BS EN 12952-3:2011

BSI Standards Publication

Water-tube boilers and auxiliary installations

Part 3: Design and calculation for pressure parts of the boiler

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National foreword

This British Standard is the UK implementation of EN 12952-3:2011. It supersedes BS EN 12952-3:2001 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee PVE/2, Water Tube And Shell Boilers.

A list of organizations represented on this committee can be obtained on request to its secretary.

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

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Chaudières à tubes d'eau et installations auxiliaires - Partie 3: Conception et calcul des parties sous pression de la chaudière

Wasserrohrkessel und Anlagenkomponenten - Teil 3: Konstruktion und Berechnung für drucktragende Kesselteile

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Foreword

This document (EN 12952-3:2011) has been prepared by Technical Committee CEN/TC 269 “Shell and water-tube boilers”, the secretariat of which is held by DIN.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by June 2012, and conflicting national standards shall be withdrawn at the latest by June 2012.

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. CEN [and/or CENELEC] shall not be held responsible for identifying any or all such patent rights.

This document supersedes EN 12952-3:2001.

This document has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association, and supports essential requirements of EU Directive(s).

For relationship with EU Directive 97/23/EC, see informative Annex ZA, which is an integral part of this document.

Annex E provides details of significant technical changes between this European Standard and the previous edition.

The European Standard series EN 12952 concerning water-tube boilers and auxiliary installations consists of the following parts:

— *Part 1: General;*

— *Part 2: Materials for pressure parts of boilers and accessories;*

— *Part 3: Design and calculation for pressure parts of the boiler;*

— *Part 4: In-service boiler life expectancy calculations;*

— *Part 5: Workmanship and construction of pressure parts of the boiler;*

— *Part 6: Inspection during construction, documentation and marking of pressure parts of the boiler;*

— *Part 7: Requirements for equipment for the boiler;*

— *Part 8: Requirements for firing systems for liquid and gaseous fuels for the boiler;*

— *Part 9: Requirements for firing systems for pulverized solid fuels for the boiler;*

— *Part 10: Requirements for safeguards against excessive pressure;*

— *Part 11: Requirements for limiting devices of the boiler and accessories;*

— *Part 12: Requirements for boiler feedwater and boiler water quality;*

— *Part 13: Requirements for flue gas cleaning systems;*

— *Part 14: Requirements for flue gas DENOX-systems using liquified pressurized ammonia and ammonia water solution;*

— *Part 15: Acceptance tests;*

— *Part 16: Requirements for grate and fluidized-bed firing systems for solid fuels for the boiler;*

— *CR 12952 Part 17: Guideline for the involvement of an inspection body independent of the manufacturer.*

NOTE 1 A Part 18 on operating instructions is currently in preparation.

Although these parts may be obtained separately, it should be recognized that the parts are inter-dependent. As such, the design and manufacture of water-tube boilers requires the application of more than one part in order for the requirements of this European Standard to be satisfactorily fulfilled.

NOTE 2 Part 4 and Part 15 are not applicable during the design, construction and installation stages.

NOTE 3 A "Boiler Helpdesk" has been established in CEN/TC 269 which may be contacted for any questions regarding the application of European Standards series EN 12952 and EN 12953, see the following website: http://www.boiler-helpdesk.din.de

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Bulgaria, Croatia, Cyprus, Czech Republic, Denmark, Estonia, Finland, France, Germany, Greece, Hungary, Iceland, Ireland, Italy, Latvia, Lithuania, Luxembourg, Malta, Netherlands, Norway, Poland, Portugal, Romania, Slovakia, Slovenia, Spain, Sweden, Switzerland and the United Kingdom.

1 Scope

This European Standard specifies the requirements for the design and calculation of water-tube boilers as defined in EN 12952-1.

The purpose of this European Standard is to ensure that the hazards associated with water-tube boilers are reduced to a minimum by the proper application of the design according to this part of EN 12952.

2 Normative references

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

EN 1092-1:2007, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 1: Steel flanges*

EN 1759-1:2004, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, Class designated — Part 1: Steel flanges NPS ½ to 24*

EN 10028-1:2007+A1:2009, *Flat products made of steels for pressure purposes — Part 1: General requirements*

EN 10164:2004, *Steel products with improved deformation properties perpendicular to the surface of the product — Technical delivery conditions*

EN 10266:2003, *Steel tubes, fittings and structural hollow sections — Symbols and definitions of terms for use in product standards*

EN 12952-1:2001, *Water-tube boilers and auxiliary installations — Part 1: General*

EN 12952-2:2011, *Water-tube boilers and auxiliary installations — Part 2: Materials for pressure parts of boilers and accessories*

EN 12952-5:2011, *Water-tube boilers and auxiliary installations — Part 5: Workmanship and construction of pressure parts of the boiler*

EN 12952-6:2011, *Water-tube boilers and auxiliary installations — Part 6: Inspection during construction; documentation and marking of pressure parts of the boiler*

EN 12952-12:2003, *Water-tube boilers and auxiliary installations — Part 12: Requirements for boiler feedwater and boiler water quality*

EN 12953-3:2002, *Shell boilers — Part 3: Design and calculation for pressure parts*

EN 13445-3:2009, *Unfired pressure vessels — Part 3: Design*

EN 13480-3:2002, *Metallic industrial piping — Part 3: Design and calculation*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in EN 12952-1:2001 apply.

4 Symbols and abbreviations

For the purposes of this document, the symbols given in EN 12952-1:2001, Table 4-1 apply. Throughout this European Standard, additional terminology and symbols have been included where necessary to meet the requirements of the specific text concerned.

5 General

5.1 Purpose

Water-tube boiler pressure parts shall be designed in accordance with the requirements of this European Standard. The resulting designs shall be reproduced in the form of approved drawings and specifications to ensure the proper application of the design requirements during the manufacturing and inspection stages.

5.2 Dimensions of pressure parts

The wall thickness and other dimensions of pressure parts sufficient to withstand the calculation pressure at calculation temperature for the design lifetime shall be determined in accordance with this European Standard using materials in accordance with EN 12952-2:2011.

The design for loadings arising from the following situations shall also be determined in accordance with this European Standard:

a) the bending of a drum or header as a beam under self weight and imposed loads;

b) local support loads on drums;

c) thermally induced forces and moments within or arising from systems of integral tubing;

d) local loading of tubes by structural attachments;

e) rapid and frequent changes of pressure and temperature.

Methods for calculating stresses caused by external loads applied to nozzles and to attachments shall be in accordance with EN 13445-3:2009.

NOTE The purpose of this part is to give specific design rules for common forms of loadings to which boiler parts are normally subjected to and general rules on how other loadings are to be considered. It does not give specific design rules for loadings other than those described in a) to e).

These design rules are adequate for boilers of established construction, installed and operated in accordance with the manufacturer's instructions.

Determination of the dimensions of pressure parts shall be given special consideration not included in this European Standard, when abnormal conditions are present, such as:

— abnormally high corrosive products of combustion;

— highly pressurized products of combustion;

— poor feedwater.

Deviations from the requirements of this European Standard by the use of alternative design methods shall be permitted, provided it can be shown that the adoption of such methods does not impair the safety of the component. A record of all deviations shall be recorded in the manufacturer’s dossier. See also EN 12952-1:2001, Clause 7.

5.3 Strength of pressure parts

The strengths of the pressure parts shall be such as to withstand the following loads:

a) internal pressure;

b) the weight of all pressure parts and their contents, the weight of components suspended from them and any superimposed slag, fuel, ash or dust;

c) loads caused by gas pressure differentials over the boiler furnace and flue gas passes;

d) loads arising at connections between the boiler system and other parts.

If applicable, the pressure parts shall be adequate to withstand wind and earthquake loads. The conditions applicable for such loads shall be determined by the customer. These determinations shall be considered by the manufacturer under his responsibility.

5.4 Design by analysis

It shall be permissible to design by analysis provided the safety and functional requirements of the components are not impaired.

The results of any stress calculations carried out for loadings not explicitly covered by equations in this Clause 5 shall be determined by using the criteria given in EN 13445-3:2009.

5.5 Cyclic loading

Boiler components are deemed to be exposed to cyclic loading if the boiler is designed for more than 500 cold start-ups. Where cylindrical or spherical pressure parts with openings are subject to cyclic loading, the following calculation for the allowable temperature change rate v_t shall be carried out:

$$v_t = \left(X - p_o \left(\frac{\alpha_m \times d_m}{n_s \times e_{ms}} - 0{,}5 \right) \right) \frac{Z}{e_{ms}^2} \tag{5.5-1}$$

where

v_t allowable rate of temperature change in K/s;

p_o is the maximum operating pressure;

d_m is the mean diameter of the shell;

e_{ms} is the minimum wall thickness;

n_s = 2 for cylindrical shells or
n_s = 4 for spherical shells;

α_m = 4 or if there is any doubt that this value is conservative, the exact value α_m for cylindrical shells taken from Figure 13.4-5 or α_{sp} for spherical shells, taken from Figure 13.4-7 shall be used;

X = 550 N/mm^2;

Z = 2 K mm^4/(Ns) for ferritic steels, or

Z = 1 K mm^4/(Ns) for austenitic and martensitic steels, or

$$Z = -\frac{0{,}5 D_{th}}{\gamma_{cyl/sp} \alpha_t \beta_t \, E/(1-\nu)} \tag{5.5-2}$$

where exact values D_{th}, β_t, E, v may be taken from Annex D, $\gamma_{cyl/sp}$ from Figures 13.4-6 or 13.4-9 and α_t from Figure 13.4-8.

If the result of this calculation is smaller than the required temperature transient at start-up, or if it is negative, then 13.4 shall apply.

For designs subject to cyclic loading, careful attention shall be paid to the design configuration in order to avoid stress raising features and to ensure good stress distribution. Stamping of materials shall not be done in critical areas.

In considering operating conditions, the design shall make adequate allowance for corrosion and fatigue.

The level of non-destructive testing adopted shall meet the acceptance criteria for main drum welds in EN 12952-6:2011.

5.6 Other design requirements

5.6.1 General

In particular, cognizance shall be taken of the following requirements in EN 12952-5:2011 and EN 12952-6:2011:

a) the design shall be such that manufacturing and welding in accordance with EN 12952-5:2011 and inspection in accordance with EN 12952-6:2011 shall be possible;

b) where partial penetration welds are to be used, the depth of the required weld preparation groove shall be specified on the drawing;

c) the welds attaching branches, nozzles, stubs and supports to drums and headers shall not involve any combination of austenitic and ferritic steel;

d) the requirements covering the attachment of nozzles and branches to drums and headers without strength welding shall be in accordance with EN 12952-5:2011, 9.3;

e) the requirements covering tube connections to drums and headers without strength welding shall be in accordance with EN 12952-5:2011, 9.4;

f) limits of operation for nodular graphite cast iron valves and fittings;

g) where random NDE of welds is permitted by EN 12952-6:2011, it shall be demonstrated that the welding is adequate for the imposed loading when a weld joint factor of 0,85 is applied;

h) the special requirements applicable to coil boilers are given in EN 12952-5:2011, Annex D;

i) the special requirements applicable to chemical recovery boilers are given in EN 12952-5:2011, Annex E.

For major components operating in the creep range, facilities shall be provided for monitoring the creep in relation to operation.

5.6.2 Access

The boiler shall be designed to ensure adequate access is provided to facilitate the internal examination of the drums and headers. The examination may be either manual or remote in accordance with the physical size of the components. The requirements and limitations of access and inspection openings shall be in accordance with EN 12952-5:2011, 9.2.

5.6.3 Drainage and venting

The boiler shall be provided with adequate means of drainage and venting in order to avoid water hammer and vacuum collapse, and to enable internal inspections to be carried out.

5.7 Design, calculation and test pressures

5.7.1 Design pressure

For the purpose of EN 12952-3 the design pressure p_d shall be equal to/or greater than the maximum allowable pressure PS.

For each compartment of the water-tube boiler, the design pressure shall be at least the highest set pressure of any safety valve mounted on that compartment.

NOTE A compartment is any pressurized section of plant which can be isolated by shut-off valves.

5.7.2 Calculation pressure

Each compartment may be divided into sections, each with its own calculation pressure p_c and calculation temperature t_c. The design of each section shall be based on one of the following:

a) for parts whose design stress has been derived from tensile strength R_m or minimum yield proof strength $R_{p0,2\ tc}$ the calculation pressure shall be the design pressure increased to the highest pressure possible when the plant is operating at the calculation temperature. Any difference between design pressure and calculation pressure might be caused by hydrostatic pressure and by pressure drop caused by fluid flow. Differences in hydrostatic height less than or equal to 0,05 MPa can be ignored;

b) for parts whose design stress has been derived from the creep rupture strength, the calculation pressure shall be the lowest set pressure of any safety valve at the superheater/reheater outlet, as appropriate, increased by the highest pressure difference possible under maximum continuous rating conditions.

A check shall be made of the thickness calculated by method b), using the calculation pressure of a) above with a design stress derived from tensile strength R_m or minimum yield/proof strength $R_{p0,2\ tc}$ at the calculation temperature used in b), and the greater thickness used.

If the minimum yield strength data at higher temperatures are not available, linear extrapolation may be allowed.

5.7.3 Calculation pressure for pressure differences

For parts with a design pressure not less than 1 MPa, which are simultaneously subject to both internal and external pressure, e.g. surface type attemperators in boiler drums, and where the design ensures that both pressures always occur together, the calculation pressure shall be the maximum pressure difference, but not less than 1 MPa. The loading occurring during hydrostatic testing shall be taken into account.

5.7.4 Hydrostatic test

5.7.4.1 General

In order to demonstrate the strength and integrity of individual components and of the completely assembled water-tube boiler, and to establish that no major error or defect has occurred, completely assembled water-tube boilers shall be hydrostatically tested to the test pressure specified in 5.7.4.2 and individual components shall be hydrostatically tested in accordance with 5.7.4.2 without any sign of weakness or defect.

The hydrostatic tests shall be carried out on welded components or the completed water-tube boiler after all welding and heat treatment has been completed, but may be carried out prior to the drilling of holes for expanded tubes in the boiler drum.

All components which are not reasonably accessible for inspection after assembly into the water-tube boiler shall be individually hydrostatically tested to the test pressure specified in 5.7.4.2 before assembly into the water-tube boiler.

5.7.4.2 Test pressure

A boiler assembly comprises of a number of components each having its own specific calculation pressure p_c and calculation temperature t_c. The test pressure p_t for components as defined in 5.7.4.1 shall be determined directly in accordance with 5.7.4.3.

As there can be only one hydrostatic test pressure for a boiler assembly or separately isolated compartment as defined in EN 12952-1:2001, 1.2, it shall be necessary to carry out a series of individual calculations on selected components throughout the assembled boiler or isolated compartment, if applicable, in accordance with 5.7.4.3, to determine the individual apparent test pressure for each selected component. The hydrostatic test pressure for the whole assembly shall be the pressure which ensures that none of the components selected shall be subjected under test conditions to a stress greater than that given in 6.3.7.

Where the hydrostatic test is harmful or impractical, other tests of a recognized value may be carried out. For tests other than the hydrostatic pressure test, additional measures, such as non-destructive tests or other methods of equivalent validity, shall be applied before those tests are carried out.

5.7.4.3 Calculation of hydrostatic test pressure

The hydrostatic test pressure for a component or completely assembled boiler shall be determined as follows:

$$p_t = 1{,}43 \cdot PS \tag{5.7-1}$$

or

$$p_t = 1{,}25 \cdot PS \cdot \frac{R_{p0,2\,20}}{R_{p0,2\,t_c}} \tag{5.7-2}$$

whichever is the greater, where

p_t is the test pressure for the component under consideration;

PS is the maximum allowable pressure or if higher the calculation pressure for the component under consideration.

The ratio $R_{p0,2\,20}/R_{p0,2\,tc}$ or in the case of austenitic steels, if its elongation after rupture exceeds 30 %, the ratio $R_{p1,0\,20}/R_{p1,0\,tc}$ to be used shall be the lowest of those permitted for the components under consideration, based on the material properties and the specific calculation temperature and should not be less than 1, see also 6.3.

If the minimum yield strength data at higher temperatures are not available, linear extrapolation may be allowed.

5.8 Metal wastage[1)]

5.8.1 Internal wastage

Internal wastage is normally small and shall not be considered for boilers operated with feedwater in accordance with EN 12952-12:2003. For components exposed to risk of greater than normal wastage (e.g. erosion by turbulence), appropriate countermeasures shall be provided.

The magnetite layer shall be protected in accordance with 13.4.1.1.

5.8.2 External wastage

External wastage of pressure parts not exposed to flue gases is normally small, and the thickness determined by this European Standard shall be adequate without further addition.

Tubes exposed to flue gases shall experience wastage to a varying extent. If the boiler design data indicates that wastage can be significant, the tubes shall be increased in thickness accordingly. In addition, other means of tube protection may occur. In this case the wall thickness allowance shall be specified by the manufacturer unless the purchaser has specified a higher allowance.

It shall be permitted to take account of wastage by means of metallurgically bonded, composite material tubing with corrosion resistant layers.

5.8.3 Requirements

Where an allowance for wastage is made, the amount shall be specified in the design documents stipulating whether this allowance is "internal or external". Strength calculations shall use the dimensions after the wastage allowances are removed.

However, for tubes designed using design strengths derived from creep rupture properties, integration over time of the effects of creep and wastage shall be permitted, so that failure can be predicted at a time not less than the design lifetime. In such cases the tube thickness towards the end of the design life might be less than required by Equations (11.2-2 to 11.2-5).

5.8.4 Stress corrosion

With boiler water quality controlled in accordance with EN 12952-12:2003, stress corrosion would not be expected to occur in ferritic tubing under normal boiler operating conditions. The risk of such corrosion in austenitic superheater materials can be satisfactorily reduced by ensuring no water droplets are carried over into the austenitic tubing. Carry over can be considered to have been adequately restricted if the steam has an enthalpy of 2 900 kJ/kg or greater, or the enthalpy corresponds to a temperature of 425 °C or higher.

Where it is predicted that exceptional conditions of chemical concentration may occur for prolonged periods of operation, the effects of stress corrosion and corrosion gouging shall be considered, and the materials selected accordingly.

NOTE It is not possible to compensate for stress corrosion by increasing the thickness of components.

5.8.5 Mechanical requirements

Where there is a likelihood of in-service relative movement or fretting between a pressure part and a non-pressure part in contact with it, consideration shall be given to wastage of the components. If necessary, wear-pads shall be welded to the pressure part, or other equivalent means shall be employed.

[1)] For the purpose of design in accordance with this European Standard, metal wastage includes oxidation, corrosion, erosion and abrasion.

5.9 Attachment on pressure parts

5.9.1 Load carrying attachments

Load carrying attachments shall be defined by the design engineer and indicated as such on the drawing.

Load carrying attachments are:

a) attachments designed for primary loads which are completely definable and are usually for support purposes, or

b) attachments which are usually provided for alignment and/or restraint purposes where the loading is not easily defined. Such attachments may be loaded by either primary or secondary loads.

Stresses caused by load carrying attachments not covered in 11.5 shall be calculated in accordance with EN 13445-3:2009.

5.9.2 Non load carrying attachments

Non load carrying attachments are attachments which carry no significant loads during manufacture, erection, testing or any operating condition.

6 Calculation temperature and nominal design stress

6.1 Calculation temperature

6.1.1 General

For the purpose of EN 12952 the maximum allowable temperature TS shall be that at the steam/hot water outlet.

The reference temperature t_{or} shall the mean fluid operation temperature of the component under consideration, which is to be expected during use.

Where steam or water flows through components in parallel, t_{or} for each component shall take account of variations in heat transfer and fluid flow between the parallel parts.

The calculation temperature t_{c} of a component may be calculated by taking account of variations in heat transfer and fluid flow in the boiler. If such calculations are not carried out then the calculation temperature t_{c} shall be composed of the reference temperature t_{or} and the temperature allowance in accordance with 6.1.2 to 6.1.10. The temperature allowances in Table 6.1-1 shall be regarded as minimum values, except where calculations of t_{c} are carried out, and is allowed by 6.1.5.

Table 6.1-1 — Reference temperatures and temperature allowances

Physical state	Reference temperature	Temperature allowances			
		Unheated components [a]	Heated components [a]		
			Heating mainly by radiation [b]	Heating mainly by convection	Protected against radiation
Water or water/steam mixture	Saturation temperature at allowable (working gauge) pressure p_{s1} or at allowable (total gauge) pressure p_{s2}	0 °C	50 °C For headers [c] (30 + 3 e_s) °C but not less than 50 °C	(15 + 2 e_s) °C but not more than 50 °C	20 °C
Superheated steam	Superheated steam, see also 6.1.3	15 °C, see also 6.1.5	50 °C	35 °C	20 °C

[a] For definitions of types of heating see 6.1.7 to 6.1.10.

[b] Platen type superheaters are treated like convection type superheaters.

[c] For definition of header see 6.1.6.

6.1.2 Circulation boilers

For circulation boilers, the reference temperature and the temperature allowance shall be in accordance with Table 6.1-1.

6.1.3 Once-through boilers, superheaters and reheaters

For once-through boilers, the reference temperature t_{or} is the temperature of the fluid. The temperature of the fluid shall be calculated by taken into account of variations in heat transfer and fluid flow in the boiler.

For superheaters and reheaters the reference temperature shall be the mean temperature expected during service, of the fluid flowing through the various boiler parts.

The temperature allowance shall be in accordance with Table 6.1-1.

6.1.4 Hot water generators

For the special case of hot water generators, where the temperature of the contained fluid is limited by thermostats[2)], the reference temperature of the components shall be the fluid temperature.

[2)] Temperature limiters manufactured and tested in accordance with EN 12952-11:2007 are considered to be reliable.

6.1.5 Temperature allowances for unheated components

For unheated components carrying superheated steam, the temperature allowance of 15 °C given in Table 6.1-1 shall be reduced to 5 °C (measuring tolerance) if it shall be ensured that the temperature required by the design cannot be exceeded.

This can be achieved by:

a) temperature control upstream of the said components;

b) the arrangement of cooling or mixing points (e.g. by headers through which the fluid flows in a longitudinal direction) upstream of the said components;

c) connection measures for heating surface arrangement or the like.

6.1.6 Headers

Tubular hollow parts with a nominal external diameter greater than 76,2 mm, into which there are three or more non-axial tube entries, shall be considered as headers.

6.1.7 Unheated components

Components shall be considered to be unheated if

a) they are behind refractory brickwork, and an intermediate space of at least 100 mm is between the brickwork and the components;

b) a gas tight welded waterwall is arranged between the components and the furnace or gas pass;

c) the components are protected by a layer of refractory bricks or refractory lining and this layer is not primarily subject to heat absorption due to radiative heat transfer (see 6.1.10); in this case, the brickwork or refractory lining shall be attached to the suspended part by means of holding devices. In the case of headers, studding can be provided for this purpose;

d) the highest possible temperature of the flue gas is less than the reference temperature of the component. If the temperature of the flue gas is only marginal above the reference temperature, the calculation temperature is not required to be defined higher than the highest possible flue gas temperature.

6.1.8 Components protected against radiation

Components shall be considered to be protected against radiation if they are screened by closely spaced tubes (maximum clear distance 3 mm) and no substantial flow of flue gases can occur between the screening tubes and the components.

6.1.9 Components heated by convection

Components shall be considered to be primarily heated by convection if

a) they are not subject to radiation (see 6.1.10);

b) the components are protected by a layer of refractory brickwork or refractory lining against radiative heat transfer (see 6.1.10). In this case, the bricks or refractory lining shall be attached to the suspended part by holding devices, which in the case of headers, can be studding;

c) the components are protected by a row of tubes with a ratio of

$$P_0 / d_o \leq 1{,}3 n^{0{,}63} \tag{6.1-1}$$

where

n is the number of the rows;

P_0 is the longitudinal pitch with Φ = 0°;

d_o is the outside diameter.

This requires a value of

$P_0/d_o \leq 1{,}3$ for one row of tubes in accordance with Figure 6.1-1;

$P_0/d_o \leq 2{,}0$ for two rows of tubes in accordance with Figure 6.1-2;

$P_0/d_o \leq 2{,}6$ for three rows of tubes in accordance with Figure 6.1-3.

d) the components are provided with closely spaced tubes in accordance with Figure 6.1-4 with a ratio of

$$\frac{P_0 \; P_{90}}{\pi \; l^2} \leq 0{,}1 \qquad (6.1\text{-}2)$$

where

P_0 is the longitudinal pitch with Φ = 0°;

P_{90} is the circumferential tube pitch on the external surface with Φ = 90°;

l is the distance between component and furnace envelope.

6.1.10 Components heated by radiation

Unscreened components shall be considered to be primarily heated by radiation, if they are subject to radiation by flue gases with a temperature > 950 °C.

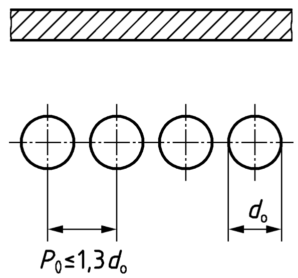


Figure 6.1-1 — Components protected by one row of tubes

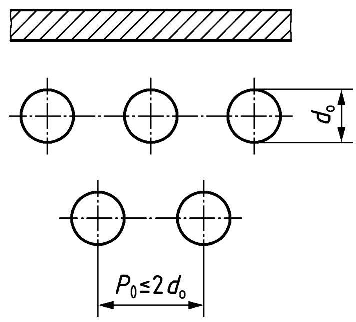


Figure 6.1-2 — Components protected by two rows of tubes

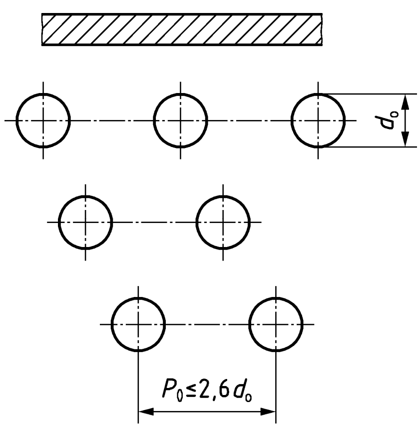
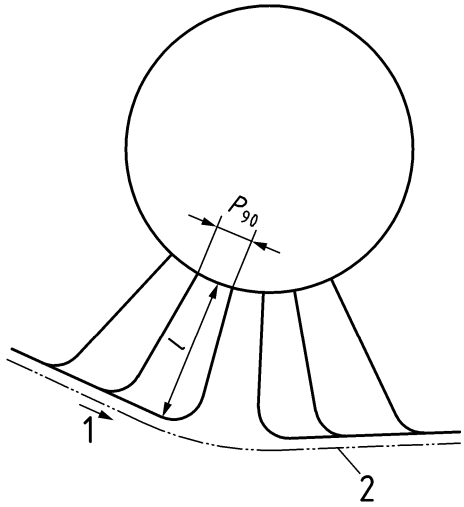


Figure 6.1-3 — Components protected by three rows of tubes



Key

1 flue gas flow

2 boundary of flue gas flow

Figure 6.1-4 — Components protected by closely spaced tubes

6.2 Maximum through-the-wall temperature difference and maximum flue gas temperature for heated drums and headers

6.2.1 Maximum through-the-wall temperature difference

The through-the-wall temperature difference (defined as outside wall temperature minus inside wall surface temperature) of heated drums and headers, shall not exceed 30 K.

NOTE Components made from ferritic steel not exceeding 32 mm wall thickness and exposed to a heat flux less or equal to 40 kW/m^2 will meet this requirement.

6.2.2 Headers exposed to flue gas

Surfaces of superheater and reheater headers shall not be exposed to flue gas temperatures in excess of 500 °C.

6.2.3 Allowable deviations

In cases where it is necessary to deviate from 6.2.1 and 6.2.2 it shall be verified that unsteady and steady thermal stresses do not lead to unacceptable stresses in the component wall.

6.3 Design stress

6.3.1 General

The design stress f to be used for the design of components primarily subjected to static loading shall be

$$f = \frac{K}{S} \tag{6.3-1}$$

where

K is the material strength value for the design conditions in accordance with the material standards and specifications given e.g. in EN 12952-2:2011 and where S is the safety factor and at the test pressure $p_{t'}$ the design stress $f_{t'}$ shall be

$$f_{t'} = \frac{K_{t'}}{S_{t'}} \tag{6.3-2}$$

where

$K_{t'}$ is the material strength value and $S_{t'}$ is the safety factor, both for the test condition.

6.3.2 Rolled and forged steels

The design stress f shall be the lowest value obtained from the equation

$$f = \min\left\{\frac{R_{\text{m 20}}}{2,4}; \frac{R_{\text{eH tc}}}{1,5} \text{ and/or } \frac{R_{\text{p0,2 tc}}}{1,5}; \frac{R_{\text{mT tc}}}{1,25}\right\} \tag{6.3-3}$$

where

$R_{\text{m 20}}$ is the tensile strength at room temperature (20 °C);

$R_{\text{eH tc}}$ is the upper yield strength at calculation temperature t_c;

$R_{\text{p0,2 tc}}$ is the 0,2%-proof strength at calculation temperature t_c and

$R_{\text{mT tc}}$ [3, 4] is the creep rupture strength for the specified lifetime T at calculation temperature t_c.

6.3.3 Austenitic steels

The design stress f shall be in accordance with the following:

— for steels with a specified minimum transverse elongation A < 30%, the lowest value obtained from the equation

$$f = min\left\{\frac{R_{\text{p0,2 tc}}}{1,5}; \frac{R_{\text{mT tc}}}{1,25}\right\} \tag{6.3-4}$$

where

$R_{\text{p0,2 tc}}$ is the 0,2%-proof strength $R_{\text{p0,2}}$ at calculation temperature t_c and
$R_{\text{mT tc}}$ [3, 4] is the creep rupture strength for the specified lifetime T at calculation temperature t_c.

— for steels with a specified minimum transverse elongation A ≥ 30 %, the lowest value obtained from equation

---

[3] T is the minimum specified design lifetime with a minimum of 100 000 h. If no lifetime is specified 200 000 h shall be taken. Should creep rupture strength values for times in excess of 100 000 h not be available, 100 000 h data may be used with S = 1,5 for internal pressure. In exceptional cases, where pressure parts are operated in the creep range for short duration (less than 10 000 h) e.g. discharge, relief lines, T may be reduced to 10 000 h with the safety factor S = 1,25.

[4] Creep rupture strength values for intermediate lifetimes shall be obtained by linear interpolation in a double logarithmic system.

$$f = min\left\{\frac{R_{\text{p1,0 tc}}}{1{,}5};\frac{R_{\text{mT tc}}}{1{,}25}\right\} \tag{6.3-5}$$

where

$R_{\text{p1,0 tc}}$ is the 1,0 %-proof strength $R_{\text{p1,0}}$ at calculation temperature t_{c} and

$R_{\text{mT tc}}$[5, 6] is the creep rupture strength for the specified lifetime T at calculation temperature t_{c}.

