upwards from the ground by a height equal to b and an upper part consisting of the remainder.

– A vertical vessel, whose height h is greater than $2b$ may be considered to be in multiple parts, comprising: a lower part extending upwards from the ground by a height not less than b; an upper part extending downwards from the top by a height not less than b and a middle region, between the upper and lower parts, which may be divided into horizontal strips, each with a height h_{strip} as shown in **Figure 7.4**.

NOTE The velocity pressure should be assumed to be uniform over each horizontal strip considered.

2.1 Wind speed and wind pressure

The fundamental value of the basic wind velocity $v_{b,0}$ is defined as the 10-minute mean wind velocity with a 0.02 annual risk of being exceeded, irrespective of direction and season, at 10 m above ground level, in open country. For the UK this value is obtained from **NA 2.4**.

The 10-minute averaging period is the meteorological standard for much of continental Europe, but the UK uses 1 hour. A factor of 1.06 has been used to adjust the measured 1-hour average data to the 10-minute period.

The fundamental value of the basic wind velocity at the site location, before the altitude correction is applied, $v_{b,map}$ should be obtained from the map in Figure NA.1. This map is similar to that in Figure 6 of BS 6399-2:1997, except that the wind velocities have been adjusted to the 10-minute mean values. The source data record has also been increased to 30 years so the map in Figure NA.1 is statistically more accurate.

The altitude factor c_{alt} is evaluated in accordance with **NA.2.5**. Where there is no significant orography, A is the altitude of the site in metres above mean sea level. Where there is significant orography, as defined by the shaded zones in Figure NA.2, A should be taken as the altitude of the upwind base of the orographic feature. The height z is equal to the height z_s as defined in Figure 6.1, or the reference height z_e of the part of the vessel under consideration, as defined in Figure 7.4.

$$c_{alt} = 1 + 0.001 \cdot A \qquad \text{for } z \leq 10 \text{ m}$$

$$c_{alt} = 1 + 0.001 \cdot A \cdot \left(\frac{10}{z}\right)^{0.2} \qquad \text{for } z > 10 \text{ m}$$

The equation for $z \leq 10$ m may be used conservatively for any building height.

The fundamental value of the basic wind velocity $v_{b,0}$ is given by **expression (NA.1)**:

$$v_{b,0} = v_{b,map} \cdot c_{alt}$$

The basic wind velocity v_b is calculated in accordance with **4.2**. For vertical vessels the directional factor c_{dir} should normally be taken as 1.0. For horizontal vessels the directional factor is obtained from **Table NA.1**. Conservatively c_{dir} may be taken as 1.0 for all directions.

The seasonal factor c_{season} is normally taken as 1.0, but if the seasonal factor is applicable for a particular design case (e.g. during the construction phase) then the value may be obtained from **Table NA.2**.

The basic wind velocity having the probability p for an annual exceedence is determined by multiplying the basic wind velocity v_b from **expression (4.1)** by the probability factor c_{prob} given by **expression (4.2)**:

$$c_{prob} = \left(\frac{1 - K \cdot \ln(-\ln(1-p))}{1 - K \cdot \ln(-\ln(0.98))}\right)^n$$

The recommended values of the shape parameter $K = 0.2$ and the exponent $n = 0.5$ are to be used (see **NA 2.8**). In the UK the standard design value for the probability p is 0.02, corresponding to a mean recurrence interval of 50 years. This gives a value of $c_{prob} = 1.0$.

The basic wind velocity v_b is calculated in accordance with **expression (4.1)** multiplied by c_{prob}:

$$v_b = c_{dir} \cdot c_{season} \cdot c_{prob} \cdot v_{b,0}$$

The roughness factor $c_r(z)$ is determined in accordance with **NA.2.11**. The classification of roughness categories has been simplified in the NA to give three terrain categories – "Sea", "Country" and "Town". As indicated in **NA.2.12**, the recommended value of the angular sector given in **4.3.2(2)** should be used, but the recommended value of upstream distance should not be used. The procedures given in the National Annex implicitly take into account the upstream distances of terrain and terrain changes.

The procedures in **Annex A.2** for the transition between roughness categories should not be used (see **NA.3.1**).

The value of $c_r(z)$ is obtained from **Figure NA.3**. Lines of constant $c_r(z)$ are plotted on the chart against the distance upwind to the shoreline in km on the abscissa (horizontal axis) and the height ($z - h_{dis}$) in m on the ordinate (vertical axis). Enter the chart with the values of the distance upwind to the shoreline and ($z - h_{dis}$), and where the two lines cross determine the value of $c_r(z)$.

The height h_{dis} takes account of the effect of closely spaced buildings and obstacles. For vessels in Town terrain, closely spaced buildings and other obstructions cause the wind to behave as if the ground level was raised to a displacement height h_{dis}. For vessels in other terrain categories $h_{dis} = 0$. The procedure in **Annex A.5** may be used to determine h_{dis} (see **NA.2.15**).

For sites in Country terrain the value of the roughness factor $c_r(z)$ obtained from **Figure NA.3** should be used. For sites in Town terrain the value of the roughness factor $c_r(z)$ obtained from **Figure NA.3** should multiplied by the roughness correction factor $c_{r,T}$ for Town terrain obtained from **Figure NA.4**. Lines of constant $c_{r,T}$ are plotted on the chart against the distance inside town terrain in km on the abscissa and the height ($z - h_{dis}$) in m on the ordinate. Enter the chart with the values of the distance inside Town terrain and ($z - h_{dis}$), and where the two lines cross determine the value of $c_{r,T}$.

The orography factor $c_o(z)$ is determined in accordance with **NA.2.13**. Further guidance is given in PD 6688-1-4.

The mean wind velocity $v_m(z)$ at a height z above the terrain is calculated in accordance with **4.3.1, expression (4.3)**:

$$v_m(z) = c_r\,(z) \cdot c_o(z) \cdot v_b$$

Design charts for $v_m(z)$ are not provided – see **NA.2.10**.

For tall vessels the wind speed will vary with the effective height z due to the factor $c_r(z)$. It is permissible to calculate the effective wind speed corresponding to the effective height at the top of the vessel, and to use this wind speed to calculate the wind force acting on the whole vessel. For tall vessels it is advantageous to divide the vessel into sections and to evaluate the effective wind speed for each section using the appropriate value of factor $c_r(z)$ corresponding to the effective height at the top of the section.

The procedure in **Annex A.4** may be used to determine the effect of large and considerably taller neighbouring structures (see **NA.2.14**). The effective wind velocity may be increased by the presence of such structures, and this is taken into account by evaluating a height z_n above the ground at which the peak velocity pressure will be determined.

The turbulence intensity $I_v(z)$ at height z is determined in accordance with **NA.2.16**. The procedure is different from that given in **4.4**, and the turbulence factor k_I is not used. The turbulence intensity $I_v(z)$ is calculated as

$$I_v(z) = I_v\,(z)_{flat}\,k_{I,T}$$

The value of $I_v(z)_{flat}$ is obtained from **Figure NA.5** using the same procedure as that given for **Figure NA.3** above.

Where orography is significant, the value obtained from **Figure NA.5** should be divided by the orography factor $c_o(z)$. The turbulence intensity is not required if the orography is not significant, or if the height $z \leq 50$ m.

The value of the turbulence correction factor $k_{I,T}$ for sites in Town terrain is obtained from **Figure NA.6** using the same procedure as that given for **Figure NA.4 above**.

For sites in Country terrain (terrain categories I and II in BS EN 1991-1-4) the factor $k_{I,T}$ is taken as 1.0.

The peak velocity pressure $q_p(z)$ at height z is determined in accordance with **NA.2.17** as follows:

a) The value of the air density ρ is taken as 1.226 kg/m^3 (see **NA.2.18**), which is appropriate for strong winds blowing off the Atlantic Ocean.

b) From **expression (4.10)** the basic velocity pressure $q_b = \frac{1}{2} \cdot \rho \cdot v_b^2 = 0.613\ v_b^2$

c) The value of the exposure factor $c_e(z)$ is obtained from **Figure NA.7** using the same procedure as that given for **Figure NA.3** above. **Figure NA.7** is also used to determine whether zone A or B will apply when evaluating the size factor c_s (see below).

d) For sites in Town terrain the value of the exposure correction factor $c_{e,T}$ is obtained from **Figure NA.8** using the same procedure as that given for **Figure NA.4** above. **Figure NA.8** is also used to determine whether zone C will apply when evaluating the size factor c_s.

e) When orography is not significant the peak velocity pressure is obtained from **expression (NA.3a)** or **(NA.3b)**:

$$q_p(z) = c_e(z) \cdot q_b \qquad \text{for sites in Country terrain}$$

$$q_p(z) = c_e(z) \cdot c_{e,T} \cdot q_b \qquad \text{for sites in Town terrain}$$

f) When orography is significant the peak velocity pressure is obtained from **expression (NA.4a)** or **(NA.4b)**:

$$q_p(z) = c_e(z) \cdot q_b \cdot \left(\frac{c_o(z) + 0.6}{1.6}\right)^2 \qquad \text{for sites in Country terrain when } z \leq 50 \text{ m}$$

$$q_p(z) = c_e(z) \cdot c_{e,T} \cdot q_b \cdot \left(\frac{c_o(z) + 0.6}{1.6}\right)^2 \qquad \text{for sites in Town terrain when } z \leq 50 \text{ m}$$

$$q_p(z) = [1 + 3.0 I_v(z)]^2 \cdot 0.5 \cdot \rho \cdot v_m^2(z) \qquad \text{for } z > 50 \text{ m}$$

Annex A to the **National Annex** contains flow diagrams for the determination of $q_p(z)$.

2.2 Wind force

2.2.1 General approach to calculation of force

The wind force F_w acting on the pressure vessel, or a section of the vessel, is obtained from **5.3(2) expression (5.3)**.

Procedures are given in **6.3.1(1)** for calculating the structural factor c_sc_d, subject to the conditions given in **6.3.1(2)**. For vertical vessels of circular cross-section whose height is less than 60 m and less than 6.5 times the vessel diameter, the value of factor c_sc_d may be taken as 1, as permitted for chimneys in **6.2(1) d)**.

As stated in **NA.2.20** the structural factor c_sc_d may be separated into a size factor c_s and a dynamic factor c_d. The size factor c_s takes into account the reduction effect on the wind action due to the non-simultaneous occurrence of the peak wind pressures on the surface, and may be determined from **Table NA.3** or **6.3.1(1) expression (6.2)**. The dynamic factor c_d takes into account the increasing effect from vibrations due to turbulence in resonance with the structure and may be determined from **Figure NA.9** or 6.3.1(1) expression (6.3).

The recommended procedure given in **Annex B** should be used to determine the factors k_p, B and R required for evaluating the size factor c_s, the dynamic factor c_d and the structural factor $c_s c_d$ in **6.3.1(1)** (see **NA.2.21**). The value of the logarithmic decrement of structural damping δ_s for use in **Figure NA.9** or **Annex B expression B(6)** is obtained from **Table F.2**.

For loading conditions where there is no liquid in the vessel, or liquid only in the bottom of the vessel, the values of the logarithmic decrement of structural damping δ_s for unlined welded steel stacks may be used. For loading conditions where there is liquid in the vessel the values of δ_s for steel stacks with one liner may be used.

The calculation for first natural frequency of vibration n_1 used in **Annex B** is based on the simple one-mass vibrating system. The vessel is assumed to be a horizontal cantilever beam, fixed at the base, with the distributed weights of the vessel sections concentrated or lumped at the centres of gravity of the sections. The deflection x_j at the centre of gravity of each section j due to the weight W_j of that section can be calculated by various methods which can be found in pressure vessel design text books.

The natural frequency of vibration n_1 is given by:

$$n_1 = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{g \cdot \Sigma w_j \cdot x_j}{\Sigma W_j \cdot x_j^2}}$$

Where g is the gravitational acceleration (9.81 m/s^2).

NOTE The deflections are in metres in the above equation.

For vessels of uniform diameter and thickness the simplified equation given in **F.2** may be used:

$$n_1 = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{g}{x}}$$

An alternative approximation which may be used for vertical pressure vessels is:

$$n_1 = \frac{0.62}{\sqrt{x_t}}$$

Where x_t is the deflection in metres at the top of the vessel.

The alternative procedure for determining the structural factor $c_s c_d$ given in **Annex C** is new and untested, and is not permitted in the UK at this stage (see **NA.3.3**).

The charts in **Annex D** for determining the value of the structural factor $c_s c_d$ should not be used (see **NA.3.4**).

For slender vertical vessels whose height is greater than 6.5 times the vessel diameter, the effect of increased turbulence in the wake of nearby structures (wake buffeting) should be taken into account (see **6.3.3**).

The wind force obtained from **5.3(2)**, **expression (5.3)** is given by:

$$F_w = c_s c_d \cdot c_f \cdot q_p(z_e) \cdot A_{ref}$$

Where z_e is the maximum height above ground of the section being considered, and c_f is the force coefficient for the vessel — see below.

The reference area A_{ref} is the area of the vessel or section under consideration projected onto a plane perpendicular to the direction of the wind. For insulated vessels the reference area must include the thickness of the insulation.

2.2.2 Force coefficient

The force coefficient c_f takes account of the overall effect of the wind on the vessel or a section of the vessel, including friction, and depends on Reynolds number Re, the equivalent roughness k of the vessel surface and the vessel diameter b. For insulated vessels the diameter must include the thickness of the insulation.

For the purpose of evaluating the Reynolds number, the peak wind velocity $v(z_e)$ at height z_e is defined in Figure 7.28, NOTE 2 as

$$v(z_e) = \sqrt{\frac{2 \cdot q_p(z_e)}{\rho}}$$

The kinematic viscosity ν of the air is taken as 15×10^{-6} m^2/s.

The value of Reynolds number is calculated in accordance with **7.9.1(1) expression (7.15)**:

$$Re = \frac{b \cdot v(z_e)}{\nu}$$

NOTE The symbol used for kinematic viscosity is the Greek letter ν, and should not be confused with the letter v used to denote velocity.

Values of the equivalent surface roughness k for different types of surface are given in **Table 7.13**.

The force coefficient $c_{f,0}$ for a circular cylinder without free end flow is determined from **Figure 7.28**. This figure is valid for smooth cylinders, but most pressure vessels have numerous projections such as nozzles and brackets. In such cases it is recommended that a value of 1.2 is used for $c_{f,0}$.

The slenderness ratio λ is determined from **Table NA.10**. The values from **Table 7.16** should not be used (see **NA.2.44**).

The end effect factor ψ_λ is obtained from **Figure 7.36** using a solidity ratio φ = 1.

The force coefficient c_f for a finite circular cylinder is determined from **7.9.2(1) expression (7.19)**:

$$c_f = c_{f,0} \cdot \psi_\lambda$$

The air flow around horizontal cylinders is influenced by the distance z_g from the bottom of the cylinder to the ground (see **Figure 7.29**). If z_g/b < 1.5 then special advice is necessary — see [1].

For vertical vessels that are close to other vessels the force coefficient $c_{f,0}$ depends on the diameter b of the vessel and the distance a between the centres of the vessels. The factor κ is obtained from **Table 7.14** and the force coefficient c_f is determined from **7.9.3 expression (7.21)**:

$$c_f = c_{f,0} \cdot \psi_\lambda \cdot \kappa$$

For spherical vessels the reference height z_e is the height to the centre of the sphere, and the horizontal (along wind) force coefficient $c_{f,x}$ is obtained from the Figure 7.30. If $z_g < b/2$, where z_g is the distance from the bottom of the sphere to the ground, the force coefficient $c_{f,x}$ is to be multiplied by 1.6 (see Figure 7.31). If $z_g < b/2$, there will also be a vertical force on the spherical vessel, and the vertical force coefficient $c_{f,z}$ = 0.6

3 Vessel attachments

Vertical vessels are often fitted with platforms and ladders, and these can have a significant effect on the total wind loading on the vessel.

For guidance, the following values may be used for the effective areas of platforms and ladders on pressure vessels, see reference [2].

The effective area of a platform may be taken as 1.1 m^2 for a round platform or 1.7 m^2 for a square or polygonal platform. These figures are based on a platform extending on each side of the vessel not more than 1 200 mm from the outside of the vessel, or insulation where applicable. For larger platforms, the effective area should be increased accordingly.

The effective area of a ladder per metre length may be taken as 0.11 m^2/m for a plane ladder or 0.33 m^2/m for a caged ladder.

A force coefficient of 1.0 should be used in conjunction with the platform and ladder areas given above.

The effects of piping and equipment attached to the vessel should also be taken into account in the evaluation of the total wind loading on the vessel.

4 Vortex-shedding and wind induced vibration

Vortex-shedding occurs when vortices are shed alternately from opposite sides of a vessel. This gives rise to a fluctuating load perpendicular to the wind direction. Structural vibrations may occur if the frequency of vortex-shedding is the same as the natural frequency of vibration of the vessel.

The information given in Annex E to BS EN 1991-1-4 should not be used. Replacement information is given in Annex A to PD 6688-1-4.

Measures which may be used to reduce the effects of wind induced vibration are discussed in PD 6688-1-4:2015, **A.1.6**.

The effects of vortex shedding need not be investigated when:

$$\frac{h}{b} \leq 6.0$$

where b as the average outside diameter of the vessel, including insulation.

The critical wind velocity $v_{\mathrm{crit},1}$ corresponding to the fundamental natural frequency of the vessel is calculated in accordance with PD 6688-1-4:2015, **equation A.2**:

$$v_{\mathrm{crit},1} = \frac{b \cdot n_1}{St}$$

The Strouhal number St should be taken as 0.18 for cylindrical vessels – see PD 6688-1-4:2015, **Table A.1**.

The effects of vortex shedding need not be investigated when

$$v_{\mathrm{crit},1} > 1.25\, v_{\mathrm{m}}$$

where v_{m} is the mean wind velocity at the top of the vessel from **expression (4.3)**.

The Reynolds number at the critical wind velocity $Re(v_{\mathrm{crit},1})$ is calculated in accordance with PD 6688-1-4:2015, **equation A.5**:

$$Re(v_{\mathrm{crit},1}) = \frac{b \cdot v_{\mathrm{crit},1}}{\nu}$$

Determine the equivalent mass per unit length m_{e}. For vessels this may be approximated by the average value of mass per unit length m over the upper third of the vessel – see **F.4 (2)**.

The Scruton number Sc is calculated in accordance with PD 6688-1-4:2015, **equation A.4**:

$$Sc = \frac{2 \cdot \delta_s \cdot m_e}{\rho \cdot b^2}$$

Two different approaches are given in PD 6688-1-4:2015 **A.1.5.2** and **A.1.5.3** for calculating the vortex excited crosswind amplitudes. Mixing of the approaches is not permitted.

Approach 1 in **A.1.5.2** includes turbulence and roughness effects, but **A.1.1** states that its use should be limited to cases where there is supporting evidence of its applicability for structures of similar dynamic and aerodynamic properties. It should not be used in cases where **A.1.5.3** is applicable.

Approach 2 in **A.1.5.3** may be used where the diameter is constant over the top half of the vessel, but it cannot be used for grouped or in-line arrangements.

PD 6688-1-4:2015 **A.1.1** provides references for sources of more general methods.

4.1 Approach 1 in PD 6688-1-4:2015 A.1.5.2:

The calculation of the largest displacement $y_{F,max}$, is an iterative procedure involving the calculation of the correlation length L_j which depends on the value of the displacement $y_{F,max}$. An initial value of the ratio L_j / b should be estimated, between 6 and 12. The correlation length L_j is then evaluated, taking the diameter b as the average outside diameter of the vessel, including insulation, over the length L_j measured from the top of the vessel.

The mode shape factor K should be taken as 0.13 for vessels – see PD 6688-1-4:2015 **Table A.5**.

*NOTE This factor K is not the same as the shape parameter used in **expression (4.2)**.*

The correlation length factor K_w for vertical vessels is calculated using the equation given in PD 6688-1-4:2015, **Table A.5** for vertical cantilevered structures. Taking $\lambda = h / b$ and $\lambda_j = L_j / b$ the equation becomes:

$$K_w = 3 \cdot \frac{\lambda_j}{\lambda} \cdot \left[1 - \frac{\lambda_j}{\lambda} + \frac{1}{3} \cdot \left(\frac{\lambda_j}{\lambda}\right)^2\right]$$

The value of K_w should not exceed 0.6, as indicated in PD 6688-1-4:2015, **equation A.8**.

NOTE The symbol h has been used here to denote the total height of the vessel in place of l used in PD 6688-1-4:2015, Table A.5, to avoid confusion with the use of l in BS EN 1991-1-4:2005+A1 to denote the length of the vessel shell.

The basic value of lateral force coefficient $c_{lat,0}$ is obtained from PD 6688-1-4:2015, **Figure A.5**. The lateral force coefficient c_{lat} is calculated using the appropriate equation from PD 6688-1-4:2015, **Table A.3**, taking the wind velocity $v_{m,Lj}$ as the mean wind velocity in the centre of the correlation length.

For vessels in a row or grouped arrangement the influence of nearby vessels should be taken into account, in accordance with PD 6688-1-4:2015, **A.1.5.2.7**.

The largest displacement $y_{F,max}$ is calculated in accordance with PD 6688-1-4:2015, **equation A.7**:

$$y_{F,max} = \frac{1}{St^2} \cdot \frac{1}{Sc} \cdot K \cdot K_w \cdot c_{lat} \cdot b$$

Taking the vibration amplitude $y_F(s_j) = y_{F,max}$ calculate the ratio $y_F(s_j)/b$ and obtain a value of the ratio L_j / b from PD 6688-1-4:2015, Table A.4. Make a revised estimate of the value of the ratio L_j / b (between 6 and 12) and repeat the procedure until the iteration converges (i.e. the calculated value of L_j / b is equal to the estimated value of L_j / b). If the average outside diameter b of the vessel over the length L_j has changed then the procedure should be repeated from the calculation of the critical wind velocity $v_{crit,1}$. If the diameter b is unchanged then the procedure should be repeated from the calculation of the correlation length factor K_w.

4.2 Approach 2 in PD 6688-1-4:2015, A.1.5.3:

In the calculations for approach 2 the diameter b should be taken as the outside diameter, including insulation, of the top half of the vessel.

Values of the aerodynamic constant C_c, the aerodynamic excitation parameter K_a and the normalized limiting amplitude a_L are obtained from PD 6688-1-4:2015 Table A.6, dependent on the Reynolds number $Re(v_{crit,1})$.

The constants c_1 and c_2 are calculated in accordance with PD 6688-1-4:2015, **equation A.16**:

$$c_1 = \frac{a_L^2}{2} \cdot \left(1 - \frac{Sc}{4 \cdot \pi \cdot K_a}\right) \qquad c_2 = \frac{\rho b^2}{m_e} \cdot \frac{a_L^2}{K_a} \cdot \frac{C_c^2}{St^4} \cdot \frac{b}{h}$$

The standard deviation of the displacement σ_y is calculated using PD 6688-1-4:2015, **equation A.15**:

$$\sigma_y = b \cdot \sqrt{c_1 + \sqrt{c_1^2 + c_2}}$$

The peak factor k_p is calculated in accordance with PD 6688-1-4:2015, **equation A.17**:

$$k_p = \sqrt{2} \cdot \left\{1 + 1.2 \cdot \tan^{-1}\left[0.75 \cdot \left(\frac{Sc}{4 \cdot \pi \cdot K_a}\right)^4\right]\right\}$$

The characteristic maximum displacement at the point with the largest movement σ_{max} is calculated in accordance with PD 6688-1-4:2015, **equation A.13**:

$$y_{max} = \sigma_y \cdot k_p$$

4.3 Forces and moments:

The fundamental flexural mode shape $\Phi_1(z)$ at height z above ground is calculated in accordance with **expression (F.13)**, taking ξ = 2.0:

$$\Phi_1(z) = \left(\frac{z}{h}\right)^2$$

where h is the height above ground to the top of the vessel.

The inertia force per unit length F_L acting perpendicular to the wind direction, at height z above the ground is calculated in accordance with PD 6688-1-4:2015, **equation A.6**. Taking $\Phi_1(z)$ from the equation given above, equation A.6 becomes:

$$F_L(z) = m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot \left(\frac{z}{h}\right)^2 \cdot y_{F,max}$$

where $m(z)$ is the mass per unit length at height z.

If the mass per unit length $m(z)$ is not constant over the height of the vessel then the length should be divided into sections, each having approximately constant mass per unit length, and the horizontal force and overturning moment should be calculated for each section.

The horizontal force acting on a section j of the vessel is given by:

$$F_j(z) = \frac{m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot y_{F,max} \cdot (z_{jt}^3 - z_{jb}^3)}{3 \cdot h^2}$$

where z_{jt} is the height to the top of the section and z_{jb} is the height to the bottom of the section. If the mass per unit length is constant over the full length then $z_{jt} = h$ and $z_{jb} = 0$.

The bending moment acting on the vessel at the bottom of the section is given by:

$$M_j(z_b) = \frac{m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot y_{F,max} \cdot (3 \cdot z_{jt}^4 + z_{jb}^4 - 4 \cdot z_{jt}^3 \cdot z_{jb})}{12 \cdot h^2}$$

The bending moment acting on the vessel at a height z less than z_{jb} due to the force acting on the section specified above is given by:

$$M_j(z) = \frac{m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot y_{F,max} \cdot [3 \cdot (z_{jt}^4 + z_{jb}^4) - 4 \cdot z \cdot (z_{jt}^3 - z_{jb}^3)]}{12 \cdot h^2}$$

If $v_m < v_{crit,1} < 1.25 \cdot v_m$ then the values of the inertia force per unit length F_L, the force $F_j(z)$ and the moments $M_j(z_b)$ and $M_j(z)$ may be reduced by dividing by $(v_{crti,1} / v_m)^2$.

The stresses in the shell and skirt are calculated in accordance with Annex B of this specification.

4.4 Fatigue assessment

A fatigue assessment should be carried out in accordance with Annex C at the following locations:

a) at the bottom of each shell cross-section where the diameter or thickness changes;

b) at the bottom of the shell;

c) at the skirt to vessel connection;

d) at the base of the skirt;

e) at the skirt to base weld;

f) for the anchor bolts.

The stress range for use in the fatigue assessment should be taken as twice the stress due to the bending moment M.

The number of load cycles N caused by vortex excited oscillation is calculated in accordance with PD 6688-1-4:2015, **equation A.10**:

$$N = \frac{2 \cdot T \cdot n_1 \cdot \varepsilon_0 \cdot \left(\frac{v_{crit,1}}{v_0}\right)^2}{\exp\left[\left(\frac{v_{crit,1}}{v_0}\right)^2\right]}$$

where:

T is the design life in seconds (equal to 3.2×10^7 multiplied by the design life in years);

ε_0 is the bandwidth factor, which may be taken as 0.3;

v_0 is $\sqrt{2}$ times the modal value of the Weilbull probability distribution assumed for the wind velocity. The value of v_0 may be taken as 20% of the mean wind velocity v_m at the top of the vessel.

5 Worked example

The following worked example illustrates the use of BS EN 1991-1-4, including the National Annex and PD 6688-1-4, for the calculation of wind loading on a vertical vessel with a skirt support.

5.1 Design data

Vessel length (tan to tan)		20 000.0 mm
Vessel outside diameter	D_o	1 000.0 mm
Vessel analysis thickness	e_{av}	13.0 mm
Vessel elastic modulus at design temperature	E_v	202 000.0 N/mm^2
Design pressure	p	1.0 N/mm^2
Thickness of insulation on vessel		50.0 mm
Skirt length to vessel bottom tan line		2 300.0 mm
Skirt analysis thickness	e_{as}	10.0 mm
Skirt elastic modulus at ambient temperature	E_s	209 000.0 N/mm^2
Thickness of insulation/fireproofing on skirt		0.0 mm
Weight of skirt	W_1	618 kg
Weight of vessel	W_2	9 671 kg
Terrain category (see **NA.2.11**)		Country
Distance upwind to shoreline		10.0 km
Fundamental value of basic wind velocity (from NA to BS EN 1991-1-4:2005+A1, **Figure NA.1**)	$v_{b,map}$	26.0 m/s
Site altitude above mean sea level	A	200.0 m
Design life		20 years

5.2 Peak velocity pressure

Vessel has 2:1 ellipsoidal ends, therefore height of top dished end, including insulation, is:

$(0.25 \times 1\ 000.0) + 50.0 = 300.0$ mm

Total height of vessel above local ground level:

$z = (2\ 300.0 + 20\ 000.0 + 300.0) / 1\ 000 = 22.6$ m

For the purpose of calculating the peak velocity pressure the vessel is divided into five parts: a lower part extending upwards from the ground by a height of 5 m; an upper part extending from a height of 20 m to the top of the vessel, and three horizontal strips in the middle region, each with a height of 5 m.

The detailed calculations for the top part of the vessel are given below, with the calculated values for the other parts tabulated at the end.

For calculating the altitude factor the value of z is taken as the height to the top of the part under consideration.

NOTE If the vessel is considered as a single part then height z is taken as 0.6 times the total height of the vessel.

Altitude factor for top part from **expression (NA.2a)**:

$c_{alt} = 1 + 0.001 \cdot A \cdot (10/z)^{0.2} = 1 + 0.001 \times 200.0 \times (10/22.6)^{0.2} = 1.170$

Fundamental value of basic wind velocity from **expression (NA.1)**:

$v_{b,0} = v_{b,map} \cdot c_{alt} = 26.0 \times 1.17 = 30.42$ m/s

Directional factor c_{dir} = 1.0 and seasonal factor c_{season} = 1.0

Using the recommended values of the shape parameter K = 0.2 and the exponent n = 0.5 gives a value of c_{prob} = 1.0.

Basic wind velocity from **expression (4.1)**, multiplied by c_{prob}:

$v_b = c_{dir} \cdot c_{season} \cdot c_{prob} \cdot v_{b,0} = 1.0 \times 1.0 \times 1.0 \times 30.42 = 30.42$ m/s

For vessels in country terrain categories h_{dis} = 0, therefore ($z - h_{dis}$) = 22.6 m

Roughness factor for top part from **Figure NA.3**, where the distance upwind to shoreline is 10.0 km:

$c_r(z)$ = 1.185

The vessel is not in Town terrain, so the roughness correction factor $c_{r,T}$ from **Figure NA.4** is not applicable.

The vessel is located on a level site, so the orography factor $c_o(z)$ = 1.0

Mean wind velocity at a height z above the terrain from **4.3.1, expression (4.3)**:

$v_m(z) = c_r(z) \cdot c_o(z) \cdot v_b = 1.185 \times 1.0 \times 30.42 = 36.06$ m/s

The procedure in **Annex A.4** may be used to determine the effect of large and considerably taller neighbouring structures (see **NA.2.14**). For this example it is assumed that there are no large neighbouring structures.

The turbulence intensity $I_v(z)$ is not required for $z \leq 50$ m.

Air density from **NA.2.18** ρ = 1.226 kg/m^3

Basic velocity pressure from **expression (4.10)**:

$q_b = \rho \cdot v_b^2/2 = 1.226 \times 30.42^2/2 = 567.3$ m/s

Exposure factor for top part from **Figure NA.7** $c_e(z)$ = 3.056

The vessel is not in Town terrain, so the exposure correction factor $c_{e,T}$ from **Figure NA.4** is not applicable.

Peak velocity pressure for top part for sites in Country terrain when orography is not significant, from **expression (NA.3a)**:

$q_p(z) = c_e(z).q_b = 3.056 \times 567.3 = 1\,733.7$ N/m^2

Summary of calculated values for each part of vessel

Part	1	2	3	4	5
z (m)	5.0	10.0	15.0	20.0	22.6
c_{alt}	1.200	1.200	1.184	1.174	1.170
v_b (m/s)	31.20	31.20	30.79	30.53	30.42
$c_r(z)$	0.906	1.031	1.107	1.161	1.185
$v_m(z)$ (m/s)	28.27	32.16	34.07	35.45	36.05
q_b (N/m^2)	596.7	596.7	581.3	571.2	567.2
$c_e(z)$	2.030	2.487	2.768	2.970	3.057
$q_p(z)$ (N/m^2)	1 211.1	1 483.9	1 609.0	1 696.8	1 733.7

5.3 Natural frequency of vibration

Vessel height $h = z = 22.6$ m

Vessel diameter $b = (1\,000.0 + 2 \times 50.0) / 1\,000 = 1.1$ m

Ratio $h / b = 22.6 / 1.1 = 20.545$

Height $h < 60$ m, but ratio $h / b > 6.5$, therefore the value of $c_s c_d$ cannot be taken as 1.

In order to determine the structural factor $c_s c_d$ it is necessary to calculate the natural frequency of vibration of the vessel.

Using the equation for the first natural frequency of vibration given in this enquiry case the deflection x_j due to the weight W_j of each section j needs to be calculated.

For this example the vessel is treated as having two sections – the skirt is section 1 and the vessel shell is section 2. More complex vessels might need to be divided into more sections. The more sections used the greater will be the accuracy of the calculation.

NOTE These sections need not be the same as the parts used in the calculations for the peak velocity pressure.