— or alternatively, and if its specified minimum transverse elongation A ≥ 35 %, the lowest value obtained from the equation

$$f = min\left\{\frac{R_{\text{m tc}}}{3};\frac{R_{\text{p1,0 tc}}}{1{,}2};\frac{R_{\text{mT tc}}}{1{,}25}\right\} \tag{6.3-6}$$

where

$R_{\text{m tc}}$ is the tensile strength at calculation temperature t_{c};

$R_{\text{p1,0 tc}}$ is the 1 %-proof strength $R_{\text{p1,0}}$ at calculation temperature t_{c};

$R_{\text{mT tc}}$[5, 6] is the creep rupture strength values for the specified lifetime T at calculation temperature t_{c}.

---

[5] T is the minimum specified design lifetime with a minimum of 100 000 h. If no lifetime is specified 200 000 h shall be taken. Should creep rupture strength values for times in excess of 100 000 h not be available, 100 000 h data may be used with S = 1,5 for internal pressure. In exceptional cases, where pressure parts are operated in the creep range for short duration (less than 10 000 h) e.g. discharge, relief lines, T may be reduced to 10 000 h with the safety factor S = 1,25.

[6] Creep rupture strength values for intermediate lifetimes shall be obtained by linear interpolation in a double logarithmic system.

6.3.4 Cast steel

The design stress f for cast steel shall be the lowest value obtained from the equation

$$f = min\left\{\frac{R_{m20}}{3,2};\frac{R_{p0,2\ tc}}{2,0};\frac{R_{mT\ tc}}{2,0}\right\} \tag{6.3-7}$$

where

$R_{m\ 20}$ is the tensile strength at room temperature (20 °C);

$R_{p0,2\ tc}$ is the 0,2 %-proof strength $R_{p0,2}$ at calculation temperature t_c;

$R_{mT\ tc}$ [7, 8] is the creep rupture strength values for the specified lifetime T at calculation temperature t_c.

6.3.5 Nodular graphite cast iron

The design stress *f* for annealed nodular graphite cast iron shall be the lowest value obtained from the equation

$$f = min\left\{\frac{R_{m20}}{4,8};\frac{R_{p0,2\ tc}}{3,0}\right\} \tag{6.3-8}$$

and for not annealed nodular graphite iron cast iron the lowest value obtained from the equation

$$f = min\left\{\frac{R_{m20}}{5,8};\frac{R_{p0,2\ tc}}{4,0}\right\} \tag{6.3-9}$$

where

$R_{m\ 20}$ is the tensile strength at room temperature (20 °C);

$R_{p0,2\ tc}$ is the 0,2 %-proof strength $R_{p0,2}$ at calculation temperature t_c.

6.3.6 Design stress for welded connections operating under creep condition

When the creep properties of the welded connection are known, the smallest of the design strengths of the welded connection and the two joined materials shall be used for loading at right angles to the weld seam.

When the creep properties of the welded connection are not known, but those of the filler material are known, the design strength for this loading shall be reduced by 20 % from the smaller of the design strengths of the joined materials. When the creep strength of the filler metal is not known, the joint strength shall be reduced by a further 20 %.

---

[7] T is the minimum specified design lifetime with a minimum of 100 000 h. If no lifetime is specified 200 000 h shall be taken. Should creep rupture strength values for times in excess of 100 000 h not be available, 100 000 h data may be used with S = 1,5 for internal pressure. In exceptional cases, where pressure parts are operated in the creep range for short duration (less than 10 000 h) e.g. discharge, relief lines, T may be reduced to 10 000 h with the safety factor S = 1,25.

[8] Creep rupture strength values for intermediate lifetimes shall be obtained by linear interpolation in a double logarithmic system.

6.3.7 Design stress for test pressure

At the test pressure $p_{t'}$, in the case of rolled and forged steels and in the case of austenitic steels with a specified minimum elongation A < 30%, the design stress $f_{t'}$ shall be

$$f_{t'} = \frac{R_{p0,2\ 20}}{S_{t'}} \tag{6.3-10}$$

where

$R_{p0,2\ 20}$ is the 0,2% proof strength at room temperature (20 °C) and

$S_{t'}$ is the safety factor for test (gauge) pressure with

2,2 for nodular graphite cast iron,

1,4 for cast steels and

1,05 for rolled and forged steels and for austenitic steels with a specified minimum elongation < 30 %,

and in the case of austenitic steels with a specified minimum elongation A ≥ 30 %, the design stress $f_{t'}$ shall be

$$f_{t'} = \frac{R_{p1,0\ 20}}{1,05} \tag{6.3-11}$$

where

$R_{p1,0\ 20}$ is the 1,0% proof strength at room temperature (20 °C) and

1,05 is the safety factor $S_{t'}$ for test (gauge) pressure for austenitic steels with a specified elongation A ≥ 30 %.

7 Cylindrical shells of drums and headers under internal pressure

7.1 Shell thickness

7.1.1 Requirements

The shell thickness after deduction of allowances, $e_{rs} = e_s - c_1 - c_2$ of drums and headers shall be the greatest of those required by the following:

a) a minimum of 9,5 mm for headers 300 mm outside diameter and above, and a minimum of 6 mm for headers below 300 mm outside diameter;

b) the requirements of 7.2 by applying 8.2 or 8.3.3 and 8.3.4;

c) the requirements of 7.3 and 7.4 (if applicable).

7.1.2 Required wall thickness including allowances

The required wall thickness including allowances shall be derived from

$$e'_s = e_{cs} + c_1 + c_2 \quad (7.1\text{-}1)$$

7.2 Basic calculation

7.2.1 Required wall thickness without allowances

The required wall thickness without allowances e_{cs}, in mm, of a cylindrical shell shall be determined by either of the following equations:

$$e_{cs} = \frac{p_c d_{is}}{(2 f_s - p_c) v} \text{ if } d_{is} \text{ is given} \quad (7.2\text{-}1)$$

or

$$e_{cs} = \frac{p_c d_{os}}{(2 f_s - p_c) v + 2 p_c} \text{ if } d_{os} \text{ is given} \quad (7.2\text{-}2)$$

where the following applies in addition to EN 12952-1:2001, Table 4-1:

— v is the minimum efficiency v_b for isolated or v_m for adjacent branches, respectively, isolated or adjacent openings in longitudinal, oblique or circumferential direction, determined in accordance with 8.2 by way of approximation or in accordance with 8.3.3 and 8.3.4.

Cladding for the purpose of metal wastage resistance shall not be taken into account.

The calculation of the required wall thickness of a main body reduced in strength by openings shall only be possible by iteration since strength reduction depends on the wall thickness.

The equivalent value of the stress in the shell shall be calculated using an inversion of Equations (7.2-1) or (7.2-2).

Thermal stress caused by through-the-wall-temperature-difference shall be taken into account in accordance with the requirements of Clause 13.

7.2.2 Different thickness

Where a shell is comprised of plates of different thickness, the plate centrelines at any cross-section shall lie on a true circle.

In determining the thickness of each plate, twice the inner radius shall be used for d_{is} in Equation (7.2-1), or twice the outer radius for d_{os} in Equation (7.2-2).

7.2.3 Fabrication tolerances

When calculating the shell thickness, the fabrication tolerances for the nominal diameter (as ordered) shall not be taken into account. When considering wall thickness reductions, it shall be assumed that these reductions occur on the inside if shells are ordered with outside diameter, and on the outside if shell are ordered with inside diameter.

7.3 Combined stress in drum or header shells

7.3.1 General

Notwithstanding the thickness of shells calculated in accordance with 7.2, calculations shall be made to ensure that in no case does the average ligament stress resulting from the combination of stresses arising from internal pressure, the self-weight of the drum or header and its contents, and all externally applied loads, exceed the design stress f (see 6.3) in any line of holes whether longitudinal, circumferential or diagonal.

The average ligament stress in ligaments between holes on a circumferential line shall be the sum of the longitudinal stresses calculated in accordance with 7.3.2 and 7.3.3, increased by $p_c/2$. The average ligament between holes on a diagonal line shall be calculated in accordance with 7.3.4.

7.3.2 Stress from longitudinal loads

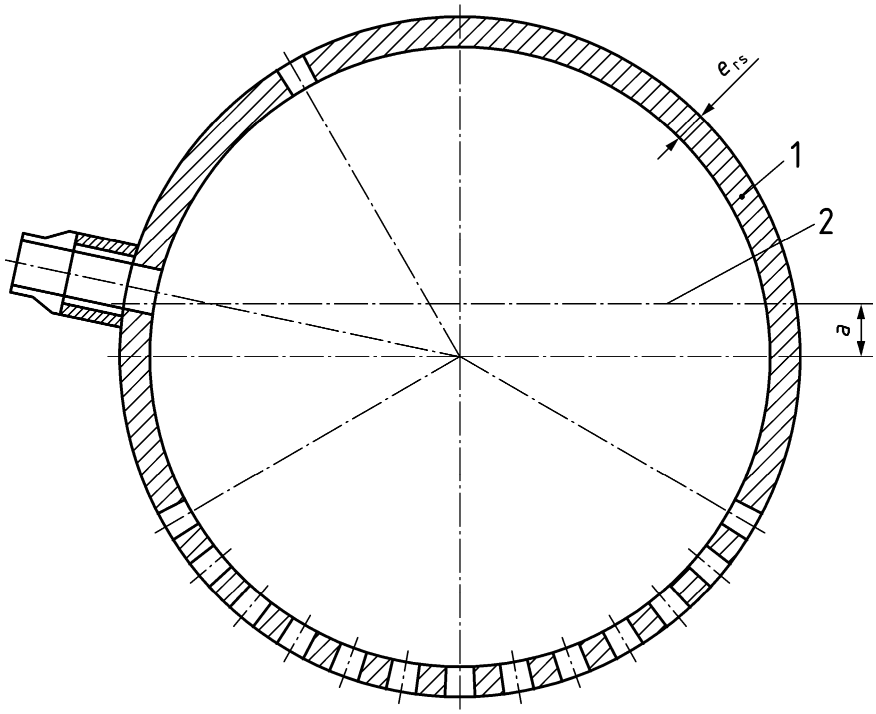
The longitudinal stress f_d averaged over the shell cross-section, caused by internal pressure and the longitudinal component of externally applied loads W, shall be calculated as follows:

$$f_d = \frac{\pi\ p_c\ d_{is}^2}{4\ A_s} + \frac{W}{A_s} \qquad (7.3\text{-}1)$$

where A_s is the shell cross-sectional area at the section considered, see Figure 7.3-1. When openings in the shell are reinforced by thickening the walls of branches in accordance with 8.1 to 8.3, the areas of compensation in the branch wall can be included in A_s.

The following condition shall apply at any circumferential section through the shell:

$$f_d + \frac{p_c}{2} \leq f_s \qquad (7.3\text{-}2)$$



Key
1 shell cross-sectional area
2 neutral axis

Figure 7.3-1 — Net area of shell cross section subject to longitudinal loads

7.3.3 Longitudinal bending stress

The bending moment M_R at any section shall be the algebraic sum of the resultant bending moments due to the eccentricity of the end pressure M_E and due to self weight, weight of water and externally applied loads M_W.

$$M_R = M_E + M_W \tag{7.3-3}$$

The resultant bending moment M_E due to the eccentricity a of the end pressure shall be calculated from:

$$M_E = \frac{p_c\, d_{is}^2\, \pi\, a}{4} \tag{7.3-4}$$

where

a is the eccentricity of the net cross section.

The resultant bending moment M_W due to externally applied loads shall be calculated by treating the shell as a beam carrying the externally applied loads including its own weight and that of the contents under working conditions. M_W shall be the bending moment at the section subjected to the greatest bending from this cause.

The stress due to bending f_M shall be calculated as follows:

$$f_M = \frac{M_R\, Y}{I_A} \tag{7.3-5}$$

where

Y is the distance from the neutral axis of the net cross section to the extreme mid point of the shell thickness in mm;

I_A is the second moment of area.

The values of M_R, Y and I_A shall depend on the direction of the neutral axis of bending. The direction shall be chosen so that the resulting stress f_M is a maximum.

Taking f_d shown in 7.3.2 into consideration the following condition shall apply:

$$f_d + f_M + \frac{p_c}{2} \leq f_s \tag{7.3-6}$$

7.3.4 Evaluation of ligament stress for inclined ligaments in drums

Where bending stresses due to weight, externally applied loads and eccentricity of end pressure are not negligible, the stress on inclined ligaments which are on a line which is at an angle Φ to the longitudinal axis of the drum as shown in Figure 8.3-3 shall be evaluated as follows:

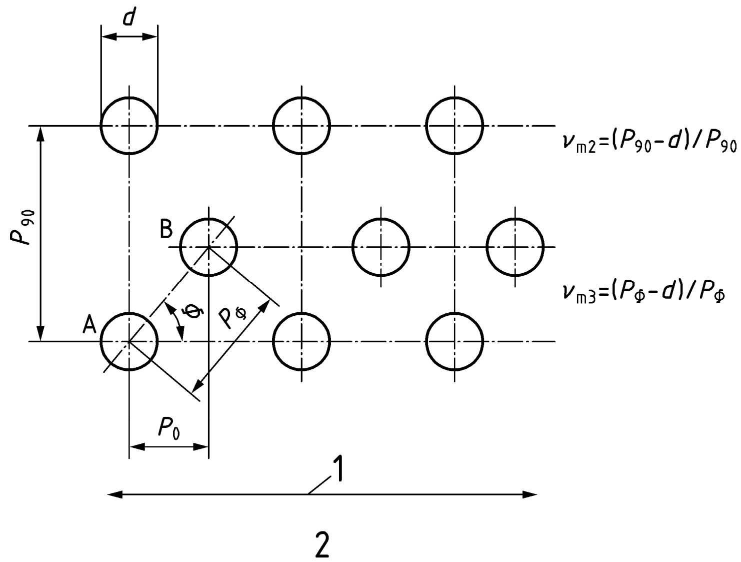
Let the circumferential stress f_1 on cross sectional area of longitudinal section be:

$$f_1 = \frac{p_c\, d_{is}}{2 e_{rs}} \tag{7.3-7}$$

Let the longitudinal stress f_2 on cross sectional area of circumferential section be:

$$f_2 = v_{m2}\left(f_d + f_M\right) \tag{7.3-8}$$

where v_{m2} is the ligament efficiency on a circumferential section at the point where f_d and f_M act, see Figure 7.3-2.



Key
1 longitudinal axis of shell
2 developed radial view on shell mid-thickness

Figure 7.3-2 — Tube holes with diagonal ligaments – efficiency factors for ligament AB

Where the ligament under consideration is between holes of equal size the apparent ligament efficiency on the diagonal line shall be given by:

$$v_{m3} = 1 - \frac{d}{P_{\Phi}} \tag{7.3-9}$$

Where holes are of unequal size, the coefficient for the diagonal line shall be given by:

$$v_{m3} = 1 - \frac{(d_1 + d_2)\cos\Phi}{2P_0} \tag{7.3-10}$$

The average direct stress f_A on the ligament shall be given by:

$$f_A = \frac{1}{v_{m3}}\left\{\frac{f_1 + f_2}{2} + \frac{(f_1 - f_2)\cos 2\Phi}{2}\right\} \tag{7.3-11}$$

The average transverse stress f_B on the ligament shall be given by:

$$f_B = v_{m3}\left\{\frac{f_1 + f_2}{2} - \frac{(f_1 - f_2)\cos 2\Phi}{2}\right\} \tag{7.3-12}$$

The average shear stress f_C on the ligament shall be given by:

$$f_C = \frac{(f_1 - f_2)\sin 2\Phi}{2v_{m3}} \tag{7.3-13}$$

The stress f_3 in MPa on the ligament shall be given by:

$$f_3 = \frac{1}{2}\left| f_A + f_B + \sqrt{(f_A - f_B)^2 + 4 f_C^2} \right| \tag{7.3-14}$$

$$f_3 + \frac{p_c}{2} \leq f_s \tag{7.3-15}$$

7.4 Boiler drum supports

Where large diameter but relatively thin drums are carrying heavy loads on saddles or slings, Equation (7.4-1) shall be used as a guide to assess whether precise calculations are needed to determine local stresses:

$$f_c = 0{,}78\left(\frac{W}{l_s e_{rs}}\right)\sqrt{\left(\frac{r_{ms}}{e_{rs}}\right)} \leq f_s \tag{7.4-1}$$

where

W is the load on the saddle or sling, in N;

l_s is the horizontal length of the saddle at right angles to the axis of the drum in mm, but not greater than the chord described by 120°.

Where application of this equation suggests that the design stress is exceeded, reference shall be made to the appropriate methods of analysis, e.g. as given in EN 13445-3:2009.

7.5 Other stresses in cylindrical shells

In addition to the requirements of 7.3 and 8.1.1 other local stresses shall also be considered.

EN 13445-3:2009 gives methods for calculating stresses caused by loads applied to nozzles and attachments.

8 Openings and branches in cylindrical shells of drums and headers and integral tubes[9)]

8.1 General

8.1.1 Requirements for the ligament efficiency of the main body with openings and branches

8.1.1.1 Ligament efficiency of the main body

For cylindrical shells with openings, the ligament efficiency of the main body shall be satisfied by the following:

a) by increasing the wall thickness of the main body compared with that of the cylindrical shell without openings. This wall thickness shall be available at least up to the length l_{rs} measured from the edge of the opening; see Figure 8.1-1 (l_{rs} see 8.1.2). Where there is a branch, the cylindrical length of the main body up to any adjacent butt weld in it shall be $l_{so} \geq e_{rs}$ (see Figures 8.1-2 and 8.1-3) or

---

9) This clause specifies the design rules for openings and branches in cylindrical shells. Dimensions do not include allowances for wall thickness negative tolerances and metal wastage.

b) by branches, measured on a length l_{b1} from the outside surface of the main body wall, which have been provided with a wall thickness in excess of that required on account of the internal pressure, without or in connection with an increase in main body wall thickness (see Figures 8.1-2 to 8.1-4). The welded joint between the main body and branch shall be a full-strength weld where in case of branches a residual gap ≤ 1,5 mm can be present. A wall thickness ratio of e_{rb}/e_{rs} up to and including 2 shall be permissible for $d_{ib} \leq 50$ mm. This shall also apply to branches with $d_{ib} > 50$ mm insofar as the diameter ratio $d_{ib}/d_{is} \leq 0,2$. For branches with $d_{ib} > 50$ mm and a diameter ratio $d_{ib}/d_{is} > 0,2$, e_{rb}/e_{rs} shall not exceed unity. These conditions do not apply to access and inspection openings.

   Expanded or set-through and seal-welded-only branches or branches attached to the main body by fillet welds with a residual root gap > 1,5 mm shall not be used in a creep rupture stress range and are not subject to dynamic loading. Such branches shall not be regarded as contributing to the reinforcement.

   The cylindrical length of branches up to the butt weld between tube and branch shall be $l_{b0} \geq e_{rb}$ (see Figures 8.1-2 and 8.1-3).

   For branches with $d_{ib}/d_{is} \geq 0,7$, calculated with a design stress derived from R_m or $R_{p0,2\ tc}$ reference shall be made to 8.3.3.4. For branches calculated with design stress derived from creep rupture strength values, the requirement shall be $d_{ib}/d_{is} \leq 0,8$ and $l_{b1} \geq l_{rb}$ and, in addition, the condition $e_{rb} \geq e_{rs} \times d_{ib}/d_{is}$ shall be satisfied for $d_{ib}/d_{is} \geq 0,5$ (for l_{rb} see 8.1.2 and for l_{b1} see Figure 8.1-2).

   In general, special emphasis shall be placed on smooth wall thickness transitions. Wall thickness transitions shall be made with an angle ≤ 30° (see Figure 8.1-2). The reinforcement of openings by inside reinforcing plates or pads shall not be permitted;

c) by increasing the wall thickness in highly loaded zones within the area of the opening (see Figures 8.1-5 and 8.1-6), which can be obtained by forging or forging and subsequent machining;

d) by reinforcing pads analogous to a) (see Figures 8.1-7 and 8.1-8).

   The requirements of Clause 13 are not applicable to this type of reinforcement. Reinforcing pads shall have close contact with the main body;

e) for branches the design drawings shall contain an indication of the required design type of the welded joint between the branch and the shell, like

   1) root branch weld either machined out or ground over or created from the inside by means of TIG;

   2) root of branch weld neither machined out nor ground over (residual gap ≤ 1,5 mm);

   3) residual gap of branch weld not specified.

8.1.1.2 Special requirements

In the case of elliptical or obround access and inspection openings, it shall be assumed that the ratio of major to minor axis does not exceed 1,5. For elliptical or obround openings in cylindrical shells the dimension extending in the direction of the shell axis shall be taken as the diameter for design purposes (for oblique nozzles see 8.3.3.3).

The calculation procedure assumes that the transitions shall show a largely notch-free surface[10]. Sharp corners shall be rounded to give a smooth transition.

---

[10] Welded joints are considered to be largely notch-free if they are free from undercutting and shrinkage grooves (root concavity), within the limits given in EN 12952-6.

Openings shall be located at an adequate distance from the welds (longitudinal and circumferential welds) of the main body. The distance shall be considered adequate if the outer edge of a branch or welded-on reinforcement, on a main body with a thickness $e_{rs} \leq 25$ mm, is at a distance of $2e_{rs}$, and on a main body with a thickness $e_{rs} > 25$ mm, the distance is at least 50 mm from the weld edge.

Machining of holes through longitudinal and circumferential welds of drums and headers shall be permitted, provided the bore of the hole clears the edge of the main seam weld, and the main seam welds have been subjected to non-destructive examination required by this European Standard. In addition, the weld cover pass of the welded nozzle, and the main seam weld local to the nozzle/hole shall be ground to remove notches.

Internal edges of nozzle hole in drums shall be rounded with a radius of ≥ 3 mm except those holes are associated with expanded tubes.

8.1.1.3 Extruded openings

Extruded openings shall only be allowed if $d_{ib}/d_{is} \leq 0{,}8$. Where such openings are to be subjected to creep rupture stresses, they shall be limited to $d_{ib}/d_{is} \leq 0{,}7$.

The efficiency factor v_b to be used in the calculation shall be in accordance with 8.1.4.3.

Extruded openings which are calculated using creep rupture strength values shall be in accordance with 8.1.4.4.

8.1.1.4 Forged branches

The restrictions imposed on extruded openings shall not apply to forged branches in accordance with Figures 8.1-5 and 8.1-6 provided that the increase in material thickness can be ensured.

8.1.1.5 Conical transitions

Conical flanges and rounded openings shall be substituted in the calculation by sectioned areas of equal cross-sectional area in accordance with Figure 8.1-9.

8.1.1.6 Set-through nozzles

The inside diameter of the branch shall not exceed one-third of the inside diameter of the shell.

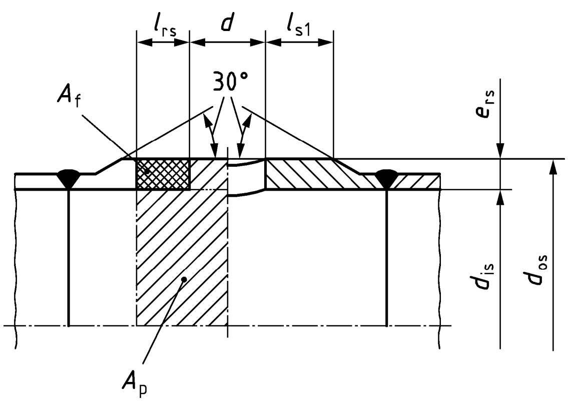


Figure 8.1-1 — Reinforcement by increased wall thickness of the main body with opening

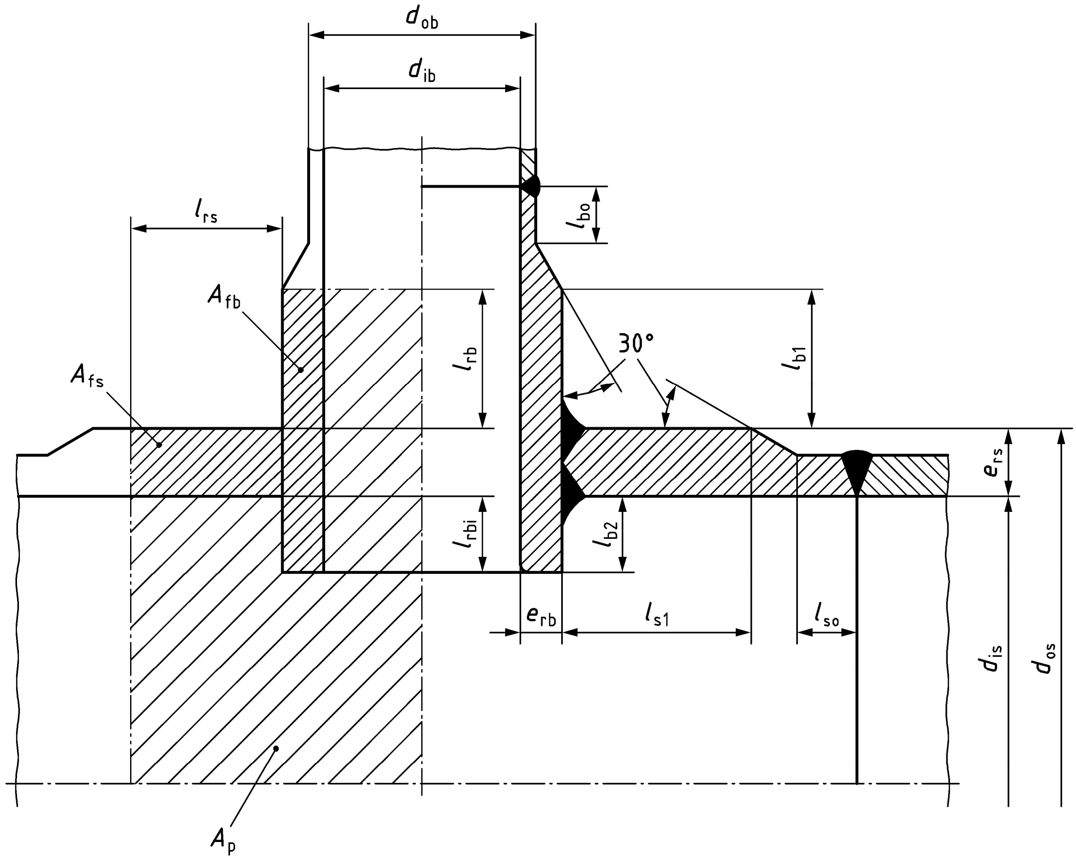
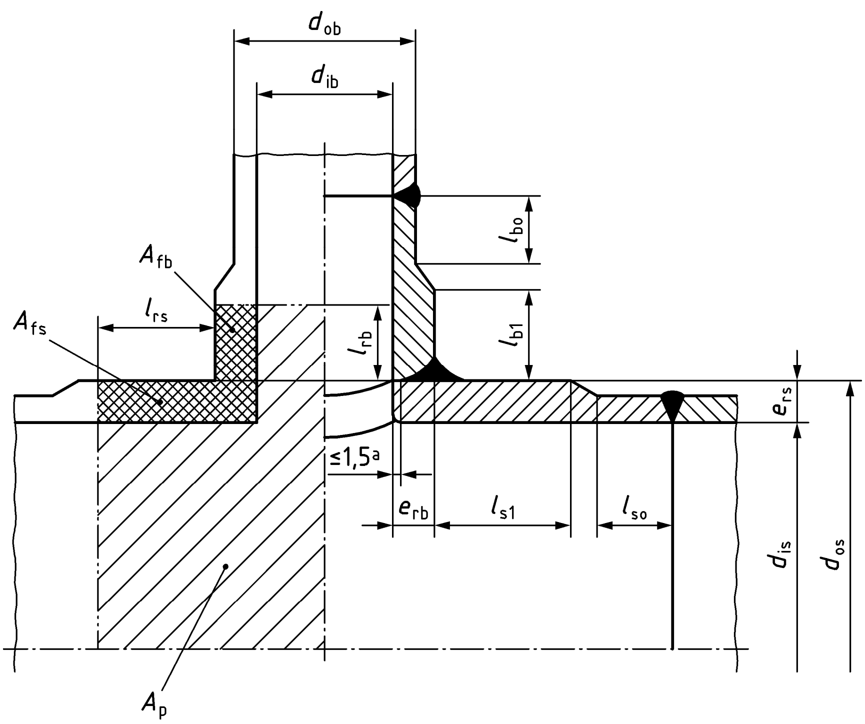