Skirt outside radius $R_{os} = 1\,000.0 / 2\,000 = 0.500$ m

Skirt inside radius $R_{is} = (1\,000.0 - 2 \times 10.0) / 2\,000 = 0.490$ m

Second moment of area:

$$I_1 = \pi \cdot \left({R_{os}}^4 - {R_{is}}^4\right)/4 = \pi \cdot \left(0.5^4 - 0.49^4\right)/4 = 0.003811 \text{ m}^4$$

Vessel outside radius $R_{ov} = 1\,000.0 / 2\,000 = 0.500$ m

Vessel inside radius $R_{iv} = (1\,000.0 - 2 \times 13.0) / 2\,000 = 0.487$ m

Second moment of area:

$$I_2 = \pi \cdot \left({R_{ov}}^4 - {R_{iv}}^4\right)/4 = \pi \cdot \left(0.5^4 - 0.487^4\right)/4 = 0.004909 \text{ m}^4$$

Weight of skirt $W_1 = 618$ kg

Length of skirt $L_1 = 2\,300 / 1\,000 = 2.3$ m

Height from underside of base to centre of gravity of skirt (taking skirt openings and attachments into account):

$L_{1b} = 1\,100 / 1\,000 = 1.1$ m

Height from centre of gravity of skirt to bottom tan line of vessel:

$L_{1t} = 2.3 - 1.1 = 1.2$ m

NOTE The calculations for the location of the centre of gravity are not included here. The weight of the base ring can be ignored for natural frequency calculations.

Weight of vessel $W_2 = 9\,671$ kg

Length of vessel (tan to tan) $L_2 = 20\,000 / 1\,000 = 20.0$ m

Height from bottom tan line to centre of gravity of vessel (taking nozzles, internals, attachments, insulation and contents into account):

$L_{2b} = 10\,969 / 1\,000 = 10.969$ m

Moment at centre of gravity of skirt due to weight of vessel:

$$M_1 = g \cdot W_2 \cdot (L_{1t} + L_{2b}) = 9.81 \times 9\,671 \times (1.2 + 10.969) = 1\,154\,504 \text{ N.m}$$

Using equations from *Roark's Formulas for Stress & Strain*, 7th Edition, Table 8.1, deflection at centre of gravity of skirt:

$$x_1 = \frac{g \cdot (W_1 + W_2) \cdot L_{1b}^3}{3 \cdot E_s \cdot 10^6 \cdot I_1} + \frac{M_1 \cdot L_{1b}^2}{2 \cdot E_s \cdot 10^6 \cdot I_1}$$

$$= \frac{9.81 \times (618 + 9\,671) \times 1.1^3}{3 \times 209\,000 \times 10^6 \times 0.003811} + \frac{1\,154\,504 \times 1.1^2}{2 \times 209\,000 \times 10^6 \times 0.003811}$$

$$= 0.000056 + 0.000877 = 0.000933\,\text{m}$$

Shear force at top of skirt $F_{ts} = g \cdot W_2 = 9.81 \times 9\,671 = 94\,873$ N

Moment at top of skirt $M_{ts} = g \cdot W_2 \cdot L_{2b} = 9.81 \times 9\,671 \times 10.969 = 1\,040\,657$ N.m

Deflection at top of skirt due to W_1:

$$x_{ts1} = \frac{g \cdot W_1 (2 \cdot L_1^3 - 3 \cdot L_1^2 \cdot L_{1t} + L_{1t}^3)}{6 \cdot E_s \cdot 10^6 \cdot I_1}$$

$$= \frac{9.81 \times 618 \times (2 \times 2.3^3 - 3 \times 2.3^2 \times 1.2 + 1.2^3)}{6 \times 209\,000 \times 10^6 \times 0.003811} = 0.000009\,\text{m}$$

Deflection at top of skirt due to W_2:

$$x_{ts2} = \frac{M_{ts} \cdot L_1^2}{2 \cdot E_s \cdot 10^6 \cdot I_1} + \frac{F_{ts} \cdot L_1^3}{3 \cdot E_s \cdot 10^6 \cdot I_1}$$

$$= \frac{1\,040\,657 \times 2.3^2}{2 \times 209\,000 \times 10^6 \times 0.003811} + \frac{94\,873 \times 2.3^3}{3 \times 209\,000 \times 10^6 \times 0.003811}$$

$$= 0.003456 + 0.000483 = 0.003939\,\text{m}$$

Total deflection at top of skirt:

$x_{ts} = x_{ts1} + x_{ts2} = 0.000009 + 0.003939 = 0.003948$ m

Slope at top of skirt due to W_1:

$$\theta_{ts1} = \frac{g \cdot W_1 \cdot (L_1 - L_{1t})^2}{2 \cdot E_s \cdot 10^6 \cdot I_1} = \frac{9.81 \times 618 \times (2.3 - 1.2)^2}{2 \times 209\,000 \times 10^6 \times 0.003811} = 0.000005\,\text{m}$$

Slope at top of skirt due to W_2:

$$\theta_{ts2} = \frac{M_{ts} \cdot L_1}{E_s \cdot 10^6 \cdot I_1} + \frac{F_{ts} \cdot L_1^2}{2 \cdot E_s \cdot 10^6 \cdot I_1}$$

$$= \frac{1\,040\,657 \times 2.3}{209\,000 \times 10^6 \times 0.003811} + \frac{94\,873 \times 2.3^2}{2 \times 209\,000 \times 10^6 \times 0.003811}$$

$$= 0.003005 + 0.000315 = 0.003320\,\text{m}$$

Total slope at top of skirt:

$$\theta_{ts} = \theta_{ts1} + \theta_{ts2} = 0.000005 + 0.003320 = 0.003325\,\text{m}$$

Deflection at centre of gravity of vessel:

$$x_2 = \frac{g \cdot W_2 \cdot L_{2b}^3}{3 \cdot E_v \cdot 10^6 \cdot I_2} + x_{ts} + \theta_{ts} \cdot L_{2b}$$

$$= \frac{9.81 \times 9\,671 \times 10.969^3}{3 \times 202\,000 \times 10^6 \times 0.004909} + 0.003948 + 0.003325 \times 10.969$$

$$= 0.042090 + 0.003948 + 0.036472 = 0.082510\,\mathrm{m}$$

NOTE Other methods are available for calculating the deflection, such as the area moment method.

Natural frequency of vibration:

$$n_1 = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{g \cdot \Sigma W_j \cdot x_j}{\Sigma W_j \cdot x_j^2}}$$

$$= \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{9.81 \times (618 \times 0.000933 + 9\,671 \times 0.082510)}{618 \times 0.000933^2 + 9\,671 \times 0.082510^2}} = 1.736\,\mathrm{Hz}$$

Using the alternative approximation, the deflection in metres at the top of the vessel is given by:

$$x_t = \frac{g.W_2 \cdot (2 \cdot L_2^3 - 3 \cdot L_2^2 \cdot L_{2t} + L_{2t}^3)}{6 \cdot E_v \cdot 10^6 \cdot I_2} + x_{ts} + \theta_{ts} \cdot L_2$$

$$= \frac{9.81 \times 9\,671 \times (2 \times 20.0^3 - 3 \times 20.0^2 \times 9.031 + 9.031^3)}{6 \times 202\,000 \times 10^6 \times 0.004909} + 0.003948$$

$$+ 0.003325 \times 20.0$$

$$= 0.094070 + 0.003948 + 0.066500 = 0.164518\,\mathrm{m}$$

where the height from centre of gravity of vessel to top tan line L_{2t} = 20.0 – 10.969 = 9.031 m

The natural frequency of vibration is given by:

$$n_1 = \frac{0.62}{\sqrt{x_t}} = \frac{0.62}{\sqrt{0.164518}} = 1.529\,\mathrm{Hz}$$

5.4 Structural factor $c_s c_d$

The vessel is insulated but does not contain liquid, therefore the value of logarithmic decrement of structural damping δ_s for unlined welded steel stacks with external thermal insulation from **Table F.2** may be used:

Logarithmic decrement of structural damping δ_s = 0.02

From BS EN 1991-1-4 :2005+A1, **Annex B**:

Reference height from **B.1** z_t = 200.0 m

Country terrain category in the UK National Annex corresponds to terrain categories I and II in BS EN 1991-1-4.

Minimum height from **Table 4.1** z_{min} = 2.0 m

Roughness length from **Table 4.1** z_0 = 0.05 m

Reference length scale from **B.1** L_t = 300.0 m

Factor from **B.1** $\alpha = 0.67 + 0.05 \cdot \ln(z_0) = 0.5202$

Reference height (see **Figure 6.1**) $z_s = 0.6 \cdot h = 0.6 \times 22.6 = 13.560$ m

Turbulence length scale for $z_{\min}$ from **expression (B.1)**:

$$L(z_{\min}) = L_t \cdot \left(\frac{z_{\min}}{z_t}\right)^{\alpha} = 300 \times \left(\frac{2.0}{200.0}\right)^{0.5202} = 27.335\,\mathrm{m}$$

For $z_s > z_{\min}$ turbulence length scale for z_s:

$$L(z_s) = L_t \cdot \left(\frac{z_s}{z_t}\right)^{\alpha} = 300 \times \left(\frac{13.56}{200.0}\right)^{0.5202} = 73.982\,\mathrm{m}$$

For $z > z_{\min}$ turbulence length scale for z:

$$L(z) = L_t \cdot \left(\frac{z}{z_t}\right)^{\alpha} = 300 \times \left(\frac{22.6}{200.0}\right)^{0.5202} = 96.501\,\mathrm{m}$$

Background factor from **expression (B.3)**:

$$B^2 = \frac{1}{1 + 0.9 \cdot \left(\frac{b+h}{L(z_s)}\right)^{0.63}} = \frac{1}{1 + 0.9 \cdot \left(\frac{1.1 + 22.6}{73.982}\right)^{0.63}} = 0.6948$$

Factors for calculating mean wind velocity at reference height z_s:

Altitude factor $c_{alt} = 1.188$

Basic wind velocity $v_{b,0} = 30.89$ m/s

Roughness factor $c_r(z_s) = 1.087$

Mean wind velocity at height z_s $v_m(z_s) = 33.60$ m/s

Non-dimensional frequency from **B.1**:

$$f_L(z_s, n_1) = \frac{n_1 \cdot L(z_s)}{v_m(z_s)} = \frac{1.736 \times 73.982}{33.60} = 3.822$$

Power spectral density function from **expression (B.2)**:

$$S_L(z_s, n_1) = \frac{6.8 \cdot f_L(z_s, n_1)}{[1 + 10.2 \cdot f_L(z_s, n_1)]^{5/3}} = \frac{6.8 \times 3.822}{(1 + 10.2 \times 3.822)^{5/3}} = 0.05559$$

Factors from **B.2**:

$$\mathit{Factor}\ \ \eta_h = \frac{4.6 \cdot h}{L(z_s)} \cdot f_L(z_s, n_1) = \frac{4.6 \times 22.6}{73.982} \times 3.822 = 5.371$$

$$\mathit{Factor}\ \ \eta_b = \frac{4.6 \cdot b}{L(z_s)} \cdot f_L(z_s, n_1) = \frac{4.6 \times 1.1}{73.982} \times 3.822 = 0.2614$$

Aerodynamic admittance functions from **expressions (B.7)** and **(B.8)**:

$$R_h = \frac{1}{\eta_h} - \frac{1}{2 \cdot \eta_h^2} \cdot \left(1 - e^{-2 \cdot \eta_h}\right) = \frac{1}{5.371} - \frac{1}{2 \times 5.371^2} \times \left(1 - e^{-2 \times 5.371}\right) = 0.1689$$

$$R_b = \frac{1}{\eta_b} - \frac{1}{2 \cdot \eta_b^2} \cdot \left(1 - e^{-2 \cdot \eta_b}\right) = \frac{1}{0.2614} - \frac{1}{2 \times 0.2614^2} \times \left(1 - e^{-2 \times 0.2614}\right) = 0.8463$$

Resonance response factor from **expression (B.6)**:

$$R^2 = \frac{\pi^2}{2 \cdot \delta_s} \cdot S_L(z_s, n_1) \cdot R_h(\eta_h) \cdot R_b(\eta_b)$$

$$= \frac{\pi^2}{2 \times 0.02} \times 0.05559 \times 0.1689 \times 0.8463 = 1.9606$$

Upcrossing frequency from **expression (B.5)**, but not less than 0.08 Hz:

$$\nu = n_1 \cdot \sqrt{\frac{R^2}{B^2 + R^2}} = 1.736 \times \sqrt{\frac{1.9606}{0.6948 + 1.9606}} = 1.492\,\text{Hz}$$

Averaging time from **B.2** T = 600 seconds

Peak factor from **expression (B.4)**, but not less than 3:

$$k_p = \sqrt{2 \cdot \ln(\nu \cdot T)} + \frac{0.6}{\sqrt{2 \cdot \ln(\nu \cdot T)}}$$

$$= \sqrt{2 \times \ln(1.492 \times 600)} + \frac{0.6}{2 \times \ln(1.492 \times 600)} = 3.850$$

Factors for calculating turbulence intensity at reference height z_s:

Turbulence intensity at height z_s $I_v(z_s)_{flat}$ = 0.167

Turbulence intensity divided by $c_o(z)$ $I_v(z_s)_{flat}$ = 0.167

Turbulence correction factor at height z_s $k_{I,T}$ = 1.0

Turbulence intensity at height z_s $I_v(z_s)$ = 0.167

Structural factor from **6.3.1(1)**:

$$c_s c_d = \frac{1 + 2 \cdot k_p \cdot I_v(z_s) \cdot \sqrt{B^2 + R^2}}{1 + 7 \cdot I_v(z_s)}$$

$$= \frac{1 + 2 \times 3.850 \times 0.167 \times \sqrt{0.6948 + 1.9606}}{1 + 7 \times 0.167} = 1.427$$

There are no nearby structures causing increased turbulence so the requirements of **6.3.3** for wake buffeting have not been considered.

5.5 Wind force

Force coefficient for a circular cylinder without free end flow as recommended in this enquiry case $c_{f,0}$ = 1.2

Total length of vessel $l = z$ = 22.6 m

Slenderness ratio from **Table NA.10**:

$$\lambda = \frac{2l}{b} \cdot \frac{2}{c_{f,0}} = \frac{2 \times 22.6}{1.1} \times \frac{2}{1.2} = 68.485$$

Solidity ratio φ = 1

End effect factor from **Figure 7.36** ψ_λ = 0.908

Force coefficient from **expression (7.19)** $c_f = c_{f,0} \cdot \psi_\lambda = 1.2 \times 0.908 = 1.090$

The vessel is not close to other vertical vessels so requirements of **7.9.3** are not applicable.

Effective wind pressure for top section of vessel:

$q_w = c_s c_d \cdot c_f \cdot q_p(z_e) = 1.427 \times 1.09 \times 1\,733.7 = 2\,696.6$ N/m^2

Reference area for top section of vessel $A_{ref} = 3.1\ m^2$

Wind force from **expression (5.3)**:

$F_w = c_s c_d \cdot c_f \cdot q_p(z_e) \cdot A_{ref} = 1.427 \times 1.09 \times 1\,733.7 \times 3.1 = 8\,359.6$ N

Height from base to centre of area $h_{bc} = 0.5 \times (22.6 + 20.0) = 21.3$ m

Bending moment at base $M_b = F_w \cdot h_{bc} = 8\,359.6 \times 21.3 = 178\,095$ N.m

Height from bottom tan line to centre of area:

$h_{tc} = 0.5 \times (22.6 + 20.0) - 2.3 = 19.0$ m

Bending moment at bottom tan line $M_t = F_w \cdot h_c = 8\,359.6 \times 19.0 = 158\,832$ N.m

Summary of forces and moments

Part	1	2	3	4	5
q_w (N/m^2)	1 883.5	2 307.9	2 502.4	2 639.0	2 696.6
A_{ref} (m^2)	5.8	6.1	6.1	6.1	3.1
F_w (N)	10 924.3	14 078.2	15 264.6	16 097.9	8 359.6
h_{bc} (m)	2.5	7.5	12.5	17.5	21.3
M_b (N.m)	2 7311	10 5587	190 808	281 713	178 095
$A_{ref,t}$ (m^2)	3.3	6.1	6.1	6.1	3.1
$F_{w,t}$ (N)	6 215.6	14 078.2	15 264.6	16 097.9	8 359.6
h_{tc} (m)	1.35	5.2	10.2	15.2	9.0
M_t (N.m)	8 391	7 3207	155 699	244 688	158 832

Total shear force at base F_b = 64 724.6 N

Total moment at base M_b = 783 514 N.m

Total shear force at bottom tan line F_t = 60 015.9 N

Total moment at bottom tan line M_t = 64 0817 N.m

5.6 Stresses in vessel and skirt

Stresses at bottom of vessel in accordance with Annex B of this specification:

Circumferential stress $\sigma_\theta = \dfrac{pR_{iv}}{e_{av}} = \dfrac{1.0 \times 487.0}{13} = 37.46\,\text{N/mm}^2$

Longitudinal tensile stress:

$$\sigma_z = \frac{pR_{iv}^2}{(2R_{iv} + e_{av})e_{av}} - \frac{W}{\pi(2R_{iv} + e_{av})e_{av}} + \frac{4M}{\pi(2R_{iv} + e_{av})^2 e_{av}}$$

$$= \frac{1.0 \times 487.0^2}{(2 \times 487.0 + 13.0) \times 13.0} - \frac{9.81 \times 9\,671}{\pi \times (2 \times 487.0 + 13.0) \times 13.0}$$

$$+ \frac{4 \times 640\,817 \times 1\,000}{\pi \times (2 \times 487.0 + 13.0)^2 \times 13.0}$$

$$= 18.48 - 2.35 + 64.43 = 80.56\,\text{N/mm}^2$$

Longitudinal compressive stress (with p = 0):

$$\sigma_z = -\frac{W}{\pi(2R_{iv} + e_{av})e_{av}} - \frac{4M}{\pi(2R_{iv} + e_{av})^2 e_{av}}$$

$$= -\frac{9.81 \times 9\,671}{\pi \times (2 \times 487.0 + 13.0) \times 13.0} - \frac{4 \times 6\,408\,170 \times 1\,000}{\pi \times (2 \times 487.0 + 13.0)^2 \times 13.0}$$

$$= -2.35 - 64.43 = -66.78\,\text{N/mm}^2$$

Longitudinal compressive stress at bottom of skirt:

$$\sigma_z = -\frac{W}{\pi(2R_{is} + e_{as})e_{as}} - \frac{4M}{\pi(2R_{is} + e_{as})^2 e_{as}}$$

$$= -\frac{9.81 \times (9\,671 + 618)}{\pi \times (2 \times 490.0 + 10.0) \times 10.0} - \frac{4 \times 783\,514 \times 1\,000}{\pi \times (2 \times 490.0 + 10.0)^2 \times 10.0}$$

$$= -3.24 - 101.79 = -105.03\,\text{N/mm}^2$$

5.7 Vortex shedding to PD 6688-1-4

Ratio h / b > 6.0, therefore the effects of vortex shedding need to be investigated

Strouhal number from PD 6688-1-4:2015, **Table A.1** St = 0.18

The critical wind velocity from **equation A.2**:

$$v_{crit,1} = \frac{b \cdot n_1}{St} = \frac{1.1 \times 1.736}{0.18} = 10.61\,\text{m/s}$$

Mean wind velocity at top of vessel v_m = 36.05 m/s

Critical wind velocity $v_{crit} < 1.25 \cdot v_m$ therefore the effects of vortex shedding need to be investigated

Reynolds number at the critical wind velocity from **equation A.5**, taking the kinematic viscosity ν of air as 15 × 10^{-6} m^2/s

$$Re(v_{crit,1}) = \frac{b \cdot v_{crit,1}}{\nu} = \frac{1.1 \times 10.61}{15 \times 10^{-6}} = 7.781 \times 10^5$$

Equivalent mass per unit length – see **F.4 (2)**:

$$m_e = \frac{9\,671}{20.0 + 0.3} = 476.4\,\text{kg/m}$$

Scruton number from **equation A.4**:

$$Sc = \frac{2 \cdot \delta_s \cdot m_e}{\rho \cdot b^2} = \frac{2 \times 0.02 \times 476.4}{1.226 \times 1.1^2} = 12.85$$

The diameter is constant over the top half of the vessel, and the vessel is not in a grouped or in-line arrangement, therefore Approach 2 in **A.1.5.3** should be used. The calculations using Approach 1 are included here for information.

Approach 1 in PD 6688-1-4:2015, A.1.5.2:

Initial estimated value of ratio L_j / b = 6

Correlation length L_j = 6 × 1.1 = 6.6 m

Mode shape factor from **Table A.5** K = 0.13

Correlation length factor for vertical cantilevered structure from **Table A.5**, but not greater than 0.6:

Taking $\lambda = h / b = 20.545$ and $\lambda = L_j / b = 6.0$, $\lambda_j / \lambda = 0.29204$

$$K_w = 3 \cdot \frac{\lambda_j}{\lambda} \cdot \left[1 - \frac{\lambda_j}{\lambda} + \frac{1}{3} \cdot \left(\frac{\lambda_j}{\lambda}\right)^2\right]$$

$$= 3 \times 0.29204 \times \left[1 - 0.29204 + \frac{0.29204^2}{3}\right] = 0.6452$$

The calculated value is greater than 0.6, therefore $K_w = 0.6$

Basic value of lateral force coefficient from **Figure A.5** for $Re(v_{crit,1}) = 7.781 \times 10^5$:

$c_{lat,0} = 0.20$

Height to the centre of correlation length $z = h - L_j / 2 = 22.6 - 6.6 / 2 = 19.3$ m

Height $(z - h_{dis}) = 19.3 - 0 = 19.3$ m

Roughness factor from **Figure NA.3** $c_r(z) = 1.155$

Mean wind velocity at centre of correlation length from **expression (4.3)**:

$v_{m,Lj} = c_r(z) \cdot c_o(z) \cdot v_b = 1.155 \times 1.0 \times 30.42 = 35.14\,\text{m/s}$

Ratio $v_{crit,1} / v_{m,Lj} = 10.61 / 35.14 = 0.302$

Lateral force coefficient from **Table A.3** for $v_{crit,1} / v_{m,Lj} \leq 0.83$:

$c_{lat} = c_{lat,0} = 0.20$

The vessel is not in a row or grouped arrangement therefore the influence of nearby vessels need not be taken into account.

Largest displacement from **equation A.7**:

$$y_{F,maz} = \frac{1}{St^2} \cdot \frac{1}{Sc} \cdot K \cdot K_w \cdot c_{lat} \cdot b$$

$$= \frac{1}{0.18^2} \times \frac{1}{12.85} \times 0.13 \times 0.6 \times 0.2 \times 1.1 = 0.0412\,\text{m}$$

Ratio $y_{F,max} / b = 0.0412 / 1.1 = 0.0375$

Value of ratio L_j / b from **Table A.4** $L_j / b = 6$

This is the same as the initial estimated value therefore further iteration is not required.

Approach 2 in PD 6688-1-4:2015, A.1.5.3:

In the calculations for approach 2 the diameter b is taken as the outside diameter, including insulation, of the top half of the vessel.

From **Table A.6** for Reynolds number $Re(v_{crit,1}) = 7.781 \times 10^5$:

$$C_c = 0.005 + \frac{\log\left[Re\left(v_{crit,1}\right)\right] - \log\left[5 \times 10^5\right]}{\log[1 \times 10^6] - \log[5 \times 10^5]} \times (0.01 - 0.005)$$

$$= 0.005 + \frac{\log\left[7.781 \times 10^5\right] - \log\left[5 \times 10^5\right]}{\log[1 \times 10^6] - \log[5 \times 10^5]} \times (0.01 - 0.005)$$

$$= 0.005 + \frac{5.891 - 5.699}{6.0 - 5.699} \times 0.005 = 0.00819$$

Aerodynamic excitation parameter $K_a = 100 \times C_c = 0.819$

Normalized limiting amplitude $a_L = 0.4$

Constants from **equation A.16**:

$$c_1 = \frac{a_L^2}{2} \cdot \left(1 - \frac{Sc}{4 \cdot \pi \cdot K_a}\right) = \frac{0.4^2}{2} \times \left(1 - \frac{12.85}{4 \times \pi \times 0.819}\right) = -0.019885$$

$$c_2 = \frac{\rho b^2}{m_e} \cdot \frac{a_L^2}{K_a} \cdot \frac{C_c^2}{St^4} \cdot \frac{b}{h}$$

$$= \frac{1.226 \times 1.1^2}{476.4} \times \frac{0.4^2}{0.819} \times \frac{0.00819^2}{0.18^4} \times \frac{1.1}{22.6} = 0.0000018919$$

Standard deviation of the displacement from **equation A.15**:

$$\sigma_y = b \cdot \sqrt{c_1 + \sqrt{c_1^2 + c_2}}$$

$$= 1.1 \times \sqrt{-0.019885 + \sqrt{0.019885^2 + 0.0000018919}} = 0.007582$$

Peak factor from **equation A.17**:

$$k_p = \sqrt{2} \cdot \left\{1 + 1.2 \cdot \tan^{-1}\left[0.75 \cdot \left(\frac{Sc}{4 \cdot \pi \cdot K_a}\right)^4\right]\right\}$$

$$= \sqrt{2} \times \left\{1 + 1.2 \times \tan^{-1}\left[0.75 \times \left(\frac{12.85}{4 \times \pi \times 0.819}\right)^4\right]\right\}$$

$$= \sqrt{2} \times \left\{1 + 1.2 \times \tan^{-1}[1.823]\right\} = 3.228$$

Characteristic maximum displacement at the point with the largest movement from **equation A.13**:

$$y_{max} = \sigma_y \cdot k_p = 0.007582 \times 3.228 = 0.02447\,\text{m}$$

Forces and moments:

Inertia force per unit length acting perpendicular to the wind direction, at the top of the vessel from **equation A.6**, taking $\Phi_1(z)$ from **expression (F.13)** with $\zeta = 2.0$:

$$F_L(z) = m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot \left(\frac{z}{h}\right)^2 \cdot y_{max}$$

$$= 476.4 \times (2 \times \pi \times 1.736)^2 \times \left(\frac{22.6}{22.6}\right)^2 \times 0.02447 = 1\,387.0\ \text{N/m}$$

Height to top of vessel shell $x_{2t} = 22.6$ m

Height to bottom of vessel shell $x_{2b} = 2.3$ m

Horizontal inertia force acting on vessel shell:

$$F_2(z) = \frac{m(z) \cdot (2 \cdot \pi \cdot n_1)^2 \cdot y_{max} \cdot \left(z_{2t}^3 - z_{2b}^3\right)}{3 \cdot h^2}$$

$$= \frac{476.4 \times (2 \times \pi \times 1.736)^2 \times 0.02447 \times \left(22.6^3 - 2.3^3\right)}{3 \times 22.6^2} = 10\,437\,\text{N}$$

Bending moment acting at the bottom tan line:

$$M_2(z_b) = \frac{m(z)\cdot(2\cdot\pi\cdot n_1)^2\cdot y_{\max}\cdot\left(3\cdot z_{2t}^4 + z_{2b}^4 - 4\cdot z_{2t}^3\cdot z_{2b}\right)}{12\cdot h^2}$$

$$= \frac{476.4\times(2\times\pi\times 1.736)^2\times 0.02447\times\left(3\times 22.6^4 + 2.3^4 - 4\times 22.6^3\times 2.3\right)}{12\times 22.6^2}$$

$$= 153\,076\ \text{N.m}$$

Bending moment at base (z = 0) due to force $F_2(z)$ on vessel:

$$M_2(z) = \frac{m(z)\cdot(2\cdot\pi\cdot n_1)^2\cdot y_{\max}\cdot\left[3\cdot\left(z_{2t}^4 + z_{2b}^4\right) - 4\cdot z\cdot\left(z_{2t}^3\cdot z_{2b}^3\right)\right]}{12\cdot h^2}$$

$$= \frac{476.4\times(2\times\pi\times 1.736)^2\times 0.02447\times\left[\left(3\times\left(22.6^4 + 2.3^4\right) - 0\right)\right]}{12\times 22.6^2}$$

$$= 177\,082\ \text{N.m}$$

Mass per unit length of skirt:

$$m(z) = \frac{618}{2.3} = 268.7\ \text{kg/m}$$

Height to top of skirt x_{1t} = 2.3 m

Height to bottom of skirt x_{1b} = 0.0 m

Horizontal inertia force acting on skirt:

$$F_1(z) = \frac{m(z)\cdot(2\cdot\pi\cdot n_1)^2\cdot y_{\max}\cdot\left(z_{1t}^3 - z_{1b}^3\right)}{3\cdot h^2}$$

$$= \frac{268.7\times(2\times\pi\times 1.736)^2\times 0.02447\times\left(2.3^3 - 0\right)}{3\times 22.6^2} = 6\ \text{N}$$

Bending moment at base (z = 0) due to force $F_1(z)$ on skirt:

$$M_1(z) = \frac{m(z)\cdot(2\cdot\pi\cdot n_1)^2\cdot y_{\max}\cdot\left[3\cdot\left(z_{1t}^4 - z_{1b}^4\right) - 4\cdot z\cdot\left(z_{1t}^3 - 3_{1b}^3\right)\right]}{12\cdot h^2}$$

$$= \frac{268.7\times(2\times\pi\times 1.736)^2\times 0.02447\times\left[3\times\left(2.3^4 - 0\right) - 0\right]}{12\times 22.6^2} = 11\ \text{N.m}$$

Total inertia force at base F = 10 437 + 6 = 10 443 N

Total bending moment at base M = 177 082 + 11 = 177 093 N.m

Stresses in vessel and skirt

Longitudinal stress at bottom of vessel due to moment:

$$\sigma_z = \frac{4M}{\pi(2R_{iv} + e_{av})^2 e_{av}} = \frac{4\times 153\,076\times 1\,000}{\pi\times(2\times 487.0 + 13.0)^2\times 13.0} = 15.39\ \text{N/mm}^2$$

Longitudinal stress at bottom of skirt due to moment:

$$\sigma_z = \frac{4M}{\pi(2R_{is} + e_{as})^2 e_{as}} = \frac{4\times 177\,093\times 1\,000}{\pi\times(2\times 490.0 + 10.0)^2\times 10.0} = 23.01\ \text{N/mm}^2$$

Fatigue assessment:

Design life $T = 3.2\times 10^7\times 20 = 6.4\times 10^8$ seconds

Bandwidth factor ε_0 = 0.3

$\sqrt{2}$ times the modal value of the Weibull probability distribution:

$v_0 = 0.2 \times v_m = 0.2 \times 36.05 = 7.21$ m/s

Number of load cycles N caused by vortex excited oscillation from **equation A.10**:

$$N = \frac{2 \cdot T \cdot n_1 \cdot \varepsilon_0 \cdot \left(\dfrac{v_{\mathrm{crit},1}}{v_0}\right)^2}{\exp\left[\left(\dfrac{{v_{\mathrm{crit},1}}^2}{v_0}\right)\right]}$$

$$= \frac{2 \times 6.4 \times 10^8 \times 1.736 \times 0.3 \times \left(\dfrac{10.609}{7.21}\right)^2}{\exp\left[\left(\dfrac{10.609}{7.21}\right)^2\right]} = 1.656 \times 10^8 \text{ cycles}$$

6 References

BSI publications

BS 6399-2, *Loadings for buildings — Part 2: Code of practice for wind loads*.

BS EN 1991-1-4:2005+A1:2010, *Eurocode 1: Actions on structures — Part 1-4: General actions — Wind actions*

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CP 3, *Code of basic data for the design of buildings — Chapter V: Loading — Part 2: Wind loads*. (Replaced by BS 6399-2).

Other publications

[1] COOK, N.J. *The designer's guide to wind loading of building structures — Part 2: Static structures.* London: Butterworth Heinemann Ltd, 1990.

[2] *Rules for Pressure Vessels*. The Hague: Dienst Voor Het Stoomwezen, 1994.

[3] YOUNG, W.C. and BUDYNAS, R.G. *Roark's Formulas for Stress and Stratin*. Seventh Edition. New York: McGraw Hill, 2002.

Enquiry Case 5500/128

Preliminary rules for a less conservative design of jacket blocking rings

Enquiry

Can the committee provide a less conservative approach than that required by **3.11.3**, for the design of blocking rings for type 2 jackets?