Figure 8.1-2 — Reinforcement by set through and full penetration welded branch



Key

[a] residual gap

Figure 8.1-3 — Reinforcement by welded on branch

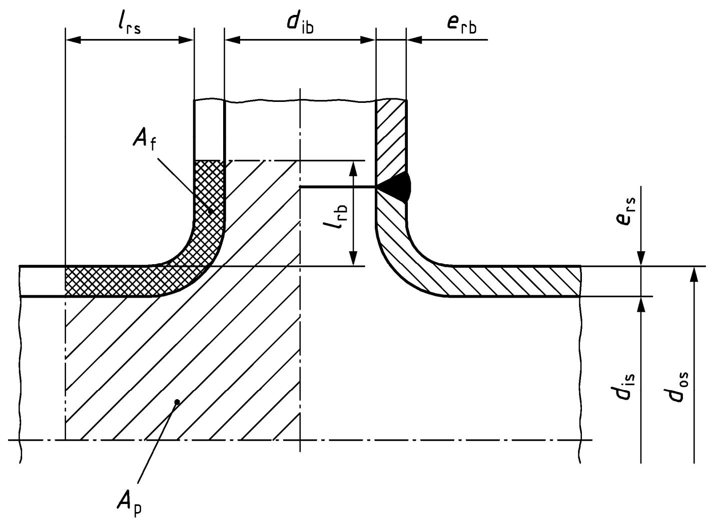


Figure 8.1-4 — Reinforcement by branch welded to extruded main body

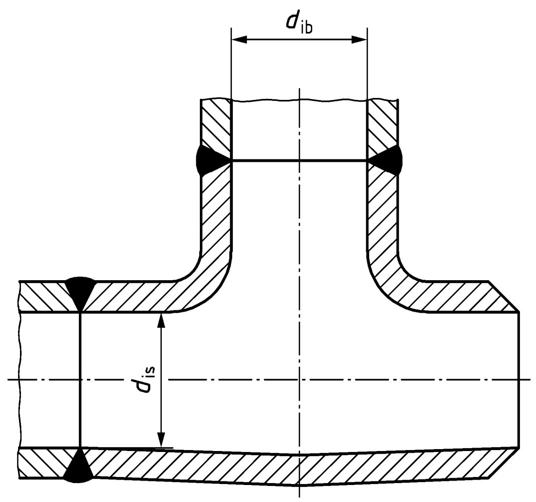


Figure 8.1-5 — Die forged branch

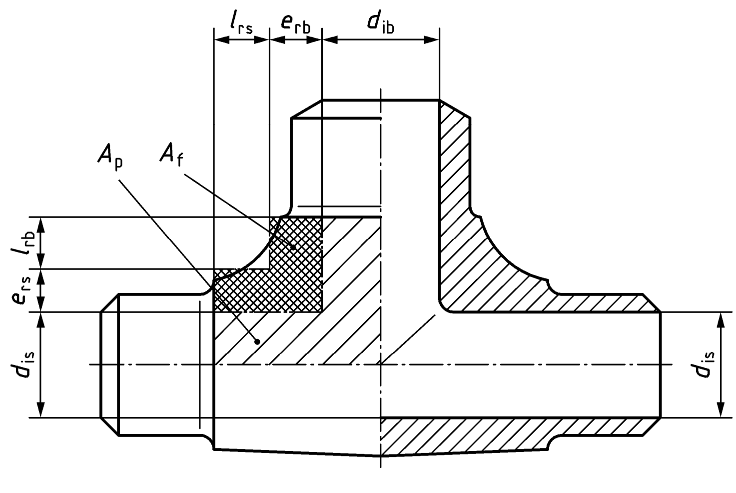


Figure 8.1-6 — Branch forged from solid material, subsequently bored and turned

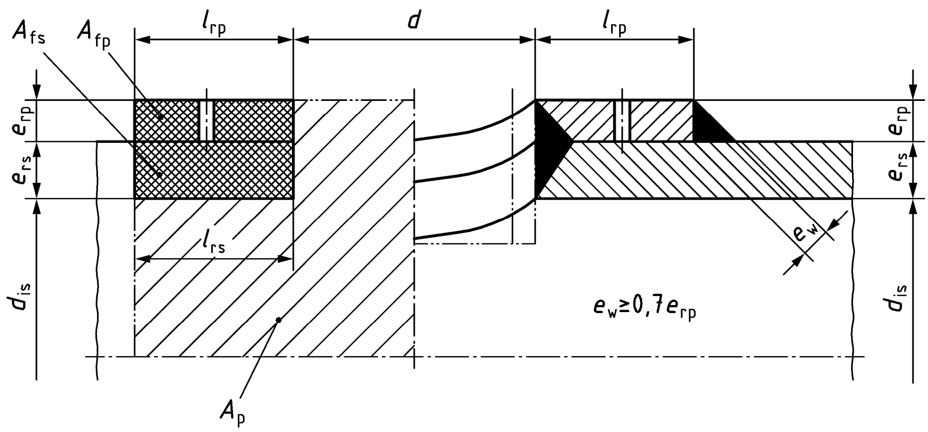
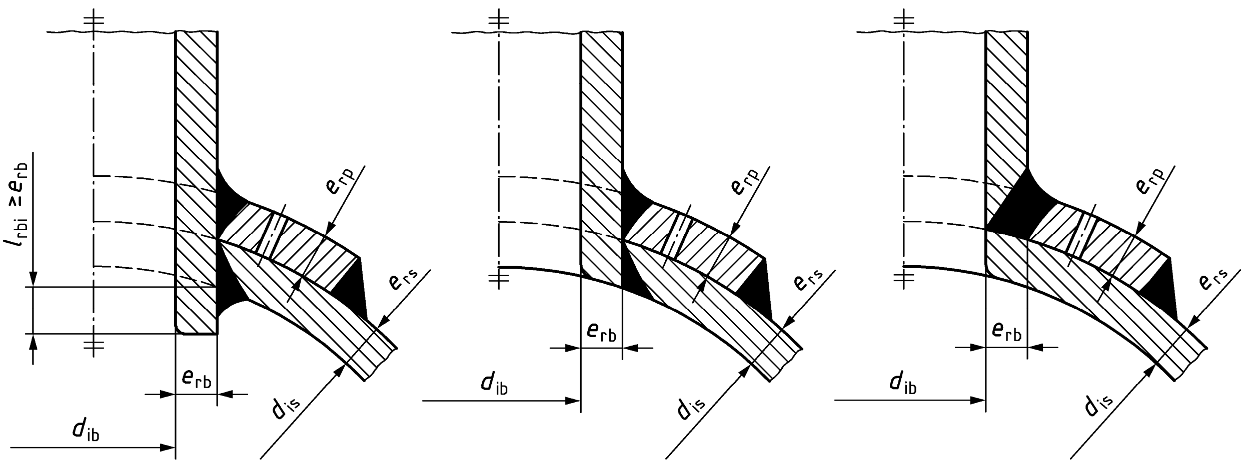


Figure 8.1-7 — Opening with reinforcing pad (allowable for t_c ≤ 250 °C)



design a	design b	design c

Figure 8.1-8 — Opening with reinforcing pad and full penetration set-through and set on welded branch (welds shown for cylindrical shells accessible from the inside)

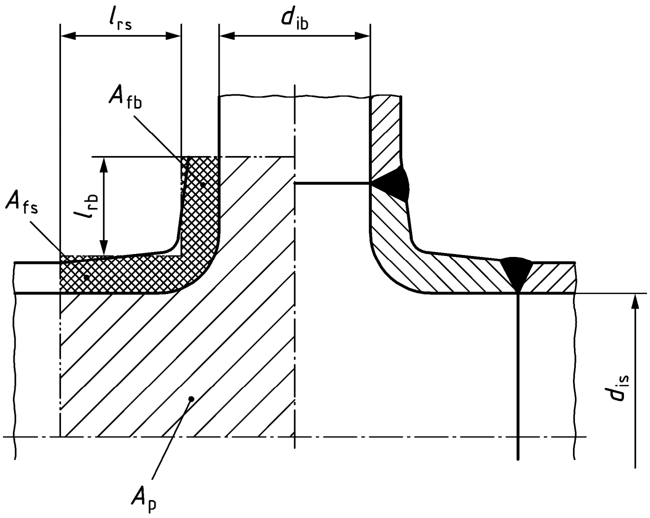


Figure 8.1-9 — Branch with conical transitions and fillets

8.1.2 Effective lengths for calculation of efficiencies of components

For the calculation of efficiencies by the approximate method, as described in 8.2 and the calculation of isolated and adjacent branches, described in 8.3, effective lengths are required which shall be used for the main body.

For openings, see Figures 8.1-1 to 8.1-3

$$l_{rs} = \min\begin{Bmatrix} \sqrt{(d_{is} + e_{rs})\, e_{rs}} \\ l_{s1} \end{Bmatrix} \qquad (8.1\text{-}1)$$

for l_{s1} see Figures 8.1-1 to 8.1-3,

and for the nozzle with $\psi \geq 45°$, see Figures 8.3-1 and 8.3-2

$$l_{rb} = \min\begin{Bmatrix} \sqrt{(d_{ib} + e_{rb})\, e_{rb}} \\ l_{b1} \end{Bmatrix} \quad \text{for external projection;} \qquad (8.1\text{-}2)$$

$$l_{rbi} = \min\begin{Bmatrix} 0{,}5\sqrt{(d_{ib} + e_{rb})\, e_{rb}} \\ l_{b2} \end{Bmatrix} \quad \text{for internal projection.} \qquad (8.1\text{-}3)$$

For openings without branches l_{rb} shall be omitted.

8.1.3 Conditions for isolated openings

Adjacent openings shall be treated as isolated openings if, for the centre distance P_Φ in accordance with Figure 8.3-3, the following condition is satisfied:

$$P_\Phi \geq \frac{\left(\dfrac{d_{ib1}}{2} + e_{rb1}\right)}{\sin\psi_1} + \frac{\left(\dfrac{d_{ib2}}{2} + e_{rb2}\right)}{\sin\psi_2} + 2\sqrt{(d_{is} + e_{rs})\, e_{rs}} \qquad (8.1\text{-}4)$$

For openings without a branch e_{rb} = 0 and ψ = 90 °.

8.1.4 Requirements for design of branches

8.1.4.1 Main body with lower design stress than the branches

If the main body, the branch and the additional reinforcement consists of materials with different design stresses and the material of the main body has the lowest design stress value f_s, this value shall be used for all materials to calculate the reinforcement of the branch.

8.1.4.2 Branches or reinforcing with lower design stress than the main body

If the material used for the branch or the reinforcing pad has a lower design stress f_b or f_p respectively than the main body design stress f_s, this design stress f_b or f_p shall be taken into account when using the equations provided for this case.

8.1.4.3 Extruded branches

If in the case of extruded branches (see Figures 8.1-4 and 8.1-5), the efficiencies for isolated branches are calculated as conventional nozzles (see Figure 8.1-3) without considering the extruded shape, the efficiency v_b shall be multiplied by 0,9.

8.1.4.4 Extruded branches in creep region

In the case of extruded branches (see Figure 8.1-4) to be used in the creep region the creep rupture strength values used in the calculation shall be multiplied by 0,9.

8.1.4.5 Special case

The hole diameter d (see Figure 8.1-1) may be used in the calculation even if the hole diameter in the main body is less than the inside diameter d_{ib} (see Figure 8.1-3) of nozzle, if the reinforcement is only compared by increasing the wall thickness of the main body.

8.1.5 Requirements for the design of reinforcing pads

8.1.5.1 General

Reinforcing pads shall not be used where there is a possibility of severe corrosion or oxidation or of large temperature gradients across the thickness of the shell. Reinforcing pads shall be not permitted on the inside of the vessel.

8.1.5.2 Pressure considerations

When reinforcing pads are used for the reinforcement of penetrations or openings, the following conditions shall be observed. Reinforcing pads shall only be used for design temperatures $\leq$ 250 °C and shall not be used for cyclic conditions. Annex B does not apply to this form of design. Such reinforcements have only little effect on bending moments acting upon the branch.

a) the ratio d_{ib}/d_{is} of the branch diameter to the cylinder diameter shall not be greater than ¼ unless the adequacy of the design is demonstrated by experience or by a hydrostatic proof test in accordance with Clause 12;

b) the width of the pad shall not be less than $l_{rs}/2$ where l_{rs} is the distance along the shell within which the shell thickness is assumed to contribute to the reinforcement. The effective width, Figures 8.1-7 and 8.1-8, shall only be used in the equations in 8.3.3.3 with a value not exceeding $l_{rp} = l_{rs}$;

c) the thickness of the pad e_{rp} shall not exceed 40 mm or the actual shell thickness e_{rs} whichever is the lower;

d) the thickness of the pad shall not be less than $e_{rs}/4$ with a maximum of 40 mm, where e_{rs} is the shell thickness;

e) the weld preparations shall be as shown in Figures 8.1-7 or 8.1-8. The pads shall have close contact with the shell.

8.1.5.3 Non-pressure considerations

When reinforcing pads are used for other than pressure considerations, such as for the connection of a support or mounting in order to avoid excessive local stresses in the drum shell, then the requirements of 8.1.5.2 shall not apply.

The leg length of the fillet weld attaching the pad to the shell shall not exceed the drum thickness e_{rs}.

8.1.5.4 Tell-tale holes

Where reinforcing plates are fitted, they shall be provided with tell-tale holes to avoid the trapping of gases during welding.

8.2 Efficiency factor, calculation by way of approximation, and maximum diameter of unreinforced openings

8.2.1 General

Exact solutions are given by the equations in 8.3.3 and 8.3.4. Application of the equations in 8.2.3 and 8.2.4, except Equation (8.2-3) may be conservative as any reinforcement from the branch is not taken into account.

8.2.2 Allowable efficiency and maximum diameter of unreinforced opening

Rearranging Equation (7.2-1) the allowable efficiency v_{all} shall be calculated for the available wall thickness e_{rs} of a main body as follows

$$v_{all} = \frac{p_c d_{is}}{(2 f_s - p_c) e_{rs}} \tag{8.2-1}$$

For this efficiency coefficient the greatest outside diameter d_{ob} of an isolated branch shall be obtained when its wall thickness can only withstand the internal pressure

$$d_{ob\ max} = 2 \left(\frac{l_{rs}}{v_{all}} - l_{rs} \right) \tag{8.2-2}$$

In this case the available average stress f_a shall be equal to the allowable stress f_s of the main body.

8.2.3 Isolated openings

The equations in this clause shall apply to single openings, or if there is more than one opening, only if Equation (8.1-4) is satisfied. In the case of more than one opening where Equation (8.1-4) is not satisfied, reference shall be made to 8.2.4.

However, isolated unreinforced openings with diameter d in cylindrical shells shall be permitted if they comply with the following conditions:

$$d \leq 0{,}14\ l_{rs} \tag{8.2-3}$$

and

$$e_{rs} \leq 0{,}1\ d_{os} \tag{8.2-4}$$

where l_{rs} shall be calculated in accordance with 8.1.2.

Equations (8.2-3) and (8.2-4) shall also be considered valid for counter bored holes and for partial penetration holes, even if the Equations (8.2-6) or (8.2-11) or the more exact calculations in accordance with 8.3.3 recommend a smaller diameter d than given in Equation (8.2-3).

Where a tube with an outside diameter d_{ob} is attached to an opening (see Figure 8.2-1) and the tube is capable of withstanding the internal calculation pressure on account of its wall thickness e_{rb}, the efficiency factor v_b of this opening in the main body shall be calculated by

$$v_b = 2l_{rs}/(2l_{rs} + d_{ob}) \tag{8.2-5}$$

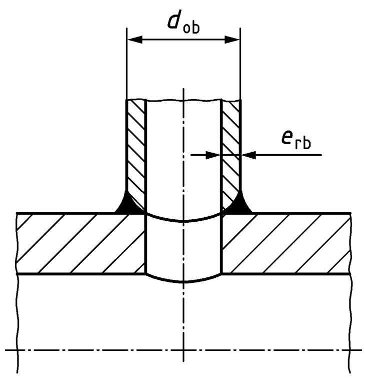


Figure 8.2-1 — Tube attached to main body

If the diameter d_{is} and the wall thickness e_{rs} of the main body are already determined, an isolated hole for a tube with a maximum outside diameter d_{ob} shall be permitted.

$$d_{ob} \leq 2l_{rs}\left(\frac{2\,e_{rs}}{d_{is}}\left(\frac{f_s}{p_c}-\frac{1}{2}\right)-1\right) \tag{8.2-6}$$

In case of expanded (see Figure 8.2-2) or set-in and seal-welded-only tubes, the tube hole diameter d in the main body shall be used in Equations (8.2-5) or (8.2-6) instead of d_{ob}.

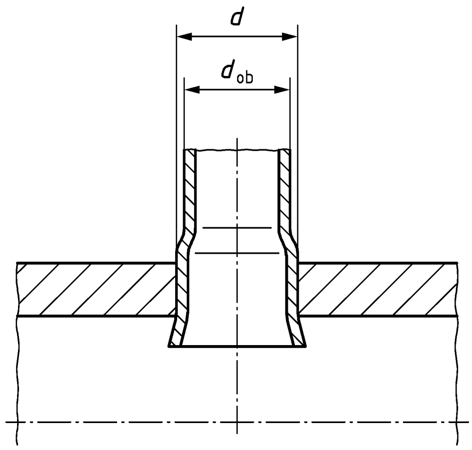
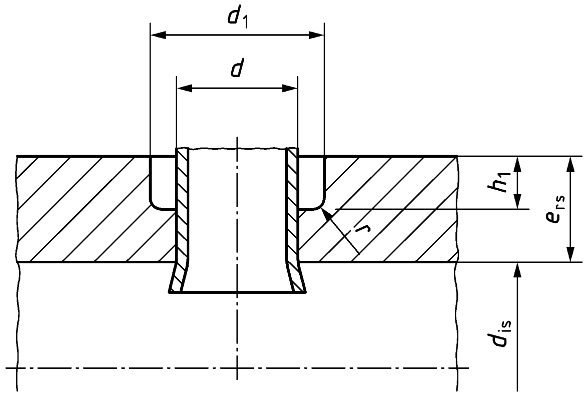


Figure 8.2-2 — Expanded tube

In the case of holes with stepwise reduced diameter, e.g. for expanded tubes in accordance with Figure 8.2-3 the equivalent diameter to be used in the efficiency calculation (8.2-5) instead of d_{ob}, shall be

$$d_{equiv} = \frac{d_1\,h_1 + d(e_{rs} - h_1)}{e_{rs}} \tag{8.2-7}$$



NOTE The counter boring may be made either from the outside, as shown, or from the inside of the shell.

Figure 8.2-3 — Counter boring for expanded tubes

The strength condition shall be satisfied, if this equivalent diameter does not exceed the limit for d_{ob} in Equation (8.2-6).

In addition, the following requirements shall be met:

a) if there is more than one opening, the condition in Equation (8.3-13) shall be fulfilled. The areas can be determined analogous to Figure 8.3-3;

b) the remaining wall thickness at the bottom of the hole shall not be less than

$$e_{rs} - h_1 \geq 0{,}4\, d_1 \sqrt{\frac{p_c}{f_s}} \tag{8.2-8}$$

and

$$e_{rs} - h_1 \geq 0{,}65\, d_1 \frac{p_c}{f_s} \tag{8.2-9}$$

c) the edge of the hole shall be radiused as shown in Figures 8.2-3 and 8.2-4.

NOTE Partial penetrating holes may be made either from the outside, as shown, or from the inside of the shell.

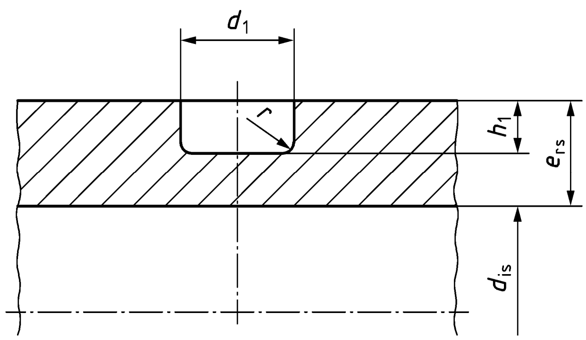


Figure 8.2-4 — Partial penetrating hole

In the case of partial penetrating holes (see Figure 8.2-4) the equivalent diameter to be used in efficiency calculation (8.2-5) instead of d_{ob} shall be

$$d_{\text{equiv}} = \frac{d_1 h_1}{e_{\text{rs}}} \tag{8.2-10}$$

Thus the allowable diameter of the hole is:

$$d_1 = \frac{2\, l_{\text{rs}}\, e_{\text{rs}}}{h_1}\left[\frac{2\, e_{\text{rs}}}{d_{\text{is}}}\left(\frac{f_{\text{s}}}{p_{\text{c}}} - \frac{1}{2}\right) - 1\right] \tag{8.2-11}$$

In addition, the requirements a), b) and c) for holes with stepwise reduced diameter as shown in Figure 8.2-3 shall be satisfied.

8.2.4 Adjacent openings

Where the condition for the centre distance P_{Φ} of adjacent openings in accordance with 8.1.3 is not met, and tubes with an outside diameter d_{ob} are connected to the openings, with the tubes only being capable of withstanding the internal pressure on account of their wall thickness e_{rb}, the efficiency factor of adjacent branches shall be derived as follows:

$$v_{\text{m}} = \frac{2\left(P_{\Phi} - d_{\text{ob}}\right)}{\left(1 + \cos^2 \Phi\right) P_{\Phi}} \leq 1 \tag{8.2-12}$$

i.e. for longitudinal pitch P_0 with $\Phi = 0°$.

$$v_{\text{m}} = \frac{P_0 - d_{\text{ob}}}{P_0} \tag{8.2-13}$$

Where the outside diameters of tubes of adjacent branches differ from each other, the following shall apply:

$$d_{\text{ob}} = \frac{d_{\text{ob}\,1} + d_{\text{ob}\,2}}{2} \tag{8.2-14}$$

8.3.4 may be used instead of the calculation by approximation used in this clause.

In the case of partially penetrating holes (see Figure 8.2-4) the equivalent diameter d_{equiv} from Equation (8.2-10) can be used in place d_{ob}, d_{ob1} or d_{ob2} as appropriate.

8.3 Design of openings and branches in cylindrical shells (efficiency and reinforcement)

8.3.1 Symbols and abbreviations

In addition to the symbols given in EN 12952-1:2001, Table 4-1, those shown in Figures 8.3-1 to 8.3-5 shall be used.

8.3.2 Requirements for application

8.3.2.1 Openings

The rules specified in 8.3.3 and 8.3.4 shall be regarded as valid for the design of circular, elliptical and obround[11] openings and nozzles (including oblique nozzles) arranged singly or in groups, in cylindrical shells, provided that the following conditions shall be satisfied.

a) Openings and nozzles normal to the shell:
The ratio of the major to minor axes of the opening does not exceed 2.

b) Oblique nozzles:
The nozzle is of circular cross section and the angle between the axis of the nozzle and a line normal to the surface does not exceed 45°.

c) All nozzles:
External forces and moments applied to the nozzle are not significant. If this is not the case, EN 13445-3:2009 shall be used to calculate and assess the resulting stresses.

The nozzle inside diameter d_{ib} (see Figure 8.1-3) of the reinforced nozzle with the exceeded wall thickness shall be used in the calculation even if the hole diameter d in the main body is less than the inside diameter d_{ib} of the nozzle.

Nozzle connections with a residual gap greater 1,5 mm or set-through and seal-welded-only branches, shall be considered as openings without branches. The pressurized area in the tube hole shall be considered except for the case of set-through tubes with an inside seal weld.

8.3.2.2 Branches

The design of branches shall be governed by the following:

a) the ability to withstand design pressure. For this purpose the minimum thickness of the branch shall be calculated in accordance with Clause 11;

b) compensation requirements for openings in the main pressure part, which shall be determined for cylindrical shells in accordance with 8.3.3 or 8.3.4;

c) the ability to withstand superimposed loading due to connected tubing or fittings. In no case shall the wall thickness of the branch be smaller than required in 8.3.3.1;

d) the minimum thickness e_{rb} of the branch without allowances shall not be less than that in accordance with Equation (8.3-1) below, where d_{ob} is the outside diameter of the branch (in mm). The requirements of this subclause shall not apply to tube stubs, which are the direct continuation of the heating tubes:

$$e_{cb} = 0{,}015\, d_{ob} + 3{,}2 \qquad (8.3\text{-}1)$$

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[11] An obround opening is an opening or hole which is neither circular nor elliptical and is formed by the bending of two radii to give a rounded opening or hole whose major and minor axes are different.

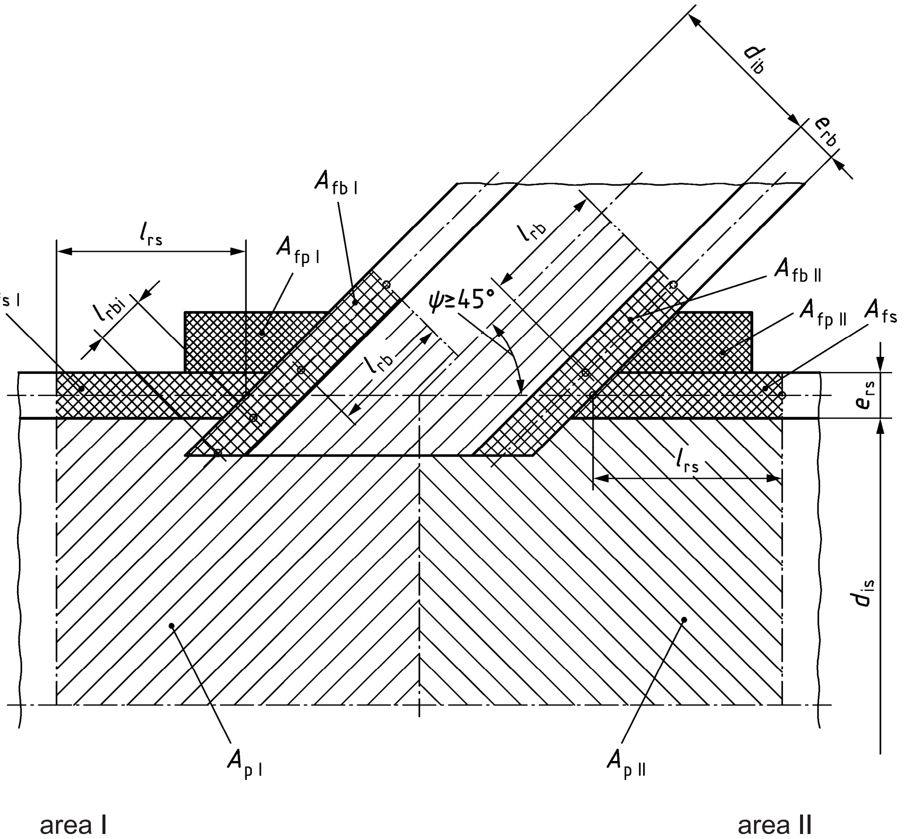
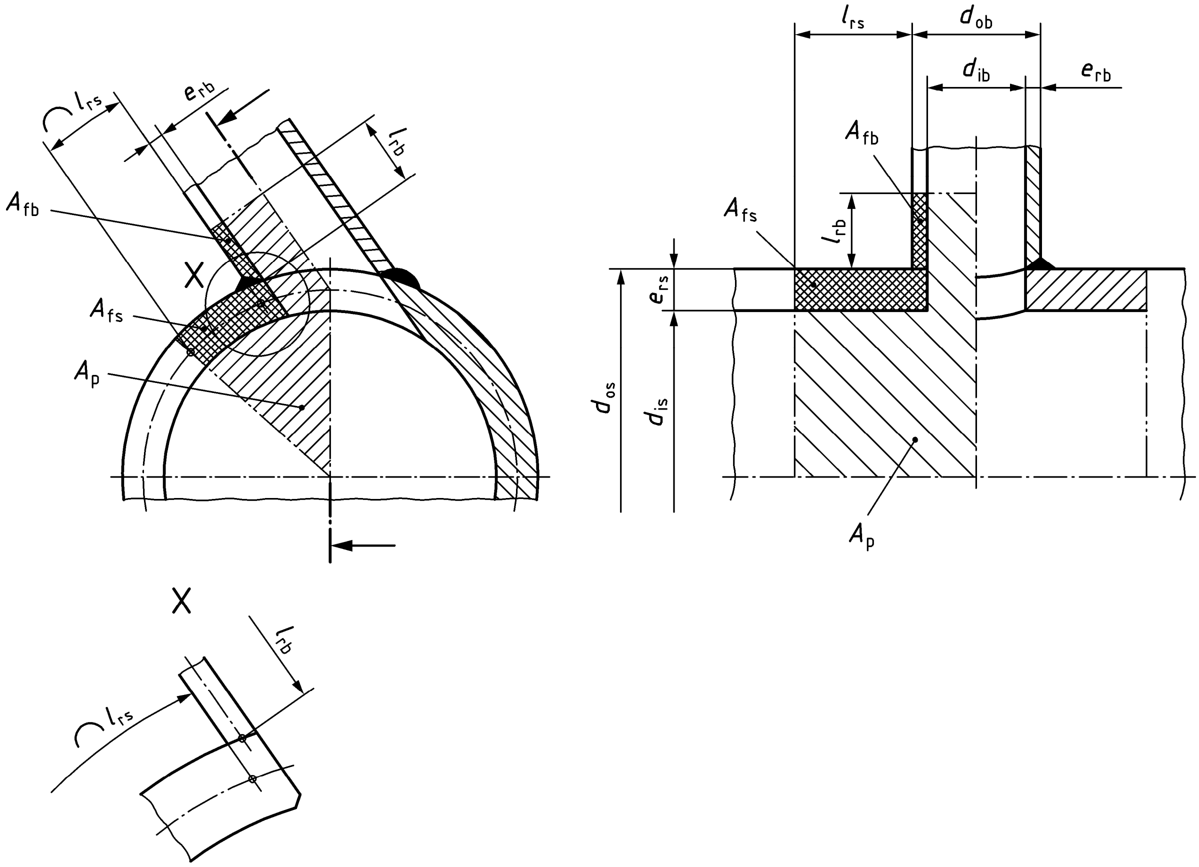
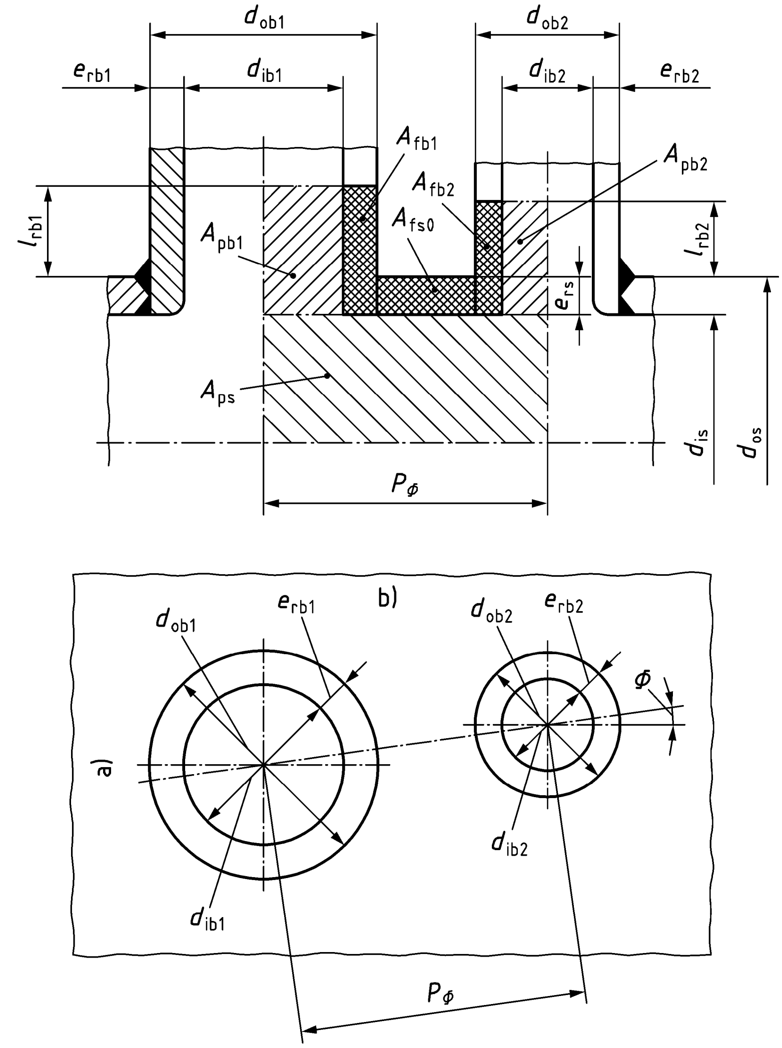


Figure 8.3-1 — Load diagram for cylindrical shell with oblique branch and reinforcing pad



NOTE Hatching and weld details in X omitted for clarity.