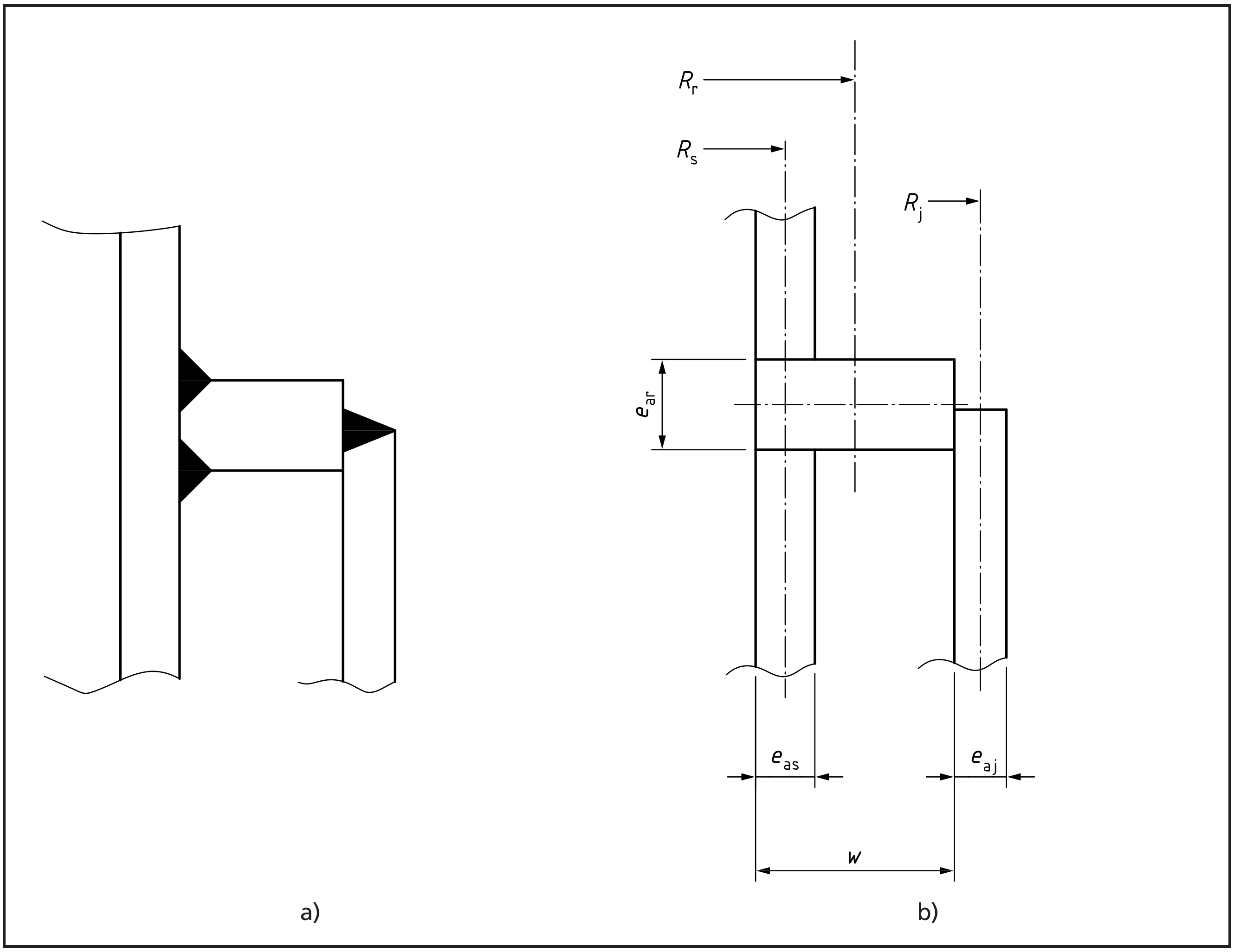
Reply

The following analysis methods may be used for jacketted vessels of type 2 construction (see Figure 3.11-1), to check the design of blocking rings, adjacent shell and jacket cylinders. A limit analysis is used to check the blocking ring and an elastic analysis used to check the shell and jacket. Both analyses should be carried out and satisfied. The blocking ring analysis is only valid when $e_{ar} \geq 2e_{as}$, this will ensure that a plastic hinge within the ring itself should not occur.

Figure 1 shows a typical physical arrangement and the corresponding geometry assumed in the analyses.

NOTE These rules are based upon the simplifying assumption that the ring does not dilate. For this, and other reasons, the rules are still slightly conservative.

Figure 1 **Physical and analysis geometry**



Notation

i is the component identifier as follows:
j – jacket cylinder,
r – blocking ring;
s – main shell;

D_i is the rotational modulus of component i [see Equation (9)];

E_i is the modulus of elasticity from Table 3.6-3 of component i;

e_i is the minimum thickness of component i to **3.5.1.2**;

e_{ai} is the analysis thickness of component i;

f_i is the design stress of component i;

K_i is the rotational stiffness of component i [see Equations (7) and (8)];

M is a moment per unit circumference;

p_j is the jacket design pressure;

p_s is the shell design pressure;

R_i is the mean radius of component i;

ν is Poisson's ratio;

w is the effective width of the blocking ring, = $(R_j - 0.5e_{aj}) - (R_s - 0.5e_{as})$;

λ_i equals $\sqrt[4]{\dfrac{3(1-\nu^2)}{(R_i e_{ai})^2}}$ for component i.

Limit analysis

The blocking ring design is assessed using a limit analysis approach to avoid a mode of failure in which the blocking ring, jacket and shell rotate as a body with plastic hinges in the jacket and shell.

The moment on the ring due to jacket pressure is calculated as follows:

$$M = \frac{p_j R_r (R_j - R_s)}{2} \quad (1)$$

The moment that can be resisted by the blocking ring acting alone is calculated as follows:

$$M_r = \frac{f_r e_{ar}{}^2 w}{4R_r} \quad (2)$$

The moment that can be resisted by the shell (above and below the ring) is calculated as follows:

$$M_s = \frac{f_s e_{as}{}^2 \left[1 - 0.5(e_s/e_{as})^2\right]}{2} \quad (3)$$

The moment that can be resisted by the jacket is calculated as follows:

$$M_j = \frac{f_j e_{aj}{}^2 \left[1 - 0.5(e_j/e_{aj})^2\right]}{4} \quad (4)$$

The resisting moment from the shear force between shell and ring is calculated as follows:

$$M_Q = \frac{f_s e_{as} e_{ar}}{4}\sqrt{\frac{e_{as}}{R_s}} \quad (5)$$

The following rule given by Equation (6) should be checked. If it is not satisfied then the thickness of the blocking ring (or shell or jacket) should be increased.

$$M \le M_r + M_s + M_j + M_Q \quad (6)$$

Elastic analysis of jacket and shell

This analysis estimates the primary and secondary stress intensities in the jacket and in the shell and compares the results with $3f_i$. Working with a simple model in which the ring is assumed not to dilate and that all of its rotation is due to the axial load from the jacket, the sources of stress in the jacket and shell are:

a) moments resulting from rotation of the ring;

b) pressure in a cylinder with one end radially constrained;

c) change of radius due to ring rotation.

Elastic analysis equations are used to calculate the stiffnesses of components and thus the sharing of the moment that is imposed on the blocking ring due to jacket pressure [see Equation (1)].

The stiffness of the blocking ring in rotation is calculated as follows:

$$K_r = \frac{E_r e_{ar}{}^2 w}{12 R_r{}^2} \tag{7}$$

The stiffness of any cylindrical shell in rotation with constraint against radial displacement is calculated as follows:

$$K_i = 2D_i \lambda_i \tag{8}$$

where

$$D_i = \frac{E_i e_{ai}{}^3}{12(1-\nu^2)} \tag{9}$$

However, the rotational stiffness of the jacket and shell attached to the ring are increased because the junction between the ring and jacket or shell is at the end of a lever of length $\frac{e_{ar}}{2}$.

This result is in an effective stiffness of the jacket of:

$$K_j = 2D_j \lambda_j \left(1 + \frac{\lambda_j e_{ar}}{2}\right)^2 \tag{10}$$

and an effective stiffness of the shell (above and below the ring) of:

$$K_s = 2D_s \lambda_s \left(1 + \frac{\lambda_s e_{ar}}{2}\right)^2 \tag{11}$$

The moments in the jacket and shell resulting from the ring moment M [see Equation (1)] are calculated as follows:

$$M_{ej} = \frac{M}{(1 + 0.5\lambda_j e_{ar})}\left(\frac{K_j}{K_j + 2K_s + K_r}\right) \tag{12}$$

$$M_{es} = \frac{M}{(1 + 0.5\lambda_s e_{ar})}\left(\frac{K_s}{K_j + 2K_s + K_r}\right) \tag{13}$$

Jacket stresses

The worst case is with a pressurized jacket and zero pressure in the shell.

Longitudinal bending stress due to the moment on the jacket M_{ej}

[see Equation (12)] is calculated as follows:

$$\sigma_{jla} = \frac{6M_{ej}}{e_{aj}{}^2} \tag{14}$$

Longitudinal membrane stress due to pressure is calculated as follows:

$$\sigma_{jlb} = \frac{p_j R_j}{2e_{aj}} \tag{15}$$

Longitudinal bending stress due to the rotational restraint of pressure dilation is calculated as follows:

$$\sigma_{jha} = \frac{1.49 p_j R_j}{e_{aj}} \tag{16}$$

NOTE Hoop stresses are of the same sign as the longitudinal stresses and can be ignored when considering the Tresca criterion.

Maximum primary plus secondary stress intensity (see **A.3.4.1.1**), is on the inside surface and is calculated as follows:

$$\sigma_{jmax} = \sigma_{jla} + \sigma_{jlb} + \sigma_{jha} \tag{17}$$

This stress intensity should be limited to $3f_j$. If this rule is not satisfied, then the blocking ring (or jacket) thickness should be increased.

Shell stresses

The shell needs only to be checked if the shell and jacket are simultaneously pressurized and if $p_s > p_j$.

Longitudinal bending stress due to the moment on the shell M_{es}

[see Equation (13)] is calculated as follows:

$$\sigma_{sla} = \frac{6M_{es}}{{e_{as}}^2} \tag{18}$$

Longitudinal membrane stress due to pressure is calculated as follows:

$$\sigma_{slb} = \frac{\left(p_s - p_j\right) R_s}{2e_{as}} \tag{19}$$

Longitudinal bending stress due to rotational restraint of pressure dilation is calculated as follows:

$$\sigma_{sha} = \frac{1.49\left(p_s - p_j\right) R_s}{e_{as}} \tag{20}$$

Hoop membrane stress due to the shell being forced radially outwards by the ring rotation is tensile and can be ignored for the calculation of stress intensity. Hoop membrane stress due to pressure is zero, as there is no radial dilation of the ring.

Maximum primary + secondary stress intensity (see **A.3.4.1.1**), is on the outside surface and;

$$\sigma_{smax} = \sigma_{sla} + \sigma_{slb} + \sigma_{sha} \tag{21}$$

This stress intensity should be limited to $3f_s$. If this rule is not satisfied, then the blocking ring (or shell) thickness should be increased.

Enquiry Case 5500/133

Flat unstayed ends of non-circular shape and associated flanges

Enquiry

Can the committee give guidance on how these flat unstayed ends of rectangular, elliptical or obround shape should be designed? Also can the committee give guidance on how openings in these ends should be designed and how associated rectangular flanges can be designed?

Reply

Yes, the following rules can be used for the above applications. These rules for flat ends with or without openings have been based upon the rules that were given in the original issue of PD 5500:2000.

1 Flat ends

1.1 Flat unstayed end of rectangular, elliptical or obround shape without openings

The minimum thickness e of a welded or bolted non-circular flat end as shown in Figure 3.5-34, or Figure 3.5-35a), Figure 3.5-35b) or Figure 3.5-35d) is given by:

$$e = CZa\sqrt{p/f}$$

where

- a is the smallest dimension of the rectangular, elliptical or obround end;
- b is the largest dimension of the rectangular, elliptical or obround end;
- C is as defined in **3.5.5.2** of the main text;
- f is the design stress for the end;
- p is the design pressure;
- Z is a coefficient derived from Figure 1.

For a bolted flat end with a full face gasket, the bolt spacing shall not exceed:

$$2 \times d_{\mathrm{b}} + \left(\frac{E}{200000}\right)^{0.25} \times \left(\frac{6e}{m + 0.5}\right)$$

where

- d_{b} is the bolt outside diameter;
- E is the modulus of elasticity of the flat end material, at the design temperature, given in Table 3.6-3 (in $\mathrm{N/mm^2}$);
- m is the gasket factor given in Table 3.8-4.

If necessary the flat end thickness shall be increased to enable this requirement to be met.

The minimum thickness e of a bolted non-circular flat end with a narrow face gasket, as shown in Figure 3.5-35c) is given by:

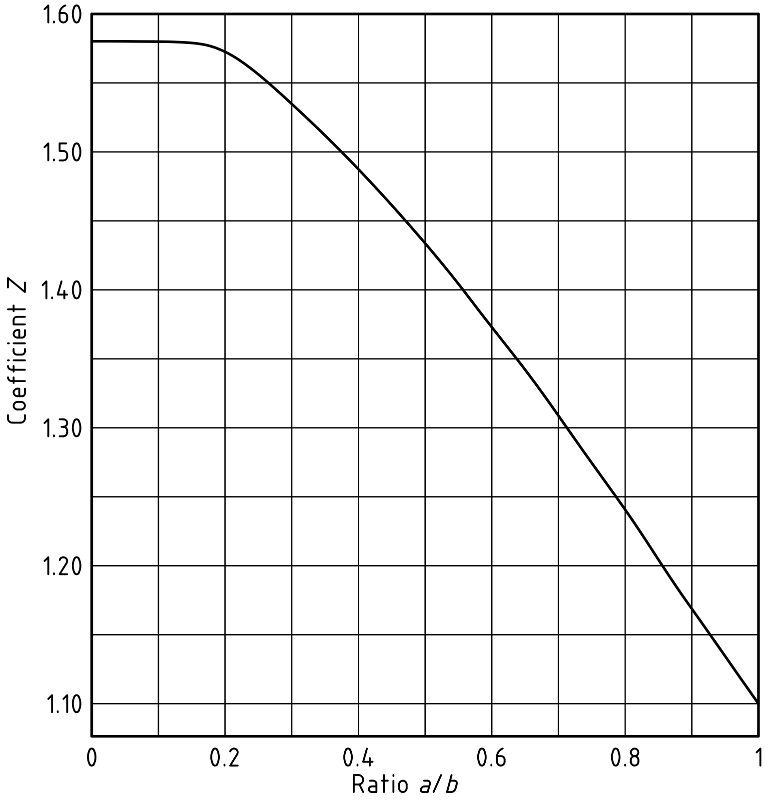
$$e = \max\left(\sqrt{\frac{6Wh_{\mathrm{G}}}{nP_{\mathrm{bmin}}S_{\mathrm{FA}}}};\ \sqrt{\frac{0.3G_{\mathrm{s}}^2pZ^2}{S_{\mathrm{FO}}} + \frac{6W_{\mathrm{m1}}h_{\mathrm{G}}}{nP_{\mathrm{bmin}}S_{\mathrm{FO}}}}\right)$$

and the minimum thickness e_1 at and beyond the gasket is given by:

$$e_1 = \max\left(\sqrt{\frac{6Wh_{\mathrm{G}}}{nP_{\mathrm{bmin}}S_{\mathrm{FA}}}};\ \sqrt{\frac{6W_{\mathrm{ml}}h_{\mathrm{G}}}{nP_{\mathrm{bmin}}S_{\mathrm{FO}}}}\right)$$

where W, S_{FO} and S_{FA} are as defined in **3.8.2** of the main text, G_s, W_{m1}, h_G and n are as defined in **2.2.2** of this Enquiry Case, and P_{bmin} is the minimum spacing between bolt centres.

Figure 1 **Value of coefficient *Z* for non-circular flat ends**



NOTE In the case of long flat ends (where a/b < 0.2) other than as shown in Figure 3.5-35, a minimum value of CZ equal to 0.71 should be used in accordance with ***3.5.5.3.1*** *to determine the thickness required.*

1.2 Flat unstayed ends of rectangular, elliptical or obround shape with openings

The following rules can be used for openings no larger than 0.5 times the smallest dimension of the end (i.e. *a*, as defined above). Holes may extend up to the inside of the shell if the end is welded, or up to the inside of the gasket if the end is bolted. A group of holes of varying hole diameter and with any orientation is permitted. For a group of holes, each pair of adjacent holes should be checked.

For long rectangular ends, with $b/a > 3$, where the reinforcement is obtained by increasing the thickness of the whole end, the minimum thickness e_p is the greater of:

$$e_p = e\sqrt{a/(a-d)}$$

and

$$e_p = e\sqrt{P/(P-d_A)}$$

where

d	is the diameter of the opening or inside diameter of a branch;
d_A	is the mean diameter of a pair of openings;

P is the distance between the centres of the pair of openings;

other terms are as defined above.

Where the pair of openings is along, i.e. parallel to, the short axis of the end, only the first equation for calculating e_p needs to be considered.

For other ends, each opening should be provided with a reinforcement area equal to 0.5*de* axisymmetrically and uniformly disposed about the centre of the opening. The same piece of material should not be used to reinforce more than one opening. The maximum length of branch material that may be included as reinforcement should be 2.5 times the lesser of the nominal thicknesses of the branch and the flat end.

2 Rectangular flanges

2.1 General

In the following notation the suffix L is used to denote lengths measured parallel to the long side of the flange, and the suffix S for lengths parallel to the short side. This notation is shown in Figure 2, Figure 3, Figure 4.

References given are to the main text unless otherwise stated.

It is assumed that the flange width measured from the inside edge to the outside edge is the same for both long and short sides, similarly the distance from the centre line of the bolts to the inside edge of the flange is assumed the same for both long and short sides, i.e. $A_L - B_L = A_S - B_S$ and $C_L - B_L = C_S - B_S$. Further it is assumed that the flange hub thicknesses g_0 and g_1 are the same for both long and short sides.

Figure 2 **Overall flange dimensions**

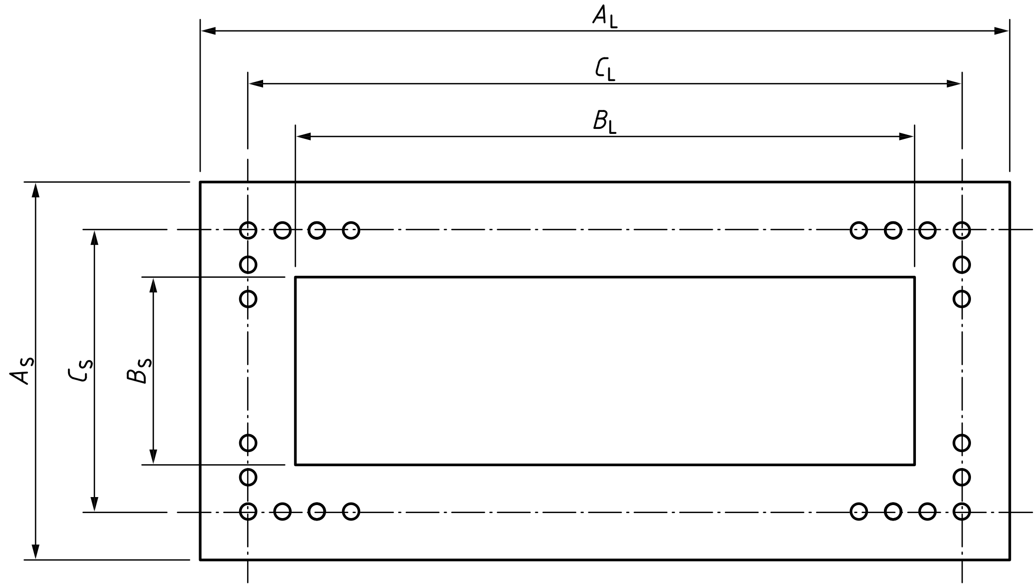


Figure 3 **Narrow face gasket flange**

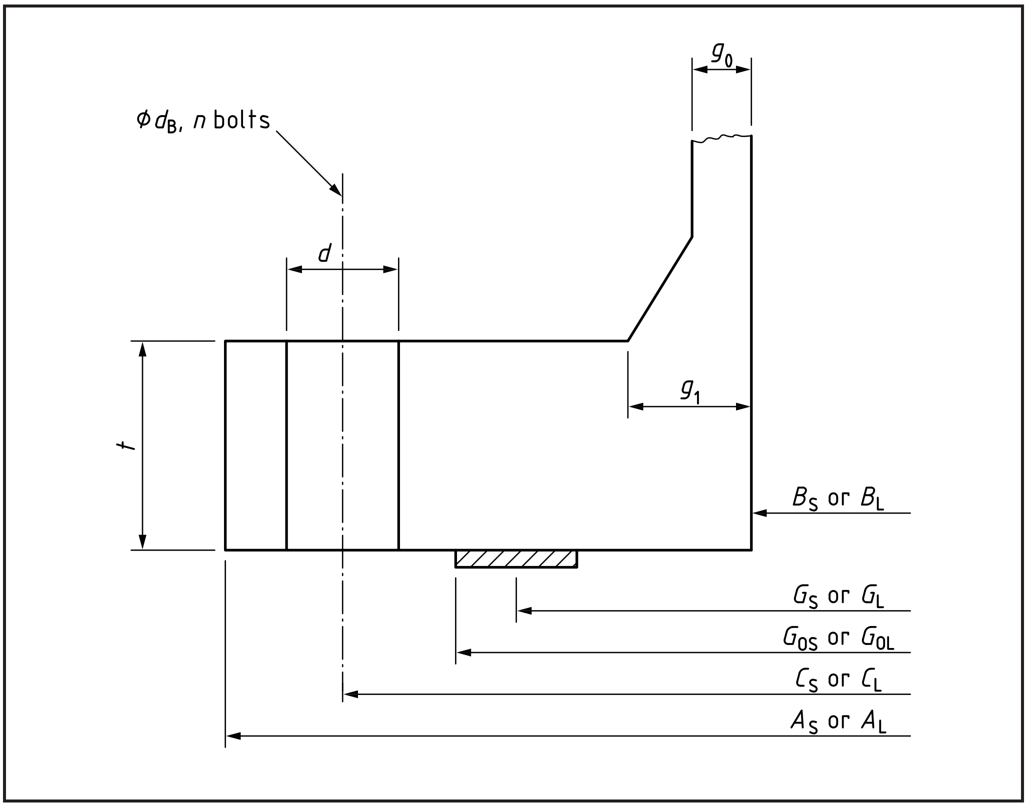
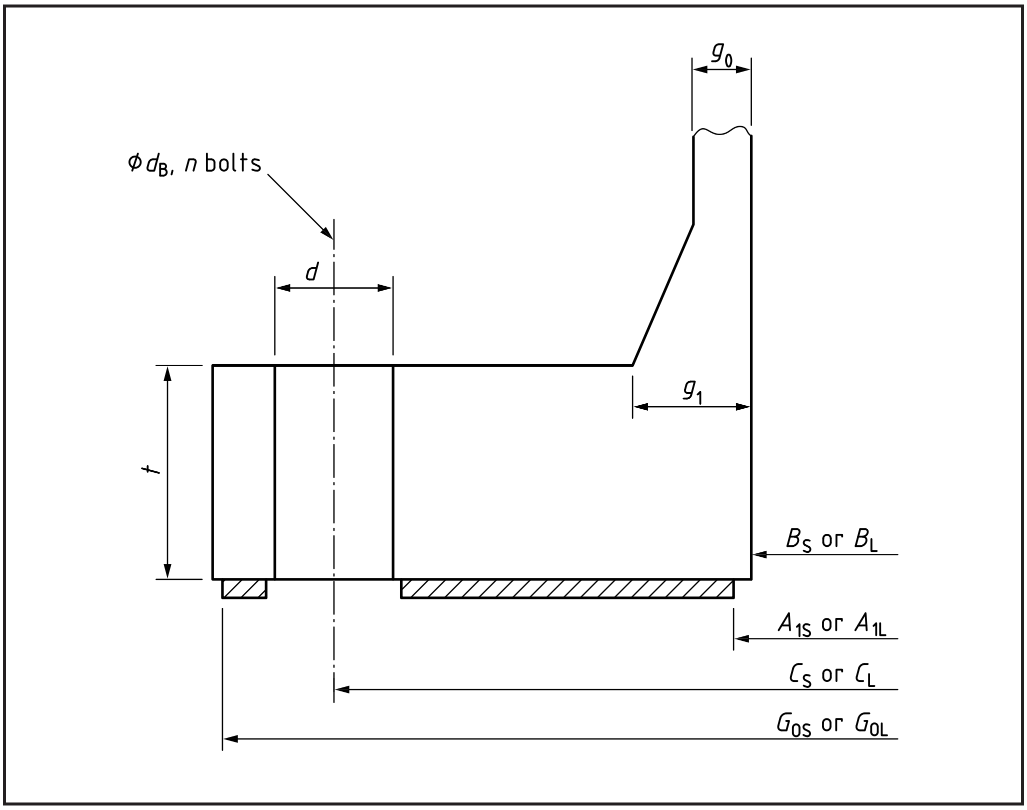


Figure 4 **Full face gasket flange**



2.2 Rectangular flange with narrow face gasket

2.2.1 General

For a circular flange calculated using the integral method, account is taken of the support from the shell, however if calculated using the loose method the flange ring is assumed to carry the full calculated moment. For a rectangular flange with a narrow face gasket the flange cannot be assumed to carry any of the moment at the mid-span points and thus the following method assumes that the rectangular shell, to which the flange is attached, carries the full moment.

NOTE These assumptions become more conservative for rectangular flanges with an aspect ratio close to a square.

The notation used in **2.2** of this Enquiry Case, if not given, is based on **3.8.2** of the main text.

2.2.2 Gasket details

It is assumed that the gasket width N is the same for both long and short sides; the gasket dimensions b and b_0 are calculated from **3.8.2** and gasket factors m and y are obtained from Table 3.8-4.

The dimensions of the gaskets are related as follows:

$$G_L = G_{0L} - 2b$$

$$G_S = G_{0S} - 2b$$

where G_0 is the outside dimension of the gasket.

2.2.3 Forces

The forces on the flange should be calculated using the following equations.

$$H = pG_L G_S$$

$$H_G = 2b(2G_L + 2G_S)mp$$

$$H_D = pB_L B_S$$

$$H_T = H - H_D$$

2.2.4 Bolting loads

The minimum required bolt loads should be calculated using the following equations.

$$W_{m1} = H_G + H$$

$$W_{m2} = 2(G_L + G_S)by$$

Check assumed bolting as per **3.8.3.2**.

2.2.5 Moments and moment arms

The moment arms and total moment acting on the flange should be calculated using the following equations.

$$h_D = \frac{C_L - B_L - g_1}{2}$$

$$h_G = \frac{C_L - G_L}{2}$$

$$h_T = \frac{2C_L - B_L - G_L}{4}$$

$$M_{op} = H_D h_D + H_G h_G + H_T h_T$$

$$M_{atm} = W h_G$$

2.2.6 Flange thickness

Using design stresses S_{FA} and S_{FO}, as defined in **3.8.2**, the flange thickness, t, is the greatest of the following:

$$\text{Operating condition, } t = \sqrt{\frac{6M_{op}}{S_{FO}[2(C_L + C_S) - nd]}}$$

$$\text{Bolting-up condition, } t = \sqrt{\frac{6M_{atm}}{S_{FA}[2(C_L + C_S) - nd]}}$$

$$\text{Bolt spacing condition, } t = \frac{(P_{bmax} - 2d_b) \times (m + 0.5)}{6 \times (E/200000)^{0.25}}$$

where

P_{bmax} is the maximum spacing between bolt centres on long or short sides;
d_b is the bolt diameter;
n is the number of bolts.

2.2.7 Shell thickness

There are no provisions in PD 5500 for the design of shells of rectangular cross-section. The required thickness of the shell plates should be determined using an appropriate method, such as that given in BS EN 13445-3:2021, Section 15.

Using design stresses S_{HA} and S_{HO}, as defined in **3.8.2**, the shell thickness, g_0, should not be less than the greater of the following:

$$\text{Operating condition, } g_0 = \sqrt{\frac{6M_{op}}{2(B_L + B_S)S_{HO}}}$$

$$\text{Bolting-up condition, } g_0 = \sqrt{\frac{6M_{atm}}{2(B_L + B_S)S_{HA}}}$$

2.3 Rectangular flange with full face gasket

2.3.1 General

For a rectangular flange with a full face gasket the bending moment is balanced by the reaction H_R outside the bolt circle, as in **3.8.4**. It is assumed that no moment is transmitted into the rectangular shell to which the flange is attached.

The notation used in **2.3** of this Enquiry Case, if not given, is based upon that of **3.8.4** of the main text, if given there, or **3.8.2** of the main text.

2.3.2 Gasket details

It is assumed that the gasket width is the same for both long and short sides, i.e. $G_{0L} - A_{1L} = G_{0S} - A_{1S}$. The gasket factors m and y are obtained from Table 3.8-4.

$$b'_o = \min[(G_{0L} - C_L); (C_L - A_{1L})]$$

$$b' = 4\sqrt{b'_o}$$

$$G_L = C_L - d - 2b''$$

$$G_S = C_S - d - 2b''$$

$2b''$ is the effective gasket pressure width, taken as 5 mm.

2.3.3 Forces

The forces on the flange should be calculated using the following equations.

$$H = p(C_L - d)(C_S - d)$$

$$H_G = 2b''(2G_L + 2G_S)mp$$

$$H_D = pB_LB_S$$

$$H_T = H - H_D$$

2.3.4 Moments and moment arms

The moment arms and total moment acting on the flange should be calculated using the following equations.

$$h_D = \frac{C_L - B_L - g_1}{2}$$

$$h_G = \frac{d + 2b''}{2}$$

$$h_T = \frac{C_L + d + 2b'' - B_L}{4}$$

$$M = H_Dh_D + H_Gh_G + H_Th_T$$

$$h_R = \frac{G_{0L} - C_L - d}{4} + \frac{d}{2}$$

$$H_R = \frac{M}{h_R}$$

2.3.5 Bolting loads

The minimum required bolting loads should be calculated using the following equations.

$$W_{m1} = H_G + H + H_R$$

$$W_{m2} = 2(C_L + C_S)b'y$$

Check assumed bolting as per **3.8.3.2**.

2.3.6 Flange thickness

Using the design stress S_{FO}, as defined in **3.8.2**, the flange thickness, t, is the greater of the following:

$$\text{Operating condition, } t = \sqrt{\frac{6M_{op}}{S_{FO}[2(C_L + C_S) - nd]}}$$

$$\text{Bolt spacing condition, } t = \frac{(P_{bmax} - 2d_b) \times (m + 0.5)}{6 \times (E/200000)^{0.25}}$$

where

P_{bmax} is the maximum spacing between bolt centres on long or short sides;
d_b is the bolt diameter;
n is the number of bolts.

Enquiry Case 5500/134

Superseded referenced standards

Enquiry

Many of the standards referenced in PD 5500 have been replaced by BS EN standards; can the committee give some guidance on how these superseded documents should be used?

Reply

The committee recognizes that many of the referenced standards have been declared obsolescent or withdrawn and have been superseded in total, or in part, by BS EN standards. Such changes are listed in the following table.