Figure 8.3-2 — Load diagram for cylindrical shell with non-radial branch

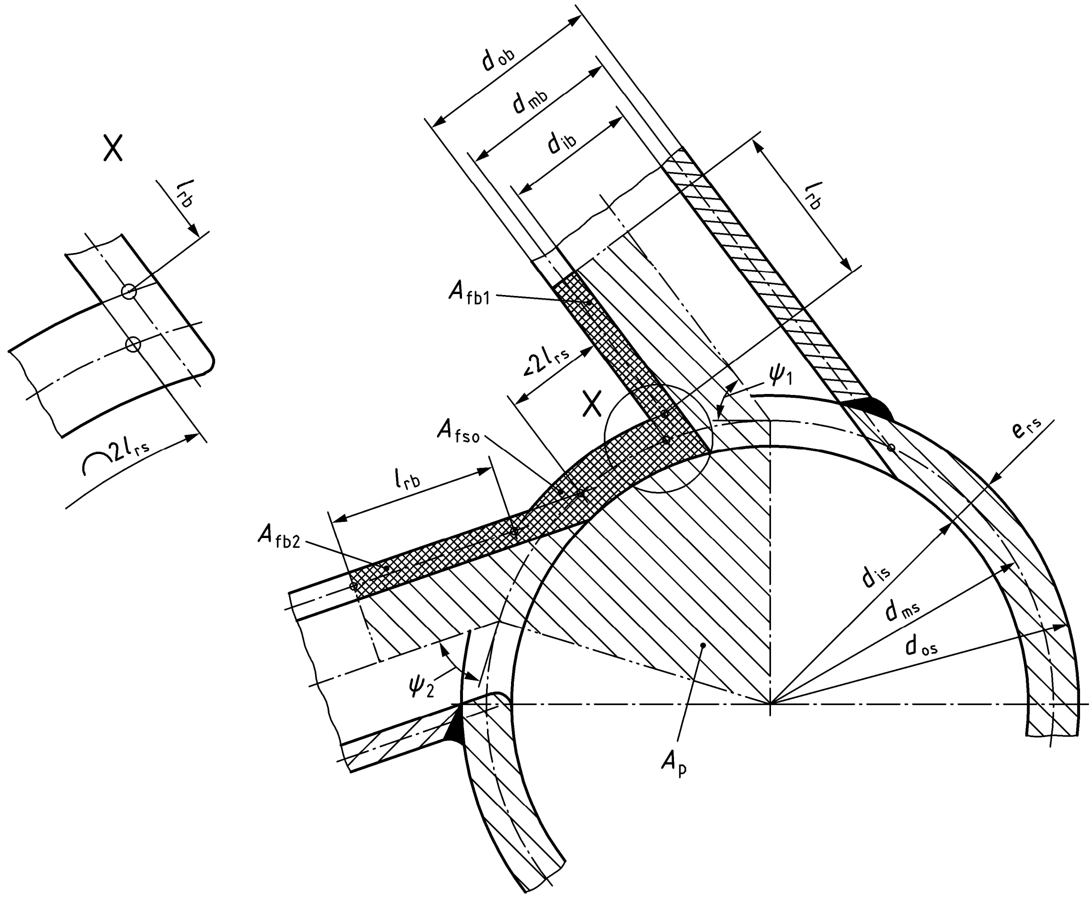


Key

a) circumferential direction

b) longitudinal direction

Figure 8.3-3 — Load diagram for cylindrical shell with adjacent branches, arranged with an angle Φ to the axis of the shell



NOTE $\psi_1 \geq 45°$,
$\psi_2 \geq 45°$,
where ψ is the angle of the axis of the branch to the axis of the shell.

Figure 8.3-4 — Load diagram for cylindrical shell with non-radial adjacent branches, arranged on the circumference

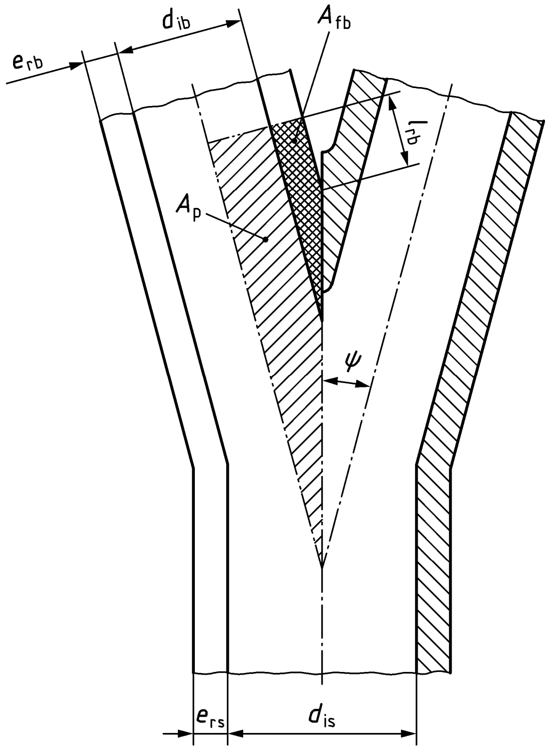


Figure 8.3-5 — Load diagram for Y-shaped branches

8.3.3 Design of isolated openings and branch connections

8.3.3.1 General

The shell thickness e_{rs} and the thickness of a branch connection e_{rb} shall not be less than that calculated for $v = 1$ in accordance with 7.2.

8.3.3.2 Isolated opening with a vertical branch

8.3.3.2.1 For isolated openings fitted with a vertical branch without additional reinforcement, 8.3.3.4 and 8.3.3.5 shall be additionally taken into account.

8.3.3.2.2 Where the following strength condition[12)]:

$$f_a = p_c\left(\frac{A_p}{A_{fs} + A_{fb}} + \frac{1}{2}\right) \le f_s \tag{8.3-2}$$

The efficiency[12)] shall be

$$v_b = \frac{d_{is}\,(A_{fs} + A_{fb})}{2\,e_{rs}\,A_p} \le 1 \tag{8.3-3}$$

[12)] The approximate calculation in accordance with 8.2 may be used instead of this calculation, in which case the reinforcing effect of the nozzles should not be considered.

8.3.3.2.3 If the design stress of the branch is less than that of the main body, the following strength condition[13] applies:

$$f_{\mathrm{a}} = \frac{p_{\mathrm{c}}\left(2\,A_{\mathrm{p}} + A_{\mathrm{fs}} + A_{\mathrm{fb}}\right)}{2\left(A_{\mathrm{fs}} + \dfrac{f_{\mathrm{b}}}{f_{\mathrm{s}}}A_{\mathrm{fb}}\right)} \leq f_{\mathrm{s}} \qquad (8.3\text{-}4)$$

and in this case, the efficiency[13] shall be

$$v_{\mathrm{b}} = \frac{d_{\mathrm{is}}\left(A_{\mathrm{fs}} + \dfrac{f_{\mathrm{b}}}{f_{\mathrm{s}}}A_{\mathrm{fb}}\right)}{e_{\mathrm{rs}}\left(2A_{\mathrm{p}} + A_{\mathrm{fb}} - \dfrac{f_{\mathrm{b}}}{f_{\mathrm{s}}}A_{\mathrm{fb}}\right)} \leq 1 \qquad (8.3\text{-}5)$$

$f_{\mathrm{b}}/f_{\mathrm{s}} > 1$ shall not be used in the calculation (see 8.1.4.1).

8.3.3.3 Isolated opening with an oblique branch and additional reinforcement

8.3.3.3.1 For isolated openings in accordance with Figure 8.3-1 the requirements for design of reinforcing pads in 8.1.5 shall be additionally taken into consideration.

8.3.3.3.2 The strength condition for area I (see Figure 8.3-1) shall be

$$f_{\mathrm{aI}} = p_{\mathrm{c}}\left(\frac{A_{\mathrm{pI}}}{A_{\mathrm{fsI}} + A_{\mathrm{fbI}} + 0{,}7A_{\mathrm{fpI}}} + \frac{1}{2}\right) \leq f_{\mathrm{s}} \qquad (8.3\text{-}6)$$

and for area II

$$f_{\mathrm{aII}} = p_{\mathrm{c}}\left(\frac{A_{\mathrm{pII}}}{A_{\mathrm{fsII}} + A_{\mathrm{fbII}} + 0{,}7A_{\mathrm{fpII}}} + \frac{1}{2}\right) \leq f_{\mathrm{s}} \qquad (8.3\text{-}7)$$

8.3.3.3.3 If the design stress of the branch material or the material of the additional reinforcement is less than that of the main body, the strength condition for area I shall be:

$$\left(f_{\mathrm{s}} - \frac{p_{\mathrm{c}}}{2}\right)A_{\mathrm{fsI}} + \left(f_{\mathrm{b}} - \frac{p_{\mathrm{c}}}{2}\right)A_{\mathrm{fbI}} + \left(f_{\mathrm{p}} - \frac{p_{\mathrm{c}}}{2}\right)0{,}7A_{\mathrm{fpI}} \geq p_{\mathrm{c}}A_{\mathrm{pI}} \qquad (8.3\text{-}8)$$

and for area II accordingly with using index II instead of index I.

8.3.3.4 Cross-section vertical to the main body axis

For branch connections calculated with elevated temperature proof strength, where $d_{\mathrm{ib}}/d_{\mathrm{is}} \geq 0{,}7$ and simultaneously $e_{\mathrm{rb}}/e_{\mathrm{rs}} < d_{\mathrm{ib}}/d_{\mathrm{is}}$, the following condition shall be met for the main body / branch transition in the cross-section vertical to the main body axis:

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[13] The approximate calculation in accordance with 8.2 may be used instead of this calculation, in which case the reinforcing effect of the nozzles should not be considered.

$$f_{ab} = \frac{p_c}{1{,}5}\left(\frac{d_{is}+e_{rs}}{2\,e_{rs}} + 0{,}2\frac{d_{ib}+e_{rb}}{e_{rb}}\sqrt{\frac{d_{is}+e_{rs}}{e_{rs}}}\right) \leq \min\left(f_s, f_b\right) \tag{8.3-9}$$

Where the main body and the branch connection consist of material with differing design stresses, the smaller value shall be used in this calculation.

8.3.3.5 Cylindrical shells with a branch not radially arranged

For cylindrical shells where the branch is not arranged in the radial direction (see Figure 8.3-2), but at an angle ψ_1 with the tangent to the main body, the higher loading may occur in the cross section of Figure 8.3-2 or in the longitudinal section of Figure 8.3-2. In both cases the strength condition as per Equation (8.3-2) applies, with the areas A_p, A_{fs} and A_{fb} shown in the respective figures to be used in the calculation. The lengths contributing to the reinforcement (effective lengths) shall only be used in the calculation of the main body in accordance with Equation (8.1-1) or of the branch connection in accordance with Equation (8.1-2) or (8.1-3) respectively.

The wall thickness of the branch e_{rb} shall not exceed the main body wall thickness e_{rs}.

8.3.3.6 Y-shaped branches

Y-shaped branches (see Figure 8.3-5) shall be made as forgings or welded construction with $\psi \geq 15°$. The strength condition for the loaded gusset area shall be

$$f_a = p_c y\left(\frac{A_p}{A_f} + \frac{1}{2}\right) \leq f_s \tag{8.3-10}$$

The effective lengths shall be used for the calculation of the branch in accordance with Equation (8.1-2). The factor y in Equation (8.3-10) shall be for branches with $\psi > 45°$,

$$y = 1 \tag{8.3-11}$$

and for branches with $d_o \leq 102$ mm and $15° \leq \psi \leq 45°$

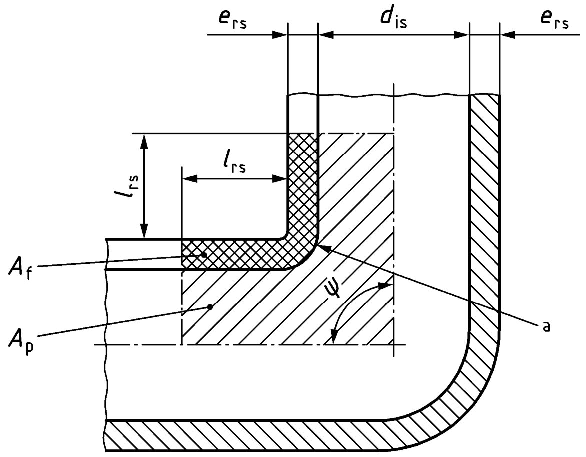
$$y = 1 + 0{,}005\,(45° - \Psi) \tag{8.3-12}$$

8.3.3.7 Cylindrical shells as forged fitting (elbows)

For forged and bored fittings with $e_{rs}/d_{is} \geq 0{,}1$ and $d_{is} \leq 90$ mm (e.g. see Figure 8.3-6) with a calculation temperature at which the design stress does not result from the creep strength $R_{m\,T\,tc}$, the following strength condition shall apply

$$p_c\left(\frac{A_p}{A_f} + \frac{1}{2}\right) \leq f \tag{8.3-13}$$

The effective lengths shall be at maximum as per Equation (8.1-1). The inclination angle ψ shall be $\geq 45°$.



Key
a) radiused

Figure 8.3-6 — Forged and bored fitting

8.3.4 Design of adjacent openings and branch connections

8.3.4.1 General

Adjacent openings shall be calculated additionally as isolated openings.

8.3.4.2 Condition of adjacent openings and branches

The calculation shall only be made if the condition for isolated openings or branch connections laid down in 8.1.3 is not met for adjacent openings or branches.

8.3.4.3 Main body with lower design stress than the branches

For adjacent openings or branch connections, the strength shall be calculated for a cross section with an angle Φ for the shell generating line in accordance with Figure 8.3-3. The following strength condition[14)] shall apply:

$$f_{a\Phi} = \frac{p_c}{2} \cdot \frac{2A_{ps}\dfrac{1+\cos^2\Phi}{2} + 2A_{pb1} + 2A_{pb2}}{A_{fs0} + A_{fb1} + A_{fb2}} + \frac{p_c}{2} \le f_s \qquad (8.3\text{-}14)$$

Diagonal or circumferential pitches shall be calculated as a longitudinal pitch in accordance with Figure 8.3-3 with a distance P_Φ. In which case the pressure area $2A_{ps}$ shall be corrected by the factor $((1 + \cos^2 \Phi)/2)$ in the strength condition in accordance with Equation (8.3-14).

The efficiency[14)] shall be

$$v_m = \frac{d_{is}}{e_{rs}} \cdot \frac{A_{fs} + A_{fb1} + A_{fb2}}{2A_{ps}\dfrac{1+\cos^2\Phi}{2} + 2A_{pb1} + 2A_{pb2}} \le 1 \qquad (8.3\text{-}15)$$

[14)] The approximate calculation in accordance with 8.2 may be used instead of this calculation, in which case the reinforcing effect of the nozzles should not be considered.

8.3.4.4 Branches with equal or lower design stress than the main body

If the design stress of one or both branches is less than that of the main body, the following condition shall apply

$$f_{a\Phi} = \frac{p_c}{2} \cdot \frac{2A_{ps}\dfrac{1+\cos^2\Phi}{2} + 2A_{pb1} + 2A_{pb2} + A_{fs} + A_{fb1} + A_{fb2}}{A_{fs} + \dfrac{f_{b1}}{f_s}A_{fb1} + \dfrac{f_{b2}}{f_s}A_{fb2}} \le f_s \tag{8.3-16}$$

In this case, the efficiency[15] shall be

$$v_m = \frac{d_{is}}{e_{rs}} \cdot \frac{A_{fs0} + \dfrac{f_{b1}}{f_s}A_{fb1} + \dfrac{f_{b2}}{f_s}A_{fb2}}{2A_{ps}\dfrac{1+\cos^2\Phi}{2} + 2A_{pb1} + 2A_{pb2} + A_{fb1} + A_{fb2} - \dfrac{f_{b1}}{f_s}A_{fb1} - \dfrac{f_{b2}}{f_s}A_{fb2}} \le 1 \tag{8.3-17}$$

if f_{b1}/f_s or f_{b2}/f_s > 1 shall not be used in the calculation.

8.3.4.5 Adjacent branches in the circumferential direction

For non-radial adjacent branches arranged on the circumference in accordance with Figure 8.3-4 the calculation procedure shall be analogous to radial branches. In this case the correction factor ((1 + $\cos^2 \Phi$)/2) shall be replaced by the factor 1.

8.4 Bolted connections

8.4.1 General

8.4 shall apply to the use and calculation of bolts which, as friction-type fasteners, are loaded by operating pressure and temperature.

To keep bolted joints as flexible as possible it is recommended that the bolts are designed as reduced shank bolts.

Highly loaded bolts shall be designed as reduced shank bolts. Reduced shank bolts shall be mandatory for:

a) a calculation temperature t_c > 300 °C, or

b) a calculation pressure p_c > 8 MPa, or

c) a root diameter d_k > 25 mm.

Bolts with a shank diameter d_s < 10 less than 10 mm are not permitted.

Flanged pressure containing joints shall not be located in the flue gas path.

Where standard flanges are used, the strength conditions for the bolts shall be deemed to have been satisfied, if in accordance with EN 13445-3:2009 bolts are used on flanges with the materials specified in this

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15) The approximate calculation in accordance with 8.2 may be used instead of this calculation, in which case the reinforcing effect of the nozzles should not be considered.

standard for the maximum allowable pressure and the corresponding operating temperature also specified therein.

The number of bolts for flanged joints shall be at least 4. The number shall be as large as practical to ensure tightness since a larger number of bolts with a smaller diameter and a correspondingly reduced pitch is more favourable than a few large diameter bolts.

To obtain low operating bolt loads, the flange gaskets shall be relatively small taking into account the operating gasket seating stress. The "tongue and groove" or "male and female" type shall be preferable to flat facings, unless metallic gaskets or combined seals are used.

The use of studded bolt connections shall be only permitted if the calculation temperature t_c does not exceed 300 °C and the calculation pressure p_c does not exceed 4 MPa. Where the main body wall thickness is inadequate to permit direct attachment, pads or saddles shall be secured to the main body by welding (see Figure 8.4-1).

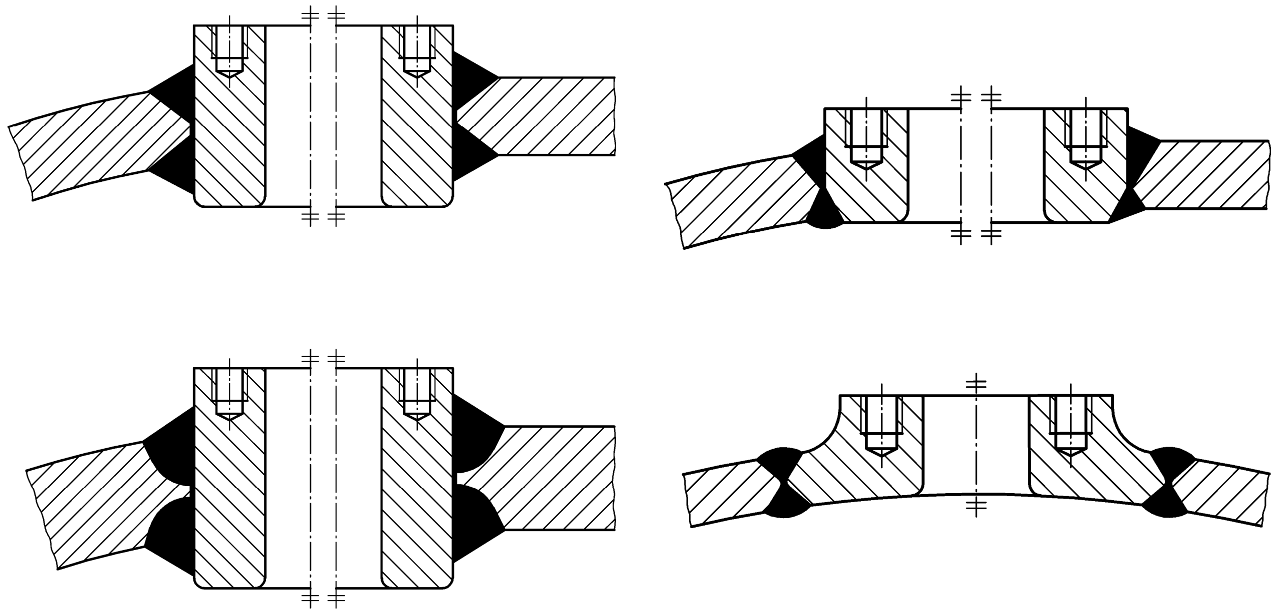


Figure 8.4-1 — Typical examples of butt-welded studded connections

8.4.2 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table 8.4-1 shall be used.Table 8.4-1 — Symbols

Symbol	Description	Unit
C_m	factor for calculating the gasket seating load	—
c_3	manufacturing tolerances	mm
d_d	mean gasket diameter	mm
d_k	root diameter	mm
d_s	shank diameter of reduced shank bolt	mm
K_D	deformation resistance of gasket material at room temperature	MPa
k_0	gasket factor for gasket seating load	mm

Table 8.4-1 *(continued)*

Symbol	Description	Unit
k_1	gasket factor for operating condition (gasket load reaction)	mm
n	number of bolts	—
P_{DB}	compression load on gasket to ensure tight joint	N
P_{DV} or $P_{DV'}$	gasket seating load	N
P_F	difference between total end force due to inside pressure and the end force due to inside pressure on area inside of flange	N
P_R	summary of end forces on tube	N
P_{Rpc}	end force due to inside pressure	N
P_{RZ}	additional end force on tube	N
P_S	bolt load (general)	N
P_{SB}	operating bolt load	N
P_{SO}	initial bolt load (bolting-up condition) prior to application of pressure	N
P_{SP}	bolt load at test pressure	N
q	quality factor	—

8.4.3 Calculation of bolt diameter

8.4.3.1 The required root diameter of a bolt shall be calculated for

a) the operating condition using the calculation pressure p_c and the calculation temperature t_c derived from the operating bolt load P_{SB},

b) the test pressure $p_{t'}$ at 20 °C and from the load P_{SP} (only required for $p_{t'}$ > 1,3 p_c),

c) the bolting-up condition prior to application of pressure with p_c = 0 and 20 °C derived from the bolt load P_{SO}.

8.4.3.2 The required bolt diameter at the root of the thread in a bolted joint with n bolts shall be

$$d_k = \sqrt{\frac{4}{\pi\, n} \cdot \frac{P_S}{\frac{K}{S} q}} + c_3 \qquad (8.4\text{-}1)$$

For reduced shank bolts the shank diameter shall be d_s = 0,9 d_k

To facilitate the calculation a coefficient

$$Z = \sqrt{\frac{4}{\pi} \cdot \frac{S}{q}} \qquad (8.4\text{-}2)$$

may be introduced (see Table 8.4-2).

The equation then becomes

$$d_k = Z\sqrt{\frac{P_S}{K_n}} + c_3 \tag{8.4-3}$$

Table 8.4-2 — Coefficient Z for calculating d_k

Quality factor q [a]	Coefficient Z			
	Operating conditions		Test pressure and bolting-up condition	
	S = 1,5	S = 1,6	S = 1,1	S = 1,2
0,75	1,60	1,65	1,37	1,43
1,00	1,38	1,43	1,19	1,24

[a] Quality factor q, see 8.4.8; safety factor S, see 8.4.7.

For reduced shank bolts the reduced shank diameter d_s shall meet the Equations (8.4-1) or (8.4-3) instead of d_k.

8.4.3.3 The following design allowance tolerances c_3 shall be used:

a) for the operating condition

root diameter 20 mm and less — c_3 = 3 mm,
root diameter 42 mm and greater — c_3 = 1 mm,
in the intermediate range — c_3 = 5 mm - 0,1d_k,

b) for the test pressure — c_3 = 0 mm,

c) for the bolting-up condition prior to application of pressure — c_3 = 0 mm.

8.4.4 Calculation of bolt load

8.4.4.1 General

The bolt load shall be determined for the operating condition, the test pressure and the bolting-up condition prior to application of pressure. The following equations shall apply to circular bolted joints.

8.4.4.2 Operating condition

The operating bolt load P_{SB} (see 8.4.3.1 a)) shall be calculated from:

$$P_{SB} = P_R + P_{DB} + P_F \tag{8.4-4}$$

The following values shall be inserted:

a) the summary P_R of end forces on tube shall be

$$P_{R} = P_{Rpc} + P_{RZ} \tag{8.4-5}$$

The end force due to inside pressure

$$P_{Rpc} = \frac{d_{i}^{2}\pi}{4} p_{c} \tag{8.4-6}$$

The additional end force P_{RZ} shall be calculated from the conditions of any connected tubing. The force P_{RZ} is equal to 0 for bolted joints to which no tubing or only tubing without additional longitudinal forces is connected. P_{RZ} may occur for normally installed connected tubing, where the transmission of additional longitudinal forces to the bolts due to thermal stresses is possible. If a calculation of the additional end force P_{RZ} is not possible, then $P_{RZ} = P_{Rpc}$ is to be introduced to the calculation. With this assumption, additional bending moments are usually also seized.

b) the compression load P_{DB} on gasket to ensure tight joint shall be

$$P_{DB} = \pi\, d_{d} 1{,}2\, k_{1}\, p_{c} \tag{8.4.7}$$

This shall ensure continuous tightness during operation. The value for k_1 shall be taken from the specification of the gasket manufacturer. For welded seals with gap, $k_1 = 0$, then P_{DB} is 0 (e.g. weld-lip seals, seal welded diaphragm plate).

c) the difference between total end force due to inside pressure and the end force due to inside pressure on area inside of flange P_F

$$P_{F} = \frac{\pi}{4}\left(d_{d}^{2} - d_{i}^{2}\right) p_{c} \tag{8.4-8}$$

This force is caused by the internal pressure p_c and is exerted on the ring area, which is formed by the gasket diameter d_d and the inside diameter d_i.

The gasket diameter d_d shall preferably be taken as the mean diameter of the gasket, since it cannot be predetermined exactly at which point of the gasket width the internal pressure is effective. For welded seals, the diameter of the outermost welded seam shall be taken, since it is assumed that the entire gap of the seal is subject to internal pressure.

8.4.4.3 Test pressure

The bolt load P_{SP} at the test pressure $p_{t'}$ (see 8.4.4.2 b)) shall be calculated from:

$$P_{SP} = \frac{p_{t'}}{p_{c}}\left(P_{R} + \frac{P_{DB}}{1{,}2} + P_{F}\right) \tag{8.4-9}$$

8.4.4.4 Bolting-up condition prior to application of pressure

The following cases shall be taken into consideration:

The bolted joint shall be tightened so that, upon installation (see 8.4.3.1 c)), the required gasket seating is ensured, any hydrostatic end forces P_{RZ} present in the tubing system can be absorbed, and that the joint remains tight under operating conditions.

To meet these requirements, the following shall be met:

$$P_{SO} = P_{DV} \quad \text{or} \quad P_{SO} = P_{DV} + P_{RZ} \tag{8.4-10}$$

where P_{DV} is the gasket seating load required to ensure an adequate fit to the contact faces. But at least:

$$P_{SO} = 1{,}1 P_{SB} = 1{,}1(P_{DB} + P_F + P_R) \tag{8.4-11}$$

$$P_{DV} = d_d \pi k_0 K_D \tag{8.4-12}$$

However, it has been shown in practice, that with non-metallic gaskets it shall not be required, to have complete gasket seating on the contact faces in accordance with Equation (8.4-12) in order to control low pressures. To avoid uneconomically large bolt diameters, shall be possible for pressures

$$p \leq \frac{k_0 K_D}{1{,}2 k_1 + \frac{d_d}{4}} \tag{8.4-13}$$

or where

$$P_{DV} \geq P_{SB} = P_{Rpc} + P_{DB} + P_F \tag{8.4-14}$$

to reduce the value P_{DV} to $P_{DV'}$

$$P_{DV'} = P_{DV} C_m + (1 - C_m)\sqrt{\left(\frac{d_i^2 \pi p_c}{4} + P_{DB} + P_F\right) P_{DV}} \tag{8.4-15}$$

where

C_m = 0,1 for liquids and

C_m = 0,2 for steam and superheated steam.

Where required, P_{RZ} shall be used in accordance with 8.4.4.2 a).

The factors k_0, k_1 and K_D depend on the type and shape of the gasket and the filling medium.

8.4.5 Calculation temperature t_c

8.4.5.1 General

The temperature t_c to be used in the bolt design shall depend on the type of the bolted joint and on the thermal insulation.

8.4.5.2 Thermal insulated joints

It may be assumed that the temperature of the bolts is less than the fluid temperature even if the temperature distribution of the complete arrangement from the fluid up to the bolts is not checked. The values given in Table 8.4-3 shall apply.

Table 8.4-3 — Maximum temperature reduction

Flange types of the joint	Maximum allowable assumption for reduction $t_{\text{fluid}} - t_{\text{c bolt}}$
loose + loose	30 °C
integral + loose	25 °C
integral + integral	15 °C

8.4.5.3 Joints without thermal insulation

At non-insulated joints, the temperature difference between fluid and bolts is greater. However, a correspondingly greater reduction of the calculation temperature t_c of the bolts shall not be allowable, since the correspondingly higher thermal stress is not taken into account. Thus, Table 8.4-3 shall also be used at non-insulated joints.

8.4.6 Design strength value K

The following K values shall apply:

For calculation temperatures t_c up to and including 350 °C the yield strength or the 0,2 % proof strength $R_{p0,2\ tc}$ (minimum value) at the calculation temperature (see 8.4.5) shall be used.

For calculation temperature t_c exceeding 350 °C, the lower of the following values shall be used:

a) the proof strength $R_{p0,2\ tc}$ (specified minimum value) at the calculation temperature t_c (see 8.4.5);

b) the creep rupture strength $R_{m\ 100\ 000\ tc}$ (mean value) at the calculation temperature t_c (see 8.4.5).

The strength parameter of bolts shall be taken from the relevant European Standards.

8.4.7 Safety factor S

The factors given in Table 8.4-4 shall be used.

Table 8.4-4 — Safety factor

Condition	For reduced shank bolts	For full diameter bolts
Operating condition	S = 1,5	S = 1,6
Test pressure Bolting-up condition prior to application of pressure	S = 1,1	S = 1,2

8.4.8 Quality factor q

The strength parameter of bolts shall be taken from the relevant European Standard. Reduced shank bolts shall be machined on all sides. For unmachined plane-parallel bearing surfaces, the quality factor q = 0,75 shall be taken. For machined bearing surfaces of mating surfaces q = 1,0 may be taken. Bearing surfaces which are not plane-parallel (e.g. on angular sections) shall not be accepted.