Superseded standard	Current status	Occurrence in PD 5500	Proposed replacement standard
BS 4-1	Withdrawn, Superseded	**W.2.2.3.6**	BS EN 10365:2017
BS 21	Withdrawn, Superseded	**3.5.4.8**	BS EN 10226
BS 1400	Withdrawn, Superseded	**Cu.2.3.1.1**, **Cu.2.3.1.3**, Table Cu.2.3-1	BS EN 1982
BS 1501	Withdrawn, Superseded	**1.6**, **3.1.5**, Table K.1-1, Table K.1-2, Table K.1-3, Table K.1-4, Table W.2-1, **W.2.2.1.5**, **W.2.2.3.6**, **W.6.2.3.1**	BS EN 10028 BS EN 10029
BS 1502	Withdrawn, Superseded	Table K.1-1, Table K.1-5	BS EN 10272 BS EN 10273
BS 1503	Withdrawn, Superseded	Table K.1-1, Table K.1-6	BS EN 10222
BS 1504	Withdrawn, Superseded	Table K.1-1, Table K.1-7	BS EN 10213
BS 1515	Withdrawn	**Z.3.2**	
BS 2870	Withdrawn, Superseded	**Cu.2.3.1.1**, **Cu.2.3.1.3**, Table Cu.2.3-1	BS EN 1653
BS 2871	Withdrawn, Superseded	**Cu.2.3.1.1**, **Cu.2.3.1.3**, Table Cu.2.3-1	BS EN 12449 and BS EN 12451
BS 2874	Withdrawn, Superseded	**Cu.2.3.1.1**, **Cu.2.3.1.3**, Table Cu.2.3-1	BS EN 12163 BS EN 12167
BS 2875	Withdrawn, Superseded	**Cu.2.3.1.1**, **Cu.2.3.1.3**, Table Cu.2.3-1	BS EN 1653
BS 2915	Withdrawn, Superseded	**3.13.3.2**	BS EN ISO 4126 Parts 2 and 6
BS 3059	Withdrawn, Superseded	Table K.1-1, Table K.1-8	BS EN 10216-1 BS EN 10216-2 BS EN 10217-1 BS EN 10217-2
BS 3072	Withdrawn	**Ni.2.1.1**, Table Ni.2.3-1, **Ni.3.1**	
BS 3074	Withdrawn	**Ni.2.1.1**, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4, **Ni.3.1**	
BS 3076	Withdrawn	**Ni.2.1.1**, Table Ni.2.3-5, **Ni.3.1**	
BS 3441	Withdrawn	**1.1.4**	
BS 3601	Withdrawn, Superseded	Table K.1-1, Table K.1-9	BS EN 10216-1 BS EN 10217-1
BS 3602-1	Withdrawn, Superseded	Table K.1-1, Table K.1-9	BS EN 10216-2 BS EN 10217-2
BS 3602-2	Withdrawn, Superseded	Table K.1-1, Table K.1-9	BS EN 10217-3 BS EN 10217-5
BS 3603	Withdrawn, Superseded	Table K.1-1, Table K.1-9	BS EN 10216-4 BS EN 10217-4
BS 3604-1	Withdrawn, Superseded	Table K.1-1, Table K.1-10	BS EN 10216-2 BS EN 10217-2

(continued)

Superseded standard	Current status	Occurrence in PD 5500	Proposed replacement standard
BS 3605-1	Withdrawn, Superseded	Table K.1-1, Table K.1-11	BS EN 10216-5
BS 3605-2	Withdrawn, Superseded	Table K.1-1, Table K.1-11	BS EN 10217-7
BS 3606	Withdrawn, Superseded	Table K.1-1, Table K.1-12	BS EN 10216-2 BS EN 10217-2
BS 3920-1:1973	Withdrawn, Superseded	**K.1.1**, **K.1.2**	BS EN 10314:2002
BS 4870-1	Withdrawn, Superseded	**5.2.3.4**, **Cu.5.2**	BS EN ISO 15614-1
BS 4870-2	Withdrawn, Superseded	**Al.5.2.3**, **Cu.5.2**	BS EN ISO 15614-2
BS 6399-2:1997	Withdrawn, Superseded	EC 5500/127	BS EN 1991-1-4:2005+A1:2010
BS EN 287-1:2011	Withdrawn, Superseded	**5.3.1**, **5.3.3**, **5.7.2**	BS EN ISO 9606-1:2017
BS EN 288	Withdrawn, Superseded	**Cu.5.2**	BS EN ISO 15614-6
BS EN 288-3	Withdrawn, Superseded	**5.2.3.4**	BS EN ISO 15614-1:2017+A1
BS EN 288-4	Withdrawn, Superseded	**Al.5.2.3**	BS EN ISO 15614-2
BS EN 10002-1:2001	Withdrawn, Superseded	**K.1.4.1.5**	BS EN ISO 6892-1:2019
BS EN 13185:2001	Withdrawn, Superseded	**5.8.8.1**	BS EN ISO 20485:2018
ISO 6303	Current, Confirmed	**2.1.2.2.2.5**	BS EN ISO 204
PD 6493:1991	Withdrawn, Superseded	**U.2.3**	BS 7910:2019

It should be noted that:

– ”Obsolescent” indicates that the standard will no longer be updated but will be retained to provide for the servicing of existing equipment that is expected to have a long working life;

– ”Withdrawn” indicates that the standard is no longer considered current (i.e. that for new equipment it would not reflect current and accepted practice). The “Withdrawn” status neither implies that a vessel built to a standard during a standard’s period of currency, nor that a vessel built today to a “Withdrawn” standard, is necessarily unsafe. The use of a “Withdrawn” standard is not prohibited, but needs careful consideration.

It is ultimately intended that the superseded standards will be removed from the main text of PD 5500 and replaced with relevant new BS EN standards. Until this occurs it is permissible to use and refer to these superseded standards. In the interim, the committee would be grateful for any comments from users identifying any technical differences, or lack of coverage, with the proposed replacements given in the above table.

The use of replacement BS ENs is permitted if the purchaser, Inspecting Authority and manufacturer are satisfied that the requirements of the superseded standard, as specified by reference in PD 5500, are covered.

Enquiry Case 5500/135

Stress concentration factors for flush nozzles in cylinders

Enquiry

In **G.2.5** there is information to determine the stress concentration factors for nozzles in spherical shells for use in fatigue assessment. Can the committee give any guidance on stress concentration factors for flush nozzles in cylindrical shells?

Reply

The below formula can be used to determine stress concentration factors for flush nozzles in cylindrical shells. The formula is presented in a paper by J.Decock [1].

$$\text{s.c.f} = \frac{2 + \frac{2d}{D}\sqrt{\frac{dt}{DT}} + \frac{1.25d}{D}\sqrt{\frac{D}{T}}}{1 + \frac{t}{T}\sqrt{\frac{dt}{DT}}}(\text{not less than } 1.0)$$

where:

d is the mean nozzle diameter;

D is the mean cylindrical shell diameter;

t is the nozzle analysis thickness;

T is the cylindrical shell analysis thickness;

and the s.c.f. is the highest peak stress referred to the membrane stress in the undisturbed shell.

This peak stress occurs at the inner crotch corner, i.e. at the junction of the inner surface of the shell with the inner surface of the nozzle. It is assumed that whilst the inner crotch corner is lightly dressed, there is not a significant machined radius. A significant machined radius, with the associated removal of reinforcing material, could increase the s.c.f. Annex C, and in particular **C.3.4.6**, gives information on how these peak stresses in nozzles should be assessed for fatigue.

In his paper, Decock describes validation of the above s.c.f. formula with reference to strain gauge measurements on 29 flush nozzles. The tested nozzles had a wide range of dimensionless parameters; the ratio *D*/*T* from 15 to 255 and the ratio *d*/*D* from 0.008 7 to 0.90. Most predicted factors were within ±10% of the observed results.

Bibliography

[1] DECOCK, J. Determination of stress concentration factors and fatigue assessment of flush and extruded nozzles in welded pressure vessels. *2nd Int. Conf. on Pressure Vessel Technology*, ASME, Oct 1973, 821–834.

Enquiry Case 5500/137

Flexibility of the nozzle-vessel interface

Enquiry

G.2.2.4, **G.2.3.5**, **G.2.4.3** and **G.2.4.4** give recommended curves for deflections and rotations caused by radial loads and moments on nozzles in cylindrical and spherical shells. For spherical shells these recommendations assume a rigid, solid nozzle attachment. For cylindrical shells the recommendations assume that the nozzle can be modelled as a patch load, or in the case of an applied moment by two patch loads. Can the committee give any guidance as to how the actual nozzle geometry can be taken into account?

Reply

The limitations in Annex G mentioned above were recognized when these were written. Recent work by Schwarz [1] now enables the committee to give the following additional guidance on how actual nozzle geometry can be used when assessing the flexibility of the nozzle-vessel interface.

1 Introduction

This enquiry case is limited to nozzles perpendicular to the shell and without reinforcement pads. The results given in the paper by Schwarz cover a reasonable range of parameters, as indicated by the limits specified for spherical and cylindrical shells. It is anticipated that future work will be carried out to extend the range.

2 Loads on nozzles in spherical shells

The notation used is based, where possible, on that given in **G.2.1.2**.

C_d	is the deflection factor;
C_r	is the rotation factor;
E	is the modulus of elasticity from Table 3.6-3 (in N/mm²);
M	is the applied moment acting on the nozzle in the spherical shell (in N·mm);
r	is the mean radius of the spherical shell (in mm);
r_o	is the mean radius of the nozzle (in mm);
t	is the analysis thickness of the spherical shell adjacent to the nozzle (in mm);
t_n	is the analysis thickness of the nozzle wall (in mm);
W	is the radial force acting on the nozzle in the spherical shell (positive inwards) (in N);
δ	is the deflection of the spherical shell due to the radial force (in mm);
i_b	is the nozzle rotation at the spherical shell due to the applied moment (in radians).

The following geometrical limit applies:

$0.2 \leq t_n/t \leq 3.33$

NOTE This limit applies to Figure 2 and Table 2. Data is given outside this range in Figure 1 and Table 1.

A non-dimensional factor u is used in Figure 1 and Figure 2 to obtain the deflection and rotation factors and thus the deflection δ and rotation i_b. The factor u is as calculated by equation (G.2.4-2):

$$u = \frac{1.82 r_o}{\sqrt{rt}}$$

The deflection and rotation for a rigid attachment on a spherical shell are evaluated in accordance with **G.2.4.3** and **G.2.4.4**. The effect of the nozzle wall thickness is taken into account by means of the factors C_d and C_r obtained from Figure 1 and Figure 2 in this enquiry case. Tabulated values are given in Table 1 and Table 2.

The deflection δ is calculated using the factor C_d from Figure 1 and the value of the parameter $\delta Et^2/Wr$ obtained from Figure G.2.4-3 using the curve $u = s$:

$$\delta = \left(\delta Et^2/Wr\right) \times C_d \times \frac{Wr}{Et^2}$$

The rotation i_b is calculated using the factor C_r from Figure 2 and the value of the parameter $\frac{\delta}{M} \cdot \frac{Et^2}{\sqrt{r\ /\ t}}$ obtained from Figure G.2.4-9 using the curve $u = s$:

$$i_b = \left(\frac{\delta}{M} \cdot \frac{Et^2}{\sqrt{r/t}}\right) \times C_r \times \frac{M\sqrt{r/t}}{r_o Et^2}$$

Figure 1 **Deflection factor C_d for spherical shell subjected to a radial load**

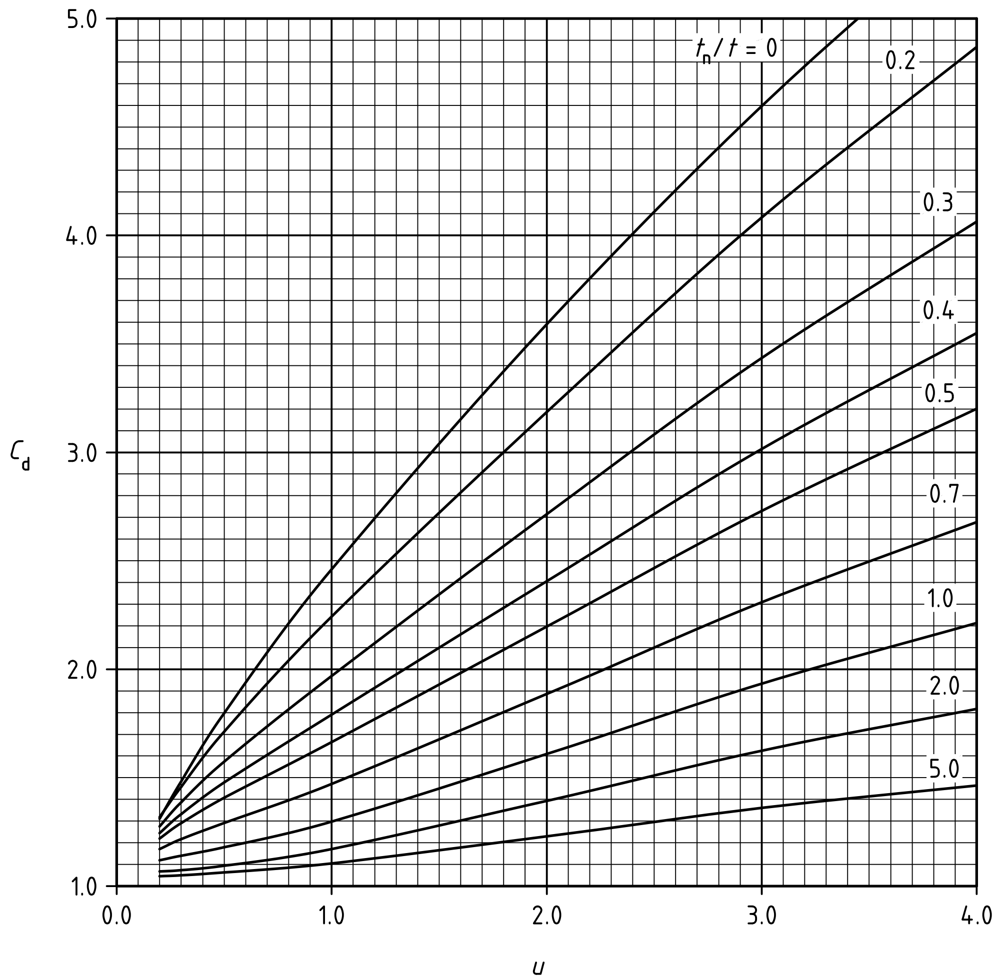


Table 1 **Deflection factor C_d for spherical shell subjected to a radial load**

	u						
t_n/t	**0.2**	**0.3**	**0.5**	**1.0**	**2.0**	**3.0**	**4.0**
0.00	1.310	1.481	1.798	2.460	3.591	4.596	5.480
0.20	1.318	1.457	1.712	2.243	3.187	4.082	4.870
0.30	1.276	1.385	1.574	1.969	2.715	3.435	4.064
0.40	1.245	1.331	1.479	1.790	2.406	3.015	3.550
0.50	1.220	1.289	1.408	1.664	2.196	2.730	3.201
0.70	1.171	1.217	1.291	1.471	1.887	2.309	2.678
1.00	1.119	1.140	1.179	1.298	1.609	1.933	2.214
2.00	1.067	1.074	1.094	1.171	1.393	1.624	1.817
5.00	1.045	1.050	1.063	1.104	1.229	1.360	1.464

Figure 2 **Rotation factor C_r for spherical shell subjected to an external moment**

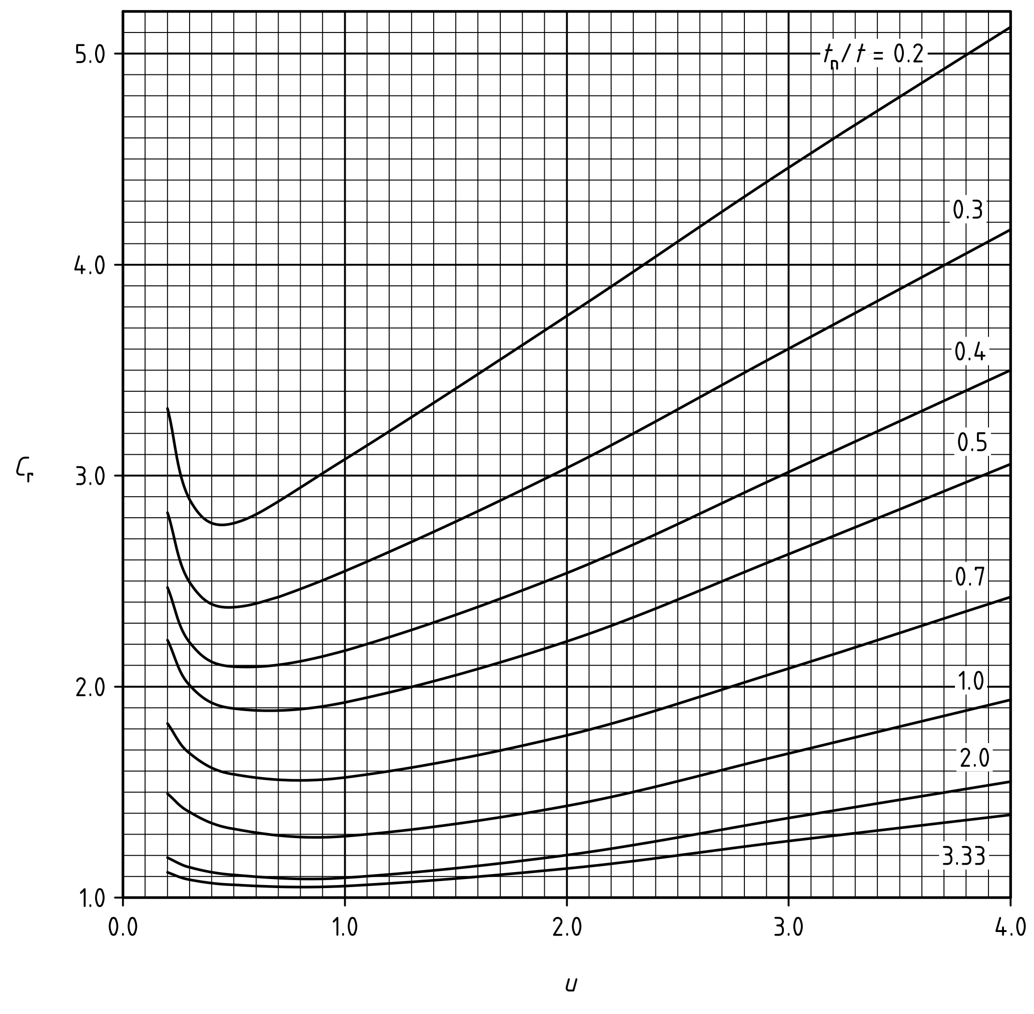


Table 2 **Rotation factor C_r for spherical shell subjected to an external moment**

	u						
t_n/t	**0.2**	**0.3**	**0.5**	**1.0**	**2.0**	**3.0**	**4.0**
0.20	3.318	2.887	2.774	3.078	3.757	4.459	5.126
0.30	2.824	2.495	2.377	2.547	3.037	3.601	4.166
0.40	2.470	2.209	2.094	2.171	2.539	3.017	3.499
0.50	2.220	2.006	1.896	1.925	2.214	2.627	3.055
0.70	1.825	1.685	1.585	1.569	1.769	2.086	2.424
1.00	1.491	1.405	1.326	1.291	1.435	1.682	1.938
2.00	1.189	1.144	1.108	1.093	1.201	1.376	1.550
3.33	1.120	1.085	1.060	1.054	1.138	1.269	1.392

3 Loads on nozzles in cylindrical shells

The notation used is based, where possible, on that given in **G.2.1.2**.

C_d is the deflection factor;
C_{rL} is the rotation factor for a longitudinal moment;
C_{rC} is the rotation factor for a circumferential moment;
E is the modulus of elasticity from Table 3.6-3 (in N/mm^2);
L is the length of the cylindrical shell (in mm);
M_L is the applied longitudinal moment on the nozzle in the cylindrical shell (in N·mm);
M_C is the applied circumferential moment on the nozzle in the cylindrical shell (in N·mm);
r is the mean radius of cylinder (in mm);
r_o is the mean radius of the nozzle (in mm);
T is the analysis thickness of the cylinder (in mm);
t_n is the analysis thickness of the nozzle wall (in mm);
W is the radial force acting on the nozzle in the cylindrical shell (positive inwards) (in N);
δ is the deflection of the cylindrical shell due to the radial force (in mm);
i_b is the nozzle rotation at the shell due to the applied moment (in radians).

The deflections and rotations due to radial loads or external moments are obtained from Figures 3 to 10 using the following geometrical parameters:

r/t is the ratio of cylinder mean radius to cylinder thickness
L/r is the ratio of cylinder length to cylinder mean radius
r_o/r is the ratio of nozzle to cylinder mean radii
t_n/t is the ratio of nozzle to cylinder thicknesses

For the case of a nozzle in a cylindrical shell, the nozzle is assumed to be located at mid-length. The following geometrical limits apply:

$0.05 \le r_o/r \le 0.5$; $1.0 \le L/r \le 5.0$; $0.5 \le t_n/t \le 2.0$.

*NOTE When a nozzle is located close to the end of a cylindrical shell (see **G.2.2.1**) the flexibility of the nozzle-vessel interface is reduced by the stiffening effect of the vessel end (e.g. dished end, or welded or bolted flat end). This effect is not taken into account in this analysis, and the deflections and rotations of nozzles located close to the vessel end will be significantly less than those predicted by the methods in this enquiry case.*

For a radial load W, the deflection parameter ($\delta Er/W$) is obtained from Figure 3 for t_n/t = 1. The effect of the nozzle wall thickness is taken into account by means of the deflection factor C_d which is obtained from Figure 4. Tabulated values are given in Table 3 and Table 4.

The deflection is calculated from the following equation:

$$\delta = (\delta Er/W) \times C_d \times \frac{W}{Er}$$

For a longitudinal moment M_L, the rotation parameter ($i_b Er^3/M_L$) is obtained from Figure 5 for $t_n/t = 1$. The variation in the rotation parameter for different values of L/r is small, so only the curves for $L/r = 1$ and 5 have been included in Figure 5. For $r_o/r = 0.05$ and 0.1 the curves in Figure 5 represent the average for the range of values of L/r from 1 to 5. The full range of data is given in Table 5.

The effect of the nozzle wall thickness is taken into account by means of the rotation factor C_{rL} which is obtained from Figure 6. Tabulated values are given in Table 6.

The rotation (in radians) is calculated from the following equation:

$$i_b = \left(i_b Er^3/M_L\right) \times C_{rL} \times \frac{M_L}{Er^3}$$

For a circumferential moment M_C, the rotation parameter ($i_b Er^3/M_C$) is obtained from Figure 7 for $t_n/t = 1$. The effect of the nozzle wall thickness is taken into account by means of the rotation factor C_{rC} which is obtained from Figure 8. Tabulated values are given in Table 7 and Table 8.

The rotation (in radians) is calculated from the following equation:

$$i_b = \left(i_b Er^3/M_C\right) \times C_{rC} \times \frac{M_C}{Er^3}$$

Table 3 **Deflection parameter $\delta Er/W$ for cylindrical shell with $t_n/t = 1$ subjected to a radial load**

***L/r* = 1**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	356	2.34×10^3	9.74×10^3	4.04×10^4	2.63×10^5	1.08×10^6
0.10	278	1.68×10^3	6.52×10^3	2.50×10^4	1.45×10^5	5.35×10^5
0.20	172	926	3.22×10^3	1.10×10^4	5.33×10^4	1.69×10^5
0.30	111	550	1.76×10^3	5.54×10^3	2.39×10^4	6.94×10^4
0.50	52.3	227	662	1.89×10^3	7.74×10^3	2.30×10^4
***L/r* = 2**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	498	3.44×10^3	1.48×10^4	6.30×10^4	4.33×10^5	1.84×10^6
0.10	413	2.67×10^3	1.10×10^4	4.39×10^4	2.76×10^5	1.09×10^6
0.20	288	1.65×10^3	6.31×10^3	2.27×10^4	1.23×10^5	4.26×10^5
0.30	208	1.05×10^3	3.79×10^3	1.26×10^4	6.03×10^4	1.87×10^5
0.50	120	475	1.50×10^3	4.58×10^3	1.81×10^4	5.03×10^4
***L/r* = 3**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	668	4.15×10^3	1.89×10^4	8.26×10^4	5.70×10^5	2.46×10^6
0.10	578	3.35×10^3	1.46×10^4	6.14×10^4	3.93×10^5	1.60×10^6
0.20	440	2.23×10^3	8.99×10^3	3.52×10^4	1.95×10^5	7.04×10^5
0.30	344	1.55×10^3	5.68×10^3	2.11×10^4	1.04×10^5	3.43×10^5
0.50	220	840	2.47×10^3	8.05×10^3	3.54×10^4	9.91×10^4
***L/r* = 5**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.06×10^3	6.13×10^3	2.43×10^4	1.11×10^5	8.13×10^5	3.46×10^6
0.10	961	5.27×10^3	1.98×10^4	8.74×10^4	6.10×10^5	2.46×10^6
0.20	787	4.02×10^3	1.37×10^4	5.51×10^4	3.50×10^5	1.27×10^6
0.30	643	3.16×10^3	9.78×10^3	3.54×10^4	2.09×10^5	7.08×10^5
0.50	411	2.06×10^3	5.68×10^3	1.63×10^4	7.80×10^4	2.46×10^5

Figure 3 **Deflection parameter $\delta Er/W$ for cylindrical shell with t_n/t = 1 subjected to a radial load**

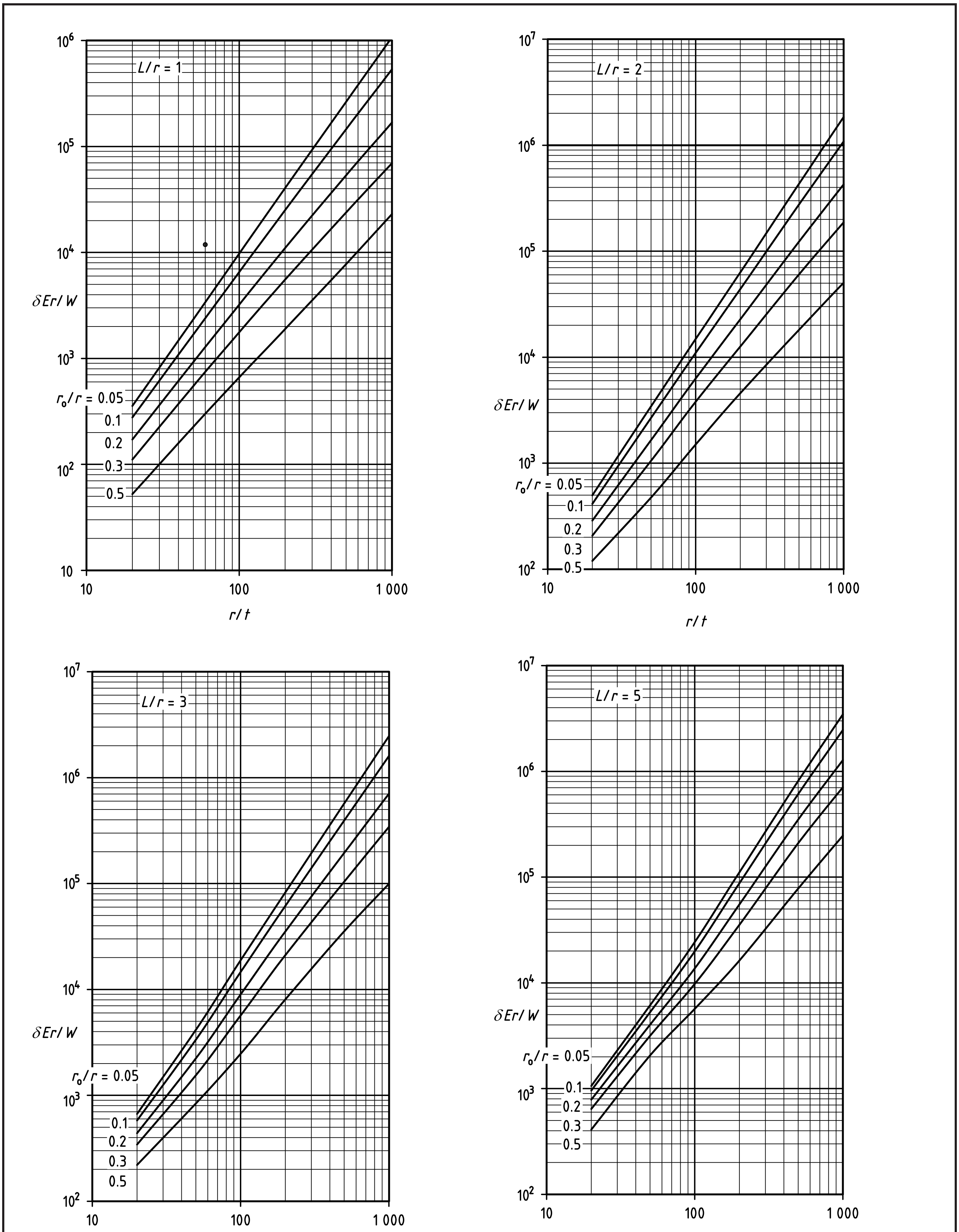


Figure 4 **Deflection factor C_d for cylindrical shell subjected to a radial load *W***

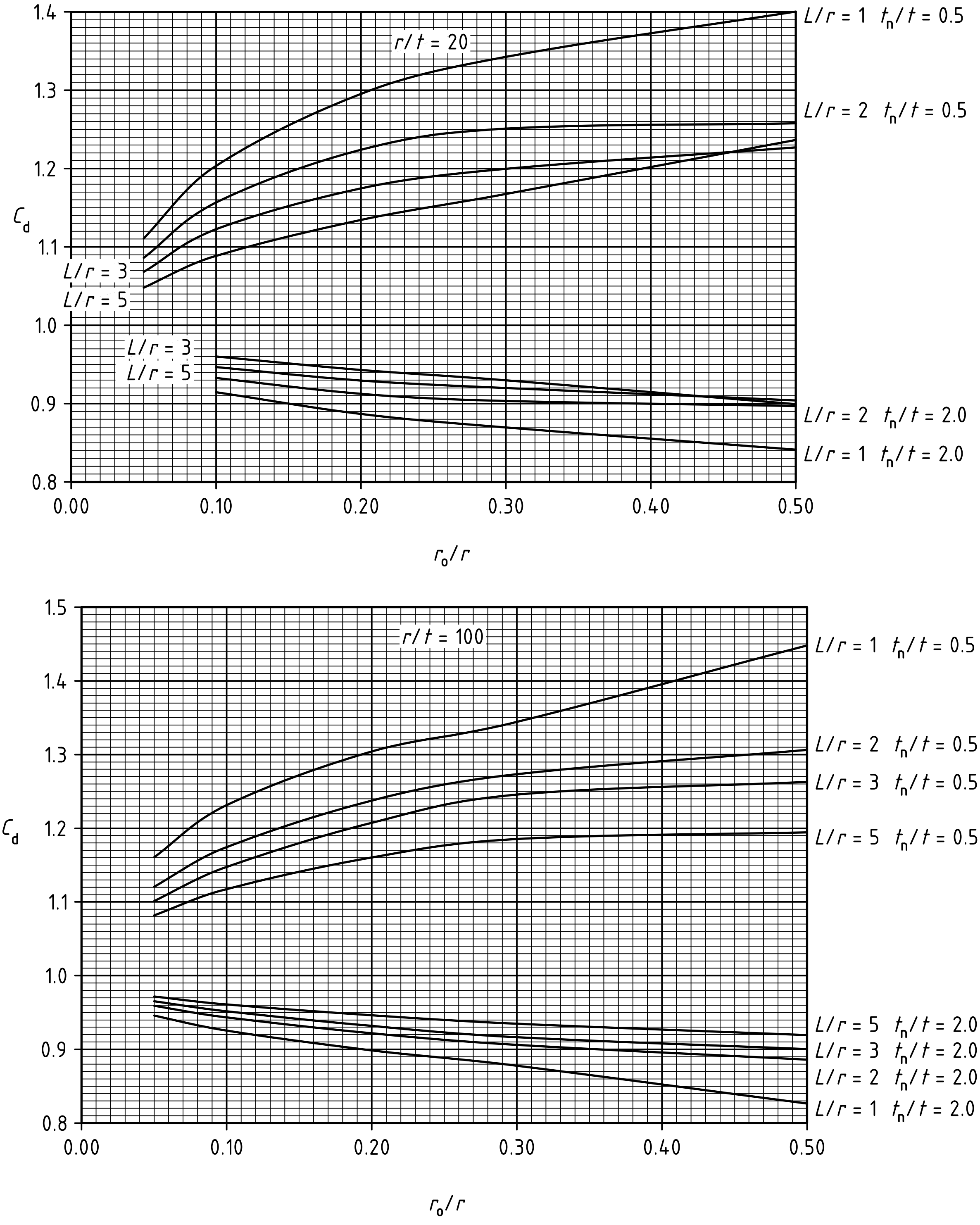


Figure 4 **Deflection factor C_d for cylindrical shell subjected to a radial load *W (continued)***

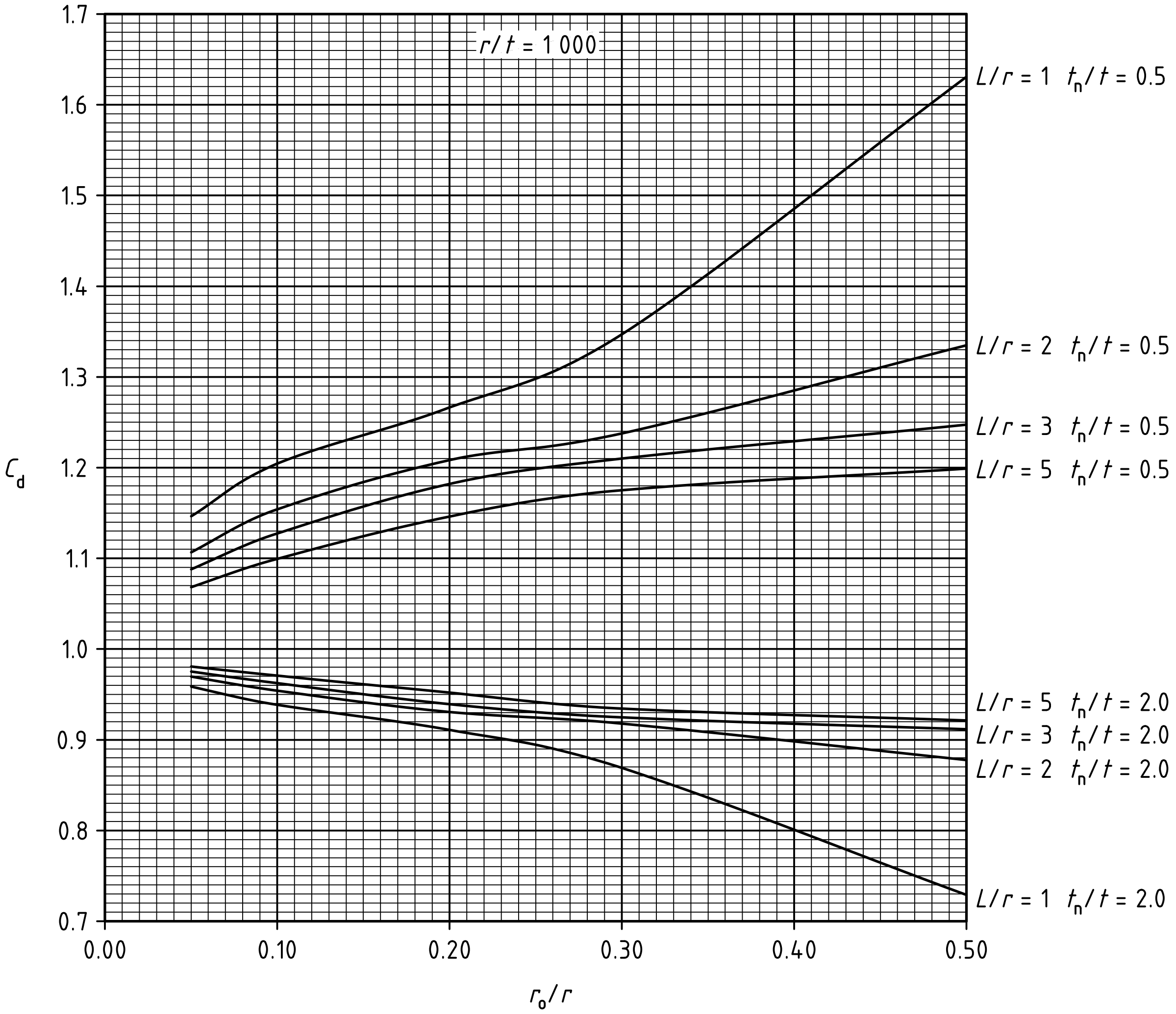


Table 4 **Deflection factor C_d for cylindrical shell subjected to a radial load**

***r/t* = 20**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.111	1.086	1.068	1.048				
0.10	1.204	1.157	1.123	1.089	0.915	0.933	0.947	0.960
0.20	1.295	1.224	1.175	1.134	0.887	0.912	0.929	0.943
0.30	1.343	1.251	1.200	1.168	0.870	0.903	0.920	0.930
0.50	1.400	1.258	1.227	1.237	0.841	0.897	0.904	0.899
***r/t* = 100**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.161	1.120	1.101	1.082	0.946	0.959	0.965	0.972
0.10	1.231	1.174	1.148	1.117	0.926	0.943	0.952	0.961
0.20	1.304	1.237	1.207	1.160	0.899	0.922	0.932	0.946
0.30	1.344	1.273	1.246	1.185	0.878	0.906	0.916	0.935
0.50	1.448	1.306	1.263	1.195	0.826	0.886	0.900	0.919
***r/t* = 1000**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.146	1.107	1.088	1.068	0.959	0.970	0.975	0.981
0.10	1.204	1.154	1.127	1.099	0.939	0.954	0.962	0.971
0.20	1.266	1.208	1.182	1.146	0.911	0.931	0.939	0.952
0.30	1.347	1.238	1.210	1.175	0.869	0.918	0.925	0.935
0.50	1.631	1.335	1.248	1.199	0.729	0.878	0.912	0.921

Table 5 **Rotation parameter $i_b E r^3 / M_L$ for cylindrical shell with t_n/t = 1 subjected to a longitudinal moment**

***L*/*r* = 1**	***r*/*t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.48×10^4	1.24×10^5	6.15×10^5	2.93×10^6	2.08×10^7	8.47×10^7
0.10	6.83×10^3	4.90×10^4	2.08×10^5	8.37×10^5	4.84×10^6	1.73×10^7
0.20	2.24×10^3	1.29×10^4	4.68×10^4	1.63×10^5	7.94×10^5	2.52×10^6
0.30	941	4.80×10^3	1.60×10^4	5.13×10^4	2.29×10^5	6.93×10^5
0.50	242	1.07×10^3	3.20×10^3	9.53×10^3	4.04×10^4	1.23×10^5
***L*/*r* = 2**	***r*/*t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.49×10^4	1.25×10^5	6.21×10^5	2.96×10^6	2.11×10^7	8.58×10^7
0.10	6.98×10^3	5.02×10^4	2.14×10^5	8.65×10^5	5.05×10^6	1.82×10^7
0.20	2.37×10^3	1.39×10^4	5.08×10^4	1.80×10^5	9.05×10^5	2.95×10^6
0.30	1.05×10^3	5.48×10^3	1.86×10^4	6.13×10^4	2.82×10^5	8.68×10^5
0.50	306	1.40×10^3	4.27×10^3	1.28×10^4	5.44×10^4	1.67×10^5
***L*/*r* = 3**	***r*/*t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.50×10^4	1.25×10^5	6.23×10^5	2.97×10^6	2.11×10^7	8.61×10^7
0.10	7.01×10^3	5.05×10^4	2.16×10^5	8.72×10^5	5.11×10^6	1.85×10^7
0.20	2.40×10^3	1.42×10^4	5.21×10^4	1.85×10^5	9.45×10^5	3.11×10^6
0.30	1.08×10^3	5.72×10^3	1.96×10^4	6.54×10^4	3.07×10^5	9.53×10^5
0.50	331	1.53×10^3	4.80×10^3	1.46×10^4	6.27×10^4	1.93×10^5
***L*/*r* = 5**	***r*/*t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.50×10^4	1.26×10^5	6.24×10^5	2.97×10^6	2.12×10^7	8.63×10^7
0.10	7.06×10^3	5.07×10^4	2.17×10^5	8.77×10^5	5.15×10^6	1.87×10^7
0.20	2.44×10^3	1.43×10^4	5.31×10^4	1.90×10^5	9.74×10^5	3.23×10^6
0.30	1.12×10^3	5.87×10^3	2.04×10^4	6.92×10^4	3.29×10^5	1.04×10^6
0.50	366	1.65×10^3	5.25×10^3	1.65×10^4	7.20×10^4	2.25×10^5

Figure 5 **Rotation parameter i_bEr^3/M_L for cylindrical shell with t_n/t = 1 subjected to a longitudinal moment**

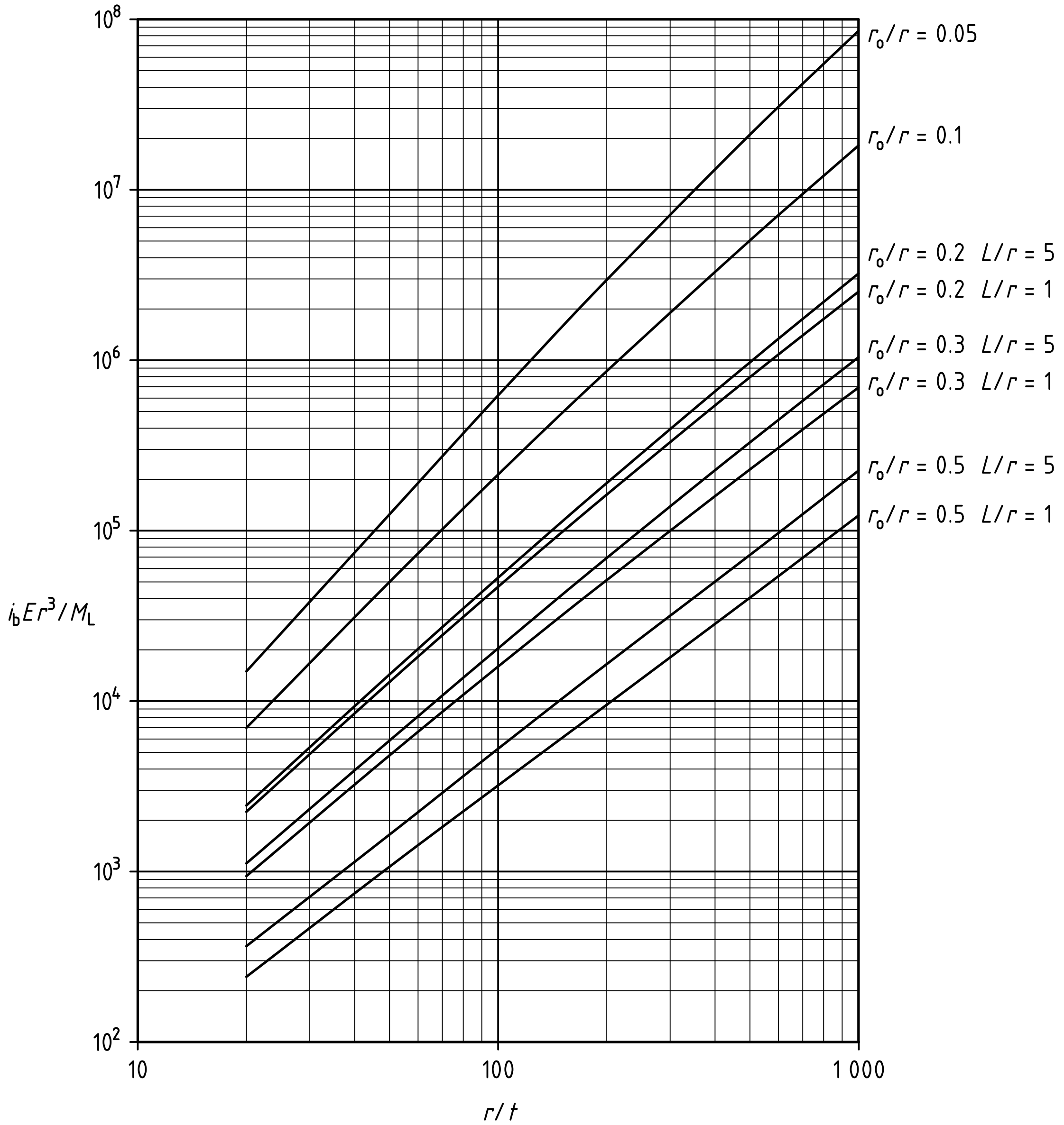


Table 6 **Rotation factor C_{rL} for cylindrical shell subjected to a longitudinal moment**

***r*/*t* = 20**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.625	1.619	1.617	1.616				
0.10	1.534	1.524	1.521	1.518	0.778	0.782	0.783	0.785
0.20	1.556	1.533	1.526	1.518	0.801	0.809	0.811	0.814
0.30	1.581	1.541	1.528	1.511	0.796	0.809	0.813	0.819
0.50	1.643	1.554	1.523	1.484	0.762	0.788	0.798	0.811
***r*/*t* = 100**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.434	1.430	1.429	1.429	0.862	0.863	0.863	0.864
0.10	1.479	1.470	1.467	1.465	0.861	0.864	0.865	0.866
0.20	1.511	1.481	1.471	1.463	0.839	0.848	0.851	0.853
0.30	1.543	1.496	1.476	1.461	0.813	0.828	0.835	0.840
0.50	1.657	1.552	1.518	1.490	0.753	0.792	0.804	0.815
***r*/*t* = 1000**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.373	1.369	1.369	1.367	0.901	0.902	0.901	0.902
0.10	1.392	1.378	1.373	1.369	0.875	0.880	0.882	0.883
0.20	1.471	1.430	1.414	1.405	0.830	0.845	0.851	0.856
0.30	1.574	1.526	1.498	1.473	0.788	0.808	0.818	0.829
0.50	1.902	1.762	1.705	1.681	0.691	0.737	0.752	0.765

Figure 6 **Rotation factor C_{rL} for cylindrical shell subjected to a longitudinal moment**

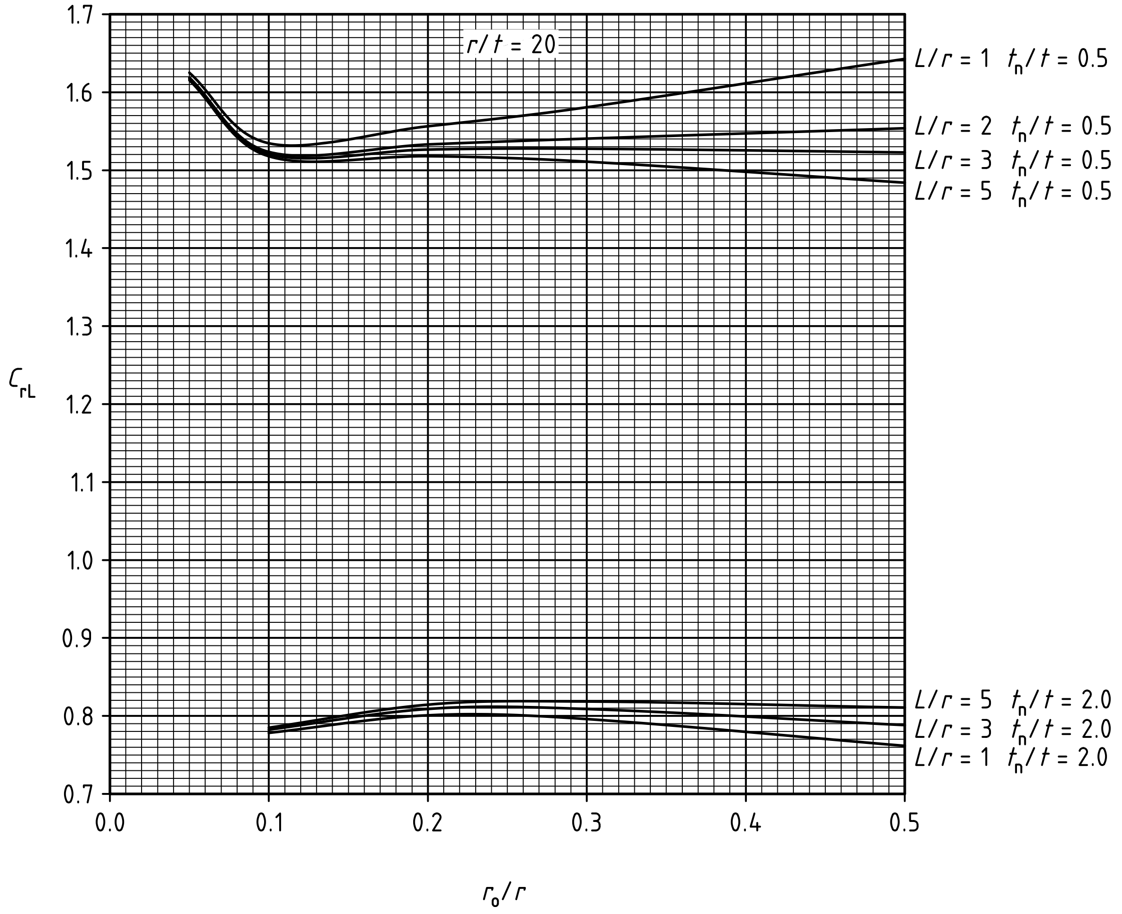


Figure 6 **Rotation factor C_{rL} for cylindrical shell subjected to a longitudinal moment *(continued)***

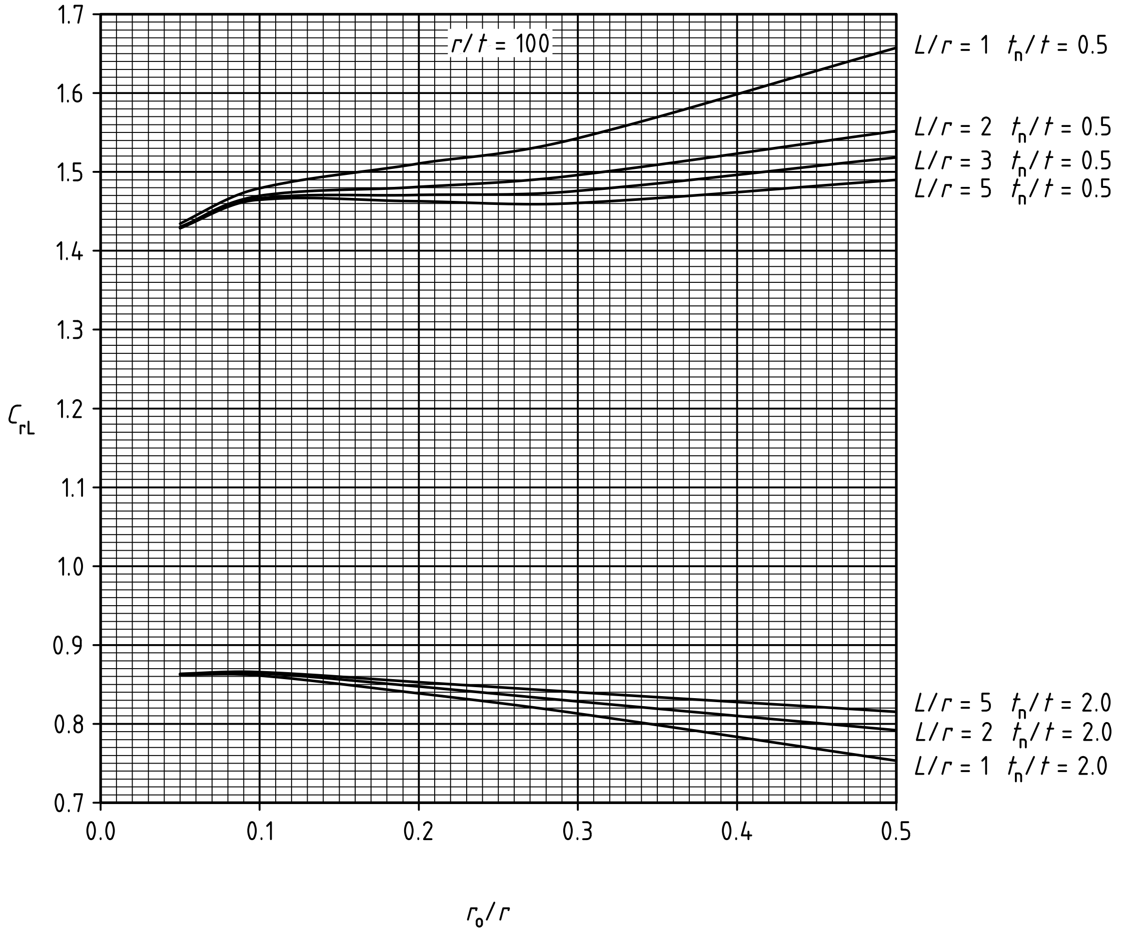


Figure 6 **Rotation factor C_{rL} for cylindrical shell subjected to a longitudinal moment *(continued)***

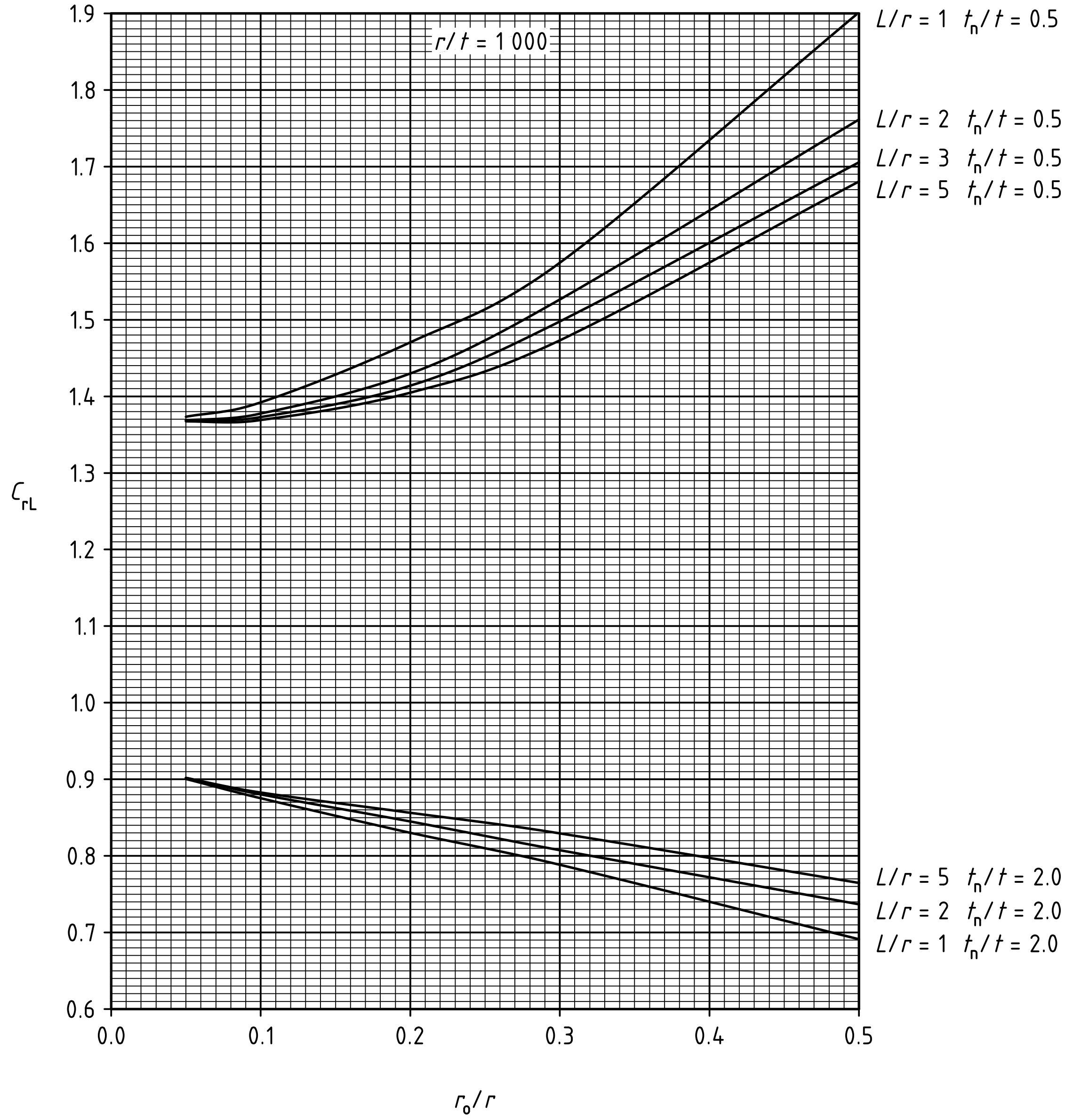


Table 7 **Rotation parameter $i_b E r^3 / M_C$ for cylindrical shell with t_n/t = 1 subjected to a circumferential moment**

***L/r* = 1**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.78×10^4	1.80×10^5	1.08×10^6	6.43×10^6	6.57×10^7	3.67×10^8
0.10	9.66×10^3	9.31×10^4	5.16×10^5	2.78×10^6	2.44×10^7	1.20×10^8
0.20	4.07×10^3	3.44×10^4	1.66×10^5	7.67×10^5	5.35×10^6	2.16×10^7
0.30	1.97×10^3	1.46×10^4	6.26×10^4	2.55×10^5	1.48×10^6	5.15×10^6
0.50	540	3.10×10^3	1.13×10^4	3.79×10^4	1.80×10^5	5.71×10^5
***L/r* = 2**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.89×10^4	1.92×10^5	1.17×10^6	7.06×10^6	7.36×10^7	4.19×10^8
0.10	1.08×10^4	1.05×10^5	6.00×10^5	3.36×10^6	3.13×10^7	1.63×10^8
0.20	5.07×10^3	4.36×10^4	2.27×10^5	1.14×10^6	8.89×10^6	3.99×10^7
0.30	2.79×10^3	2.12×10^4	1.01×10^5	4.62×10^5	3.07×10^6	1.19×10^7
0.50	973	5.98×10^3	2.30×10^4	9.03×10^4	4.69×10^5	1.52×10^6
***L/r* = 3**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.94×10^4	2.00×10^5	1.20×10^6	7.33×10^6	7.70×10^7	4.42×10^8
0.10	1.12×10^4	1.12×10^5	6.35×10^5	3.62×10^6	3.45×10^7	1.84×10^8
0.20	5.44×10^3	5.03×10^4	2.56×10^5	1.34×10^6	1.12×10^7	5.22×10^7
0.30	3.10×10^3	2.67×10^4	1.24×10^5	5.95×10^5	4.35×10^6	1.77×10^7
0.50	1.14×10^3	9.00×10^3	3.49×10^4	1.35×10^5	8.08×10^5	2.77×10^6
***L/r* = 5**	***r/t***					
r_o/r	**20**	**50**	**100**	**200**	**500**	**1000**
0.05	1.94×10^4	2.06×10^5	1.26×10^6	7.61×10^6	8.08×10^7	4.68×10^8
0.10	1.12×10^4	1.18×10^5	6.92×10^5	3.88×10^6	3.79×10^7	2.07×10^8
0.20	5.50×10^3	5.59×10^4	3.07×10^5	1.57×10^6	1.38×10^7	6.85×10^7
0.30	3.15×10^3	3.14×10^4	1.66×10^5	7.81×10^5	6.09×10^6	2.75×10^7
0.50	1.17×10^3	1.14×10^4	5.81×10^4	2.37×10^5	1.37×10^6	5.26×10^6

Figure 7 **Rotation parameter $i_b Er^3/M_C$ for cylindrical shell with t_n/t = 1 subjected to a circumferential moment**

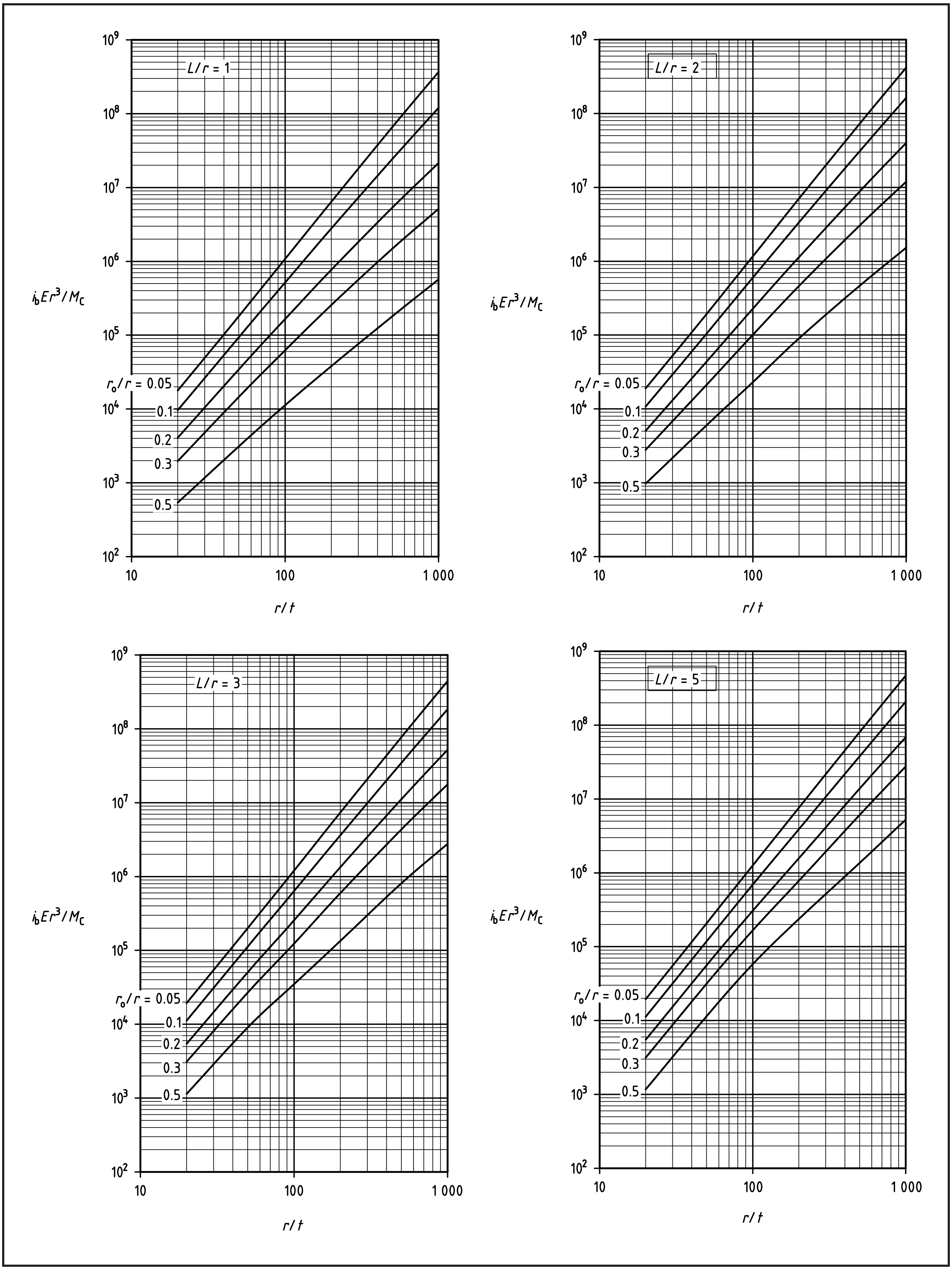


Figure 8 **Rotation factor C_{rC} for cylindrical shell subjected to a circumferential moment**

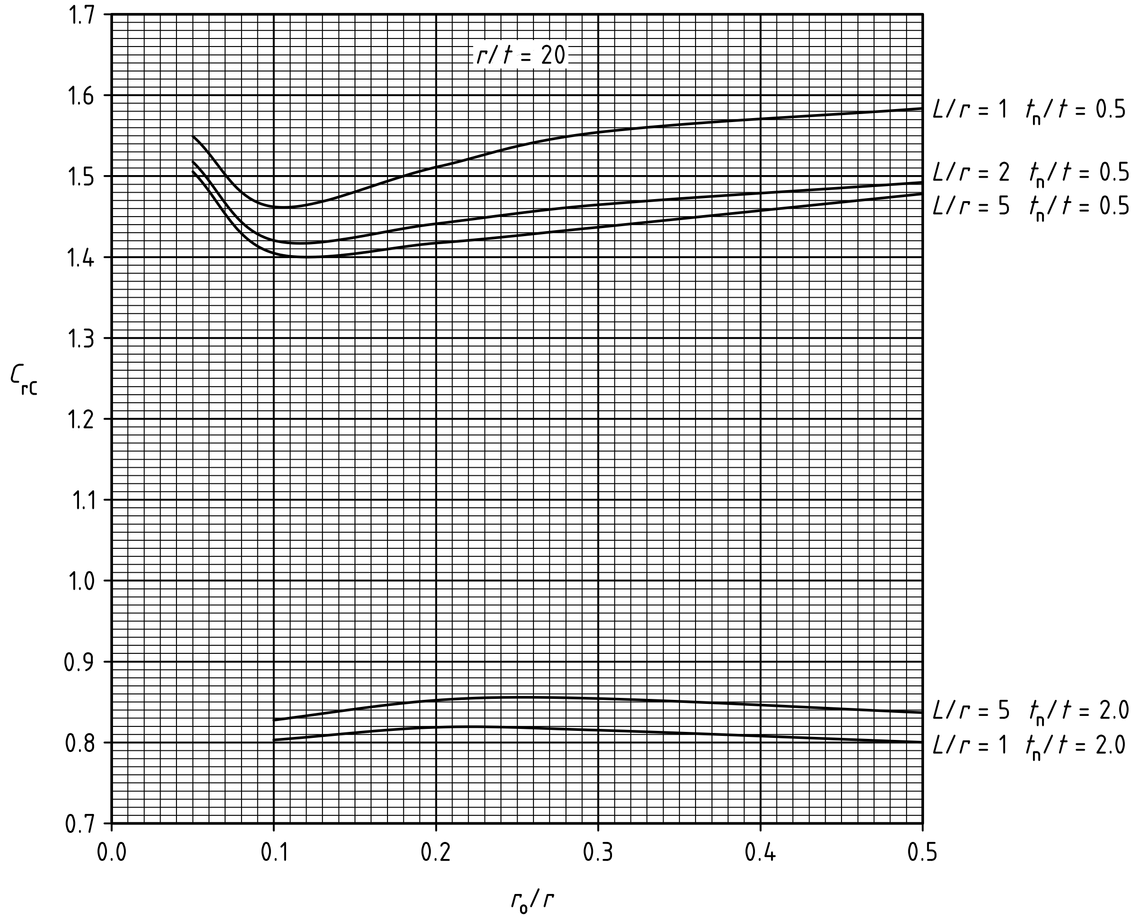
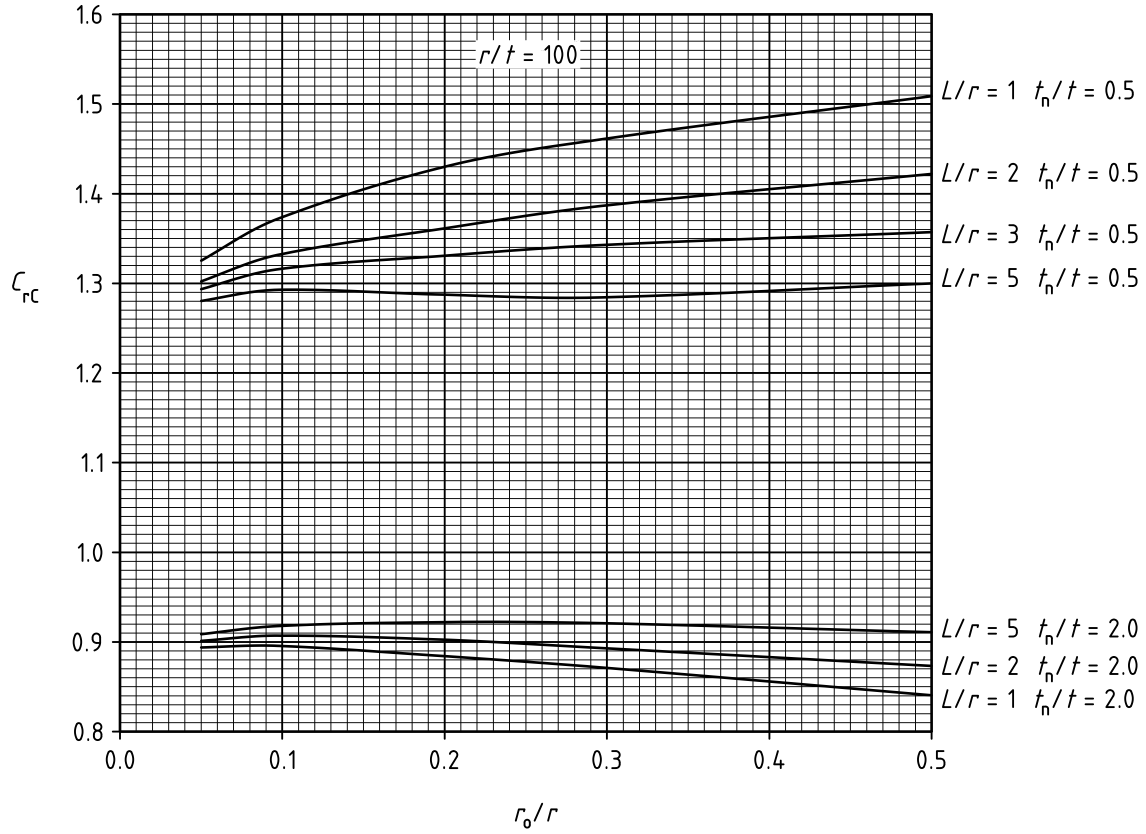