8.5 Screwed and socket welded connections

8.5.1 Screwed connections into the belt

In addition, screwed connections into the shell shall be permitted only if the following conditions are met:

a) the outside diameter of the connection shall not exceed 60,3 mm;

b) the calculation pressure shall not exceed 4 MPa;

c) the calculation temperature shall not exceed 400 °C;

d) seal welding shall be applied if the temperature is greater than 220 °C or the pressure greater than 2 MPa;

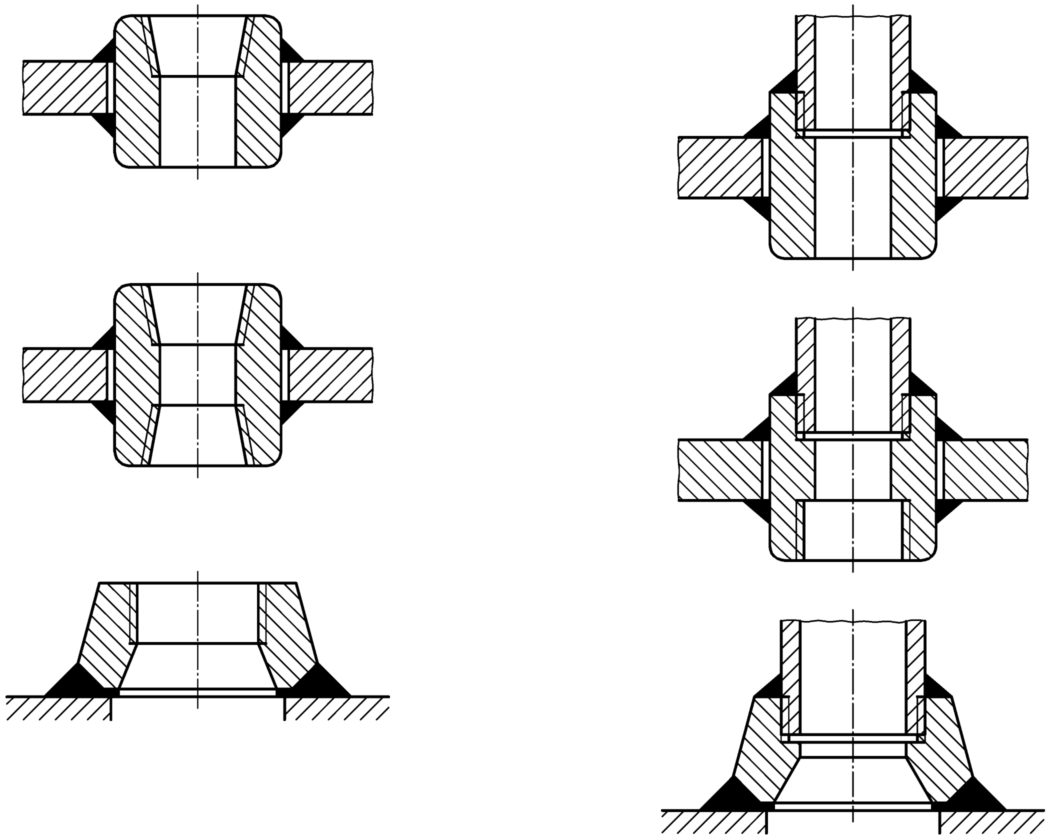
e) in the case of seal welding the steel of the screwed component shall not contain more than 0,25 % carbon.

The design of connections shall conform to with 8.3.2; the minimum thickness shall be measured at the root of the thread.

The main pressure part shall be designed as having an uncompensated opening with a diameter equal to that of the root of the thread in the hole.

8.5.2 Screwed socket connections

Screwed or screwed and seal welded socket connections shall not be used in service where fatigue, severe erosion, crevice corrosion or shock is expected to occur, or for tubes exceeding 60,3 mm outside diameter. The maximum operating pressure shall not exceed 4 MPa and the design temperature shall not exceed 400 °C. The threading shall be in accordance with EN 10266:2003 (see Figure 8.5-1).



a) screwed b) screwed and seal welded

Figure 8.5-1 — Screwed and screwed and seal welded connections

The main pressure part shall be designed as having an uncompensated opening.

8.5.3 Socket welded connections

Socket welded joints (see Figure 8.5-2) shall not be used in service where:

a) pressure is exceeding 2 MPa;

b) metal temperatures are above 350 °C;

c) outside diameter for tubes is exceeding 60,3 mm (see Figure 8.5-3).

Screwed flanges shall not be used. Flanges shall not be used where the bolts would be exposed to products of combustion.

The thickness of socket welded fittings shall not be less than 2,6 times the nominal thickness of the pipe. Leg lengths of the fillet weld shall be as shown in Figure 8.5-3 so as to achieve the throat dimension not less than the nominal thickness of the tube. The material shall be compatible with the associated tubing.

Socket welded fittings shall be of forged steel in accordance with the requirements of EN 12952-2:2011. The dimensions, clearances, ratings and weld joints shall be in accordance with the requirements of the appropriate European Standards.

NOTE In the absence of an appropriate European Standard, guidance should be taken from appropriate national standard e.g. BS 3799 [1].

The main pressure part shall be designed as having an uncompensated opening.

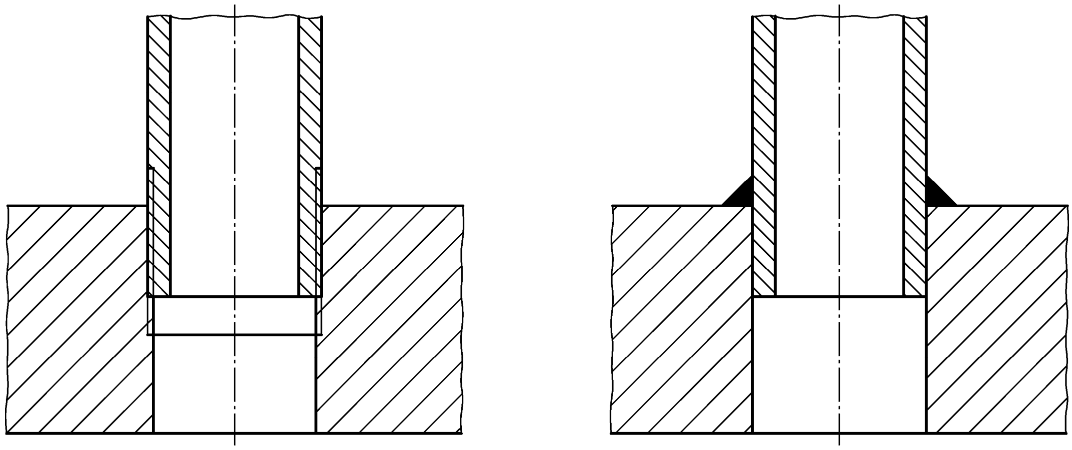
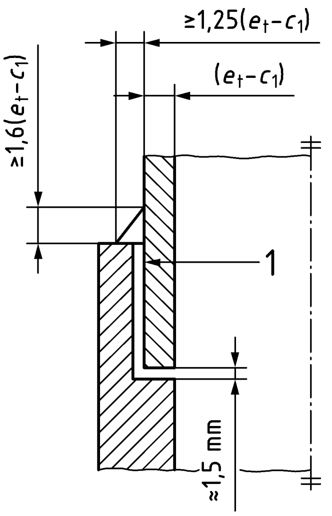


Figure 8.5-2 — Screwed and socket welded connections



Key

1 diametral clearance 1,00 mm max.

Figure 8.5-3 — Socket weld

9 Headers and plain tubes of rectangular section

9.1 General

This applies to the calculation of rectangular-section tubes and headers with rounded corners, manufactured by forging, rolling or drawing and without longitudinal welds.

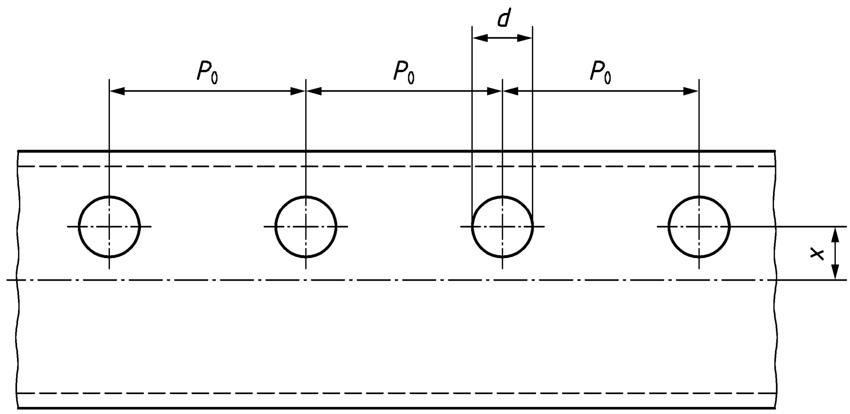
Longitudinally welded rectangular tubes shall only be permitted if the longitudinal weld has a weld factor of 1 and has been tested in accordance with the provisions of circular tubes, see EN 12952-6:2011.

Rectangular-section headers or tubes shall have the same thickness on all sides. The sides shall have no openings, or have circular openings of identical dimensions located on a single line (Figure 9.1-1, a)) or on two parallel lines (Figure 9.1-1, b)). Elliptical openings of identical dimensions shall only be located on one line. The lines of openings shall be parallel to the longitudinal axis.

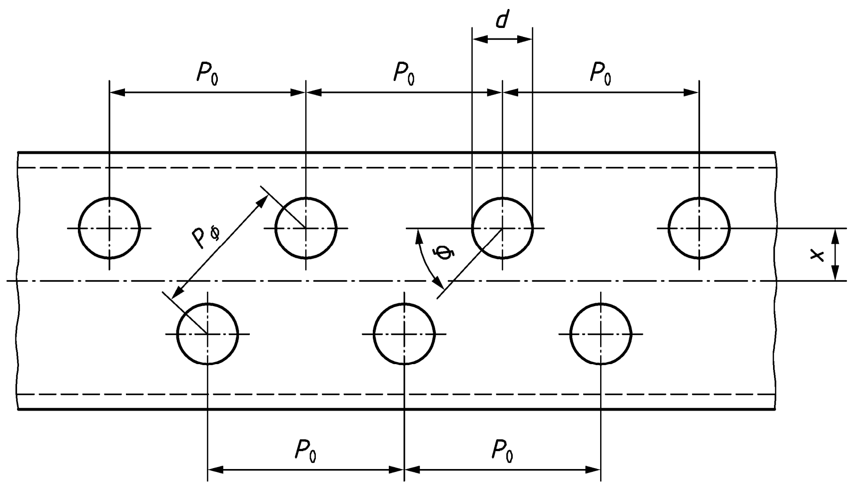
In the case of staggered openings the effect of the off-set shall be investigated by considering the ligament efficiency for diagonal ligaments (Figure 9.1-1, b)).

NOTE These rules only consider loadings due to internal pressure. These rules do not apply to rectangular-section headers subjected to furnace radiation nor to rectangular-section tubes and headers with a wall thickness exceeding 30 mm which are in direct contact with the furnace gases, the anticipated temperature of which exceeds 650 °C.

In the case of superheated steam headers with heated walls the flue gas temperature shall not exceed 500 °C.



a) openings located on a single line



b) openings located on two parallel lines

Figure 9.1-1 — Symbols for ligaments in rectangular headers

9.2 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table 9.2-1 shall be used.

Table 9.2-1 — Symbols

Symbol	Description	Unit
d	diameter of opening, or in the case of elliptical openings, the opening diameter measured along the header longitudinal axis	mm
d'	for elliptical openings, the diameter measured perpendicular to the header longitudinal axis (see Figure 9.2-2)	mm
e_c	required wall thickness of rectangular-section header or tube without allowances (see Figure 9.1-1)	mm
m	One-half of the internal width of the rectangular-section header or tube, in parallel to that referred to in the calculation (see Figure 9.2-1)	mm
n	one-half of the internal width of the rectangular-section header or tube, perpendicular to that referred to in the calculation (see Figure 9.2-1)	mm
P_0	pitch of one row of openings parallel to the header longitudinal axis (see Figure 9.1-1)	mm
P_Φ	centre distance of two adjacent openings for an oblique ligament at an angle Φ (see Figure 9.1-1)	mm
r_i	internal corner radius of rectangular-section header or tube	mm
x	distance between the opening under consideration or the row of openings and the centre line of the wall (see Figure 9.1-1)	mm
Y	coefficient	mm^2
Φ	angle between the line connecting two openings with diagonal pitch and the header longitudinal axis (see Figure 9.1-1)	degree
v_{bb}	ligament efficiency for an individual opening referring to bending stress	—
v_{bt}	ligament efficiency for an individual opening referring to tensile stress	—
v_{mt}	ligament efficiency for a row of openings parallel to the header longitudinal axis	—
v_{mb}	ligament efficiency for two adjacent openings with diagonal pitch at the angle α	—

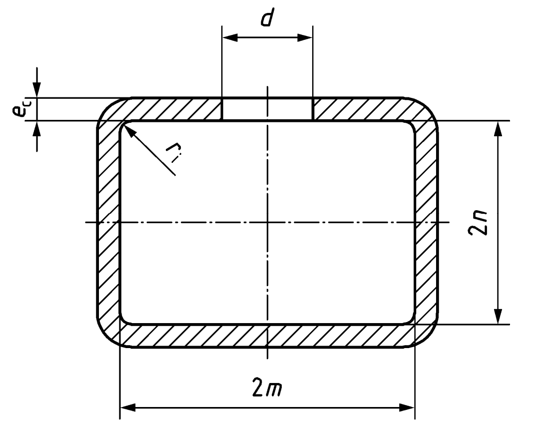
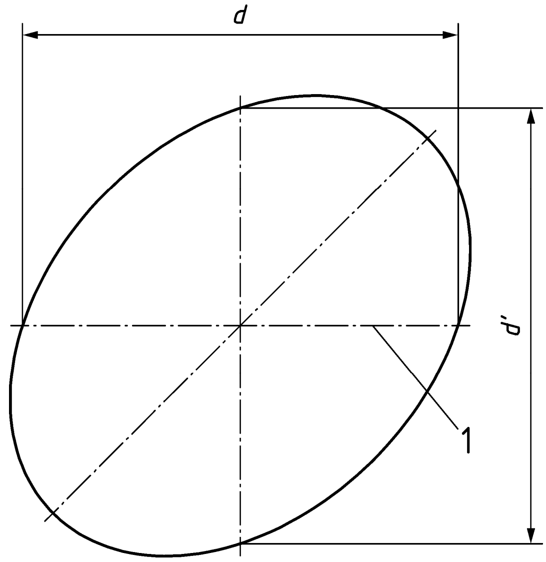


Figure 9.2-1 — Symbols for rectangular headers



Key

1 longitudinal axis of header

Figure 9.2-2 — Definition of the diameters *d* and *d'* for an elliptical opening

9.3 Required wall thickness

9.3.1 General

To determine the minimum wall thickness e_c of a straight tube or header of rectangular section without allowance for metal wastage and fabrication tolerance, the thickness required at the centre of the sides, at the corners, at the line of holes parallel to the longitudinal axis and at the staggered row of holes shall be calculated. The minimum wall thickness e_c required for the header shall be the greatest wall thickness determined, and shall apply to all four sides.

The actual wall thickness shall be

$$e = e_c + c_1 + c_2 \quad (9.3\text{-}1)$$

The smallest allowable wall thickness e shall be 3 mm.

9.3.2 Minimum wall thickness at the centre of one side

The thickness e_c required at the centre of a side wall of a rectangular-section tube or header with an internal width 2 m shall be determined in accordance with:

$$e_c = \frac{p_c n}{f} + \sqrt{\frac{4Yp_c}{f}} \tag{9.3-2}$$

The coefficient Y to be used in determining the minimum wall thickness required shall be determined in accordance with

$$Y = \left| \frac{1}{3}\left(\frac{m^3 + n^3}{m+n} \right) - \frac{m^2}{2} \right| \quad \text{(absolute value of this expression)} \tag{9.3-3}$$

9.3.3 Minimum wall thickness at the corners

The minimum thickness required at the corners of a rectangular section tube or header with an internal width 2 m shall be determined in accordance with Equation (9.3-2).

The coefficient Y to be used in determining the minimum wall thickness required shall be determined in accordance with

$$Y = \frac{1}{3}\left(\frac{m^3 + n^3}{m+n} \right) \tag{9.3-4}$$

To avoid excessive stresses at the header corners, the following condition

$$r_i \geq \frac{1}{3} e \tag{9.3-5}$$

but not less than 8 mm, shall be satisfied.

9.3.4 Minimum thickness at a line of openings

The thickness required of the side wall of a rectangular-section header with an internal width 2 m at a line of openings parallel to the header longitudinal axis shall be determined in accordance with:

$$e_c = \frac{p_c n}{f v_{mt}} + \sqrt{\frac{4Yp_c}{f v_{mb}}} \tag{9.3-6}$$

The coefficient Y to be used in determining the thickness required shall be determined in accordance with

$$Y = \left| \frac{1}{3}\left(\frac{m^3 + n^3}{m+n} \right) - \frac{m^2 - x^2}{2} \right| \quad \text{(absolute value of this expression)} \tag{9.3-7}$$

The distance x of the line of holes to the centre line of the wall shall not exceed $m/2$.

The ligament efficiencies, v_{mt} referring to tensile stress and v_{mb} referring to bending stress, used in the equation for determining the wall thickness shall be derived in accordance with:

$$v_{\mathrm{mt}} = \frac{P_0 - d}{P_0} \qquad (9.3\text{-}8)$$

and:

$$v_{\mathrm{mb}} = \frac{P_0 - d}{P_0} \text{ or } \frac{P_0 - d'}{P_0} \quad \text{if } d \text{ or } d' < 0{,}6\, m \qquad (9.3\text{-}9a)$$

or:

$$v_{\mathrm{mb}} = \frac{P_0 - 0{,}6m}{P_0} \quad \text{if } d \text{ or } d' \geq 0{,}6\, m \qquad (9.3\text{-}9b)$$

9.3.5 Minimum wall thickness at staggered opening arrangement (diagonal pitch)

The required minimum thickness required of the side wall of a rectangular header with an internal width of $2m$ within the area of adjacent openings staggered at an angle Φ shall be as determined in accordance with Equation (9.3-6).

The coefficient Y to be used in determining the thickness required shall be determined in accordance with

$$Y = \left| \frac{1}{3}\left(\frac{m^3 + n^3}{m + n}\right) - \frac{m^2}{2} \right| \cos\Phi \text{ (absolute value of this expression)} \qquad (9.3\text{-}10)$$

The ligament efficiencies v_{mt} referring to tensile stress and v_{mb} referring to bending stress used in the equation for determining the wall thickness shall be derived from Equation (9.3-8) and Equation (9.3-9a) or (9.3-9b).

The distance x of the line of openings to the centre line of the wall shall not exceed $m/2$ in the case of staggered openings (diagonal pitch).

9.3.6 Minimum wall thickness at isolated openings

Lines of openings and adjacent openings with diagonal pitch at an angle α shall be treated like isolated openings if the tube pitch P_0 or the centre distance P_Φ exceeds the internal width $2m$ of the rectangular header parallel to the wall to be calculated.

The wall thickness required for the side wall at isolated openings or at lines of openings or adjacent openings with diagonal pitch, which meet the calculation requirement for isolated openings, shall be determined in accordance with:

$$e_{\mathrm{c}} = \frac{p_{\mathrm{c}} n}{f v_{\mathrm{bt}}} \sqrt{\frac{4Y p_{\mathrm{c}}}{f v_{\mathrm{bb}}}} \qquad (9.3\text{-}11)$$

The coefficient Y to be used in determining the thickness required shall be determined in accordance with Equation (9.3-7) for isolated openings and lines of openings without diagonal pitch.

The ligament efficiencies, v_{bt} referring to tensile stress, and v_{bb} referring to bending stress, used in the equation for determining the wall thickness shall be derived as follows:

$$v_{\mathrm{bt}} = 1 - \frac{d}{m} \qquad (9.3\text{-}12)$$

and

$$v_{bb} = 1 - \frac{d}{2m} \quad \text{or} \quad 1 - \frac{d'}{2m} \qquad \text{if } d \text{ or } d' < 0{,}6m; \tag{9.3-13a}$$

or

$$v_{bb} = 0{,}7 \qquad \text{if } d \text{ or } d' \geq 0{,}6m. \tag{9.3-13b}$$

10 Ends and spherical shells

10.1 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table 10.1-1 shall be used.

Table 10.1-1 — Symbols

Symbol	Description	Unit
a	major axis of the flat, elliptical or rectangular end	mm
b	minor axis of the flat, elliptical or rectangular end	mm
d_L	bolt circle diameter for flanges	mm
e_1	residual wall thickness of flat end at root of relief groove	mm
e_2	wall thickness of flat end at transition to cylindrical section	mm
e_k	wall thickness of knuckle	mm
α_1, α_2	angle at wall thickness transitions	degree
β_k	shape factor for knuckle	—
Φ_1	centre angle (see Figure 10.2-7)	rad

10.2 Spherical shells and dished heads

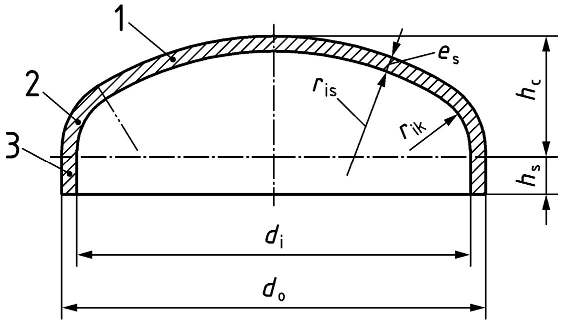
10.2.1 General

This applies to the design and dimensions of spherical shells and dished ends provided with openings, if any. It does not consider the effect of external forces and moments which shall be additionally taken into account, if required.

NOTE EN 13445-3:2009 gives methods for calculating stresses arising from external forces and moments.

The equations shown in this clause shall be applied provided the dimensional limits of Figures 10.2-1 and 10.2-2 are adhered to:

$r_{is} \leq d_o$
$r_{ik} \geq 0{,}1\ d_o$
$r_{ik} \geq 2\ e_s$
$h_s \geq 50$ mm, except when shell diameter $d_o \leq 80$ mm
$h_c \geq 0{,}18\ d_o$
$e_s - c_1 \geq 0{,}005\ d_o$



Key
1 spherical shell
2 knuckle
3 cylindrical section

Figure 10.2-1 — Torispherical end

$h_s \geq 50$ mm, except when shell diameter $d_o \leq 80$ mm
$h_c \geq 0{,}18\ d_o$
$e_s - c_1 \geq 0{,}005\ d_o$

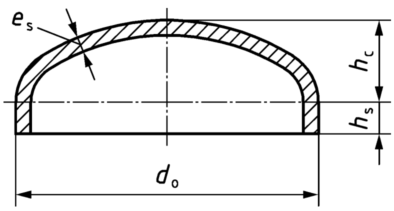


Figure 10.2-2 — Semi-ellipsoidal end

Semi-ellipsoidal ends shall be treated as torispherical ends with the same height h_c.

The following radius shall be used (see Figure 10.2-2):

$$r_{is} = \frac{d_o^2}{4 \cdot h_c} \tag{10.2-1}$$

10.2.2 Calculation formula

10.2.2.1 The ordered wall thickness shall be at least

$$e_s \geq e_{cs} + c_1 + c_2 \tag{10.2-2}$$

where

c_1 is to compensate for negative fabrication tolerances;

c_2 is to compensate for wall thickness reduction due to corrosion;

$c_2 = 0$ for ferritic steels in contact with feedwater in accordance with EN 12952-12:2003.

In the case of severe corrosion conditions an increased c_2 value shall be chosen.

The minimum wall thickness e_{rs} shall be chosen to ensure that the equivalent stress calculated in accordance with the following equation does not exceed the design stress at the calculation temperature when operating at the calculation pressure.

$$f_a = \left(\frac{A_p}{A_f} + \frac{1}{2} \right) p_c \le f_s \tag{10.2-3}$$

If the ligament efficiency v is known the required wall thickness e_{cs} without allowances shall be calculated as follows:

$$e_{cs} = r_{is} \left(\sqrt{1 + \frac{2 p_c}{(2 f_s - p_c) v}} - 1 \right) \tag{10.2-4}$$

For unpierced ends, the efficiency shall be v = 1,0.

For pierced ends 10.2.3 shall apply.

The ligament efficiency may be calculated in accordance with Table 10.2-1.

Table 10.2-1 — Calculation formulas for ends with branches

Equivalent stress in ends with single branches [a]: $$f_a = \frac{p_c (2A_p + A_{fs} + A_{fb})}{2\left(A_{fs} + \frac{f_b}{f_s} A_{fb} \right)} \le f_s$$
Equivalent stress in ends with adjacent branches [a]: $$f_a = \frac{p_c (2A_p + A_{f_s} + A_{f_{b1}} + A_{f_{b2}})}{2\left(A_{f_s} + \frac{f_{b1}}{f_s} A_{f_{b1}} + \frac{f_{b2}}{f_s} A_{f_{b2}} \right)} \le f_s$$
Ligament efficiency in ends with single branches [a b]: $$v_b = \frac{r_{is}\left(A_{f_s} + A_{f_b} \frac{f_b}{f_s} \right)}{e_{rs}\left(2A_p + A_{f_b} - \frac{f_b}{f_s} A_{f_b} \right)\left(1 + \frac{e_{rs}}{2r_{is}} \right)} \le 1$$
Ligament efficiency in ends with adjacent branches [a b]: $$v_m = \frac{r_{is}\left(A_{f_s} + \frac{f_{b1}}{f_s} A_{f_{b1}} + \frac{f_{b2}}{f_s} A_{f_{b2}} \right)}{e_{rs}\left(2A_p + A_{f_{b1}} + A_{f_{b2}} - \frac{f_{b1}}{f_s} A_{f_{b1}} - \frac{f_{b2}}{f_s} A_{f_{b2}} \right)\left(1 + \frac{e_{rs}}{2r_{is}} \right)} \le 1$$
a The ratio f_b / f_s shall be taken as unity if it is greater than 1,0.
b The ligament efficiency shall be taken as unity if it is greater than 1,0.

For thin-walled ends with $e_{cs} \leq 0{,}1\ r_{is}$ the required wall thickness without allowances may be calculated as follows:

$$e_{cs} = \frac{r_{is}\, p_c}{(2 f_s - p_c) v} \tag{10.2-5}$$

If the wall thickness has been determined, the equivalent stress in the end may be calculated using:

$$f_a = p_c \left[\frac{r_{is}}{2 e_{rs} \left(1 + \frac{e_{rs}}{2 r_{is}} \right) v} + \frac{1}{2} \right] \leq f_s \tag{10.2-6}$$

10.2.2.2 Depending on the circumstances the maximum loading in dished ends under internal pressure may occur in the knuckle or adjacent to openings. The calculation shall therefore be made for both locations.

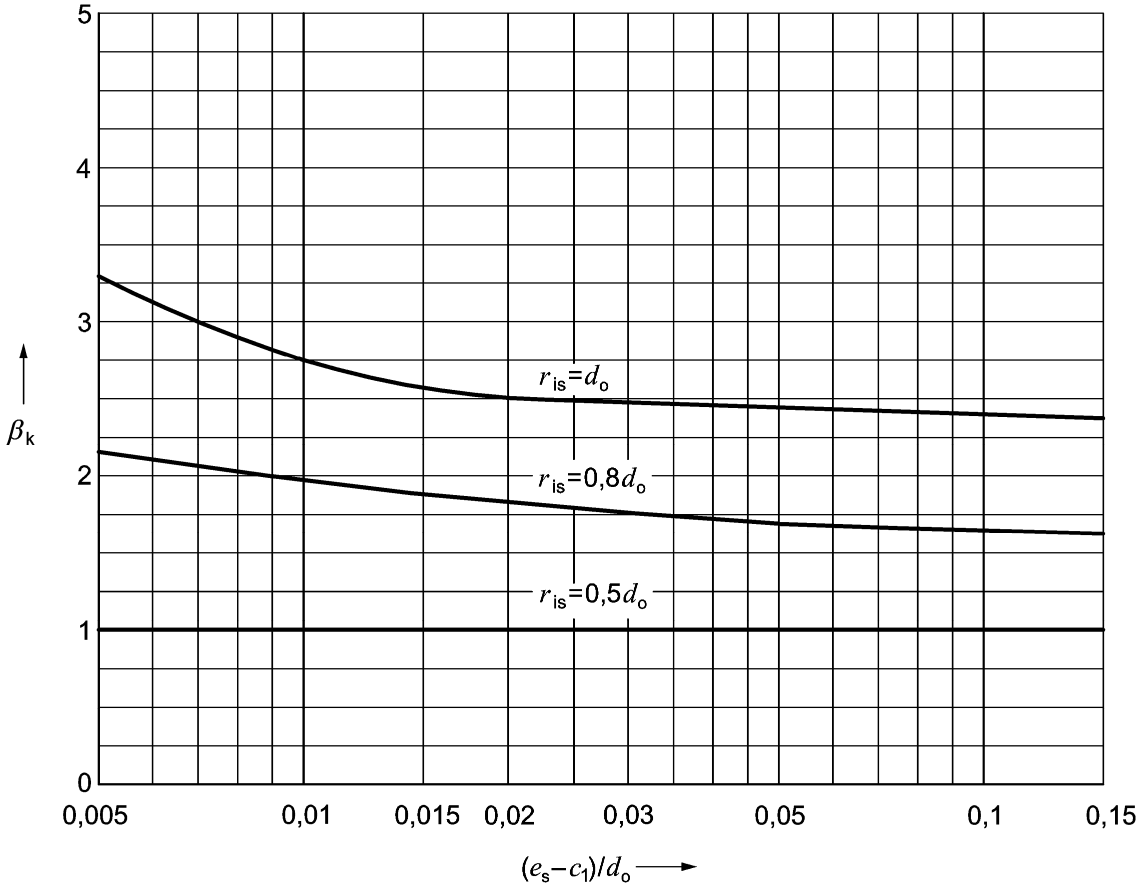
The required wall thickness without allowances at the knuckle shall be

$$e_k = \frac{p_c\, d_o\, \beta_k}{4 f_s} \tag{10.2-7}$$

β_k shall be taken from Figure 10.2-3. Linear interpolation between the curves shall be used.

The minimum wall thickness e_k shall be at least 0,005 d_o.

The equivalent stress in a knuckle with a known wall thickness shall be calculated by transposing the Equation (10.2-7).



The curves shall be defined by the following equations, where x = log $[(e_s\text{-}c_1)/d_o]$ with 0,005 ≤ $[(e_s\text{-}c_1)/d_o]$ ≤ 0,15:

for $r_{is} = d_o$:

β_k = - 0,593 8 x^3 - 2,096 4 x^2 - 2,510 8 x + 1,384 4;

for r_{is} = 0,8 d_o:

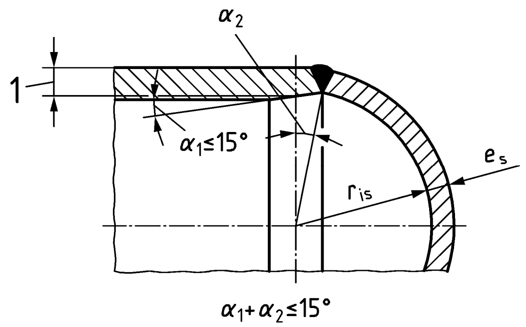
β_k = -0,059 1 x^5 - 0,531 4 x^4 - 2,039 8 x^3 - 3,766 8 x^2 - 3,455 x + 0,392 7.

Figure 10.2-3 — Shape factor β_k for torispherical heads

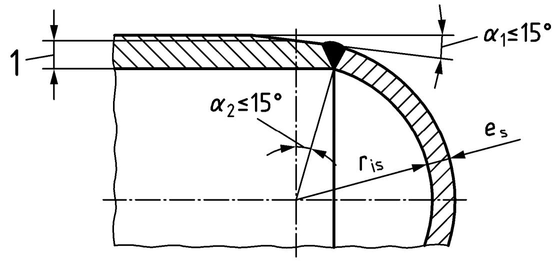
10.2.2.3 The wall thickness of the cylindrical skirt of the knuckle shall be at least equal to that of an unpierced cylindrical shell of the same diameter.

$$e_c = \frac{d_i\, p_c}{2f - p_c} \tag{10.2-8}$$

10.2.2.4 For hemispherical ends without a cylindrical skirt as shown in Figure 10.2-4 the wall thickness of an unpierced end shall be at least 10 % thicker than that required by Equation (10.2-4). The angle of the slope at the shell/end transition shall not exceed 15°.



type a



type b

Key

1 minimum wall thickness of cylindrical shell in accordance with Equation (10.2-8)

Figure 10.2-4 — Hemispherical end

10.2.3 Openings in dished ends and spherical ends

Openings in dished ends shall be located as shown in Figure 10.2-5. The ratio of branch thickness e_{rb} to end thickness e_{rs} shall be

$$\frac{e_{\mathrm{rb}}}{e_{\mathrm{rs}}} \le \begin{cases} 2 \text{ for } d_1 \le 0{,}4\, r_{\mathrm{is}} \\ 1 \text{ for } d_1 \ge 1{,}4\, r_{\mathrm{is}} \end{cases} \qquad (10.2\text{-}9)$$

Intermediate values shall be subject to straight line interpolation.

The requirements of Equation (10.2-3) shall be met. When determining the pressure and stress loaded areas in accordance with Figure 10.2-6 only the following effective lengths shall be considered:

a) in the spherical portion (crown) of the shell. In which case the wall thickness e_{rs} without reinforcing pad, if any, shall be used.

$$l_{\mathrm{rs}} \le \sqrt{(2r_{\mathrm{is}} + e_{\mathrm{rs}})e_{\mathrm{rs}}} \qquad (10.2\text{-}10)$$

b) in the nozzle:

$$l_{\mathrm{rb}} \le \sqrt{(d_{\mathrm{ib}} + e_{\mathrm{rb}})e_{\mathrm{rb}}} \qquad (10.2\text{-}11)$$

Nozzles shall only be considered in the calculation as contributing to the reinforcement if it is a full penetration weld (see Figure 10.2-6 e)). Screwed-in or rolled-in nozzles which are only seal-welded shall not be considered as contributing to the reinforcement. This shall also apply to screwed covers and closures.

Adjacent openings shall be calculated as isolated openings if the following requirements for the centre angle Φ_{b} (see Figure 10.2-7) are satisfied:

$$\Phi_b \ge arc\sin\frac{\frac{d_{ib1}}{2}+e_{rb1}}{r_{is}+\frac{e_{rs}}{2}} + arc\sin\frac{\frac{d_{ib2}}{2}+e_{rb2}}{r_{is}+\frac{e_{rs}}{2}} + 2\frac{\sqrt{(2r_{is}+e_{rs})e_{rs}}}{r_{is}+\frac{e_{rs}}{2}} \qquad (10.2\text{-}12)$$

NOTE Angle Φ_{b} measured in radians.

Adjacent openings where the effective lengths overlap shall be subject to a common calculation of areas in accordance with Figure 10.2-7 a) and 10.2-7 b) respectively.

Isolated unreinforced openings in spherical shells shall be permitted subject to the following provisions:

$$d_{\max} = 0{,}14\, l_{\text{rs}} \tag{10.2-13}$$

and

$$\frac{e_{\text{rs}}}{d_{\text{os}}} \leq 0{,}1 \tag{10.2-14}$$

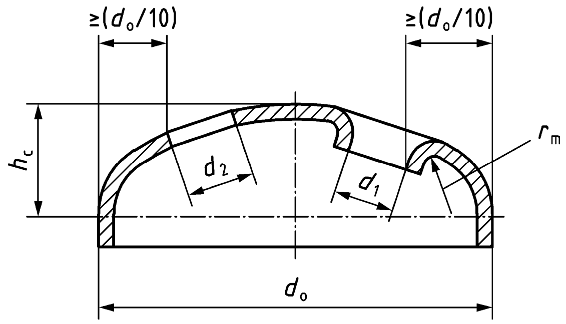
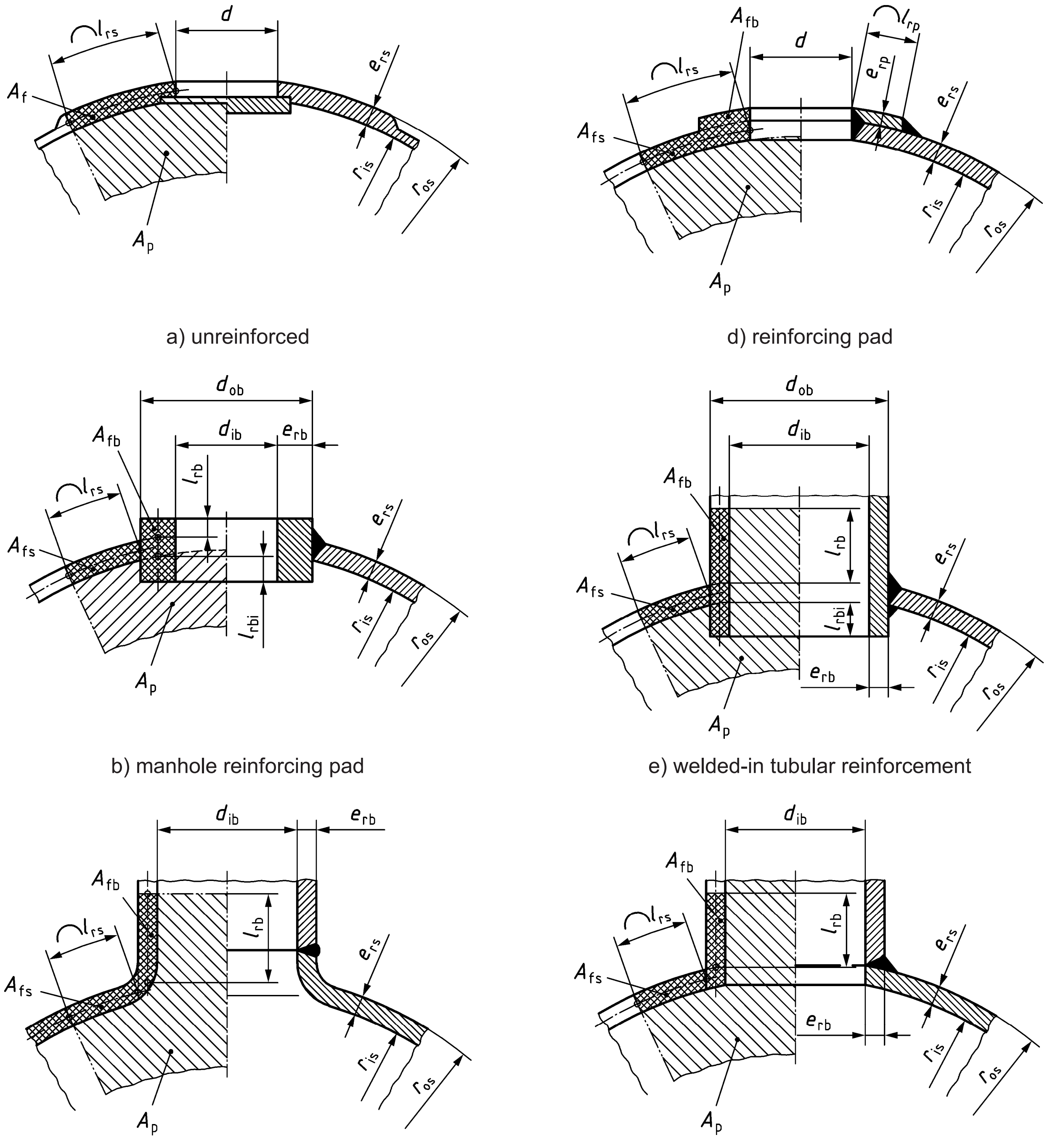


Figure 10.2-5 — Openings in ends



a) unreinforced

d) reinforcing pad

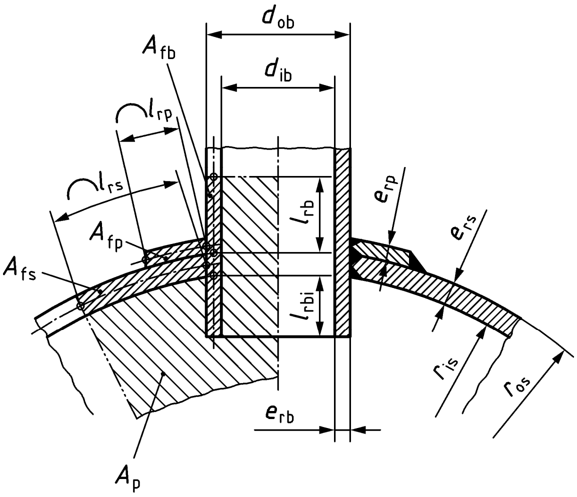
b) manhole reinforcing pad

e) welded-in tubular reinforcement

c) extruded

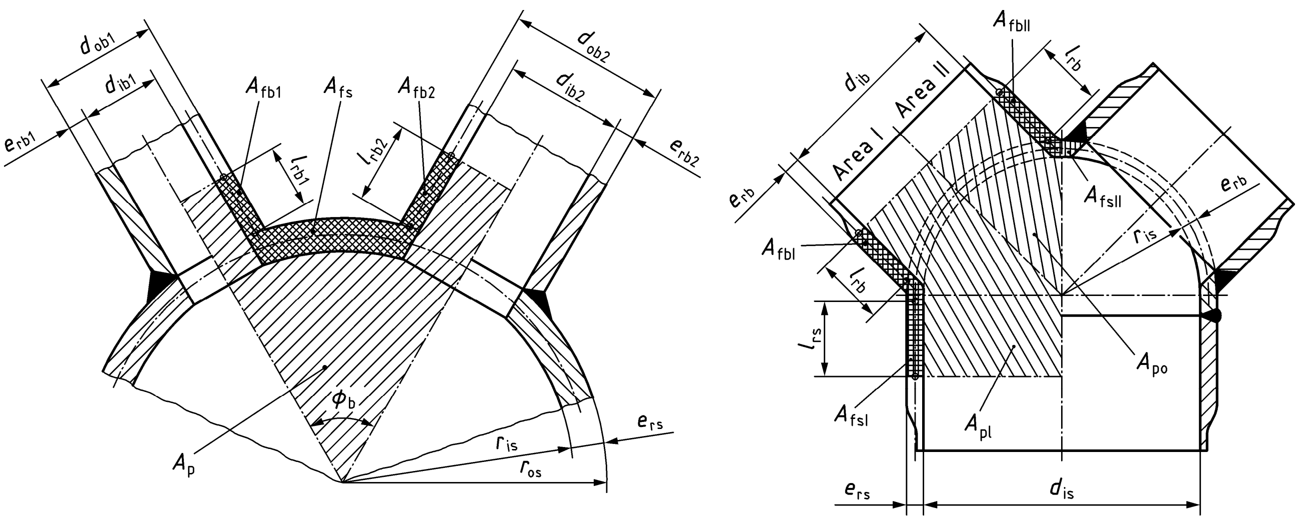
f) welded-on tubular reinforcement

Figure 10.2-6 — Determination of pressure and stress loaded areas in spherical shells and dished ends



g) isolated opening with pad-type and tubular reinforcement

Figure 10.2-6 — Determination of pressure and stress loaded areas in spherical shells and dished ends *(continued)*



a) adjacent branches

b) Y shaped branches

Figure 10.2-7 — Load diagram of spherical shell

Pad-type reinforcement shall only be permitted for operating temperatures up to and including 250 °C. It shall have close contact to the shell and be welded so that the weld throat thickness equals at least 70 % of the pad thickness. The stressed area of the reinforcing pad shall only be considered as 70 % effective in the strength condition in accordance with Equation (10.2-3).

If the main body, the reinforcement and the branch consist of materials with differing design stresses f and if the main body has the lowest design stress, this shall be used in Equation (10.2-3). If the main body has the highest design stress, the following condition shall be satisfied instead of Equation (10.2-3):

$$\left(f_{\mathrm{s}} - \frac{p_{\mathrm{c}}}{2}\right) A_{\mathrm{fs}} + \left(f_{\mathrm{b}} - \frac{p_{\mathrm{c}}}{2}\right) A_{\mathrm{fb}} + \ldots \geq p_{\mathrm{c}} A_{\mathrm{p}} \tag{10.2-15}$$

For spherical shells with Y-shaped branches in accordance with Figure 10.2-7 b), the two strength conditions for areas I and II shall be satisfied simultaneously.

Spherical shells with non-radial branches shall be calculated in accordance with Figure 10.2-8.

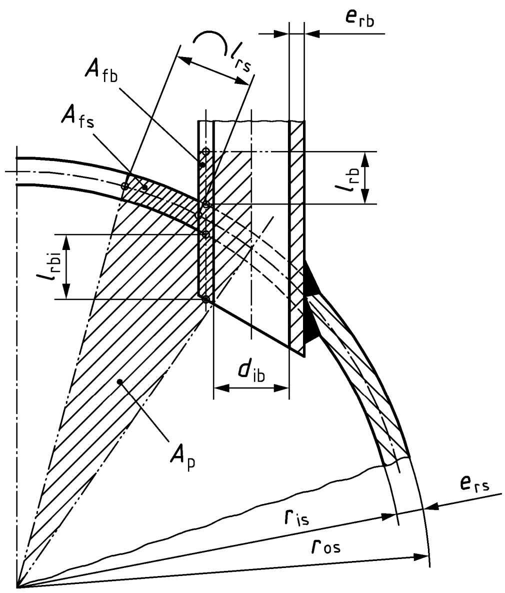


Figure 10.2-8 — Load diagram for a spherical shell with non-radial branch

10.3 Unstayed flat ends

10.3.1 General

This applies to the design and dimensions of unstayed flat ends within the following limits. Flat stayed ends shall be designed in accordance with EN 12953-3:2002.

Referring to Figure 10.3-1, in addition to any limits shown in the sketches, the following limitations shall also apply:

a) Ends formed by the hot closure of a cylindrical shell and forged end caps butt welded to the shell:

$d_i \leq 600$ mm;

$r_{ik} \geq 0{,}3\ e_{rh}$, but also $r_{ik} \geq 5$ mm.

b) Ends forged or machined from a billet and butt welded to the shell (see Figure 10.3-1 a) and Figure 10.3-1 b)):

$r_{ik} \geq 0{,}3\ e_{rh}$, but also $r_{ik} \geq 5$ mm.

c) Flat ends totally set into the shell, which has a rebate.

$d_i \leq 600$ mm

(Use restricted to steel group 1 with R_m not greater than 470 MPa).

d) Flat ends totally set into the shell, which does not have a rebate:

$d_i \leq 600$ mm

(Use restricted to steel group 1 with R_m not greater than 470 MPa).

Ends made from a plate and machined and butt welded to the shell (see Figure 10.3-1 e) and Figure 10.3-1 f)) plate material with Z35-quality in accordance with EN 10164:2004 shall be used.

e) Flat ends with a machined relief groove, butt welded to the shell:

$r_{ik} \geq 0{,}2\ e_{rh}$, but also $r_{ik} \geq 5$ mm.

For the ends machined from a plate materials with a Z35 quality in accordance with EN 10164:2004 shall be used.

For plates additional NDE of the edge area local to the weld preparation is required to ensure that no lamellar tearing occurs.

The end thickness at the groove shall not be less than required by 10.3.2.

f) Flat ends with a rebate, welded to the end of the shell:

$d_i \leq 600$ mm

For the ends machined from a plate materials with a Z35 quality in accordance with EN 10164:2004 shall be used.

(Use restricted to steel group 1 with R_m not greater than 470 MPa).

g) Flat end totally set into the shell with no rebate, full penetration welded from both inside and outside, with the inner weld having a notch-free profile:

$r_{ik} \geq 0{,}5\ e_{rh}$, but also $r_{ik} \geq 5$ mm.

The types c), d) and f) are not recommended for use if cyclic loading is expected to occur

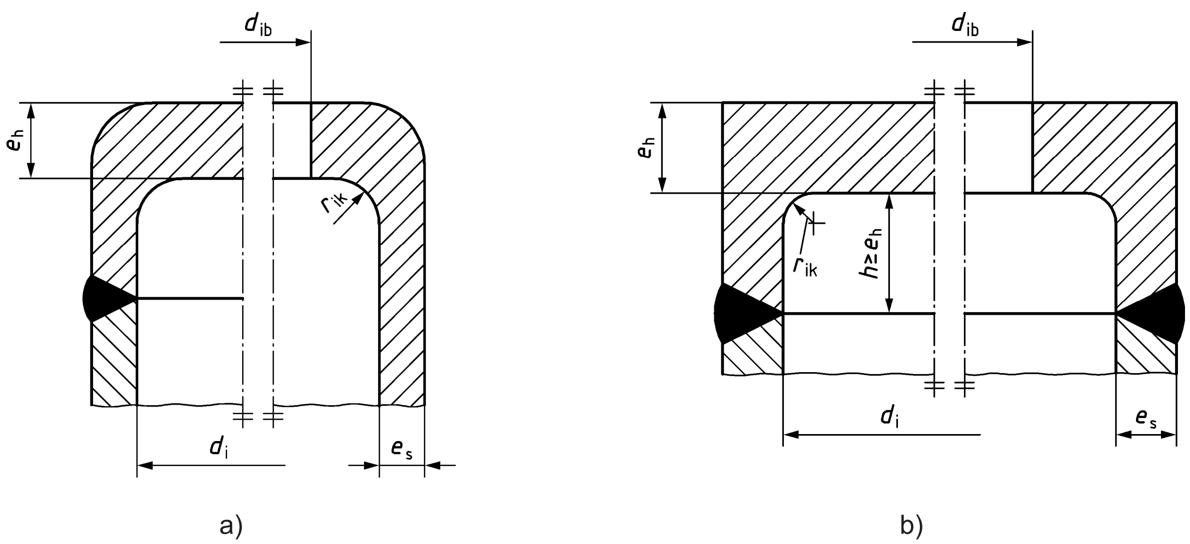


Figure 10.3-1 — Unstayed flat ends

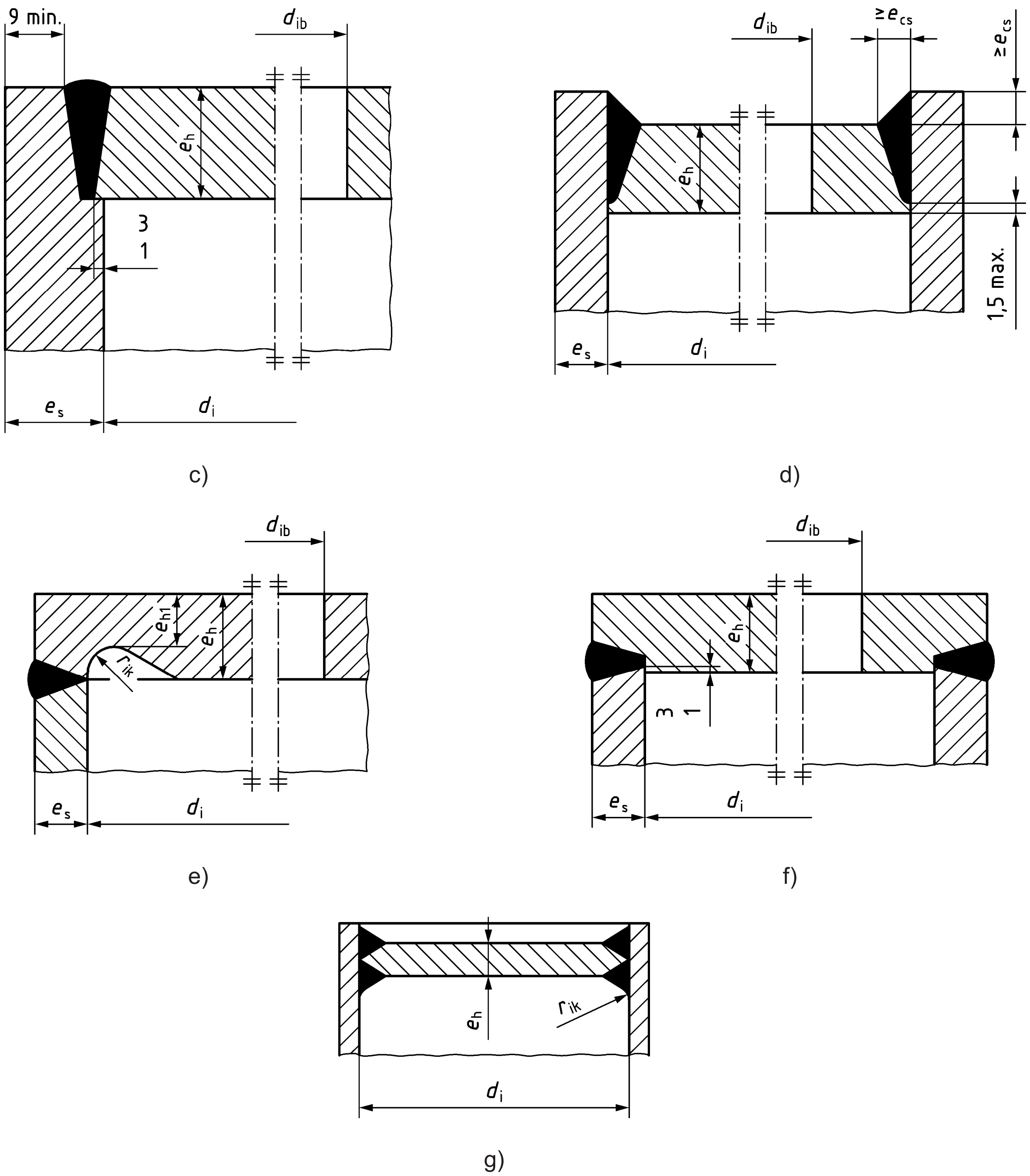


Figure 10.3-1 — Unstayed flat ends *(continued)*

10.3.2 Thickness of circular unstayed flat ends

The required wall thickness shall be at least

$$e_{h'} = e_{ch} + c_1 + c_2 \tag{10.3-1}$$

For ends of the type shown in Figure 10.3-1, e), the wall thickness shall meet the above requirement except at the bottom of the groove, where the thickness may be reduced to not less than

$$e_{h1} = e_{ch1} + c_1 + c_2 \tag{10.3-2}$$

The allowance c_1 is to cover negative fabrication tolerances. The allowance c_2 is to cover metal wastage. Normally, c_2 may be taken as zero (see 5.8).

The wall thickness remote from any relief groove shall be calculated as follows

$$e_{\mathrm{ch}} = C_1 C_2 C_3 d_{\mathrm{i}} \sqrt{p_{\mathrm{c}}/f} \tag{10.3-3}$$

and at the bottom of the groove

$$e_{\mathrm{ch1}} \geq 1{,}3\, p_{\mathrm{c}} \left(d_{\mathrm{i}}/2 - r_{\mathrm{ik}}\right)/f \qquad \text{but also} \qquad e_{ch1} \geq e_{rs} \tag{10.3-4}$$

The welds in ends to Figure 10.3-1 c), 10.3-1 d), 10.3-1 e), 10.3-1 f) and 10.3-1 g) shall be regarded as fully loaded joints (see 6.3.2) when selecting a value for f in Equation (10.3-3). If f is so reduced below that for the end base material, f_{b}, the minimum value for C_1 shall be lowered from 0,41 to $0{,}41 \cdot \sqrt{f / f_{\mathrm{b}}}$.

The value of C_1 shall be taken from Figure 10.3-2 for ends in accordance with Figure 10.3-1, a), b), e) and g).

For ends in accordance with Figure 10.3-1, c), d) and f), Figure 10.3-3 shall be used

for circular ends: $C_2 = 1$,

for unpierced ends: $C_3 = 1$.

For pierced ends, C_3 shall be taken from Figure 10.3-4. Excentric openings shall be treated as concentric openings. $d_{\mathrm{ib}}/d_{\mathrm{i}}$ shall not exceed 0,8. The opening shall not encroach on any weld between end and shell.

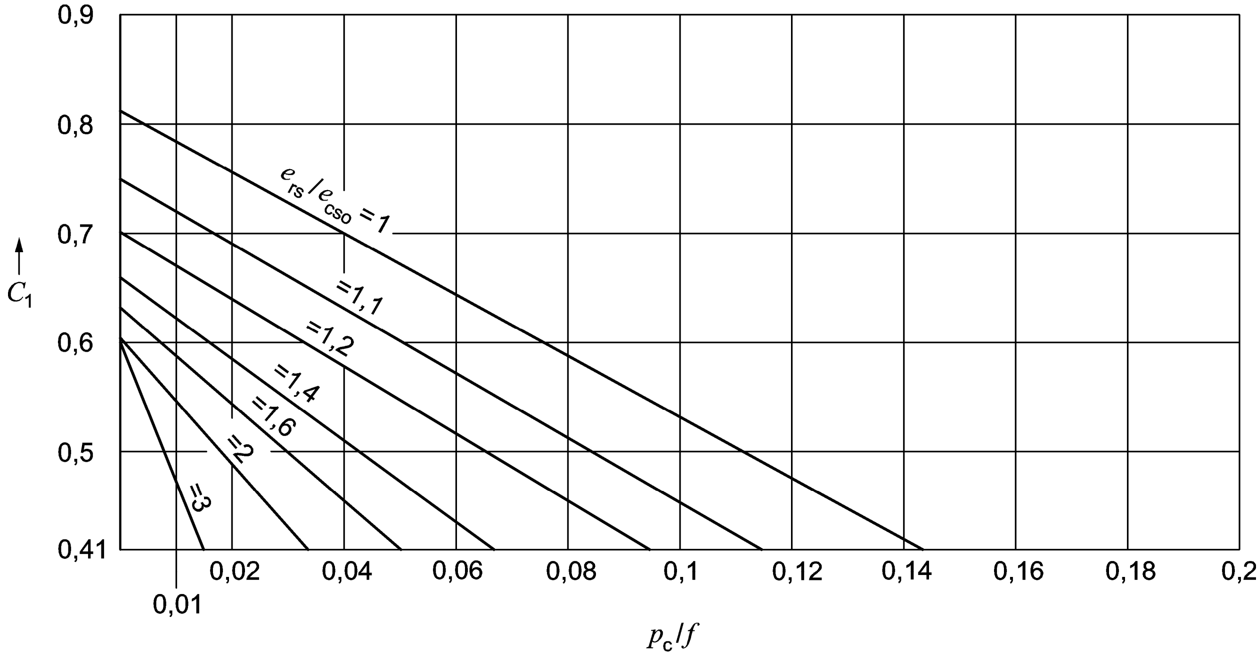
10.3.3 Thickness of non-circular and rectangular unstayed flat ends

For oval ends with major and minor axes a and b measured inside the shell, the requirements of 10.3.2 shall apply, but with b taken in place of d_{i}, and with

$$C_2 = \sqrt{3{,}4 - 2{,}4\, b/a} \text{ , but not greater than 1,6} \tag{10.3-5}$$

For rectangular ends of major and minor internal dimensions a and b, the requirements of 10.3.2 shall apply, but with b taken in place of d_{i}, and with

$$C_2 = 1{,}1\sqrt{3{,}4 - 2{,}4\, b/a} \text{ , but not greater than 1,6} \tag{10.3-6}$$



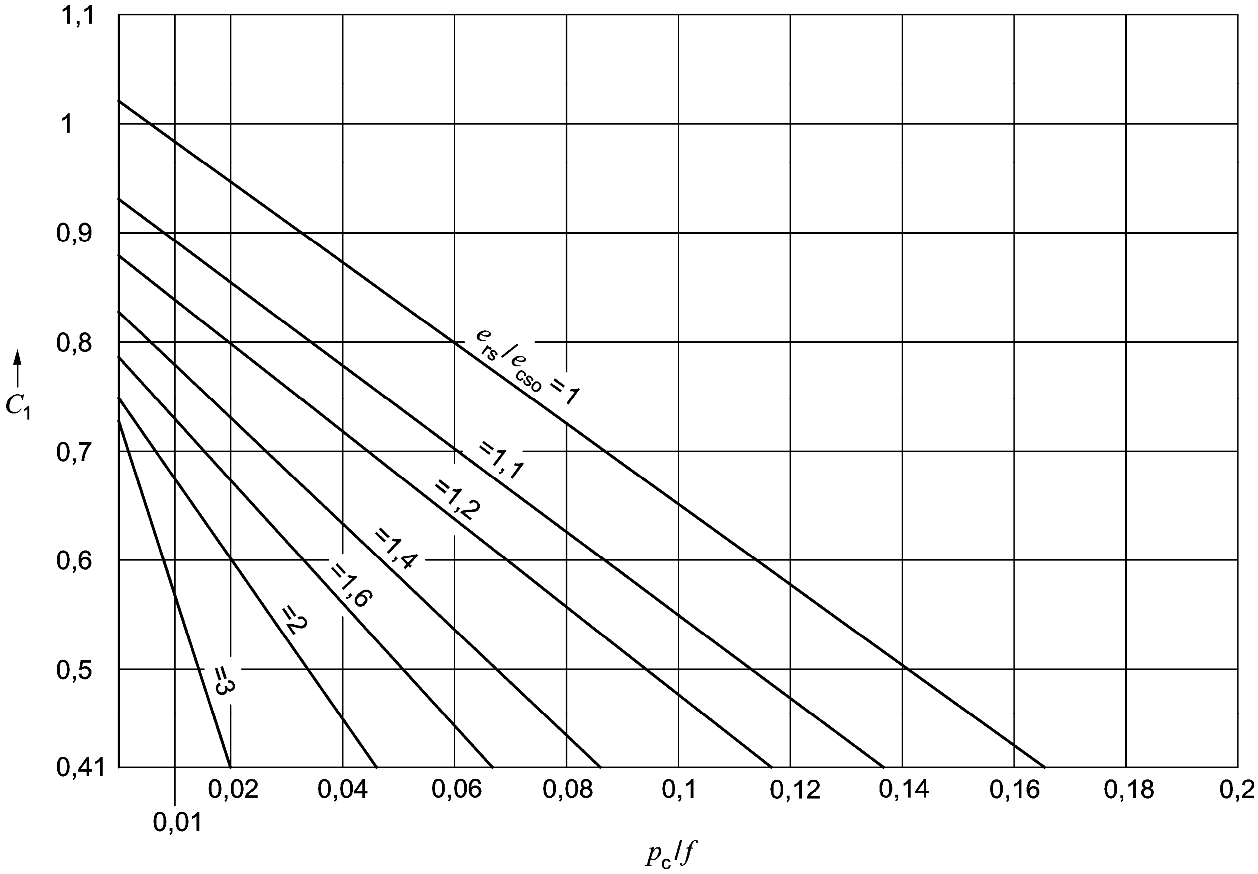
e_{cso} = calculated shell thickness with v = 1 (refer Equation (7.2-1) or (7.2-2))

$C_1 \geq 0{,}41$

$$C_1 = (-\,1{,}057\,25 - 1{,}608\,40\,x + 0{,}116\,245\,x^2 - 0{,}288\,657\,x^3)\,p_c/f + 0{,}54 + 0{,}324\,245\,x^{-1} - 0{,}668\,534\,x^{-2} + 0{,}634\,377\,x^{-3}$$

with $x = e_{rs}/e_{cso}$

Figure 10.3-2 — Factor C_1 for header ends with the butt weld in the cylindrical shell in accordance with Figure 10.3-1, a), b), e) and f)