Figure 8 **Rotation factor C_{rC} for cylindrical shell subjected to a circumferential moment *(continued)***



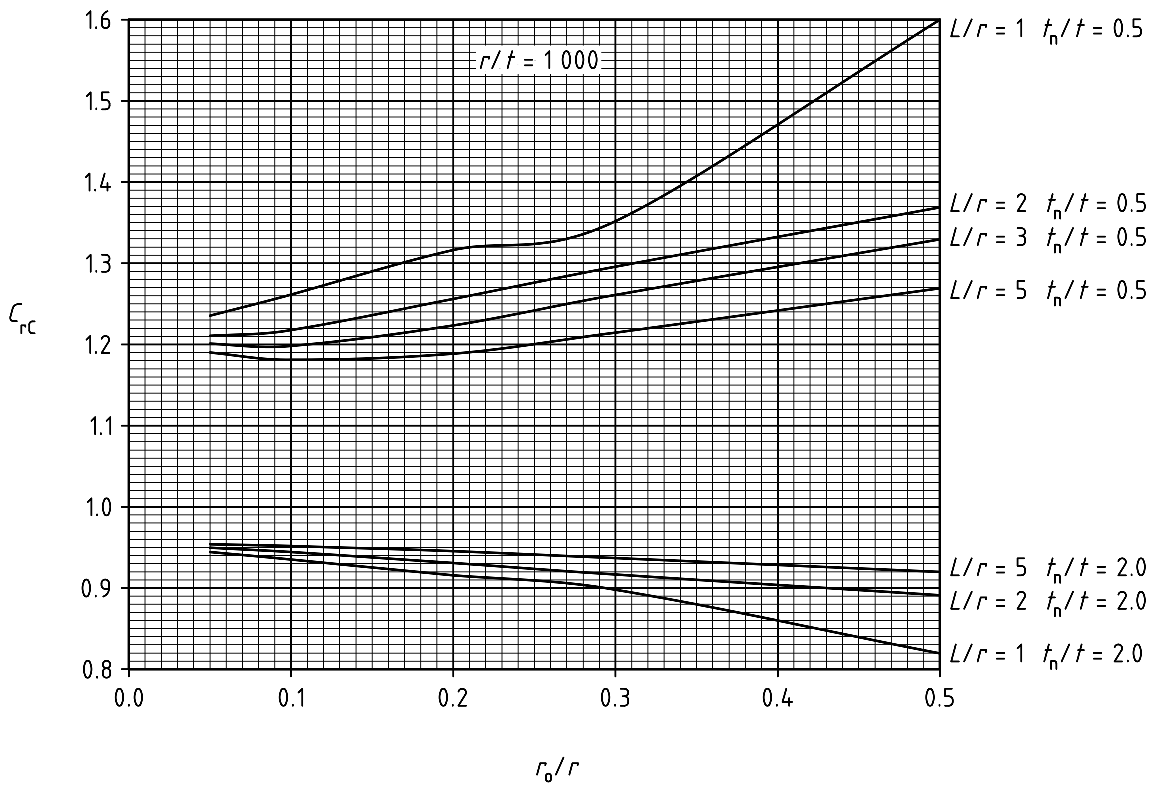


Table 8 **Rotation factor C_{rC} for cylindrical shell subjected to a circumferential moment**

***r/t* = 20**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.549	1.518	1.507	1.505				
0.10	1.462	1.420	1.408	1.405	0.803	0.821	0.826	0.828
0.20	1.511	1.441	1.420	1.417	0.819	0.844	0.851	0.852
0.30	1.554	1.465	1.441	1.437	0.815	0.846	0.853	0.854
0.50	1.584	1.492	1.480	1.478	0.800	0.835	0.837	0.837
***r/t* = 100**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.325	1.302	1.294	1.280	0.894	0.901	0.904	0.908
0.10	1.374	1.333	1.316	1.293	0.895	0.907	0.912	0.918
0.20	1.430	1.361	1.331	1.287	0.884	0.903	0.911	0.922
0.30	1.461	1.387	1.343	1.285	0.871	0.893	0.905	0.921
0.50	1.509	1.422	1.357	1.300	0.841	0.873	0.892	0.911
***r/t* = 1000**	**t_n/t = 0.5**				**t_n/t = 2.0**			
	L/r				**L/r**			
r_o/r	**1**	**2**	**3**	**5**	**1**	**2**	**3**	**5**
0.05	1.236	1.210	1.201	1.190	0.944	0.950	0.952	0.954
0.10	1.261	1.218	1.198	1.181	0.935	0.944	0.949	0.952
0.20	1.316	1.256	1.224	1.189	0.916	0.931	0.938	0.945
0.30	1.352	1.296	1.261	1.214	0.898	0.917	0.926	0.937
0.50	1.600	1.369	1.329	1.269	0.819	0.891	0.903	0.920

4 References

[1] SCHWARZ, M.M. *Flexibility analysis of the vessel-piping interface*. ICPVT-10, Vienna, Austria, 2003.

Enquiry Case 5500/138

Guidance on alternative rules for nozzle reinforcement

Enquiry

3.5.4.3 and **3.5.4.9** give alternative design methods for calculating the amount of compensation to be provided at an opening or nozzle connection. Can the committee give any guidance as to how the compensation required by the two methods compares?

Reply

The committee recognizes that the two alternative nozzle compensation rules in **3.5.4** have different bases and can lead to different results. PD 6550-2:1989 discusses how the rules for **3.5.4.3** are based on various stress analysis studies reported between 1961 and 1974, and on photo-elastic tests reported by Money in 1966.

The alternative method given in **3.5.4.9** is a more empirical method which is based on a simple limit load analysis. The load due to pressure acting over the area in the vicinity of the nozzle is balanced by the load bearing capacity of the geometry and the material. Although only recently included as a method in PD 5500, this method has been used for many years in continental European pressure vessel codes and has been adopted by BS EN 13445-3.

With many years of good experience with both these methods it can be concluded that a safe nozzle design can be provided whichever method is used.

A comprehensive comparison of the two methods has recently been carried out for flush and protruding nozzles in cylindrical shells and spherical shells. The calculation procedures used in the two methods are completely different, so the comparison has been based on the maximum permissible design pressures for specific geometries calculated using the two methods.

The analysis was carried out for a comprehensive range of values of nozzle to shell diameter ratio d/D from 0.05 to 1.0 for flush nozzles in cylindrical shells, 0.05 to 0.3 for protruding nozzles in cylindrical shells, and 0.05 to 0.5 for flush and protruding nozzles in spherical shells. The range of shell diameter to thickness ratio D/e_{as} covered was 10 to 5 000, though this was restricted in many cases by the limitations specified in PD 5500. For example, the method in **3.5.4.3** is only applicable to large flush nozzles ($d/D > 1/3$) in cylindrical shells when D/e_{ns} does not exceed 200.

Comparison of the Stress Based Method in 3.5.4.3 and the Pressure Area Method in 3.5.4.9

The notation used in this enquiry case is given in **3.5.4.1**, with the following additions:

p_1 is the maximum permissible pressure using the stress based method in **3.5.4.3**;

p_2 is the maximum permissible pressure using the pressure area method in **3.5.4.9**.

The analysis was carried out for the full range of values of e_{ab}/e_{as}, d/D and D/e_{as} where both **3.5.4.3** and **3.5.4.9** are applicable. It should be noted that some of these geometries are unlikely to occur in practice. Very thin shells (D/e_{as} greater than 1 000) are not generally used due to manufacturing problems and lack of structural stability, and it is unusual for the ratio e_{ab}/e_{as} to be less than about half the ratio d/D for cylindrical shells or less than $2d/D$ for spherical shells. The limitations on maximum nozzle wall thickness for reinforcement given in **3.5.4.3.3**i) and Figure 3.5-31 have been incorporated in the analysis.

In the analysis to **3.5.4.3** the value of the factor *C* has been taken as 1.1 (nozzle with negligible external loads). If factor *C* is taken as 1.0 (nozzle loads not negligible, or vessel operating in the creep range) then the values of the ratio of the permissible pressures p_1/p_2 are reduced by dividing by the factor 1.1.

Limit on value of parameter ρ

In **3.5.4.3.3** it states that extrapolation of Figures 3.5-9, 3.5-10 and 3.5-11 is not permitted, and that e_{rs} has to be increased where necessary in order to reduce ρ to an acceptable value. In the charts for flush nozzles in cylindrical shells with $d/D < 0.3$, flush nozzles in spherical shells, and protruding nozzles in cylindrical and spherical shells the maximum value of D/e_{as} is limited in many cases by the maximum value of ρ in the relevant figure in PD 5500.

In such cases the use of the pressure area method in **3.5.4.9** may give an acceptable design without having to increase the shell thickness e_{rs}.

Flush nozzles in cylindrical shells

For flush nozzles in cylindrical shells the stress based method (using $C = 1.1$) permits a higher pressure than that permitted by the pressure area method when e_{ab}/e_{as} is larger than about 0.4. This applies over the whole range of values of d/D. If $C = 1.0$ the stress based method permits a higher pressure when e_{ab}/e_{as} is larger than about 0.5 to 0.6.

For small nozzles ($d/D < 0.2$) the ratio p_1/p_2 increases as the ratio D/e_{as} is reduced, but for large nozzles ($d/D > 0.3$) the ratio p_1/p_2 increases as the ratio D/e_{as} is increased. This is due to the use of different design curves in the stress based method for $d/D < 0.3$ (Figure 3.5-11) and $d/D > 0.2$ (Figure 3.5-12).

The pressure area method permits a considerably higher pressure than that permitted by the stress based method for large diameter thin nozzles (i.e. small values of e_{ab}/e_{as} and $d/D \geq 0.3$). In many cases the ratio e_{ab}/e_{as} is below the value likely to be encountered in practice (i.e. $< 0.5d/D$). An analysis of the stress concentration factors on which the calculation method is based suggests that the stress based method may be conservative in this region, rather than the pressure area method being unconservative.

For large diameter nozzles in thin shells ($d/D \geq 0.3$ and $D/e_{as} > 100$) the stress based method permits a considerably higher pressure than that permitted by the pressure area method for values of e_{ab}/e_{as} greater than about 0.4.

Flush nozzles in spherical shells

For flush nozzles in spherical shells the stress based method permits a higher pressure than the pressure area method for thicker shells, and the pressure area method permits a higher pressure for thin shells, particularly for very thin or very thick nozzles ($e_{ab}/e_{as} < 0.5$ or > 1.0).

Protruding nozzles in cylindrical shells

For protruding nozzles in cylindrical shells the stress based method (using $C = 1.1$) permits a higher pressure than that permitted by the pressure area method when e_{ab}/e_{as} is larger than about 0.3. This applies over the whole range of values of d/D. If $C = 1.0$ the stress based method permits a higher pressure when e_{ab}/e_{as} is larger than about 0.4 to 0.6. The ratio p_1/p_2 increases as the ratio D/e_{as} is reduced. For small nozzles ($d/D \leq 0.1$) the ratio p_1/p_2 increases as e_{ab}/e_{as} is increased.

Protruding nozzles in spherical shells

For protruding nozzles in spherical shells the results are very similar to those for protruding nozzles in cylindrical shells, which is due to the fact that they are based on the same figure in **3.5.4.3**.

Figure 1 **Ratio p_1/p_2 for flush nozzle in cylinder with d/D = 0.1**

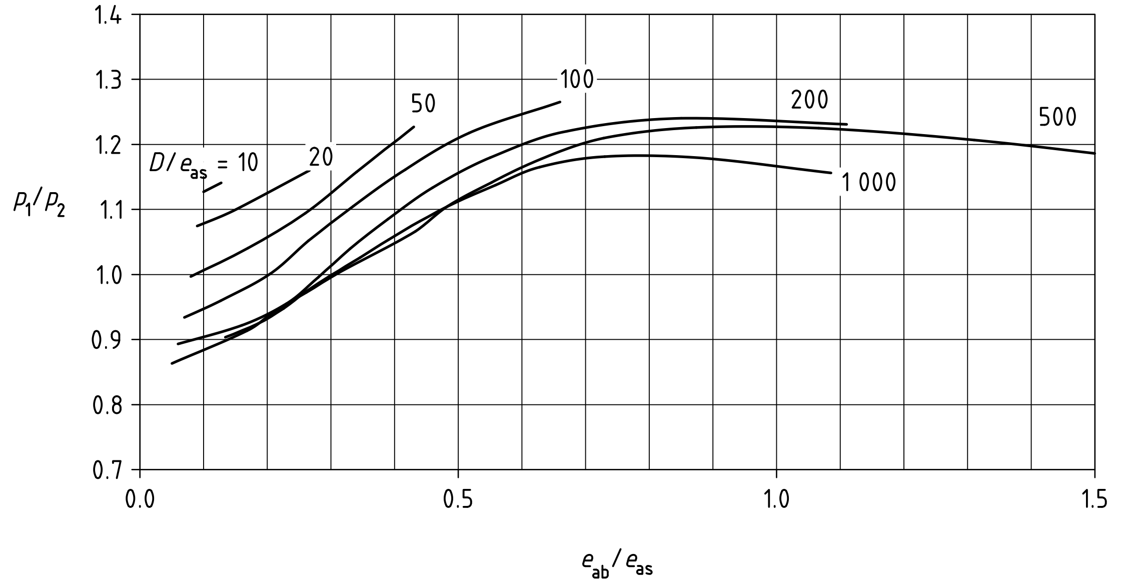
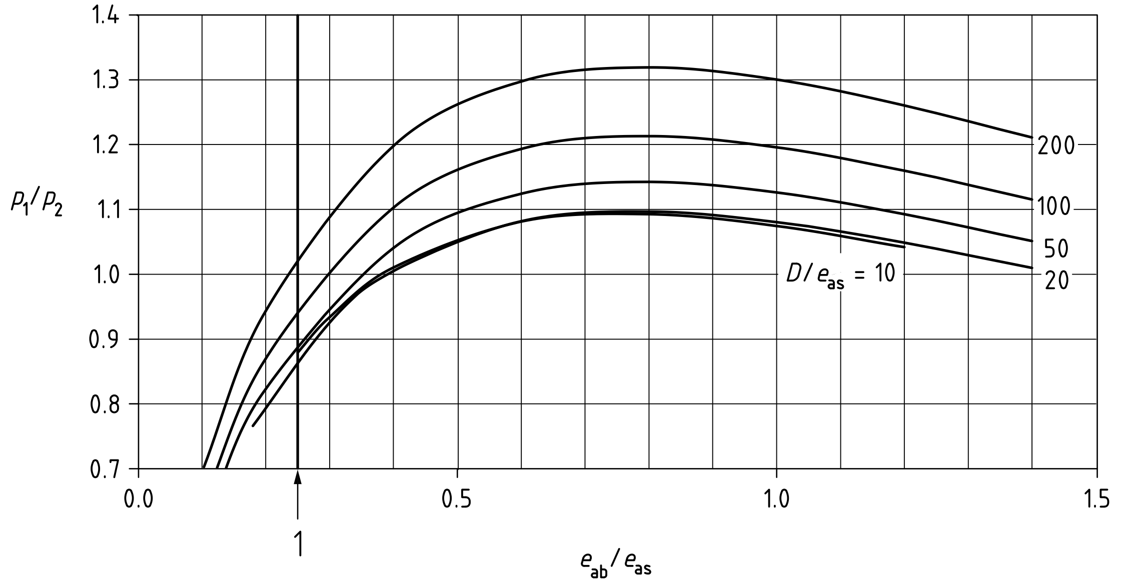


Figure 2 **Ratio p_1/p_2 for flush nozzle in cylinder with d/D = 0.5**



Key

1 $e_{ab}/e_{as} = 0.5d/D$

Figure 3 **Ratio p_1/p_2 for flush nozzle in sphere with d/D = 0.1**

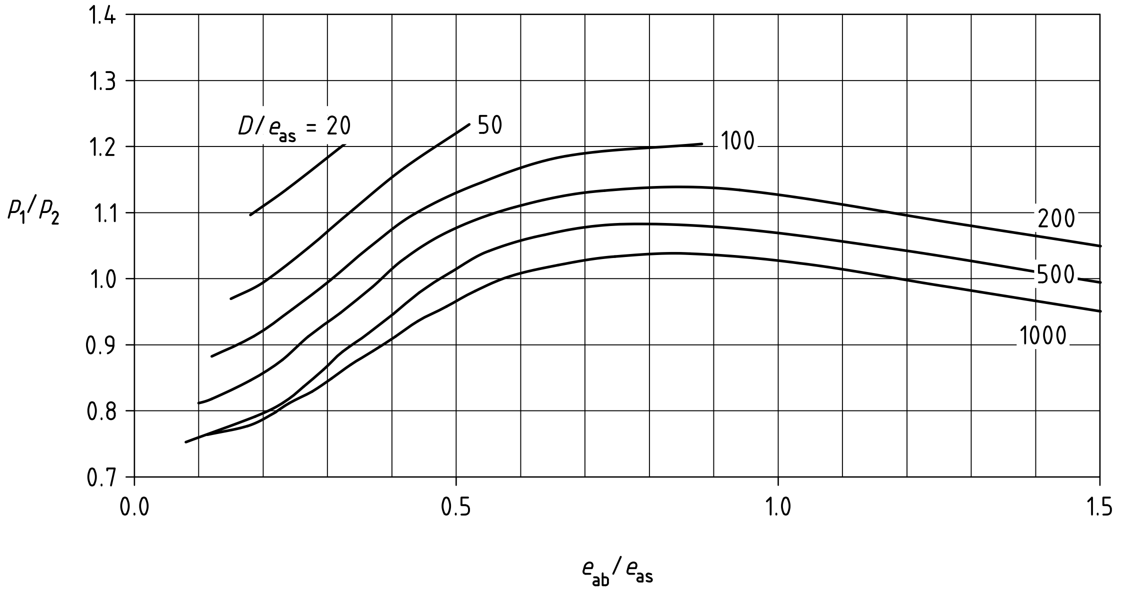


Figure 4 **Ratio p_1/p_2 for flush nozzle in sphere with d/D = 0.3**

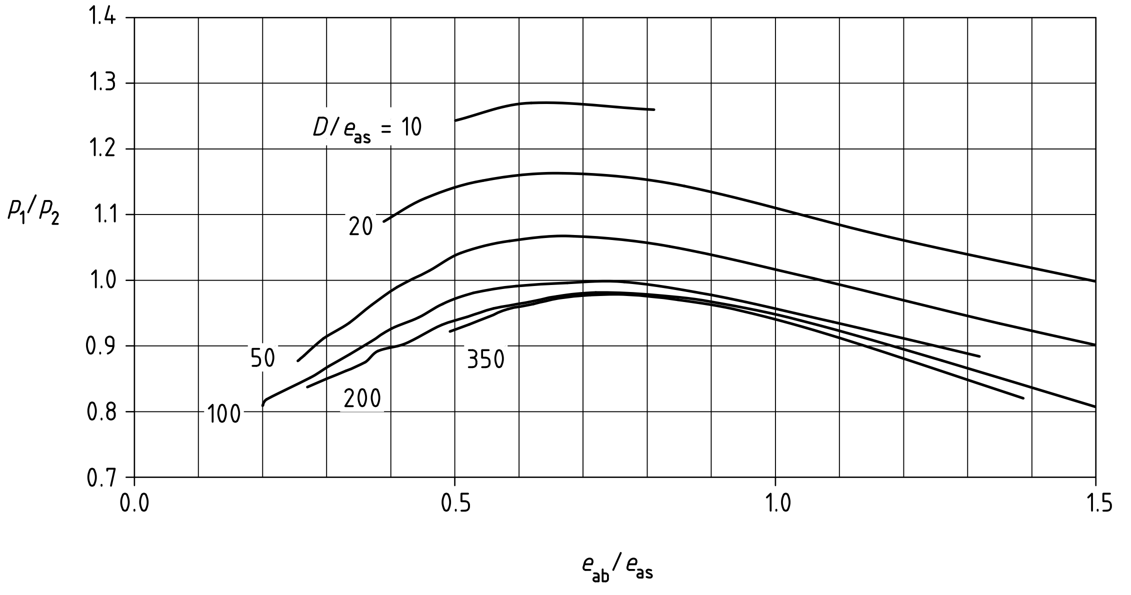


Figure 5 **Ratio p_1/p_2 for protruding nozzle in cylinder with d/D = 0.1**

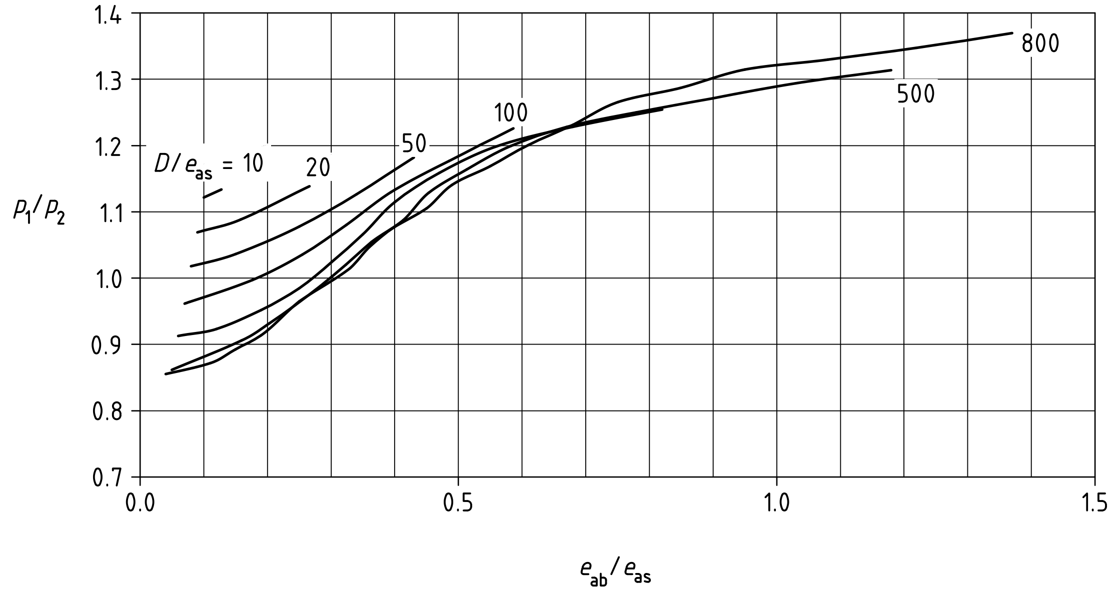


Figure 6 **Ratio p_1/p_2 for protruding nozzle in cylinder with d/D = 0.3**

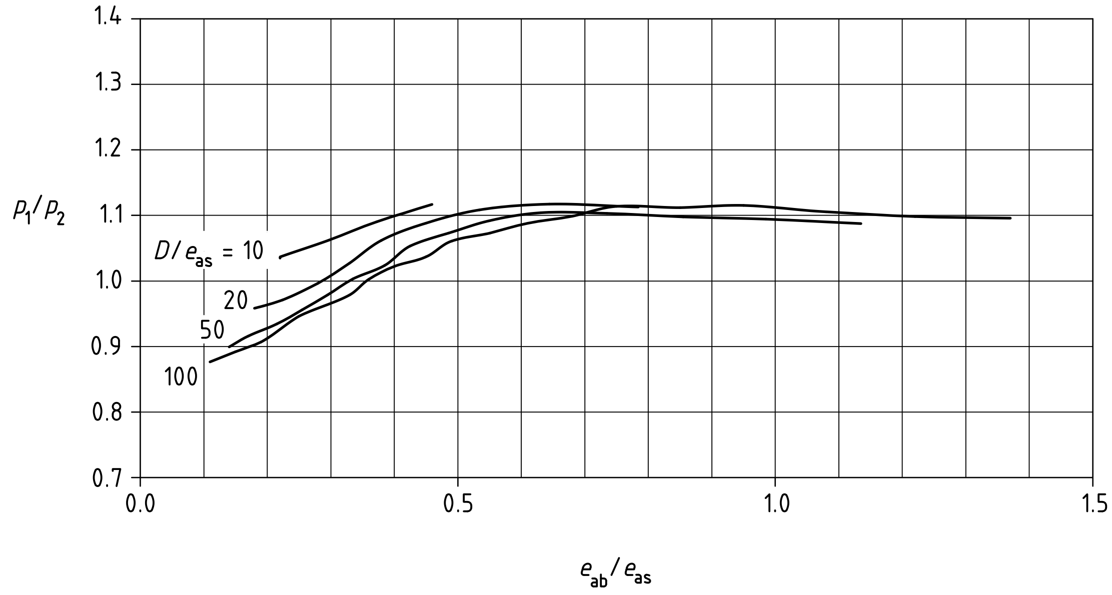


Figure 7 **Ratio p_1/p_2 for protruding nozzle in sphere with d/D = 0.1**

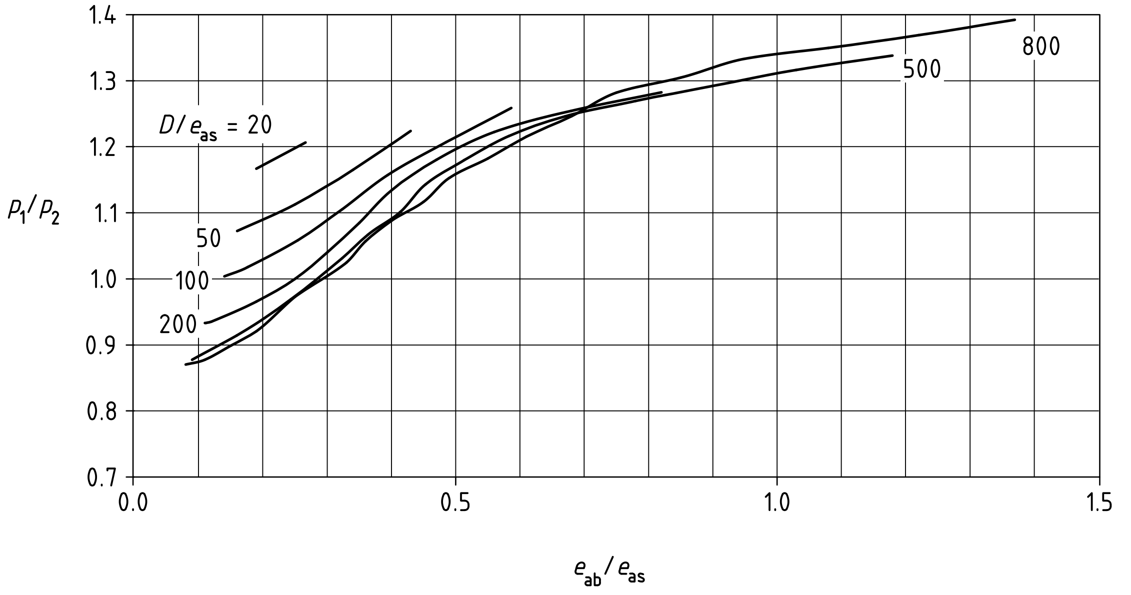
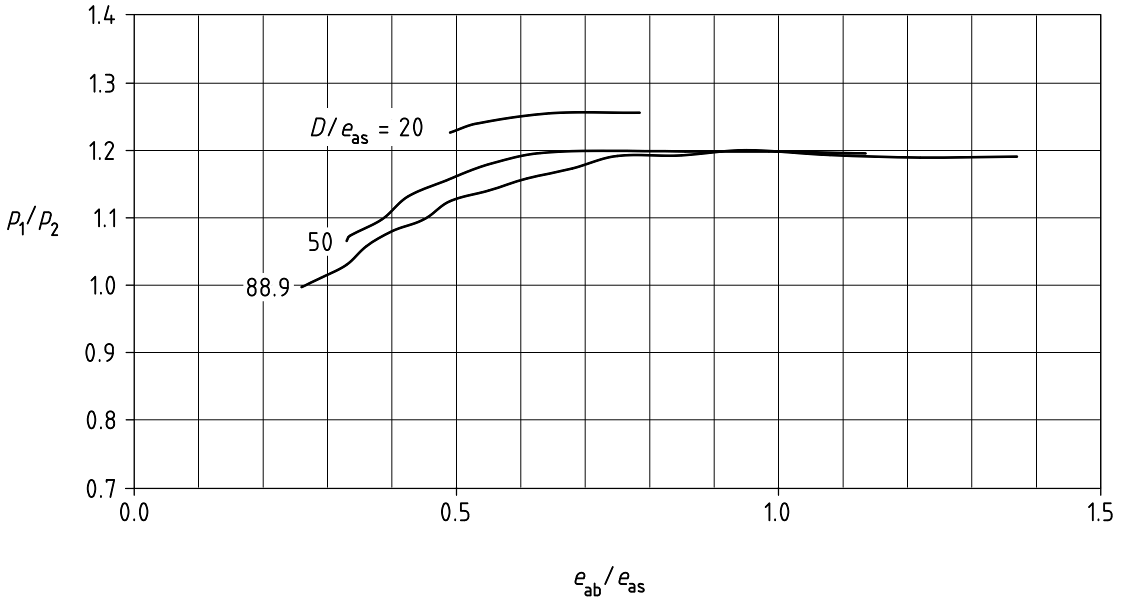


Figure 8 **Ratio p_1/p_2 for protruding nozzle in sphere with d/D = 0.3**



Enquiry Case 5500/139

ISO 16528-2:2007 Conformance Tables

Enquiry

Can the committee give any guidance on the conformance of PD 5500 to ISO 16528.

Reply

The following tables give details of the conformance of PD 5500:2024 to ISO 16528-1:2007, *Boiler and pressure vessels – Part 1: Performance requirements*.

Table 1 **Failure mode summary**

STANDARD[a]: PD 5500:2024.	
FAILURE MODE SUMMARY[b]	
Failure Modes according to ISO 16528 Part 1 clause 6.3	**Addressed (Y/N/P)[c]**
Brittle fracture	Y
Ductile failure	Y
Excessive deformation leading to leakage or other loss of function	P
Elastic or elastic-plastic instability (buckling)	Y
Additional Failure Modes according to ISO 16528 Part 1, 6.2	**Addressed (Y/N/P)[c]**
Creep rupture	Y
Creep: excessive deformations at mechanical joints or resulting in unacceptable transfer of load	P
Creep instability	P
Erosion, corrosion	Y
Environmentally assisted cracking, e.g. stress corrosion cracking, hydrogen induced cracking, etc.	P
Progressive plastic deformation	Y
Alternating plasticity	Y
Fatigue under elastic strains (medium and high cycle fatigue) or under elastic-plastic strains (low cycle fatigue)	Y
Environmentally assisted fatigue, e.g. stress corrosion cracking or hydrogen induced cracking	P

[a] Provide full title of the standard and revision or addenda level.

[b] Failure modes addressed by this form. (See ISO 16528 Part 1)

[c] Y — failure mode addressed by standard.
N — failure mode not addressed by standard.
P — failure mode recognized by standard but complete details not addressed.

Table 2 **Detailed failure mode checklists**

STANDARD: PD 5500:2024.
FAILURE MODE[a]**: Brittle fracture**
EXPLICIT DESIGN[d]
References[b]**:** Clause **3.1.7** and Annex D – Requirements for vessels designed to operate below 0 °C. Clause **Du.3.1.7** and Annex Du.D.
Comments[c]**:** Annex D is based on fracture mechanics principles and operating experience. Its application is limited to ferritic steels in material groups 1, 2 and 4.
IMPLICIT DESIGN[e]
References: Annex U – Guidance on the use of fracture mechanics analyses.
Comments: Annex U may be used as a basis for determining the suitability of particular vessels for their intended duty by carrying out a fracture mechanics analysis following the principles of BS 7910.
FABRICATION DETAILS[f]
References: Annex D, Table D.4 – Reference thickness of weld joint components. Annex D, clause **D.8** – Design, manufacture and workmanship.
Comments: Table D.4 gives reference thicknesses for use in the determination of impact testing requirements. Clause **D.8** specifies design and fabrication requirements for vessels subject to low temperature.
MATERIAL REQUIREMENTS[g]
References: Clause **2.1.2** – Materials for pressure parts, Clause **2.2** – Materials for low temperature applications, and Table 2.2-1 — Bolting materials for low temperature, Annex D, clauses **D.6** – Required impact parameters, and **D.8.3** – Heat treatment of components after forming. Aluminium, Copper, Nickel, Titanium and Duplex supplements, clauses **Al.2.2**, **Cu.2.2**, **Ni.2.2**, **Ti.2.2** and **Du.2.2**. Annex Du.D clauses **Du.D.2** – Scope and **Du.D.6** – Required impact parameters.
Comments: Clause **2.1.2** specifies general requirements for materials that ensure sufficient ductility and toughness to prevent brittle fracture at room temperature. Clause **2.2** gives specific requirements for materials for low temperature applications, and Table 2.2-1 specifies bolting materials for low temperature. Clause **D.6** specifies the required impact test temperature and impact energy values, and **D.8.3** gives requirements for normalizing heat treatment after cold forming. Clauses **Al.2.2**, **Cu.2.2**, **Ni.2.2** and **Ti.2.2** state that no special provisions are necessary for alumnium and alumninium alloys, copper and copper alloys, and nickel and nickel alloys at temperatures down to -196 °C, and titanium and titanium alloys at temperatures down to -100 °C. Annex Du.D specifies requirements for duplex and super duplex steels at temperatures below 0 °C.

Table 2 **Detailed failure mode checklists** *(continued)*

EXAMINATION REQUIREMENTS[h]
References: Annex D, clause **D.7** – Material impact testing requirements.
Comments: Clause **D.7.1** covers the testing of plates, forgings, castings and tubes; and clause **D.7.2** covers the testing of welds and heat affected zones.
TESTING REQUIREMENTS[i]
References: Clause **5.8.2.4**.
Comments: Clause **5.8.2.4** gives information regarding the test temperature for vessels which may be susceptible to brittle fracture.
USE/APPLICATION LIMITS[j]
References: Table 3.4-1 – Construction Categories. Annex D.
Comments: Category 3 construction (visual examination only) is not permitted for carbon steel and carbon manganese steel vessels at temperatures below 0°C. The use of Annex D is limited to ferritic steels in material groups 1, 2 and 4.

STANDARD: PD 5500:2024.
FAILURE MODE: Ductile failure
EXPLICIT DESIGN
References: Section 3 – Design. Clause **2.3** – Nominal design strength. Clause **3.4.2.2** – Category 3. Clause **3.4.2.3** – Additional limit for statically cast components. Annex A, clause **A.3.1.1** – Gross plastic deformation. Aluminium, Copper, Nickel, Titanium and Duplex supplements, sections **Al.3**, **Cu.3**, **Ni.3**, **Ti.3** and **Du.3** – Design. Clauses **Al.2.3**, **Cu.2.3**, **Ni.2.3**, **Ti.2.3** and **Du.2.3** and clauses **Al.3.4.2.2** and **Cu.3.4.2.2**.

Table 2 **Detailed failure mode checklists** *(continued)*

Comments:
With the exception of sub-section **3.6**, all the design methods in Section 3 are based on limiting the stresses in the vessel component to prevent ductile rupture. Clause **2.3** specifies the procedure for evaluating the nominal design strength for use in the design rules, based on the yield strength and tensile strength of the material. Clause **3.4.2.2** gives design stress limits for construction category 3 (visual examination only). Clause **3.4.2.3** contains requirements for the design stress for statically cast components. Annex A, clause **A.3.1.1** specifies the margin against gross plastic deformation to be used in stress analysis. Sections **Al.3**, **Cu.3**, **Ni.3**, **Ti.3** and **Du.3** give specific design requirements for aluminium, copper, nickel and titanium alloys, and duplex and super duplex steels respectively. Clauses **Al.2.3**, **Cu.2.3** and **Ni.2.3**, **Ti.2.3** and **Du.2.3**, and clauses **Al.3.4.2.2** and **Cu.3.4.2.2** specify requirements for nominal design strengths for aluminium, copper, nickel and titanium alloys, and duplex and super duplex steels.
IMPLICIT DESIGN
References:
Section 3 – Design. Clause **4.2.4.2.3** – Circularity (out-of-roundness and peaking).
Comments:
Section 3 limits stresses to prevent ductile rupture. Clause **4.2.4.2.3** limits out-of-roundness and peaking to avoid excessive local strains.
FABRICATION DETAILS
References:
Clauses **4.2.3** – Assembly tolerances and **4.2.4** – Tolerances for vessels subject to internal pressure.
Comments:
Clauses **4.2.3** and **4.2.4** specify maximum acceptable manufacturing tolerances to avoid local increases in stress due to misalignment, out-of-roundness, etc.
MATERIAL REQUIREMENTS
References:
Clause **2.1.2.2.2.3**.
Comments:
Clause **2.1.2.2.2.3** specifies requirements for mechanical properties and minimum percentage elongation at fracture.
EXAMINATION REQUIREMENTS
References:
Clause **3.4.1** – Construction categories. Clause **5.6** – Non-destructive testing.
Comments:
Clause **3.4.1** specifies requirements for the selection of construction category and the corresponding level of NDT. Clause **5.6** specifies requirements for non-destructive testing of parent material and welded joints.
TESTING REQUIREMENTS
References:
Clause **5.8** – Pressure testing.

Table 2 **Detailed failure mode checklists** *(continued)*

Comments:
Clause **5.8** specifies requirements for pressure testing, including limits on maximum membrane stress during the test.
USE/APPLICATION LIMITS
References:
Clause **1.1.4**, Section 3. Clause **3.2.2** – Design criteria.
Comments:
Clause **1.1.4** lists the types of vessel not covered by PD 5500. Clause 3.2.2 Note 1 states that the equations in section 3 are based on mean diameter rules and are not necessarily applicable when the ratio of the outside diameter of the vessel to the inside diameter of the vessel D_o/D_i exceeds 1.3. Limitations are specified for the application of the design rules in various parts of Section 3.

STANDARD: PD 5500:2024.
FAILURE MODE: Excessive deformation leading to leakage or other loss of function
EXPLICIT DESIGN
References:
Sub-section **3.8** – Bolted flanged connections. Aluminium, Copper and Titanium supplements, clauses **Al.3.8.1**, **Cu.3.8** and **Ti.3.8**.
Comments:
The requirements in sub-section **3.8** are based on the Taylor Forge flange design method, which was derived from considerations of strength. Gasket factors *y* and *m* are used to determine the required bolt loads for gasket seating during assembly and gasket sealing during operation. Clauses **Al.3.8.1**, **Cu.3.8** and **Ti.3.8** give specific requirements for aluminium, copper and titanium alloy flanges.
IMPLICIT DESIGN
References:
Clause **3.8.1** – General.
Comments:
Clause 3.8.1 Note 4 advises that special consideration may be needed in the design of flanges where a specific degree of leak-tightness is required.
FABRICATION DETAILS
References:
Clause **3.8.1.6** – Machining. Table 3.8-3 – Recommended surface finish on gasket contact faces for body flanges and flanges fitted with covers.
Comments:
Clause **3.8.1.6** specifies machining requirements for flanges. Table 3.8-3 gives the recommended surface finish for the gasket contact faces of flanges for different types of gasket.
MATERIAL REQUIREMENTS
References:
Table 3.8-1 – Recommended design stress values for flange bolting materials. Table 3.8-4 – Gasket materials and contact facings: gasket factors (*m*) for operating conditions and minimum design seating stress (*y*).
Comments:

Table 2 **Detailed failure mode checklists** *(continued)*

EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References: Clause **5.8.5.8** Clause **5.8.8** – Leak testing
Comments: Clause **5.8.5.8** specifies requirements where leakage of a gasketted component under test conditions is considered to be of concern. Clause **5.8.8** specifies requirements for leak testing.
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Elastic or elastic-plastic instability (buckling)
EXPLICIT DESIGN
References: Sub-section **3.6** – Vessels under external pressure. Clause **3.9.5** – Allowable shell and tube longitudinal stresses. Clause **3.11.2.3** – External pressure design for inner jacketted cylinder. Annex A clause **A.3.5** – Limits for longitudinal compressive general membrane stress in a vessel. Annex G, clauses **G.2.3.7** – Nozzle strength, **G.2.5.8** – Nozzle strength, **G.2.8.2.6** – Nozzle strength, and **G.2.8.3.6** – Nozzle strength. Annex M – Requirements for establishing the allowable external pressure for cylindrical sections outside the circularity limits specified in **3.6**. Aluminium, Copper, Nickel, Titanium and Duplex supplements, sub-sections **Al.3.6**, **Cu.3.6**, **Ni.3.6**, **Ti.3.6** and **Du.3.6**.
Comments: Sub-section **3.6** gives rules for the design of stiffened and unstiffened cylinders and cones, spherical components and dished ends. These rules are based on a combination of theoretical collapse pressures and results from a large number of collapse tests, and incorporate a safety factor of 1.5. Clause **3.9.5** gives a procedure for calculating the maximum allowable compressive stress in the tubes of a shell and tube heat exchanger. Clause **3.11.2.3** gives a procedure for the design of an inner jacketed cylinder. Annex A, clause **A.3.5** gives two alternative procedures for evaluating the maximum allowable longitudinal compressive stress in a cylindrical or conical section subject to an axial load or a bending moment. Annex G, clauses **G.2.3.7**, **G.2.5.8**, **G.2.8.2.6** and **G.2.8.3.6** specify requirements for the maximum applied bending moment and axial compressive force on a nozzle. Annex M – provides a procedure to determine the allowable external pressure for cylinders outside the circularity limits specified in **3.6**. Sub-sections **Al.3.6**, **Cu.3.6**, **Ni.3.6**, **Ti.3.6** and **Du.3.6** and give specific requirements for aluminium, copper, nickel and titanium alloys, and duplex and super duplex steels respectively.
IMPLICIT DESIGN
References: Annex A, clause **A.3.1.3** – Buckling. Annex A, clause **A.3.3.3** – Additional stress limits for attachments, supports, nozzles and openings.

Table 2 **Detailed failure mode checklists** *(continued)*

Comments: Clause **A.3.1.3** gives general requirements for components or loadings associated with substantial compressive stress. Clause **A.3.3.3** gives stress limits for local compressive stresses at nozzles and attachments.
FABRICATION DETAILS
References: Clause **3.6.1** – Vessels under external pressure – General. Clause **3.6.4** – Spherical shells. Clause **3.6.8** – Procedure by which the departure from the mean circle may be obtained.
Comments: Clause **3.6.1** a) gives circularity tolerance for cylindrical and conical shells subject to external pressure. Clause **3.6.4** gives circularity tolerance for spherical shells subject to external pressure. Clause **3.6.8** gives a procedure for evaluating the departure from the mean circle based on measurements of the fabricated cylinder.
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References: Clauses **5.8.2.7**, **5.8.5.5** and **5.8.5.6**.
Comments: Clause **5.8.2.7** gives pressure testing requirements for multi-compartment vessels designed for vacuum conditions. Clause **5.8.5.5** specifies requirements for pressure testing of single wall vessels subject to vacuum. Clause **5.8.5.6** gives requirements for testing of jacketed vessels subject to vacuum conditions.
USE/APPLICATION LIMITS
References: Clause **3.6.1** – Vessels under external pressure – General. Clause **3.6.4** – Spherical shells. Annex M.
Comments: The design rules in sub-section **3.6** are not applicable when the circularity tolerance exceeds the limit specified in clause **3.6.1** a) for cylindrical and conical shells, or clause **3.6.4** for spherical shells. Annex M provides a procedure to determine the allowable external pressure for cylinders outside the circularity limits specified in **3.6**.

STANDARD: PD 5500:2024.
FAILURE MODE: Creep rupture
EXPLICIT DESIGN
References: Clause **2.3.4** – Time-dependent design strength. Aluminium supplement, clause **Al.2.3.1.2**, Nickel supplement clause **Ni.2.3.1.2** and Titanium supplement, clause **Ti.2.3.3**.
Comments: Clause **2.3.4** gives the basis for evaluating the nominal design strength for use in the design rules, based on the stress required to produce rupture in the design lifetime. Clauses **Al.2.3.1.2**, **Ni.2.3.1.2** and **Ti.2.3.3** give the basis for the time-dependent design strengths for aluminium, nickel and titanium alloys.
IMPLICIT DESIGN
References: Clause **3.2.2** – Design criteria. Clause **3.5.4.3.1** – Design of isolated openings and nozzle connections – General. Clause **3.8.1** – Bolted flanged connections – General.
Comments: Clause 3.2.2 Note 2 gives recommendations for cases where the design strength is time dependant and internal pressure is not the dominant loading. Clause **3.5.4.3.1** gives requirements for nozzle reinforcement for vessels operating in the creep range. Clause 3.8.1 Note 4 advises that special consideration may be needed in the design of flanges operating at high temperatures.
FABRICATION DETAILS
References:
Comments:
MATERIAL REQUIREMENTS
References: Clause **2.1.2.2.2.5**. Table 2.3-1 – Temperature above which time dependent properties shall be considered. Clause **3.10.4** – Welded joints in time dependent applications. Annex K, clause **K.1.3** and Tables K.1-2 to K.1-12.
Comments: Clause **2.1.2.2.2.5** gives requirements for stress rupture properties for materials to be used in the creep range. Table 2.3-1 specifies the temperatures above which time dependent properties shall be considered for various material types. Clause **3.10.4** covers long term ductility of weld material and heat affected zones where the design strength is time dependent. Clause **K.1.3** gives requirements for the extension of lifetimes for vessels operating in the creep range. Tables K.1-2 to K.1-12 give values of design stress for BS materials at temperatures in the creep range.
EXAMINATION REQUIREMENTS
References: Clause **3.2.4** – Maximum design temperature. Clause **5.7.2.1** – Category 1 and 2 constructions.

Table 2 **Detailed failure mode checklists** *(continued)*

Comments:
Clause 3.2.4 Note gives recommendations for in-service inspection of vessels operating in the creep range. Clause 5.7.2.1 Note recommends that acceptance criteria for weld defects in vessels to be operated in the creep range may require special consideration.
TESTING REQUIREMENTS
References:
Clause **5.8.5.1**.
Comments:
Clause **5.8.5.1** gives requirements for the test pressure for vessels to be operated in the creep range.
USE/APPLICATION LIMITS
References:
Copper supplement, clause **Cu.2.3.1.5**.
Comments:
Clause **Cu.2.3.1.5** states that time-dependent design stresses are not currently available for copper and copper alloys for use in pressure vessel calculations.

STANDARD: PD 5500:2024.
FAILURE MODE: Creep: excessive deformations at mechanical joints or resulting in unacceptable transfer of load
EXPLICIT DESIGN
References:
Comments:
IMPLICIT DESIGN
References: Clause **3.8.1** – Bolted flanged connections – General.
Comments: Clause 3.8.1 Note 4 advises that special consideration may be needed in the design of flanges operating at high temperatures due to creep of the bolts and flanges.
FABRICATION DETAILS
References:
Comments:
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Creep instability
EXPLICIT DESIGN
References:
Comments:
IMPLICIT DESIGN
References: Clause **3.2.2** – Design criteria
Comments: Clause 3.2.2 Note 2 gives recommendations for cases where there is a possibility of creep deformation leading to instability.
FABRICATION DETAILS
References:
Comments:
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Erosion, corrosion
EXPLICIT DESIGN
References: Clause **3.3.2** – Additional thickness to allow for corrosion.
Comments: Clause **3.3.2** specifies that additional thickness is required to cover the total amount of corrosion on both surfaces.
IMPLICIT DESIGN
References: Clause **3.3.1** – Corrosion, erosion and protection – General. Clause **3.3.4** – Wear plates. Annex C, clause **C.1.2.2** – Corrosion.
Comments: Clause **3.3.1** gives recommendation regarding the effects of corrosion. Clause **3.3.4** gives advice regarding wear plates for conditions of erosion and abrasion. Clause **C.1.2.2** gives recommendations for fatigue assessment in corrosive service.

Table 2 **Detailed failure mode checklists** *(continued)*

FABRICATION DETAILS
References: Clauses **3.5.4.5.3** and **3.6.2.2** – Stiffening rings for cylindrical shells, general requirements. Clause **3.7.2.4.1**. Clause **3.12** – Manholes, inspection openings and quick release openings. Clause **4.3.5.4** – Limpet coils. Clause **4.3.6.2** – Backing strips. Annex E, clause **E.2.3** – Selection of detail. Annex E, clause **E.2.5.10**.
Comments: These clauses give requirements and recommendations regarding various fabrication details for vessels subject to erosion or corrosion.
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References: Clause **5.8.5** – Standard test pressure.
Comments: Clause **5.8.5.1** gives the formula for the calculation of the test pressure, including the effect of the corrosion allowance.
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Environmentally assisted cracking, e.g. stress corrosion cracking, hydrogen induced cracking, etc.
EXPLICIT DESIGN
References:
Comments:
IMPLICIT DESIGN
References: Clause **3.4.2.1** – Categories 1 and 2.
Comments: Clause **3.4.2.1** covers the use of lower design stresses to reduce the risk of stress corrosion cracking.
FABRICATION DETAILS
References: Clause **4.3.5.4** – Limpet coils.
Comments: Clause **4.3.5.4** gives requirements for limpet coil welds where there is a risk of stress corrosion cracking.

Table 2 **Detailed failure mode checklists** *(continued)*

MATERIAL REQUIREMENTS
References: Clause **3.3.1** – Corrosion, erosion and protection – General. Clauses **4.5.3.1** and **4.5.5.1**, and Table 4.5-1. Clause **4.5.1.1**.
Comments: Clause **3.3.1** gives requirements for the purchaser to specify where the fluid could give rise to stress corrosion cracking. Clauses **4.5.3.1** c), 4.5.5.1 Note, and Table 4.5-1 Note c cover requirements for post-weld heat treatment where there is a risk of stress corrosion cracking and/or hydrogen cracking. Clause 4.5.1.1 Note 1 gives guidance on preheat requirements where there is a risk of hydrogen cracking.
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Progressive plastic deformation
EXPLICIT DESIGN
References: Clause **3.5.4.3** – Design of isolated openings and nozzles. Annex G, clause **G.2.6** – Spherical shells: shakedown loads for radial nozzles.
Comments: The procedure specified in clause **3.5.4.3** for isolated openings and nozzles in vessels is based on considerations of shakedown under pressure loading as described in PD 6550-2:1989. Clause **G.2.6** contains a procedure for performing a shakedown analysis for nozzles in spherical shells subject to pressure and mechanical loading.
IMPLICIT DESIGN
References: Annex A, clause **A.3.1.2** – Incremental collapse. Annex A, clause **A.3.4.2.5** – Thermal stress.
Comments: Clause **A.3.1.2** gives recommendations for shakedown analysis. Clause **A.3.4.2.5** gives guidance on general thermal stresses which may produce incremental distortion.
FABRICATION DETAILS
References:
Comments:

Table 2 **Detailed failure mode checklists** *(continued)*

MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References: Clause **3.5.4.2** – Application. Annex G, clause **G.2.6.1.1** – Introduction.
Comments: Clause **3.5.4.2** specifies limits for the application of the requirements in **3.5.4**. Clause **G.2.6.1.1** gives limits on the applicability of the calculation procedure in **G.2.6**.

STANDARD: PD 5500:2024. **FAILURE MODE: Alternating plasticity**
EXPLICIT DESIGN
References:
Comments:
IMPLICIT DESIGN
References: Annex C, clause **C.3.3.5** – Elastic-plastic conditions
Comments: Clause **C.3.3.5** specifies requirements for applying a plasticity correction factor in a fatigue assessment if the calculated pseudo-elastic stress range exceeds twice the yield strength of the material.
FABRICATION DETAILS
References:
Comments:
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References:
Comments:

STANDARD: PD 5500:2024.
FAILURE MODE: Fatigue under elastic strains (medium and high cycle fatigue) or under elastic-plastic strains (low cycle fatigue)
EXPLICIT DESIGN
References: Annex C – Assessment of vessels subject to fatigue. Aluminium, Nickel, Titanium and Duplex supplements, sections **Al.C**, **Ni.C**, **Ti.C** and **Du.C**.
Comments: Annex C covers the assessment of vessels subject to fatigue and contains requirements to ensure that the vessel is designed to have a fatigue life which is at least as high as the required service life. The assessment is based on a number of S-N curves, each of which refers to a class of weld details. A detailed fatigue analysis of a vessel, or a component or bolting, need not be carried out if the criteria given in clause **C.2** are satisfied. Sections **Al.C**, **Ni.C**, **Ti.C** and **Du.C** give specific requirements for fatigue assessment of vessels which are fabricated from aluminium, nickel and titanium alloys, and duplex and super duplex steels.
IMPLICIT DESIGN
References: Annex G, clause **G.3.3.2.8** – Maximum stress for use in fatigue assessment Annex G, sub-section **G.4** – Simplified method for assessing transient thermal stress at a pressure vessel nozzle. Annex R – Guidance on additional information for flat ends and flat plates.
Comments: Clause **G.3.3.2.8** gives a procedure for the determination of a maximum stress at the horn of a saddle support for use in a fatigue assessment. Sub-section **G.4** contains a procedure for evaluating transient thermal stresses at a nozzle for use in a fatigue assessment. Annex R gives a method for calculating the maximum stress in a cylinder attached to a flat end; this stress may be used in a fatigue assessment.
FABRICATION DETAILS
References: Clause **4.3.6.2** – Backing strips. Annex C, clause **C.3.4.6.4** – Deviations from design shape. Annex E, clauses **E.2.5.12** and **E.2.7.2.3**.
Comments: Clause **4.3.6.2** prohibits the use of permanent backing strips for butt welded joints when the vessel is subject to fatigue. Clause **C.3.4.6.4** gives a procedure for determining the effect in a fatigue assessment of the additional bending stress due to departure from the design shape. Clause **E.2.5.12** is a note applicable to Figures E.9 to E.44 and indicates when a weld is not recommended for fatigue service. Clause **E.2.7.2.3** is a note applicable to Figures E.9 to E.23 and gives recommendations for the radiusing of the internal edges of the bores of nozzles which are in fatigue service.
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References: Annex C, Table C.2 – Classification of weld details
Comments: Table C.2 indicates where welds shall be proved free from significant defects by non-destructive testing.

Table 2 **Detailed failure mode checklists** *(continued)*

TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References: Annex C, clause **C.1.2.3** – Temperature.
Comments: Clause **C.1.2.3** limits the use of the fatigue curves to temperatures up to 350 °C for ferritic steels in material groups 1 to 6, 9 and 11, 430 °C for austenitic steels in material group 8, 100 °C for aluminium alloys in material groups 21 to 23, 300 °C for duplex steels in material sub-group 10.1, 250 °C for super duplex steels in material sub-group 10.2, 450 °C for nickel alloys in material groups 41 to 48 and 150 °C for titanium alloys in material groups 51 to 54.

STANDARD: PD 5500:2024.
FAILURE MODE: Environmentally assisted fatigue, e.g. stress corrosion cracking or hydrogen induced cracking
EXPLICIT DESIGN
References:
Comments:
IMPLICIT DESIGN
References: Annex C, clause **C.1.2.2** – Corrosion. Annex C, clause **C.3.4.2** – Assessment of weld defects.
Comments: Clause **C.1.2.2** gives guidance on corrosion fatigue. Clause **C.3.4.2** refers to BS 7910 which addresses failure due to corrosion, erosion and environmentally assisted cracking.
FABRICATION DETAILS
References:
Comments:
MATERIAL REQUIREMENTS
References:
Comments:
EXAMINATION REQUIREMENTS
References:
Comments:
TESTING REQUIREMENTS
References:
Comments:
USE/APPLICATION LIMITS
References: Aluminium, Nickel, Titanium and Duplex supplements, clauses **Al.C.3.2.4**, **Ni.C.3.2.4**, **Ti.C.3.2.4** and **Du.C.3.2.4**
Comments: Clauses **Al.C.3.2.4**, **Ni.C.3.2.4**, **Ti.C.3.2.4** and **Du.C.3.2.4** state that expert advice should be sought on the effects of corrosive environments on fatigue strength for aluminium, nickel and titanium alloys and duplex and super duplex steels.

[a] Failure mode addressed by this form. (See Table 1.)

[b] Provide specific clause or paragraph references (including the title, if any) indicating where relevant rules can be found. These references need not be exhaustive, but should be detailed enough to establish that the standard adequately addresses the selected failure mode.

[c] Provide explanatory comments indicating the background for the approach employed or other material that might be useful. For example, brief description of failure theory(ies) used should be provided. References to academic papers and empirical testing methods used to establish rules are encouraged.

[d] Reference(s) to rules or requirements that directly affect how the standard addresses the selected failure mode, e.g. formulas for sizing wall thickness of components for resisting ductile burst.

[e] This section may be used to provide references and comments when design tables, empirically based rules or other approaches are employed whose derivation is not obvious. It may also be used to provide general information on design margins (safety factors) on material properties, etc. Many successful standards do not provide explicit design rules for certain failure modes yet do employ combinations of material control, temperature limits or other means to provide adequate protection against failure. This section may be used to provide information on how his standard indirectly addresses certain failure modes when explicit rules are not provided.

[f] References for fabrication details relevant to the selected failure mode, e.g. control of cylinder ovality, weld profiling, control of tolerances, etc. For example, control of cylinder ovality is important for prevention of buckling of externally loaded vessels. This section should be used to describe such fabrication controls relevant for the designated failure mode.

[g] Relevant requirements for base and welding materials, e.g. control of YS/UTS ratios, provisions for addressing strain hardening, applications of heat treatment, etc. Assuring that fabrication processes have not adversely affected material properties beyond acceptable limits can be important for preventing certain types of failures. This section should be used to describe such controls.

[h] References for NDT or visual inspection relevant to the selected failure mode. (If NDT is correlated to design factors, this should be noted.)

[i] Provisions for final testing, i.e. hydrostatic or leak tests should be noted with specific information on normal test pressures and control of test lower and upper test pressures.

[j] An explanation shall be provided defining the limitations in the standard's scope or application relative to Part 1 Clause **6.2** failure mode(s).

Table 3 **Detailed technical requirements checklist**

ISO 16528-1 subclause	Description	PD 5500 clause(s)	Description	Comments
7.2.1	Materials – General	**2.1**	Selection of materials	
7.2.2	Specification of materials	**2.1.1 Al.2.1.1 Cu.2.1.1 Ni.2.1.1 Ti.2.1.1**	Selection of materials – General	
		2.1.2 Al.2.1.2 Cu.2.1.2 Ni.2.1.2 Ti.2.1.2 Du.2.1.2	Materials for pressure parts	
		2.2 Al.2.2 Cu.2.2 Ni.2.2 Ti.2.2 Du.2.2	Materials for low temperature applications	
		3.3.1 Ti.3.3	Corrosion, erosion and protection – General	
		Annex D Annex Du.D	Requirements for vessels designed to operate below 0 °C	Specifies required impact properties
7.2.3	Material certification	**4.1.2**	Material identification	
7.3.1	Design – loadings and other design considerations	**3.2.1**	Consideration of loads	
		3.2.3	Design pressure	
		3.2.4	Maximum design temperature	
		3.2.5	Minimum design temperature	
		3.2.6	Thermal loads	
		3.2.7	Wind and earthquake loads	
		Annex **B.5**	Calculation of wind loading	
		Annex **B.6**	Calculation of earthquake loading	
7.3.2	Design methods	**3.2.2**	Design criteria	Addresses design by rule, design by analysis and design by testing
		Section 3	Design	Contains methods for design by rule
		Annex A	Requirements for design not covered by Section 3	Gives design criteria for design by analysis
		5.8.6	Proof hydraulic test	Specifies method for design by testing
7.3.3	Design margins	**2.3 Al.2.3.1.3 Cu.2.3.1.3 Ni.2.3.1.2 Ti.2.3.2 Ti.2.3.3**	Nominal design strength	Refers to factors on material properties

Table 3 **Detailed technical requirements checklist** *(continued)*

ISO 16528-1 subclause	Description	PD 5500 clause(s)	Description	Comments
7.3.4	Design factors	**3.4 Al.3.4 Cu.3.4 Ni.3.4 Ti.3.4 Du.3.4**	Construction categories and design stresses	Specifies requirements for construction categories 1, 2 and 3. See Annex **Z.3** regarding weld joint efficiency
7.3.5	Means for examination	**3.12** **5.1 Al.5.1**	Manholes, inspection openings and quick release openings Inspection and testing — General	Clause **3.12** refers to BS 470 – Inspection, access and entry openings for pressure vessels
7.3.6	Draining and venting	**5.8.2.3 5.8.3.3**		Clause **5.8.3.3** refers to venting before hydraulic testing and **5.8.2.3** refers to venting and draining after testing
7.3.7	Corrosion and erosion	**3.3**	Corrosion, erosion and protection	
7.3.8.1	Overpressure protection – general	**3.13**	Protective devices for excessive pressure or vacuum	
7.3.8.2	Types of devices	**3.13**	Protective devices for excessive pressure or vacuum	
7.3.8.3	Safety accessories	**3.13**	Protective devices for excessive pressure or vacuum	Refers to BS EN ISO 4126 and BS 1123 and for specific requirements for safety accessories
7.4.1	Manufacture – methods	Section 4 **Al.4 Cu.4 Ni.4 Ti.4 Du.4**	Manufacture and workmanship	ISO 16528-1 clause **7.4.1** is a general statement which is addressed in various clauses in PD 5500 Section 4
7.4.2	Identification of materials	**4.1.2**	Material identification	
7.4.3	Preparation of parts	**4.2 Al.4.2 Cu.4.2 Ni.4.2 Ti.4.2 Du.4.2**	Cutting, forming and tolerances	
7.4.4	Welding	**4.3 Al.4.3 Cu.4.3 Ni.4.3 Ti.4.3 Du.4.3**	Welded joints	
7.4.5	Welding procedure qualifications	**5.2 Al.5.2 Cu.5.2 Ni.5.2 Ti.5.2 Du.5.2**	Approval testing of fusion welding procedures	
7.4.6	Welder qualifications	**5.3 Al.5.3 Cu.5.3 Ni.5.3 Ti.5.3 Du.5.3**	Welder and operator approval	

Table 3 **Detailed technical requirements checklist** *(continued)*

ISO 16528-1 subclause	Description	PD 5500 clause(s)	Description	Comments
7.4.7	Welder identification	**5.3.4**		
7.4.8	Heat treatment	**4.5 Al.4.5 Cu.4.5 Ni.4.5 Ti.4.5 Du.4.5**	Heat treatment	
7.4.9	Tolerances	**4.2.3**	Assembly tolerances	
		4.2.4	Tolerances for vessels subject to internal pressure	
		4.2.5 **3.6.1** a) **3.6.4**	Tolerances for vessels subject to external pressure	Clause **3.6.1 a)** gives circularity tolerance for cylindrical and conical shells subject to external pressure. Clause **3.6.4** gives circularity tolerance for spherical shells subject to external pressure
		Annex L	Guidance on structural tolerances	
7.5.1	Inspection & examination (I&E) – General	**5.1 Al.5.1**	Inspection and testing – General	
7.5.2	I&E methods	**5.6 Al.5.6 Cu.5.6 Ti.5.6**	Non-destructive testing	
		5.6.5 Al.5.6.5 Cu.5.6.6	Choice of non-destructive test methods for welds	
		5.6.6 Al.5.6.6	Non-destructive testing techniques for welds	
7.5.3	I&E procedures	**1.5.2.2**	On completion of construction	Clause **1.5.2.2** f) specifies I&E procedures to be supplied by the manufacturer
		5.6.6.7	Reporting of non-destructive testing examinations	Clause **5.6.6.7** specifies information required in I&E reports
7.5.4	I&E personnel qualification	**5.6.6.1 Al.5.6.6.1 Cu.5.6.6.1 Ti.5.6.6.1**	Radiographic techniques	Clause **5.6.6.1** refers to BS EN ISO 17636-1 which contains requirements for personnel qualifications
		5.6.6.2 Al.5.6.6.2	Ultrasonic techniques	Clause **5.6.6.2** refers to BS EN ISO 17640 which contains requirements for personnel qualifications
7.5.5	Evaluation of indications & acceptance criteria	**5.7 Al.5.7 Cu.5.7 Ti.5.7**	Acceptance criteria for weld defects revealed by visual examination and non-destructive testing	

Table 3 **Detailed technical requirements checklist** *(continued)*

ISO 16528-1 subclause	Description	PD 5500 clause(s)	Description	Comments
7.5.6	Disposition of unacceptable imperfections	**5.7.2 Al.5.7.3 Cu.5.7 Ti.5.7.2**	Assessment of defects	
		5.7.3 Al.5.7.4 Du.5.7.3	Repair of welds	
7.6.1	Final inspection	**5.8.10 Al.5.8.10**	Final inspection	
7.6.2	Final pressure test	**5.8 Al.5.8.6**	Pressure tests	
7.7	Marking/labelling	**5.8.9**	Vessel nameplate	
8	Conformity assessment	**1.4.4**	Certificate of Conformance	

Enquiry Case 5500/140

Use of copper, copper alloy, nickel and nickel alloy materials covered by European approval of materials and ASME or ASTM standards

Enquiry

The British Standards for copper and copper alloys listed in Table Cu.2.3-1 have been withdrawn and the replacement European EN standards do not cover all the alloys previously included in the British Standards; and British Standards for nickel and nickel alloys listed in Table Ni.2.3-1, Table Ni.2.3-2, Table Ni.2.3-3, Table Ni.2.3-4 and Table Ni.2.3-5 have been withdrawn and have not been replaced. Can the committee give guidance on the use of copper, copper alloy, nickel and nickel alloy materials covered by EAMs (European approval of materials) and ASME or ASTM standards.