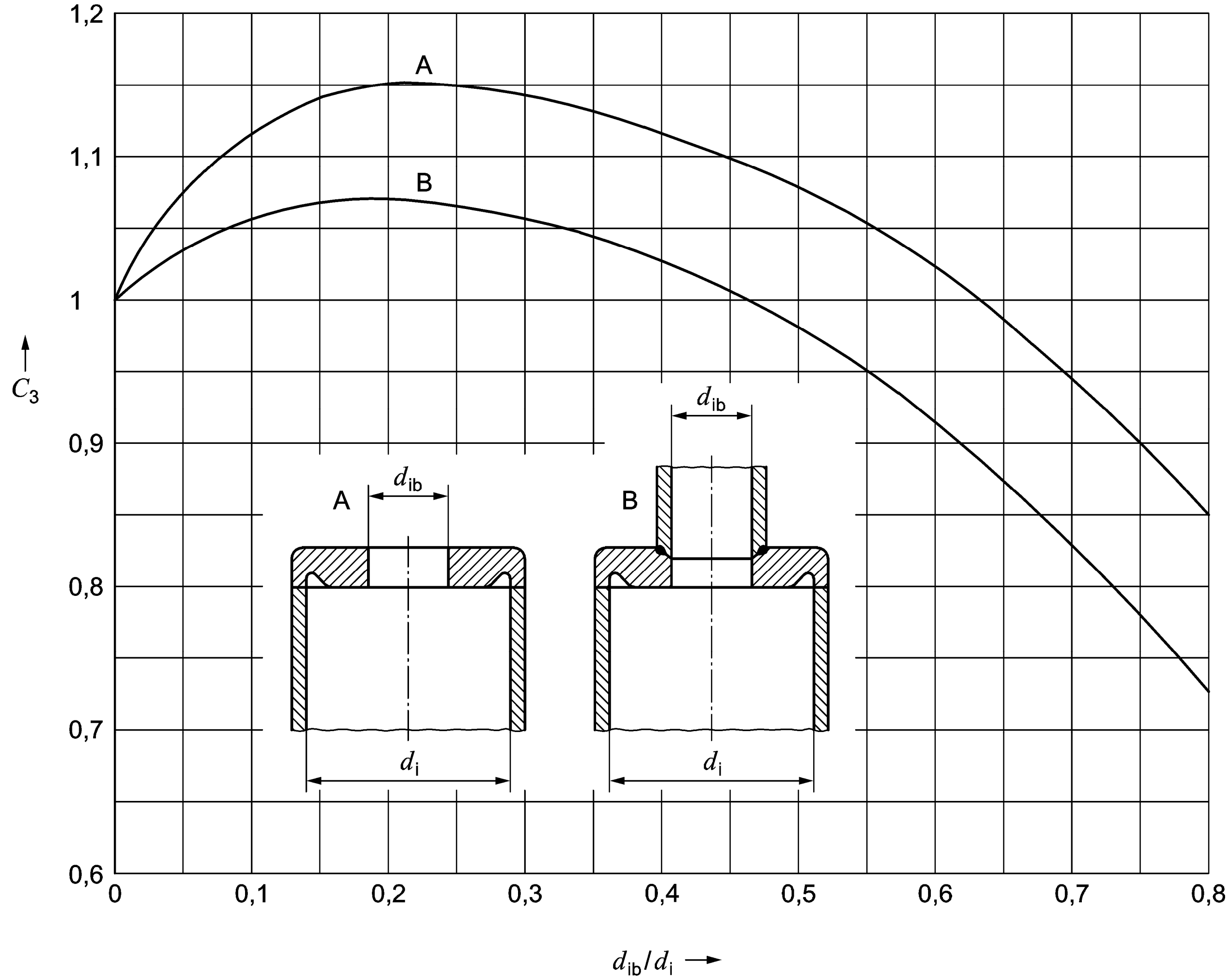
e_{cso} = calculated shell thickness with v = 1 (refer Equation (7.2-1) or (7.2-2))

$C_1 \geq 0,41$

$C_1 = (-\ 1{,}138\ 50 - 2{,}806\ 82\ x + 0{,}841\ 379\ x^2 - 0{,}517\ 434\ x^3) p_c/f + 0{,}68 + 0{,}231\ 694\ x^{-1} - 0{,}440\ 285\ x^{-2} +$
$+\ 0{,}550\ 637\ x^{-3}$

with $x = e_{rs}/e_{cso}$

Figure 10.3-3 — Factor C_1 for header ends with the tee butt weld in accordance with Figure 10.3-1 c), d) and g)



The curves are defined by the following equations, where $x = d_{ib}/d_i$

Curve A: $C_3 = 1 + 1{,}98\,x - 9{,}02\,x^2 + 18{,}53\,x^3 - 19{,}31\,x^4 + 7{,}54\,x^5$

Curve B: $C_3 = 1 + 0{,}944\,x - 4{,}31\,x^2 + 8{,}39\,x^3 - 9{,}21\,x^4 + 3{,}69\,x^5$

Figure 10.3-4 — Design factor C_3

10.4 Flat unstayed closures

For calculation pressures exceeding 2 MPa, only internally fitted doors may be used. For calculation pressures of 2 MPa and below, external closures of the blank flange type shall be used.

The inside diameter of the opening shall be limited to a maximum of 460 mm.

The thickness shall be determined in accordance with the requirements of 10.3, with d_i in accordance with Figure 10.4-1.

When the closure is external, C_1 shall be taken as 0,41, except where closures of the type shown in Figure 10.4-1 f) are used, where the bolting adds to the bending moment in the plate. In such cases, the values for C_1 given in Table 10.4-1 shall apply.

Table 10.4-1 — Values for C_1

d_L/d_i	C_1
1,0	0,45
1,1	0,50
1,2	0,55
1,3	0,60

When internal doors of the type shown in Figure 10.4-1 a) are used, factor C_1 shall be taken as 0,45. Account shall also be taken of the additional bending moment in the plate caused by the bolting. If no further exact calculation is carried out, p_c shall be multiplied by a load factor of min. 1,5.

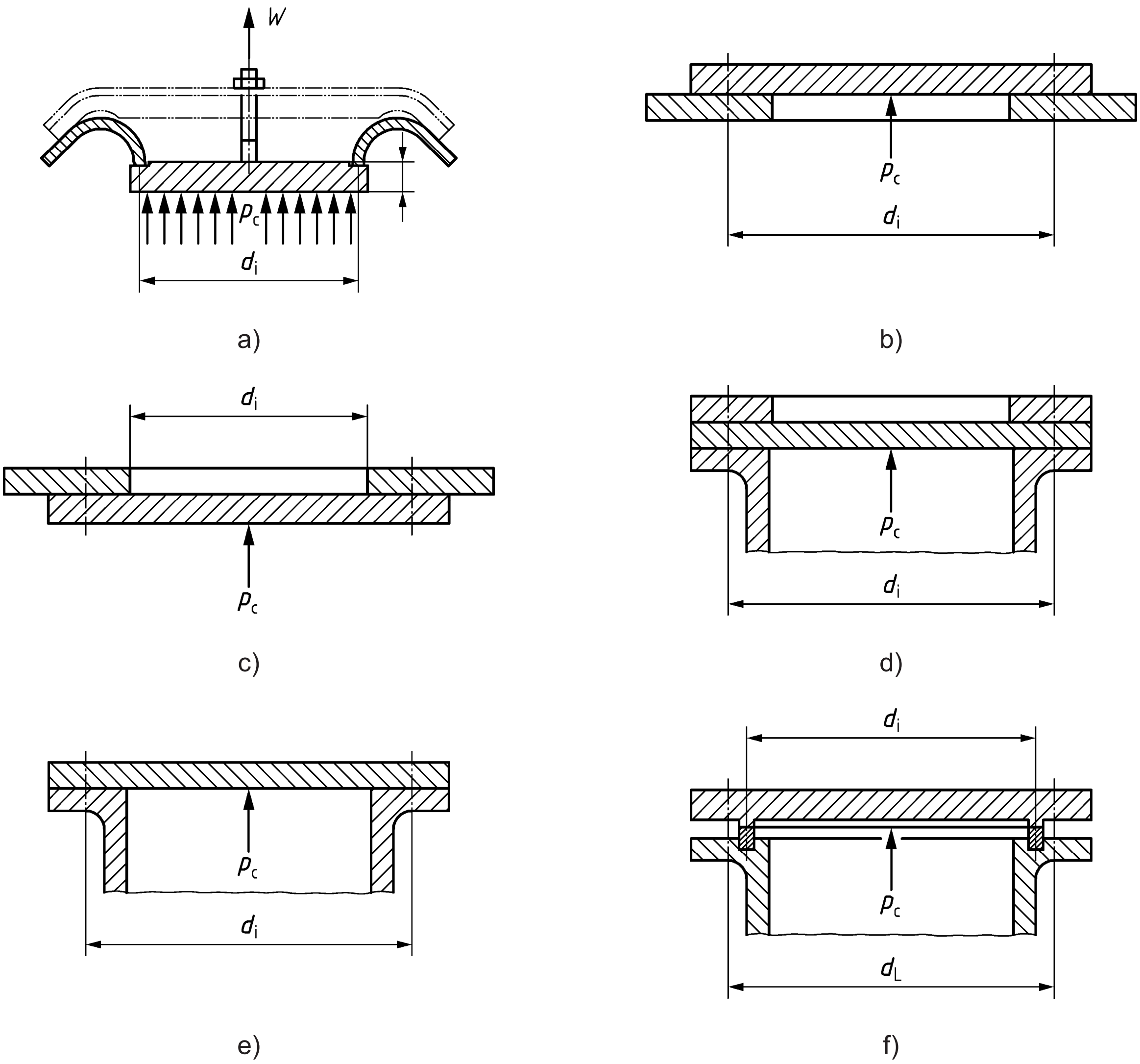


Figure 10.4-1 — Flat unstayed removable closure

11 Tubes

11.1 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table 11.1-1 shall be used.

Table 11.1-1 — Symbols

Symbol	Description	Unit
e_{cti}	Required wall thickness on the intrados of the tube bend, without allowances, in accordance with Figure 11.1-1	mm
e_{cto}	Required wall thickness on the extrados of the tube bend, without allowances, in accordance with Figure 11.1-1	mm
$e_{ti'}$	Required wall thickness on the intrados of the tube bend, with allowances	mm
$e_{to'}$	Required wall thickness on the extrados of the tube bend, with allowances	mm
r_b	Curvature radii of the tube bend, in accordance with Figure 11.1-1	mm

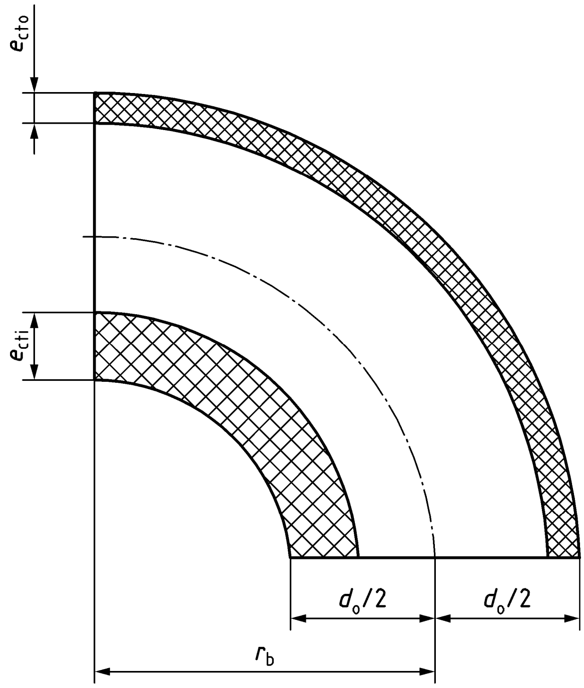


Figure 11.1-1 — Notations used for tube bends

11.2 Thickness of straight boiler tubes

11.2.1 Required wall thickness with allowances

The required wall thickness with allowances for the straight tube shall be derived as follows

$$e_t = e_{ct} + c_1 + c_2 \tag{11.2-1}$$

Cladding for the purpose of metal wastage resistance shall not be considered as part of the pressure thickness.

For tubes heated from all sides where $d_o \leq 44{,}5$ mm and where the circumferential stress is decisive in the design of the tubes, the allowance may be $c_1 = 0$, provided that the allowable negative tolerance in the European Standard for the tube is not exceeded. Where the axial stress is decisive for the design of the tubes the appropriate value of c_1 shall be added.

11.2.2 Required wall thickness without allowances

The required thickness without allowances for the straight tube shall be determined in accordance with one of the following equations:

$$e_{ct} = \frac{p_c d_o}{(2f - p_c)v + 2p_c} \tag{11.2-2}$$

or for $v = 1$ (tubes without opening)

$$e_{ct} = \frac{p_c\, d_o}{2f + p_c} \tag{11.2-3}$$

or

$$e_{ct} = \frac{p_c\, d_i}{(2f - p_c)v} \tag{11.2-4}$$

or for $v = 1$

$$e_{ct} = \frac{p_c\, d_i}{2f - p_c} \tag{11.2-5}$$

where v is the efficiency defined in 8.2.

11.2.3 Minimum thickness

The thickness $e_t - c_1$ (nominal thickness minus allowance c_1) shall not be less than the minimum value given in Table 11.2-1.

NOTE The diameters used for calculation purposes are nominal diameters i.e. manufacturing tolerances are not applied to diameters used in the calculations.

Table 11.2-1 — Minimum thickness of straight tubes

Nominal outside diameter mm	Minimum thickness $e_t - c_1$ mm
≤ 38	1,7
> 38 ≤ 51	2,2
> 51 ≤ 70	2,4
> 70 ≤ 76	2,6
> 76 ≤ 95	3,0
> 95 ≤ 102	3,3
> 102	3,5

11.2.4 Circumferentially butt welded tubes

To accommodate the levels of NDE required by EN 12952-6:2011 the following calculation shall be carried out to ensure the tube thickness at the butt weld is adequate for the loading imposed when the appropriate weld joint coefficient v_{NDE} is applied

$$e_{wt} \geq \frac{d_i}{2}\left\{\sqrt{1+\frac{p_c + 4Q/\left(\pi\, d_i^2\right)}{\left(f - p_c/2\right) v_{NDE}}} - 1\right\} \tag{11.2-6}$$

or

$$e_{wt} \geq \frac{d_o}{2}\left\{1-\sqrt{1-\frac{p_c + 4\,Q/\left(\pi\, d_o^2\right)}{p_c+\left(f - p_c/2\right) v_{NDE}}}\right\} \tag{11.2-7}$$

where

e_{wt} is the minimum required thickness of the tube at the butt joint,

Q is the axial force on the circumferential butt weld caused by forces additional to pressure

and

v_{NDE} = joint coefficient 1,0 for 100 % NDE and 0,85 for 10 % NDE.

11.3 Thickness of tube bends and elbows

11.3.1 General

Where tubes are to be bent, the following requirements shall be taken into account at the design stage.

11.3.2 Required wall thickness with allowances

The required wall thickness for bends and elbows shall be:

for the intrados

$$e_{\text{ti'}} = e_{\text{cti}} + c_1 + c_2 \tag{11.3-1}$$

for the extrados

$$e_{\text{to'}} = e_{\text{ct0}} + c_1 + c_2 \tag{11.3-2}$$

11.3.3 Required wall thickness without allowances

If a more exact calculation in accordance with Annex A is not carried out, the required wall thickness without allowances for bends and elbows shall not be less than the following:

for the extrados

$$e_{\text{cto}} = e_{\text{ct}} \frac{\dfrac{2\,r_{\text{b}}}{d_{\text{o}}} + 0{,}5}{\dfrac{2\,r_{\text{b}}}{\text{d}_{\text{o}}} + 1} \tag{11.3-3}$$

for the intrados

$$e_{\text{cti}} = e_{\text{ct}} \frac{\dfrac{2\,r_{\text{b}}}{d_{\text{o}}} - 0{,}5}{\dfrac{2\,r_{\text{b}}}{d_{\text{o}}} - 1} \tag{11.3-4}$$

where e_{ct} is the required wall thickness for a straight tube in accordance with 11.2.2.

For practical reasons the calculation to determine the thickening on the intrados of tube bends as given in Equation (11.3-4) shall not be required for bends on tubes ≤ 80 mm diameter.

11.3.4 Departure from circularity of tube bends

The departure from circularity *u* of tube and tube bends u shall be:

$$u = 2\frac{\hat{d}_o - \check{d}_o}{\hat{d}_o + \check{d}_o} \cdot 100\% \tag{11.3-5}$$

where

$\hat{d}_o$ is the maximum outside diameter measured at the tube bend apex,

$\check{d}_o$ is the minimum outside diameter measured at the same cross section as $\hat{d}_o$.

and shall not exceed the limits given in EN 12952-5:2011, 7.3.7.

11.4 Flexibility of integral tubing systems

11.4.1 General

The tubes shall be arranged so that the system is sufficiently flexible to absorb the whole of its own expansion and that of the connecting equipment.

Account shall also be taken of additional loadings in the design of branches to drums, headers, etc. in accordance with 7.5.

11.4.2 Analysis

A flexibility analysis shall be used if there is any doubt as to the ability of the system to satisfy the specified requirements. Such a method of analysis is given in EN 13480-3:2002.

The designer shall be responsible for performing the necessary analysis, unless the system complies with one of the following criteria:

a) the tubing system duplicates a successfully operating installation or replaces a system with a satisfactory service record;

b) the tubing system can be assessed adequately by comparison with previously analysed systems.

NOTE The tubes may be pre-stressed (cold pull) to reduce end effects under hot conditions of working and also to reduce hot stress levels. (For recommended methods, reference should be made to EN 13480-3).

11.5 Structural attachments to tubes

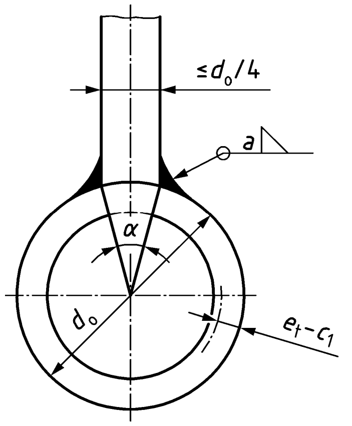
11.5.1 General

Where structural attachments such as lugs and brackets, are welded to tubes for load-carrying purposes, see 5.9, the material of the attachment and of the weld shall be suitable for the environment and be compatible with the tube material.

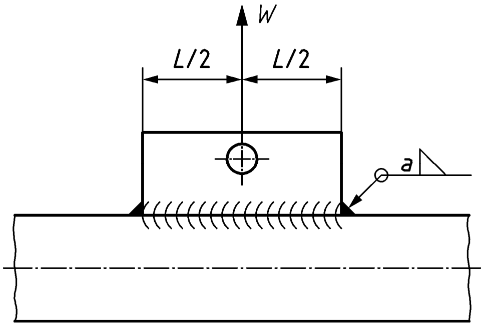
The weld shall be continuous around the attachment without interruption, except in the case of U profiles in accordance with Figure 11.5-1 e).

The thickness of the attachment section, measured in the tube circumferential direction, shall not exceed one-quarter of the tube diameter at the point of attachment to the tube.

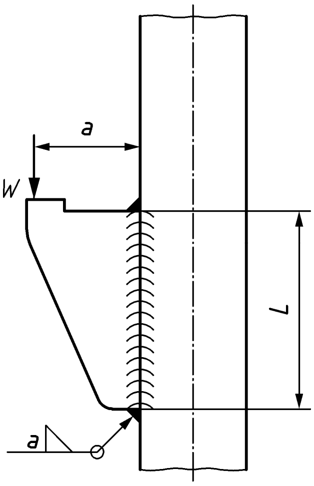
NOTE Figure 11.5-1 is intended to illustrate typical details for the clarification of design requirements.



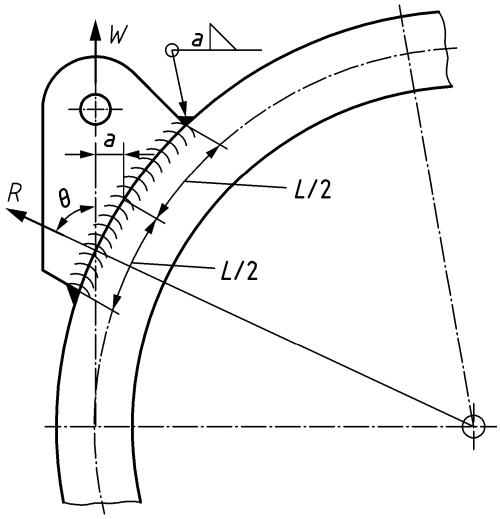
a) Structural attachment welded to tube (section)



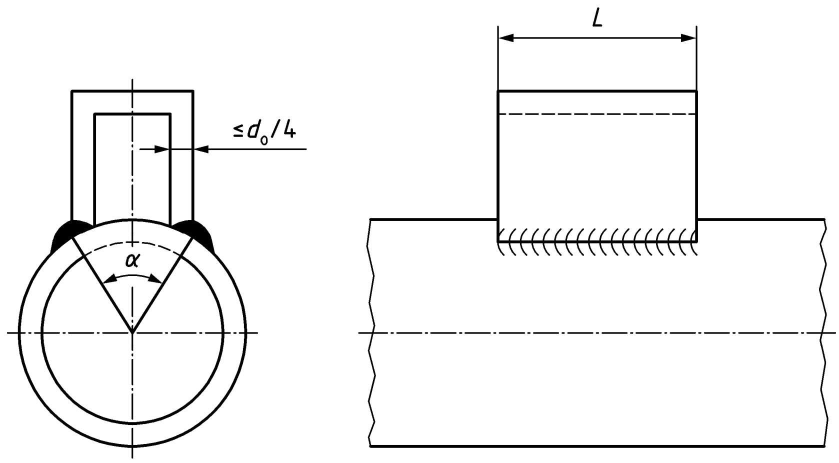
b) Horizontal tube vertical symmetrical support



c) Support on vertical tube



d) Curved tube vertical support



e) Double leg structural attachment

Figure 11.5-1 — Typical structural attachments to tubes

11.5.2 Attachments welded on tube bends

Where such attachments are welded on tube bends the design shall ensure that the stresses resulting from the loading remain within acceptable limits. This shall be determined by stress analysis complying with the criteria of EN 13445-3:2009 or EN 13480-3:2002, or justified by documented evidence of previous satisfactory operating experience with similar designs.

11.5.3 Length of attachments

Where attachments are welded on straight tubes the length of the attachment, measured along the axis of the tube, shall comply with 11.5.4 and 11.5.5, or be determined by stress analysis conforming to the criteria of EN 13445-3:2009 or EN 13480-3:2002 or justified by documented evidence of previous satisfactory operating experience with similar designs.

11.5.4 Limit of intensity in the case of radial loading

The intensity of radial loading shall not exceed that given by Figure 11.5-2 for tensile or Figure 11.5-3 for compressive attachment loading

where

q is the greatest intensity of radial load, in Newtons per millimetre;

f is the design stress of the tube material as used in 11.1, in megapascal;

$e_t - c_1$ is the minimum tube thickness i.e. nominal thickness less allowance, in millimetres;

d_o is the outside diameter of the tube in millimetres;

α is the angle subtended by the attachment at the tube centre in degrees (see Figure 11.5-1).

11.5.5 Calculation of intensity in the case of radial loading

For an attachment welded longitudinally along a tube, the greatest occurring tensile and compressive radial load intensities shall be calculated from the following equation:

$$q = \frac{R}{L} \pm \frac{6\,W\,a}{L^2} \tag{11.5-1}$$

where

W is the load carried by the attachment in Newtons;

a is the eccentricity of the line of action of load W about the line of attachment to the tube in millimetres;

R is the radial component of the load W in Newtons, $R = \cos\theta \cdot W$;

L is the length of the attachment in millimetre.

NOTE Where eccentric loading occurs, it may be necessary to check both the radial compressive and tensile load intensities, which occur at opposite ends of the attachment. W, R and q are positive when tensile and negative when compressive.

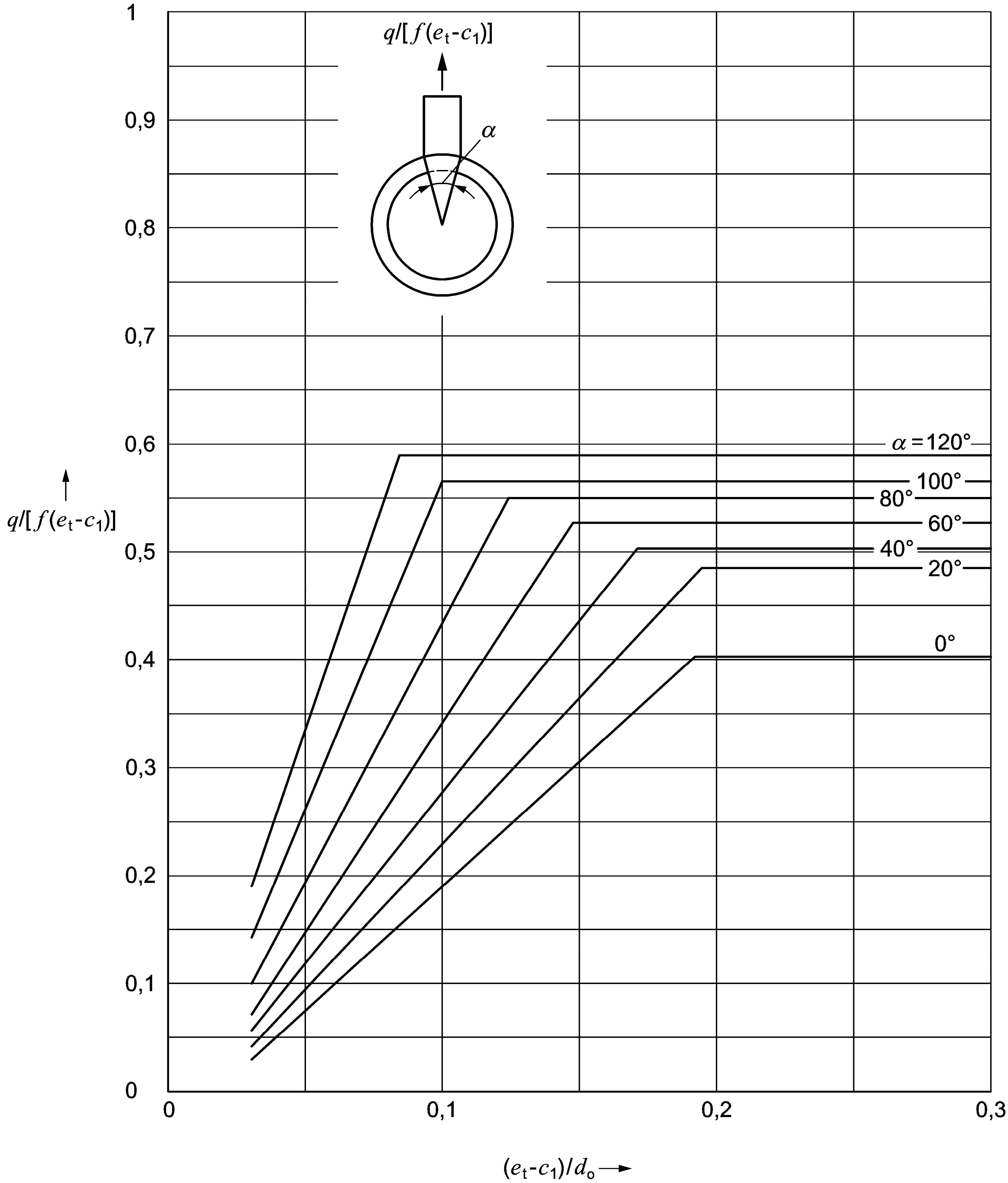


Figure 11.5-2 — Allowable tensile loading of the attachment

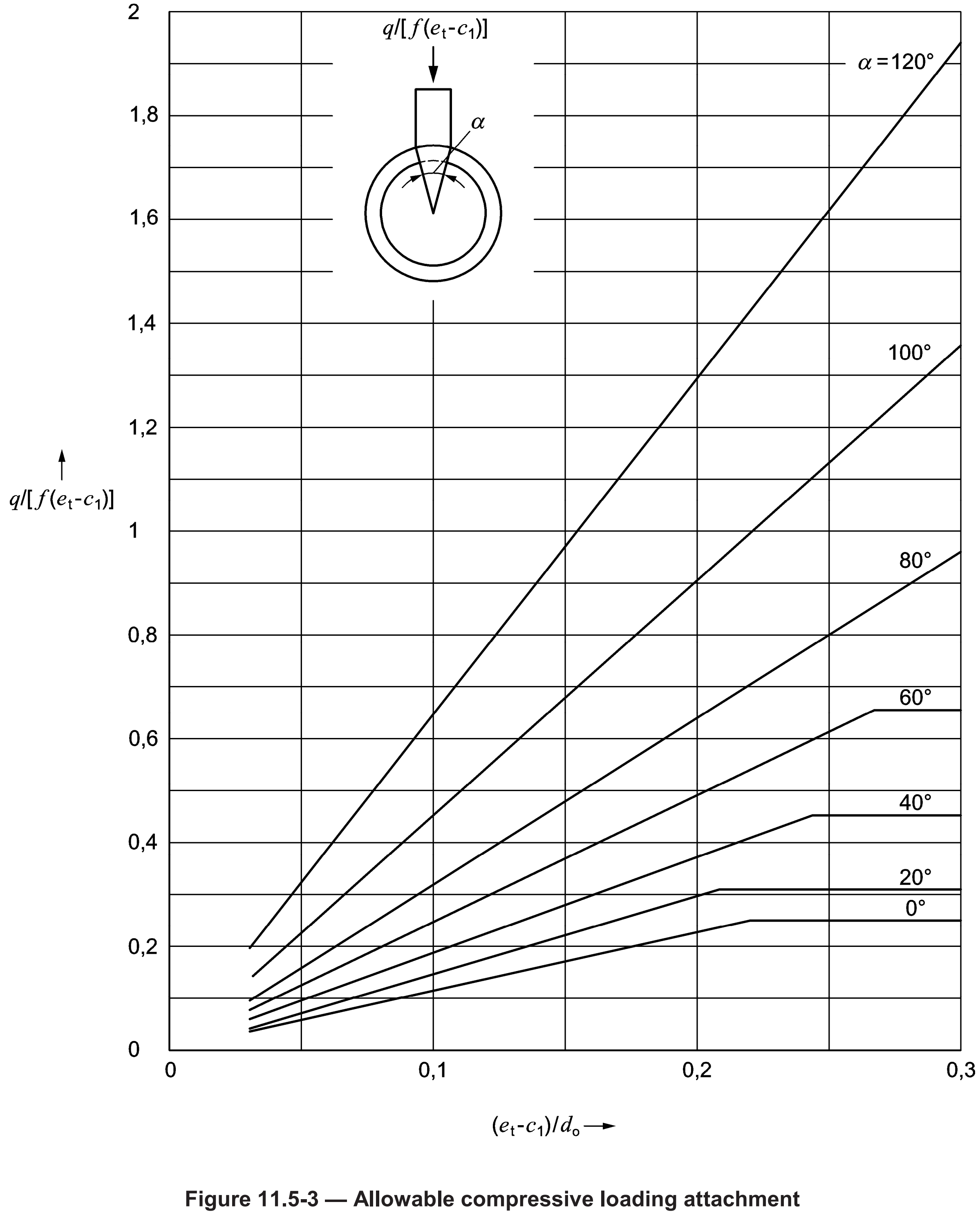


Figure 11.5-3 — Allowable compressive loading attachment

11.5.6 Strength of welds

In calculating the strength of welds for structural attachments, the allowable design stress shall correspond to that of the weaker of the tube or structural attachment, multiplied by the following weld strength reduction factors v_w:

0,7 for fillet welds;

0,75 for partial penetration welds (with or without superimposed fillet welds);

1,0 for full penetration.