Reply

The committee confirms that it is acceptable to use ASME or ASTM copper and copper alloy materials listed in Table 1, subject to conformity to Cu.2.1.2.1c).

Table 1 **Designations of some ASME and ASTM copper alloys and nearest BS and EN equivalents**

UNS No.	Group	BS alloy	EN No.	EN name	Alloy name
C12200	31	C 106	CW024A	Cu-DHP	Copper
C26130	32.2	CZ 126	CW707R	CuZn30As	Arsenical brass
C36500	32.2	CZ 123	CW610N	CuZn39Pb0.5	60/40 brass (low lead)
C44300 C44400 C44500	32.2 32.2 32.2	CZ 111	CW706R	CuZn28Sn1As	Admiralty brass
C46400 C46500	32.2 32.2	CZ 112	CW712R	CuZn36Sn1Pb	Naval brass
C61400	35	CA 106	CW303G	CuAl8Fe3	7% Aluminium bronze
C63000	35	CA 105	CW304G	CuAl9Ni3Fe2	10% Aluminium bronze
C63280	35	CA 104	CW307G	CuAl10Ni5Fe4	10% Aluminium bronze
C68700	32.2	CZ 110	CW702R	CuZn20Al2As	Aluminium brass
C70610	34	CN 102	CW352H	CuNi10Fe1Mn	90/10 Copper-nickel
C71500	34	CN 107	CW354H	CuNi30Mn1Fe	70/30 Copper-nickel
C71640	34	CN 108	CW353H	CuNi30Fe2Mn2	

The design strengths for the materials listed in Table 1 may be derived in accordance with Cu.2.3.1.3, except that the design strength at design temperatures exceeding 50 °C should not exceed $R_{p0.2}/1.6$, where values of 0.2% proof stress $R_{p0.2}$ are taken as being equal to S_y from ASME II Part D, Table Y-1.

The committee confirms that it is acceptable to use the nickel and nickel alloy materials covered by European approval of materials listed in Table 2 below.

Table 2 **Nickel and nickel alloy materials covered by EAMs as of April 2017**

EAM Reference.	Short name	Product form	BS alloy	UNS No.	Group
0879-1:2001/05	EAM-Nickel 201-1	Hot and cold rolled plates, sheets and strips	NA12	N02201	41
0879-2:2001/05	EAM-Nickel 201-2	Forgings	NA12	N02201	41
0879-3:2001/05	EAM-Nickel 201-3	Bars	NA12	N02201	41
0879-4:2001/05	EAM-Nickel 201-4	Seamless tubes	NA12	N02201	41

The committee also confirms that it is acceptable to use ASME or ASTM nickel and nickel alloy materials listed in Table 3 below, subject to conformity to **Ni.2.1.2.1**c).

Table 3 **Designations of some ASME and ASTM nickel alloys**

UNS No.	Group	BS alloy	EN No.	EN name	Trade names[a]
N02200	41	NA11	2.4066	Ni99.2	Nickel 200
N02201	41	NA12	2.4068	LC Ni99	Nickel 201
N04400	42	NA13	2.4360	NiCu30Fe	Monel 400 [b], Nicorros [c]
N04405	42	—	—	—	Monel R-405 [b]
N06002	43	—	2.4665	NiCr22Fe18Mo	Hastelloy X [d], Nicrofer 4722Co [c]
N06007	43	—	2.4618	NiCr22Mo6Cu	Hastelloy G [d], Nicrofer 4520hMo [c], Illium F [e]
N06022	44	—	2.4602	NiCr21Mo14W	Hastelloy C-22 [d], Nicrofer 5621hMoW [c]
N06030	45	—	—	—	Hastelloy G30 [d]
N06035	43	—	2.4643	NiCr33Mo8	Hastelloy G35 [d]
N06045	45	—	2.4889	NiCr28FeSiCe	Alloy 45 TM, Nicrofer 45TM [c]
N06059	43	—	2.4605	NiCr23Mo16Al	Alloy 59, Nicrofer 5923hMo[c]
N06200	43	—	2.4675	NiCr23Mo16Cu	Hastelloy C2000 [d]
N06210	43	—	—	—	—
N06230	43	—	2.4733	NiCr22W14Mo	Haynes 230 [d]
N06455	43	—	2.4610	NiMo16Cr16Ti	Hastelloy C-4 [d], Nicrofer 6616hMo [c]
N06600	43	NA14	2.4816	NiCr15Fe	Inconel 600 [b], Nicrofer 7216 [c]
N06600	43	—	2.4817	LC NiCr15Fe	Inconel 600L [b], Nicrofer 7216LC [c]
N06601	43	—	2.4851	NiCr23Fe	Inconel 601 [b], Nicrofer 6023 [c]
N06617	46	—	2.4663	NiCr23Co12Mo	Haynes 617 [d], Inconel 617 [b], Nicrofer 5520Co [c]
N06625	43	NA21	2.4856	NiCr22Mo9Nb	Haynes 625 [d], Inconel 625 [b], Nicrofer 6020hMo [c]
N06686	43	—	2.4606	NiCr21Mo16W	Inconel 686 [b], Nicrofer 6032 [c]
N06690	43	—	2.4642	NiCr29Fe	Inconel 690 [b], Nicrofer 6030 [c]
N06975	45	—	—	—	Hastelloy G-2 [d]
N06985	45	—	2.4619	NiCr22Mo7Cu	Hastelloy G3 [d], Nicrofer 4823hMo [c], Nicrofer 4023hMo [c]
N08020	45	—	2.4660	NiCr20CuMo	Alloy 20
N08024	45	—	—	—	—
N08026	45	—	—	—	—
N08028	45	—	1.4563	X1NiCrMoCu31-27-4	Alloy 28
N08031	45	—			—
N08120	45		2.4854	NiFe33Cr25Co	Haynes HR-120 [d]
N08330	45	—	—	—	—
N08800	45	NA15	1.4876 1.4558	X10NiCrAlTi32-20 X2NiCrAlTi32-20	Incoloy 800 [b], Nicrofer 3220 [c]
N08801	45	—	—	—	Incoloy 801 [b]
N08810	45	NA15(H)	1.4958	X5NiCrAlTi31-20	Incoloy 800H [b], Nicrofer 3220H [c]

Table 3 Designations of some ASME and ASTM nickel alloys *(continued)*

UNS No.	Group	BS alloy	EN No.	EN name	Trade names[a]
N08825	45	NA16	2.4858	NiFe30Cr21Mo3	Incoloy 825 [b], Nicrofer 4221 [c]
N08904	45	—	1.4539	X1CrNiMoCu25-20-5	—
N08925	45	—	1.4529	X1NiCrMoCu25-20-7	—
N10001	44	—	2.4810	NiMo3	Hastelloy B [d]
N10003	44	—	—	—	—
N10242	44	—	—	—	Haynes 242 [d]
N10276	43	—	2.4819	NiMo16Cr15W	Hastelloy C276 [d], Nicrofer 5716hMoW[c]
N10629	44	—	2.4600	NiMo29Cr	Alloy B4, Nimofer 6629 [c]
N10665	44	—	2.4617	NiMo28	Hastelloy B2 [d], Nimofer 6928 [c]
N10675	44	—	2.4600	NiMo29Cr	Hastelloy B3 [d]
N12160	46	—	2.4880	NiCo29Cr28Si	Haynes HR-160 [d]

[a] Trade names or trademarks of products given in the final column of this table are examples of products available commercially. Equivalent products may be used if they can be shown to lead to the same results. This information is given for the convenience of users of this specification and does not constitute an endorsement by BSI of these products.

[b] Monel, Inconel, Nimonic and Incoloy are trade names of products supplied by Special Metals.

[c] Nicorros and Nicrofer are trade names of products supplied by VDM (Outokumpu).

[d] Hastelloy and Haynes are the trade names of products supplied by Haynes International.

[e] Illium is the trade name of products supplied by Stainless Foundry and Engineering.

Materials N08904 and N08925 are classified as group 8.2 austenitic stainless steels in PD CEN ISO/TR 20173:2018, Table 1, but for this specification they should be treated as group 45 nickel-iron-chromium alloys, and the requirements of the nickel supplement should be applied for vessels or components designed and manufactured from these materials.

The design strengths for the materials listed in Table 3 may be derived in accordance with **Ni.2.3.1.2**. ASME and ASTM materials do not have specified elevated temperature values of yield strength $R_{e(T)}$; the values of yield strength S_y from ASME II Part D, Table Y-1 may be used to evaluate the time-independent design strength using **2.3.3.3b)**.

Values of elastic modulus for ASME and ASTM materials that do not have a BS alloy equivalent may be obtained from ASME II Part D, Table TM-4.

The committee also confirms that it is acceptable to use the ASME and ASTM materials listed in Table 4 below, subject to conformity to **2.1.2.1**c). These materials are classified as group 8.2 austenitic stainless steels in PD CEN ISO/TR 20173:2018, Table 1. The main body of this specification should be applied for vessels or components manufactured from these materials.

Table 4 **Designations of some ASME and ASTM nickel alloys classified as Group 8.2**

UNS No.	Group	EN No.	EN Name
N08320	8.2	–	–
N08367	8.2	–	–
N08700	8.2	–	–

Enquiry Case 5500/141

Alternative limits for longitudinal compressive general membrane stress in a vessel

Enquiry

The alternative limits specified in **A.3.5.1.3** are based on the ECCS European Recommendations *Buckling of steel shells* 4th Edition, 1988. This publication has been superseded by the 5th Edition, September 2013. Can the committee give guidance on the application of the 5th Edition of the ECCS recommendations to pressure vessels.

Reply

The following procedure can be used as an alternative to the rules given in **A.3.5.1.3** to calculate the allowable longitudinal compressive stress.

The 5th Edition of the ECCS recommendations quotes extensively from the Eurocode EN 1993-1-6:2007 and is completely compatible with that standard. The following procedure is based on the requirements of EN 1993-1-6:2007, *Eurocode 3 – Design of steel structures – Part 1-6: Strength and Stability of Shell Structures*, **8.4**, **8.5** and Annex D.1, with additional procedures from the French pressure vessel code, CODAP Division 2: 2015, Part F.

1 Calculations

Cylinders need not be checked against longitudinal buckling, and the permissible longitudinal compressive stress may be taken as being equal to the design stress f if the following condition is satisfied:

$$\frac{R}{e} \leq 0.03 \frac{E}{sf} \tag{1}$$

The following procedure shall be used to find the permissible longitudinal compressive stress in a cylindrical shell when the condition in Equation (1) is not satisfied.

The methods for measuring tolerances are given in clause 2 below.

1) Calculate the length parameter

$$\omega = \frac{L}{\sqrt{Re}} \tag{2}$$

2) Calculate the factor C_x

For short cylinders ($\omega \leq 1.7$)

$$C_x = 1.36 - \frac{1.83}{\omega} + \frac{2.07}{\omega^2} \tag{3}$$

for medium length cylinders ($1.7 \leq \omega \leq 0.5R/e$)

$$C_x = 1.0 \tag{4}$$

for long cylinders ($\omega > 0.5R/e$) C_x is the greater of:

$$C_x = 1.0 + \frac{0.2}{C_{xb}}\left[1 - 2\omega\frac{e}{R}\right] \quad \text{or} \quad C_x = 0.6 \tag{5}$$

where factor C_{xb} is obtained from Table 1

Table 1 **Factor C_{xb}**

Case	Boundary condition	Factor C_{xb}
1	cylinders that are restrained in the axial direction at both ends	6.0
2	cylinders that are restrained in the axial direction at one end (e.g. the bottom of the skirt on a vertical vessel that is fixed with anchor bolts or a shell welded to a girth flange)	3.0
3	cylinders that are not restrained in the axial direction at either end	1.0

NOTE 1 The end of a cylinder that is restrained in the axial direction is one where the axial displacement at the end is constant around the circumference.

NOTE 2 Taking C_{xb} = 1.0 is a conservative assumption.

3) Calculate

$$K = \frac{C_x p_e}{p_{yss}} \tag{6}$$

where p_e and p_{yss} are defined in Equations (3.6.4-2) and (3.6.4-1) respectively, in **3.6.4**.

4) Determine the fabrication tolerance quality class using the procedures given in clause 2 and obtain the value of the fabrication quality parameter Q from Table 2.

Table 2 **Fabrication quality parameter *Q***

Fabrication tolerance quality class	Fabrication quality parameter *Q*
Class A (Excellent)	40
Class B (High)	25
Class C (Normal)	16

5) Calculate the characteristic imperfection amplitude

$$\Delta w_k = \frac{\sqrt{Re}}{Q} \tag{7}$$

6) Calculate the elastic imperfection reduction factor

$$\alpha = \frac{0.62}{1 + 1.91(\Delta w_k/e)^{1.44}} \tag{8}$$

7) The longitudinal squash limit slenderness λ_{x0} shall be taken as:

$$\lambda_{x0} = 0.2 \tag{9}$$

8) Calculate the plastic limit relative slenderness

$$\lambda_p = \sqrt{2.5\alpha} \tag{10}$$

9) Calculate the shell relative slenderness for longitudinal buckling

$$\lambda_x = \sqrt{\frac{1}{K}} \tag{11}$$

10) Calculate the buckling reduction factor

$$\Delta = 1 \qquad \text{when } \lambda_x \leq \lambda_{x0} \tag{12}$$

$$\Delta = 1 - 0.6\left(\frac{\lambda_x - \lambda_{x0}}{\lambda_p - \lambda_{x0}}\right) \qquad \text{when } \lambda_{x0} < \lambda_x < \lambda_p \tag{13}$$

$$\Delta = \alpha K \qquad \text{when } \lambda_x \leq \lambda_x \tag{14}$$

11) Calculate the maximum allowable longitudinal compressive stress

$$\sigma_{z,\text{allow}} = \frac{\Delta \psi s f}{1.5} \tag{15}$$

where factor ψ is obtained from clause 2, step 10, below.

2 Tolerances

The fabrication tolerance quality class shall be chosen as Class A, Class B or Class C according to the tolerance definitions given in the following procedure. The tolerance class shall be determined separately for the out of roundness, misalignment and profile irregularity tolerances: the lowest fabrication tolerance quality class obtained shall then be used to determine the value of the fabrication quality parameter Q from Table 1.

NOTE 1 The lowest fabrication tolerance quality class is that which gives the lowest value of the fabrication quality parameter Q.

If none of the tolerances exceeds the relevant maximum recommended value given in Tables 3, 4, 5 or 6 for fabrication tolerance class C then factor ψ = 1.0. If any of the tolerances exceeds the relevant maximum recommended value for fabrication tolerance class C then the fabrication quality parameter Q shall be obtained from Table 2 for tolerance class C and factor ψ shall be obtained from step 10 below.

At the design stage the fabrication tolerance quality class may be chosen based on the fabrication tolerances that are expected for the completed vessel. After fabrication is complete the tolerances shall be measured and the actual fabrication tolerance quality class shall be determined. If this is lower than that assumed at the design stage then the maximum allowable longitudinal compressive stress shall be recalculated using the actual fabrication tolerance quality class, and the design re-assessed to ensure that the compressive stresses are acceptable.

NOTE 2 Using fabrication tolerance quality class C and taking factor ψ = 0.75 is a conservative assumption for vessels that satisfy the manufacturing tolerance requirements of PD 5500, Section 4.

1) Evaluate the out of roundness

$$U_r = \frac{(D_{max} - D_{min})}{D_{nom}} \times 100 \tag{16}$$

where D_{max} and D_{min} are the maximum and minimum internal diameters measured at any one cross-section and D_{nom} is the nominal internal diameter.

2) Determine the fabrication tolerance quality class so that the relevant maximum out of roundness $U_{r,max}$ from Table 3 satisfies the following condition:

$$U_r \leq U_{r,max}$$

Table 3 **Maximum out of roundness $U_{r,max}$**

Diameter range	**$D_{nom} \leq 500$**	**$500 < D_{nom} < 1\,250$**	**$1\,250 \leq D_{nom}$**
Fabrication tolerance quality class	**Recommended value of $U_{r,max}$ [%]**		
Class A (Excellent)	1.4	$0.7 + 0.000\,933(1\,250 - D_{nom})$	0.7
Class B (High)	2.0	$1.0 + 0.001\,333(1\,250 - D_{nom})$	1.0
Class C (Normal)	3.0	$1.5 + 0.002\,000(1\,250 - D_{nom})$	1.5

NOTE 1 The nominal shell internal diameter D_{nom} is in mm in the above table.

*NOTE 2 For fabrication purposes the maximum permissible out of roundness for vessels subject to external pressure is specified in **3.6.1**, and for vessels subject to internal pressure in **4.2.4.2.3** a).*

3) Determine the non-intended misalignment at circumferential welds

$$d_n = \max\left(d_1 - \frac{e_2 - e_1}{2}; 0\right) \tag{17}$$

where d_1, is the maximum measured offset between the middle lines of adjacent plates, and e_1 and e_2 are the analysis thicknesses of the thinner and thicker plates being joined.

4) Determine the fabrication tolerance quality class so that the relevant maximum non-intended misalignment $d_{n,max}$ from Table 4 satisfies the following condition:

$$d_n \leq d_{n,max}$$

Table 4 **Maximum non-intended misalignment $d_{n,max}$**

Fabrication tolerance quality class	**Recommended value of $d_{n,max}$**
Class A (Excellent)	2 mm
Class B (High)	3 mm
Class C (Normal)	4 mm

*NOTE For fabrication purposes the maximum permissible misalignment at circumferential welds is specified in **4.2.3.1** and **4.2.3.2**.*

5) Determine the non-intended misalignment parameter

$$U_n = \frac{2d_n}{e_1 + e_2} \tag{18}$$

6) Determine the fabrication tolerance quality class so that the relevant maximum non-intended misalignment parameter $U_{n,max}$ from Table 5 satisfies the following condition:

$$U_n \leq U_{n,max}$$

Table 5 **Maximum non-intended misalignment $U_{n,max}$**

Fabrication tolerance quality class	**Recommended value of $U_{n,max}$**
Class A (Excellent)	0.14
Class B (High)	0.20
Class C (Normal)	0.30

7) The depth w of local irregularities in the shell shall be measured in both the longitudinal and circumferential directions using templates as shown in Figure A-3:

a) a straight bar of length $l_x = 4\sqrt{Re}$ but no longer than 95% of the distance between circumferential welds;

b) a circular template bent to the radius of the outside surface of the shell with a length l_θ which is the same as length l_x in a) but no longer than 95% of the distance between longitudinal welds;

c) for circumferential and longitudinal welds a straight bar or circular template of length $l_w = 25e_{min}$ (where e_{min} is the thinner of the adjacent parts at the weld), but no longer than 500 mm.

8) Determine the value of the profile irregularity parameter U_0

$$U_0 = \max\left(\frac{w_x}{l_x};\ \frac{w_\theta}{l_\theta};\ \frac{w_w}{l_w}\right) \qquad (19)$$

where w_x is the depth measured in a) above, w_θ is the depth measured in b) above, and w_w is the depth measured in c) above.

9) Determine the fabrication tolerance quality class so that the relevant maximum profile irregularity parameter $U_{0,max}$ from Table 6 satisfies the following condition:

$$U_0 \leq U_{0,max}$$

Table 6 **Maximum profile irregularity parameter $U_{0,max}$**

Fabrication tolerance quality class	Recommended value of $U_{0,max}$
Class A (Excellent)	0.006
Class B (High)	0.010
Class C (Normal)	0.016

*NOTE For fabrication purposes the maximum permissible irregularities in profile are specified in **4.2.4.2.3** b) and c).*

10) If any of the tolerances exceeds the relevant maximum recommended value given in Tables 3, 4, 5 or 6 for fabrication tolerance class C then determine the value of factor ψ from Tables 7, 8, 9 and 10 for each case where the tolerance exceeds the relevant maximum recommended value, and take the smallest value of ψ for use in equation (16).

Table 7 **Correction factor ψ for out of roundness**

Nominal internal diameter (mm)	Correction factor ψ	
	$\lambda_x < 1.5$	$\lambda_x \geq 1.5$
$D_{nom} \leq 500$	$\psi = 1 - \frac{\lambda_x}{3}\left(\frac{U_r}{3} - 1\right)$	$\psi = 1.5 - 0.5\frac{U_r}{3}$
$500 < D_{nom} < 1\,250$	$\psi = 1 - \frac{\lambda_x}{3}\left(\frac{U_r}{K_D} - 1\right)$	$\psi = 1.5 - 0.5\frac{U_r}{K_D}$
$D_{nom} \geq 1\,250$	Not permitted	Not permitted

where the factor $K_D = 1.5 + 0.002(1\,250 - D_{nom})$

NOTE The nominal shell internal diameter D_{nom} is in mm in the above table.

Table 8 **Correction factor ψ for non-intended misalignment**

Shell relative slenderness	**Correction factor ψ**
$\lambda_x < 1.5$	$\psi = 1 - \frac{\lambda_x}{3}\left(\frac{d_n}{4} - 1\right)$
$\lambda_x \geq 1.5$	$\psi = 1.5 - 0.5\frac{d_n}{4}$

NOTE The offset d_n is in mm in the above table.

Table 9 **Correction factor ψ for non-intended misalignment**

Shell relative slenderness	**Correction factor ψ**
$\lambda_x < 1.5$	$\psi = 1 - \frac{\lambda_x}{3}\left(\frac{U_n}{0.3} - 1\right)$
$\lambda_x \geq 1.5$	$\psi = 1.5 - 0.5\frac{U_n}{0.3}$

Table 10 **Correction factor ψ for profile irregularity**

Shell relative slenderness	**Correction factor ψ**
$\lambda_x < 1.5$	$\psi = 1 - \frac{\lambda_x}{3}\left(\frac{U_0}{0.016} - 1\right)$
$\lambda_x \geq 1.5$	$\psi = 1.5 - 0.5\frac{U_0}{0.016}$

Enquiry Case 5500/142

Cylindrical shells of varying thickness under external pressure

Enquiry

The calculation procedure in **3.6.2** covers cylindrical shells of constant thickness. Can the committee give guidance on the application of these rules to shells of varying thickness.

Reply

The following procedure can be used to calculate the effective unsupported shell length *L* and thickness *e* for unstiffened cylinders, or cylindrical shells between points of support, where the cylindrical shell consists of two sections of different thicknesses.

The procedure is based on the method given in EN 1993-1-6:2007 *Eurocode 3 – Design of steel structures – Part 1-6: Strength and Stability of Shell Structures*, Annex D.2.

1 Notation

The notation is as given in **3.6.1.1**, except as follows:

e_1	is the analysis thickness of the thinner shell section;
e_2	is the analysis thickness of the thicker shell section;
L_{eff}	is the effective length of the shell of varying thickness;
L_{tot}	is the total length of both shell sections;
L_1	is the length of the thinner shell section;
K_L	is a factor used to determine the effective length of the shell;
m	is a factor used to determine K_L;
R_1	is the mean radius of the thinner shell section;
R_2	is the mean radius of the thicker shell section.

2 Calculations

The following procedure is only applicable for shell lengths in the range:

$$0.1 \leq \left(\frac{L_1}{L_{tot}}\right) \leq 0.5$$

If $L_1 > 0.5\ L_{tot}$ then calculate the allowable external pressure p_a in accordance with **3.6.2.1** using:

$$e = e_1 \quad L = L_{tot} \quad R = R_1$$

If $L_1 < 0.1\ L_{tot}$ then assume that $L_1 = 0.1\ L_{tot}$ in the following calculations.

1) Calculate the factor m

$$m = 2.593 - 4.8\left(\frac{L_1}{L_{tot}}\right)^2 \tag{1}$$

2) Calculate the factor K_L

$$K_L = \max\left[\left(\frac{L_{tot}}{L_1}\right) \cdot \left(\frac{e_1}{e_2}\right)^m ; 1.0\right] \tag{2}$$

3) Calculate the effective length L_{eff}

$$L_{eff} = L_1 K_L \tag{3}$$

4) Calculate the allowable external pressure p_{a1} in accordance with **3.6.2.1** using:

$$e = e_1 \quad L = L_{eff} \quad R = \left(\frac{R_1 + R_2}{2}\right)$$

5) Calculate the allowable external pressure p_{a2} in accordance with **3.6.2.1** using:

$$e = e_2 \quad L = L_{tot} \quad R = R_2$$

6) The allowable external pressure p_a for the shell of varying thickness should be taken as the smaller of p_{a1} or p_{a2}.

3 Tolerances

The cylinders should be circular to within the tolerance specified in **3.6.1** a).

The misalignment at the circumferential weld between the two shell sections should be within the tolerances specified in **4.2.3.1** and **4.2.3.2**.

4 Permissible longitudinal compressive stress

For cylindrical shells of varying thickness the maximum value of the longitudinal compressive general membrane stress in each shell section should not exceed the permissible longitudinal compressive general membrane stress in that section.

If the permissible longitudinal compressive general membrane stress is calculated in accordance with either **A.3.5.1.2** or **A.3.5.1.3** the length of the shell section is not considered.

If the permissible longitudinal compressive general membrane stress is calculated in accordance with Enquiry Case 5500/141 each cylindrical shell section of length L_j and thickness e_j should be treated as an equivalent cylinder of length $L = L_{tot}$ and thickness $e = e_j$. For cylinders classified as long the value of the factor C_{xb} should be conservatively taken as 1.0.

Enquiry Case 5500/144

Post Brexit UK pressure equipment legislation

Enquiry

The UK has left the EU, and the transition period after Brexit finished at the end of December 2020. Can the committee give guidance on how will this affect pressure vessels designed, manufactured, inspected and tested in accordance with PD 5500.

Reply

Following the UK's withdrawal from the EU, the Pressure Equipment Directive 2014/68/EU (PED), as implemented by The Pressure Equipment (Safety) Regulations 2016 (PER), is no longer applicable for vessels placed on the market in the UK. The new regulations and requirements are summarized below.

1 Regulations

For pressure vessels placed on the market in Great Britain (England, Scotland and Wales), the requirements of the PER, as amended by Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019 apply. CE marking is replaced by UKCA marking and the Declaration of Conformity with the PED is replaced by the UK Declaration of Conformity with the PER. This ensures that the equipment has freedom of movement within Great Britain. For mandatory conformity assessments, the conformity assessment body shall be an approved body or user inspectorate appointed by the UK government. These requirements apply to pressure vessels manufactured in Great Britain or imported from outside the UK, but not to vessels manufactured in Northern Ireland.

For pressure vessels placed on the market in Great Britain and manufactured in Northern Ireland, special arrangements apply. The manufacturer shall complete a Declaration of Conformity with the PED and may then affix the CE mark when applicable. If a notified body or user inspectorate appointed by a member state of the European Union is used for mandatory conformity assessments, the equipment has freedom of movement within Great Britain, Northern Ireland and the European Union. If an approved body or user inspectorate appointed by the UK government is used for mandatory conformity assessments, an additional UK(NI) mark is required which ensures that the equipment has freedom of movement in Great Britain and Northern Ireland, but not in the European Union.

For pressure vessels placed on the market in Northern Ireland, the requirements of the PER, as amended by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020 apply. The manufacturer shall complete a Declaration of Conformity with the PED and may then affix the CE mark when applicable. If a notified body or user inspectorate appointed by a member state of the European Union is used for mandatory conformity assessments, the equipment has freedom of movement within the UK and the European Union. If the conformity assessment body is an approved body or user inspectorate appointed by the UK government, an additional UK(NI) mark is required and the equipment has freedom of movement within the UK, but not the European Union. These requirements apply to pressure vessels manufactured in Northern Ireland or imported into Northern Ireland, including those manufactured in Great Britain. Equipment carrying only the UKCA mark is not valid for the Northern Ireland market.

For pressure vessels placed on the market in the European Union, the requirements of the PED apply. The manufacturer shall complete a Declaration of Conformity with the PED and may affix the CE mark when applicable. This ensures that the equipment will have freedom of movement within the European Union (and Northern Ireland). For mandatory conformity assessments, the conformity assessment body shall be a notified body or user inspectorate appointed by a member state of the European Union.

CE marked pressure vessels that meet the requirements of the PED, while these match the requirements of the PER, can continue to be placed on the market in Great Britain until 31 December 2024.

NOTE 1 The UKCA mark or CE mark can only be affixed to vessels that fall within the scope of the relevant regulations (PER or PED) and which conform with all the relevant requirements. The UKCA mark or CE mark should not be affixed to vessels that are classified as being manufactured according to sound engineering practice – see Regulation 8 in Part 1 of the PER and Article 4, paragraph 3 of the PED.

NOTE 2 The UK government intends to extend recognition of the CE marking for placing pressure equipment on the market in Great Britain indefinitely, beyond December 2024.

2 Conformity assessment bodies

In Great Britain, approved bodies and user inspectorates are appointed by the UK government for carrying out specified duties under the PER, as amended by Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

In Northern Ireland, approved bodies and user inspectorates are appointed by the UK government for carrying out specified duties under the PER, as amended by Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020.

In the European Union, notified bodies and user inspectorates are appointed by member states of the EU for carrying out specified duties under the EU PED, or any national implementation thereof.

3 Inspecting Authority

When PD 5500 is being used to satisfy requirements of the UK PER or the EU PED, or any national implementation thereof, the approved body, notified body or user inspectorate can also act as the Inspecting Authority, and for permanent joining a Recognised Third Party Organization (RTPO) can act as the Inspecting Authority.

4 Materials for pressure parts

For pressure vessels placed on the market in Great Britain, the use of materials covered by European approval for material does not provide compliance with the PER, as amended by Schedule 24 of the Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019.

For pressure vessels placed on the market in the EU, paragraph 4.3 of Annex I of the PED states that, where a material manufacturer has an appropriate quality-assurance system, certified by a competent body established within the Union and having undergone a specific assessment for materials, certificates issued by the manufacturer are presumed to certify conformity with the relevant requirements of this point. For pressure vessels placed on the market in Northern Ireland, "within the Union" is replaced by "in a relevant state" (i.e. Northern Ireland or any EEA state). For pressure vessels placed on the market in Great Britain, the equivalent paragraph 31(8) of Schedule 2 of the PER is omitted (see section 5 below), so certificates issued by the material manufacturer do not provide presumption of conformity with paragraph 31 of Schedule 2 of the PER.

5 Essential safety requirements and Annex Z

Changes to the Essential Safety Requirements in Schedule 2 of the PER:

a) in paragraphs 21(4), 31(4)(b)(i) and 35(1) for "harmonised" substitute "designated";

b) in paragraph 29(1) for "CE" substitute "UK";

c) omit paragraphs 31(4)(b)(ii) and 31(8).

When PD 5500 is being used to satisfy requirements of the PER, as modified by either Schedule 24 of The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019 or Schedule 2 of The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020, the manufacturer is responsible for compliance with the provisions of the Regulations. On completion of all the design, manufacturing, inspection and testing procedures, and after verification of conformity, the manufacturer can make a Declaration of Conformity with the PER.

Annex Z has been amended to provide guidance on the use of PD 5500 to satisfy the Essential Safety Requirements in Schedule 2 of the PER.

6 Additional information

Attention is drawn to the existence of the following UK Government websites that give the text of the statutory instruments and guidance on their application:

Statutory Instruments

The Pressure Equipment (Safety) Regulations 2016:
www.legislation.gov.uk/uksi/2016/1105

The Product Safety and Metrology etc. (Amendment etc.) (EU Exit) Regulations 2019:
www.legislation.gov.uk/uksi/2019/696/

The Pressure Vessels (Amendment) (Northern Ireland) (EU Exit) Regulations 2020:
www.legislation.gov.uk/uksi/2020/678/made

Guidance

www.gov.uk/government/publications/pressure-equipment-safety-regulations-2016

www.gov.uk/guidance/placing-manufactured-goods-on-the-market-in-great-britain

www.gov.uk/guidance/using-the-ukca-marking

www.gov.uk/guidance/trading-and-moving-goods-in-and-out-of-northern-ireland

www.gov.uk/guidance/placing-manufactured-goods-on-the-eu-market

www.gov.uk/guidance/conformity-assessment-and-accreditation

www.gov.uk/government/publications/the-border-operating-model

www.gov.uk/guidance/ukca-marking-conformity-assessment-and-documentation

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