The loaded area of the weld shall be taken as the throat thickness multiplied by the length of weld. For the purpose of stress calculations the throat thickness shall be taken as

a) 0,7 of the leg length for fillet welds;

b) the depth of the groove for partial penetration welds;

c) the thickness of the structural attachment for full penetration welds.

For compound welds, the effective throat thickness shall be the sum of the throat thickness of the constituent parts.

11.6 Fitting and joining of heated tubes

11.6.1 Fitting of tubes

Fitting of tubes to drums or headers shall be in accordance with EN 12952-5:2011, 8.11 or 9.4.

11.6.2 Joining of heated tubes

Joining of heated tubes shall be done by butt welds in accordance with EN 12952-5:2011, 8.11. Socket welds, sleeve welds, flanges and screwed joints shall not be used.

11.7 Joining of unheated tubes

11.7.1 General

Unheated tubes shall be joined by welded or integral flanges, butt welds, socket welds, or screwed and seal welded joints. Screwed flanges and sleeve welds shall not be used. Flanges shall not be used where the bolts would be exposed to products of combustion. See also EN 12952-5:2011, 8.11 and 9.4.

11.7.2 Flanges and bolting

The flanges and bolting for ordinary bolted flange joints shall be in accordance with EN 1092-1:2007 or EN 1759-1:2004. For welded-on flanges the weld penetrations and flange types shall be selected in accordance with EN 1092-1:2007 or EN 1759-1:2004. Gasket selection (e.g. material and dimensions) shall be such that the gasket factor and seating stresses are consistent with the flanges and bolting requirements.

NOTE Special joints and special types of flanges may be used provided that they are shown to be suitable for the design conditions.

11.7.3 Butt welded joints

Butt welded joints shall be in accordance with EN 12952-5:2011, 8.11.

11.7.4 Socket welded joints

Socket welded joints shall be in accordance with 8.5.3.

11.7.5 Screwed or screwed and seal welded joints

Screwed or screwed and seal welded joints shall be in accordance with 8.5.2.

12 Pressure parts of irregular shape

12.1 Hydrostatic test for determining the allowable internal pressure

12.1.1 General

This applies to components for which no design rules exist and where the strength of a pressure part cannot be safely calculated or where a theoretical stress analysis is unsuitable.

The components shall satisfy the requirements of 12.1.2 or 12.1.3 as appropriate.

These requirements shall not apply to components which are exposed to significant cyclic loading as given in 5.5, or where creep is critical, or where there are significant loadings in addition to internal pressure. In these cases, appropriate tests including these effects shall be carried out.

Programmes shall be prepared for all tests, and included in the documentation submitted for design approval. Approval shall be obtained prior to testing.

12.1.2 Proof test to produce yielding

This method shall only be permitted for materials with adequate ductility and designs where buckling is not a criterion for failure.

A full-sized test component shall be made which shall be suitable for hydrostatic testing. The component shall be capable of examination at a sufficient number of locations to ensure that all critical areas can be adequately observed and measured. This test will only be applied when the product of the maximum allowable pressure PS and the volume V is less than 6 000 bar litres.

The component shall be tested by gradually increasing the hydrostatic pressure until significant general yielding is first observed. The test shall then be discontinued and the hydrostatic test pressure recorded. This pressure p_{ty} shall be used in Equation (12.1-1) to determine the maximum allowable pressure PS for the component.

NOTE The terms significant general yielding is intended to apply to the type of yielding where a substantial portion of the component exceeds the yield point of the material. It is not intended to apply to the type of yielding which occurs at the first application of test pressure because of stress redistribution at points of stress concentration.

The test may be discontinued prior to the achievement of yielding if the hydrostatic test pressure has attained a value p_{ty} equivalent to the required maximum allowable pressure PS calculated from a transposition of Equation (12.1-1).

Maximum allowable pressure PS

$$PS = a_1\, a_2\, p_{ty} \tag{12.1-1}$$

where

a_1 is the factor for considering corrosion, erosion and wear

$$a_1 = \left(\frac{e - c_2}{e_{\mathrm{a}}}\right)^2, \tag{12.1-2}$$

if the bending stress across the wall thickness is decisive for judging the strength,
or

$$a_1 = \frac{e - c_2}{e_a}, \tag{12.1-3}$$

if the membrane stress is decisive

a_2 is the factor for considering the difference between the assessment and the test condition.

$$a_2 = \frac{f}{\overline{R}_e} \tag{12.1-4}$$

e is the minimum manufactured wall thickness specified in the design at the location controlling the design. If this is not known, the minimum specified wall thickness at the weakest location shall be taken;

e_a is the actual thickness of the test component, measured before the test at the same location as e;

f is the design stress at the assessment condition (in accordance with 6.3);

P_{ty} is the gauge pressure at which the test was discontinued;

$\overline{R}_e$ is the mean value of actual yield strength values at room temperature, obtained by means of tensile test specimens.

12.1.3 Proof test to destruction

This method shall be applicable to all materials.

A full-size test piece shall be made which shall be suitable for hydrostatic testing. The test piece shall be pressurized to failure and the hydrostatic pressure at failure shall be recorded. The maximum allowable pressure PS shall be determined as follows:

$$PS = a_1\, a_3\, p_{tB} \tag{12.1-5}$$

$$a_3 = \frac{f}{\overline{R}_m} \tag{12.1-6}$$

where

a_1 is the same factor as in Equation (12.1-2);

a_3 is the factor for considering the difference between the assessment and the test condition;

p_{tB} is the pressure recorded at failure;

$\overline{R}_m$ is the mean value of actual tensile strength values at room temperature, obtained by means of tensile test specimens.

12.2 Numerical methods

12.2.1 General

The following applies to components for which no design rules exist or for which the strength cannot be safely determined by means of the equations given in this European Standard.

12.2.2 Methods

One of the following calculation methods shall be used:

a) freebody method;

b) finite differences method;

c) finite element method.

12.2.3 Evaluation of stress

The evaluation of stress shall be carried out in accordance with EN 13445-3:2009.

13 Fatigue

13.1 General

13.1.1 Procedure

When pressure parts are subject to significant fluctuating loading, as described in 13.1.2, the need for a fatigue analysis shall be assessed using 13.3.

If a fatigue analysis is required, the fluctuating stresses shall be calculated in accordance with 13.1.3. The fatigue damage shall be calculated using Annex B and assessed against the criteria given in 13.1.4.

Due to the simplicity of this analysis, the results may be conservative with respect to life prediction. More complex methods, e.g. finite element analysis, may be applied to obtain more exact life predictions.

Isolated holes not larger than 20 mm in diameter need not be subjected to fatigue analysis.

13.1.2 Fatigue loading

The numbers and types of transient events, such as start-ups and load changes, to be experienced by the boiler during its design life shall be specified by the purchaser. If they are not specified, 2000 cold start-ups shall be assumed, with increased fatigue damage margins as defined in 13.1.4.

During service, the pressure parts shall be subjected to cyclic or repeated stresses (see Figure B.1), caused by fluctuating loads, which taken together over the design life, shall form the fatigue loading. The fatigue loading shall comprise different loading events, i.e. well defined loading sequences which may be repeated during the design life. Realistic assessment of the fatigue loading is crucial to the calculation of fatigue damage. Thus, the loading assumed for the fatigue assessment shall represent an upper bound estimate of the fluctuating loads to be experienced by the boiler or component during the full design life.

Sources of fluctuating loading acting on the boiler or component shall be identified. Account shall be taken of all likely operational or environmental effects arising from the reasonably foreseeable usage of the boiler during its design life.

Examples are as follows

a) application of fluctuations in pressure (including testing);

b) temperature transients;

c) restrictions of expansion or contraction during normal temperature variations;

d) forced vibrations;

e) variations in external loads.

The effect of fatigue loading shall be represented by its stress history, i.e. the stress variation at a given point in the boiler during the fatigue loading event (see Figure B.3).

13.1.3 Calculation of fluctuating stress

13.4 shall give an approximate but acceptable method of calculating the fluctuating stresses caused by pressure and temperature at cylinder to cylinder and cylinder to sphere intersections. The stresses so calculated may be used in Annex B without consideration of the notch factors in B.5.

If a more accurate analysis is desired, a detailed stress analysis shall be carried out.

13.1.4 Fatigue assessment

The load cycle history shall be classified into n_1 cycles with the same applied fluctuating stress $2f_{\text{va1}}$, mean fluctuating stress $\overline{f}_{\text{v1}}$ and reference temperature t_1^*; n_2 with stresses $2f_{\text{va2}}$ and $\overline{f}_{\text{v2}}$, temperature t_2^*; and so on.

For each set of n, $\overline{f}_{\text{v}}$, $2f_{\text{va}}$ and t^*, Annex B shall be used to determine the maximum allowable number of load cycles N.

The fatigue damage shall be assessed using the linear cumulative damage hypothesis as follows:

$$\sum_k \frac{n_k}{N_k} = \left(\frac{n_1}{N_1} + \frac{n_2}{N_2} + \ldots \frac{n_k}{N_k} \right) \tag{13.1-1}$$

This summation is termed the usage factor, and shall not exceed 1,0 where the fatigue loading has been specified. When this is not the case and 2 000 cold starts have been assumed, the usage factor shall not exceed 0,4.

13.2 Symbols and abbreviations

In addition to symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table 13.2-1 shall be used.

Table 13.2-1 — Symbols and abbreviations

Symbol	Description	Unit
B	coefficient from Figure 13.3-2 or 13.3-3	° C
C_{v}	coefficient from Figure 13.3-6	—
C_{Θ}	coefficient from Figure 13.3-5	—
c_{p}	mean specific heat of metal during temperature transient	J/(kg · K)

Table 13.2-1 *(continued)*

Symbol	Description	Unit
D_{th}	thermal diffusivity of metal = $k/\rho_m\ c_p$ (see NOTE below)	m^2/s
$\bar{e}$	mean thickness at the point under consideration	mm
e_a	smaller thickness at a thickness change	mm
e_b	larger thickness at a thickness change	mm
e_{mb}	mean wall thickness of branch, measured value if available	mm
e_{ms}	mean wall thickness of main body, measured value if available	mm
f_1, f_2, f_3	principal stresses	MPa
f_{tang}	stress tangential to the main body and tangential to the branch	MPa
$f_{tang\ p}$	pressure component of f_{tang}	MPa
$f_{tang\ t}$	thermal component of f_{tang}	MPa
f_{ax}	stress axial to the main body	MPa
f_{rad}	stress radial to the main body	MPa
$\Delta f_{12}, \Delta f_{22}, \Delta f_{31}$	differences of principal stresses	MPa
$\Delta \check{f}_{12}, \Delta \hat{f}_{12}$	minimum and maximum value of Δf_{12} that occurs during operation time	MPa
Δf_{tang}	total range of circumferential (principle) stress	MPa
$\Delta f_{tang\ t}$	range of circumferential thermal stress	MPa
Δf_v	reference stress amplitude	MPa
g_s	factor for partition of the thermal stress range	—
h	mean heat transfer coefficient between internal fluid and metal during transient	$W/(m^2 \cdot K)$
k	mean thermal conductivity of metal during transient	$W/(m \cdot K)$
N_B	Biot no. = $h\bar{e}/k$ (see NOTE below)	—
N_F	Fourier no. at end of transient = $D_{th} t / \bar{e}^{-2}$ (see NOTE below)	—
N	number of load cycles to be expected during operation	—
p_o	operating pressure	MPa
p_{max}	upper pressure level of a load cycle	MPa
p_{min}	lower pressure level of a load cycle	MPa
T_f	time during which the internal fluid temperature is changing	s
t^*	decisive temperature of a load cycle	°C
Δt	wall temperature difference, defined as integral mean wall temperature minus inside wall surface temperature	K

Table 13.2-1 *(continued)*

Symbol	Description	Unit
Δt_f	temperature change between two steady state conditions	K
v_t	rate of temperature change	Ks^{-1}
α	stress concentration factor	—
α_m	stress concentration factor due to pressure for openings in cylindrical main body	—
α_{sp}	stress concentration factor due to pressure for openings in spherical shell	—
α_t	stress concentration factor due to thermal stress	—
β_L	coefficient of linear thermal expansion	K^{-1}
γ	temperature dependent factor	—
γ_{cyl}	shape factor for cylindrical shell	—
γ_{sp}	shape factor for spherical shell	—
ρ_m	density of metal	kg/m^3
v	Poisson's ratio	—
Θ	temperature (see Figure 13.3-1)	°C
$\Delta\Theta$	temperature difference (see 13.3.4 e))	K
NOTE The units shown are those normally used. Conversion may be necessary for use in the dimensionless equations.		

13.3 Exemption rule for fatigue analysis

13.3.1 General

A fatigue analysis shall not be required if the conditions for materials according to 13.3.2 and for the loadings according to 13.3.3 are satisfied.

13.3.2 Materials

Welded connections are made between materials with similar coefficients of expansion, i.e. the connected materials derive of only one of the following three classes:

a) carbon or ferritic alloy steels (Cr < 3 %),

b) high-chromium steels,

c) austenitic stainless steels.

13.3.3 Loadings

It can be demonstrated from previous experience of similar materials, geometries and operating conditions that the loading conditions imposed will cause only insignificant fatigue, or, it concerns exceptional operating conditions, hydrostatic test etc. or

each of the following conditions a) to f) is fulfilled:

a) the component has been designed for sustained pressure in accordance with the rules in EN 12952-3;

b) the total number of cold starts is less than 3 000. Other starts and load changes, where the pressure change exceeds 50 % of the maximum operational pressure, shall also be classed as cold starts;

c) the total number of starts and load changes, where the pressure change does not exceed 50 % of the maximum operational pressure, shall be less than 10 000;

d) the mechanical loading on branches shall be limited so that the limiting value of parameter λ is not exceeded. The limiting value is the lesser of 1 and that required to satisfy the Equation (13.3-3).

$$\lambda = \frac{M}{f\,z} \tag{13.3-1}$$

where

λ is the ratio of the total stress, exclusive of stress concentrations, caused by mechanical loads, to $3f$;

f is the design stress at calculation temperature t_c;

M is the total moment applied to the branch in Newton millimetres;

and

z is calculated from

$$z = \pi \left(r_{mb}\right)^{\frac{3}{2}} \left(r_{ms}\right)^{-\frac{1}{6}} \left(e_{ms}\right)^{\frac{5}{3}} \tag{13.3-2}$$

where

e_{ms} is the mean thickness of the main body in millimetres;

r_{mb} is the mean radius of the branch in millimetres;

r_{ms} is the mean radius of the main body in millimetres;

e) The temperature differences between two close points $\Delta\Theta$ (see Figure 13.3-1) during transients of the type in b) above, and $\Delta\Theta'$ during transients of the type in c) above shall satisfy:

$$0{,}3(\Delta\Theta)^3 + (\Delta\Theta')^3 < B^3 \tag{13.3-3}$$

where B shall be given by Figure 13.3-2 for carbon and other ferritic steels and by Figure 13.3-3 for austenitic steels, as a function of external load parameter λ (see d) above), design stress f and temperature factor γ.

Temperature factor γ is given in Figure 13.3-4, as a function of t^*, the reference temperature defined in Annex B.

The temperature differences shall be calculated in accordance with 13.3.5.

Two points shall be considered close if their distance apart does not exceed $2\sqrt{r\bar{e}}$, where r is the mean mid-wall radius and $\bar{e}$ is the mean wall thickness between the points considered (see Figure 13.3-1).

f) The temperature differences in Kelvin, $\Delta\Theta$ and $\Delta\Theta'$ shall be

$$\Delta\Theta, \Delta\Theta' < \left\{ \begin{array}{ll} 0{,}625 f & \text{for carbon or other ferritic steels} \\ 0{,}454 f & \text{for austenitic steels} \end{array} \right\} \tag{13.3-4}$$

where f is the design stress in megapascal.

13.3.4 Temperature differences during transient operating conditions

For purposes of applying the fatigue analysis exemption criteria, the temperature differences $\Delta\Theta$, $\Delta\Theta'$ between two close points shall be calculated as follows:

a) The temperature differences shall be the maximum values obtained by considering conditions at:

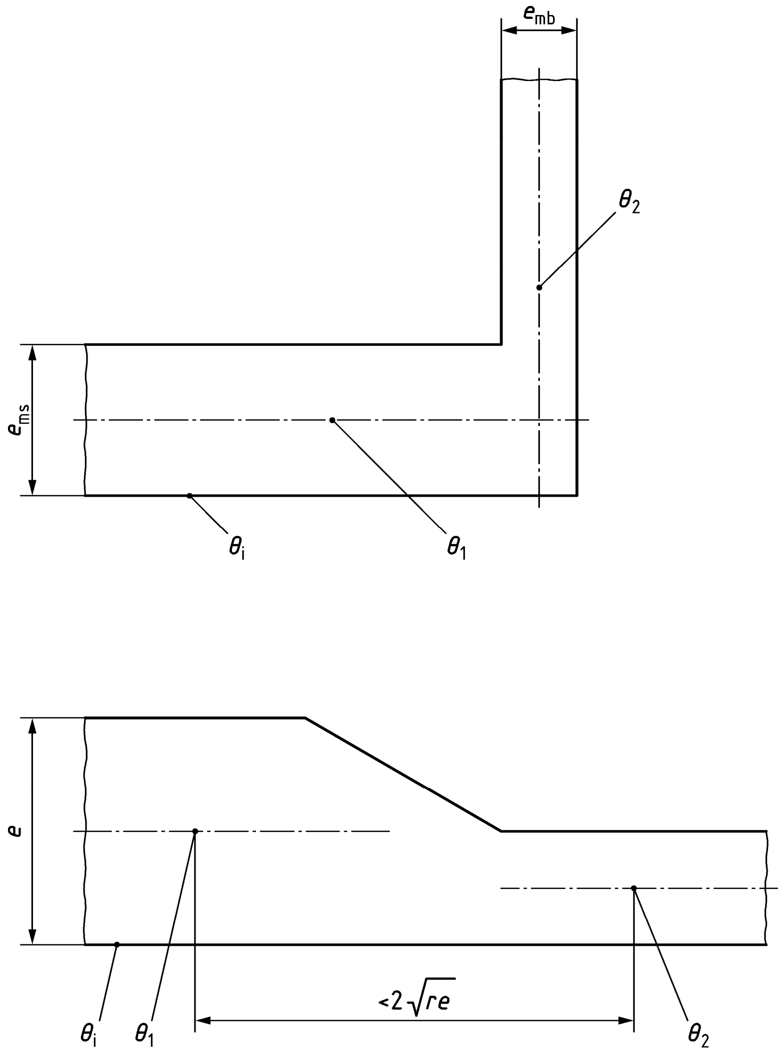
1) the point of maximum wall thickness;

2) the region of maximum change in wall thickness.

b) At either point, the temperature difference shall be given by:

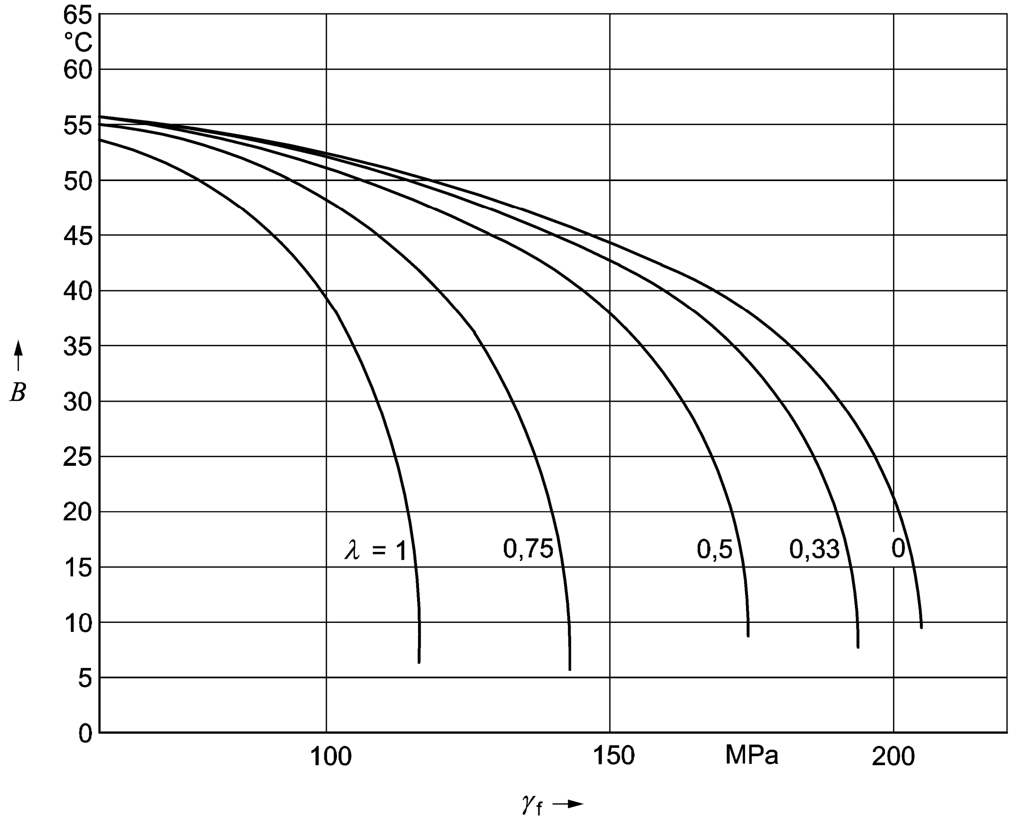
$$\Delta\Theta = \min\left\{ \Delta t_{\mathrm{f}} C_{\Theta}; \frac{\Delta t_{\mathrm{f}}}{N_{\mathrm{F}}} C_{\mathrm{v}} \right\} \tag{13.3-5}$$

When considering conditions at the point of maximum wall thickness, $e_{\mathrm{b}}/e_{\mathrm{a}} = 1$ (see Figures 13.3-5 and 13.3-6).



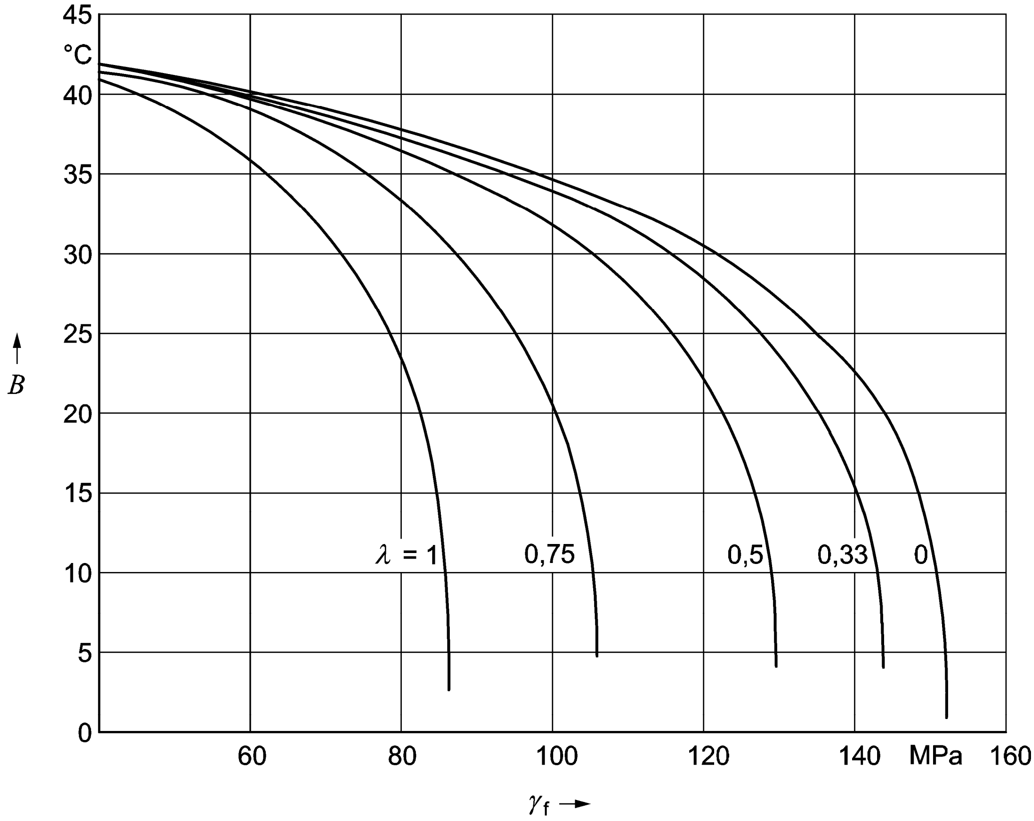
Θ_1 and Θ_2 are the mean temperatures through the walls; $\Delta\Theta = \max\{|\Theta_1 - \Theta_2|;|\Theta_i - \Theta_2|\}$

Figure 13.3-1 — Definition of *ΔΘ*



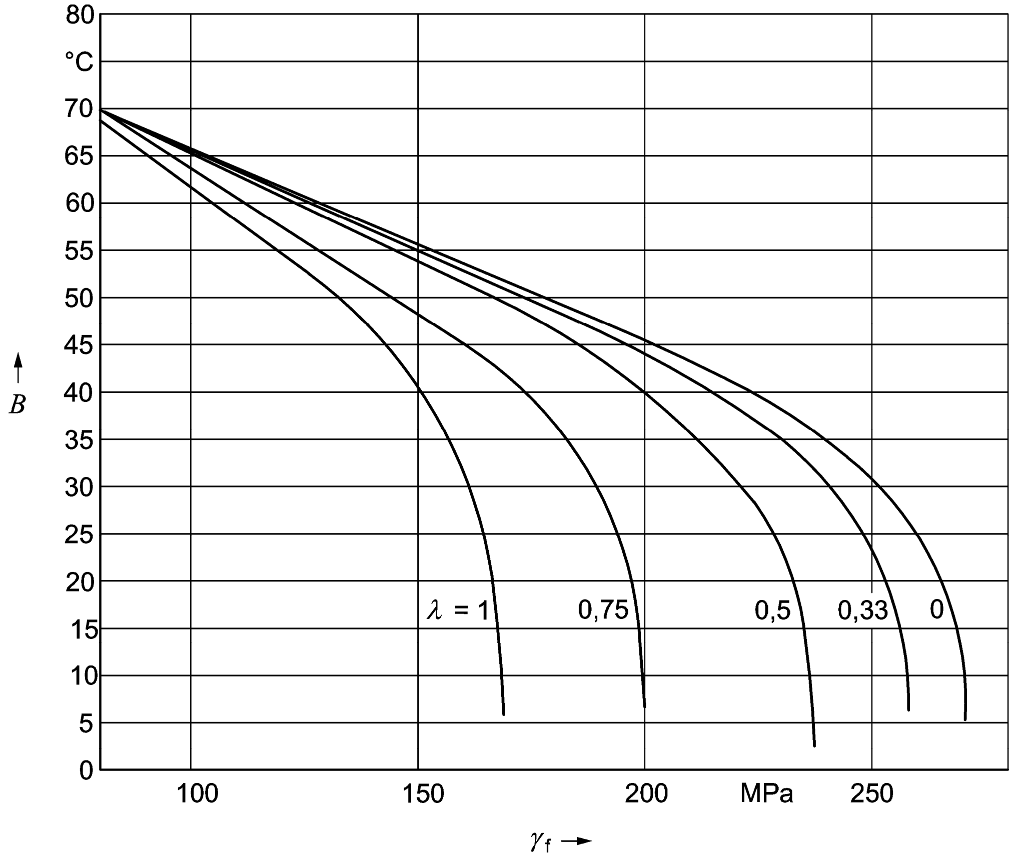
a) full penetration weld

Figure 13.3-2 — Values of B for carbon or ferritic alloy steels



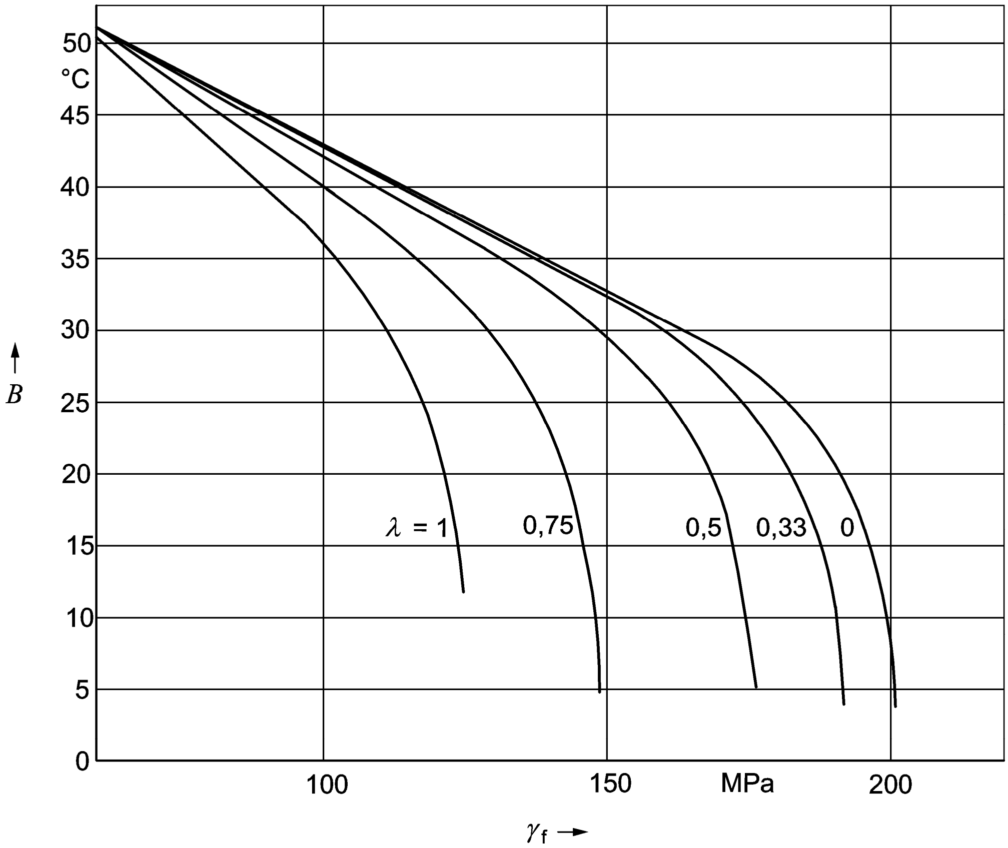
b) partial penetration weld

Figure 13.3-2 — Values of B for carbon or ferritic alloy steels *(continued)*



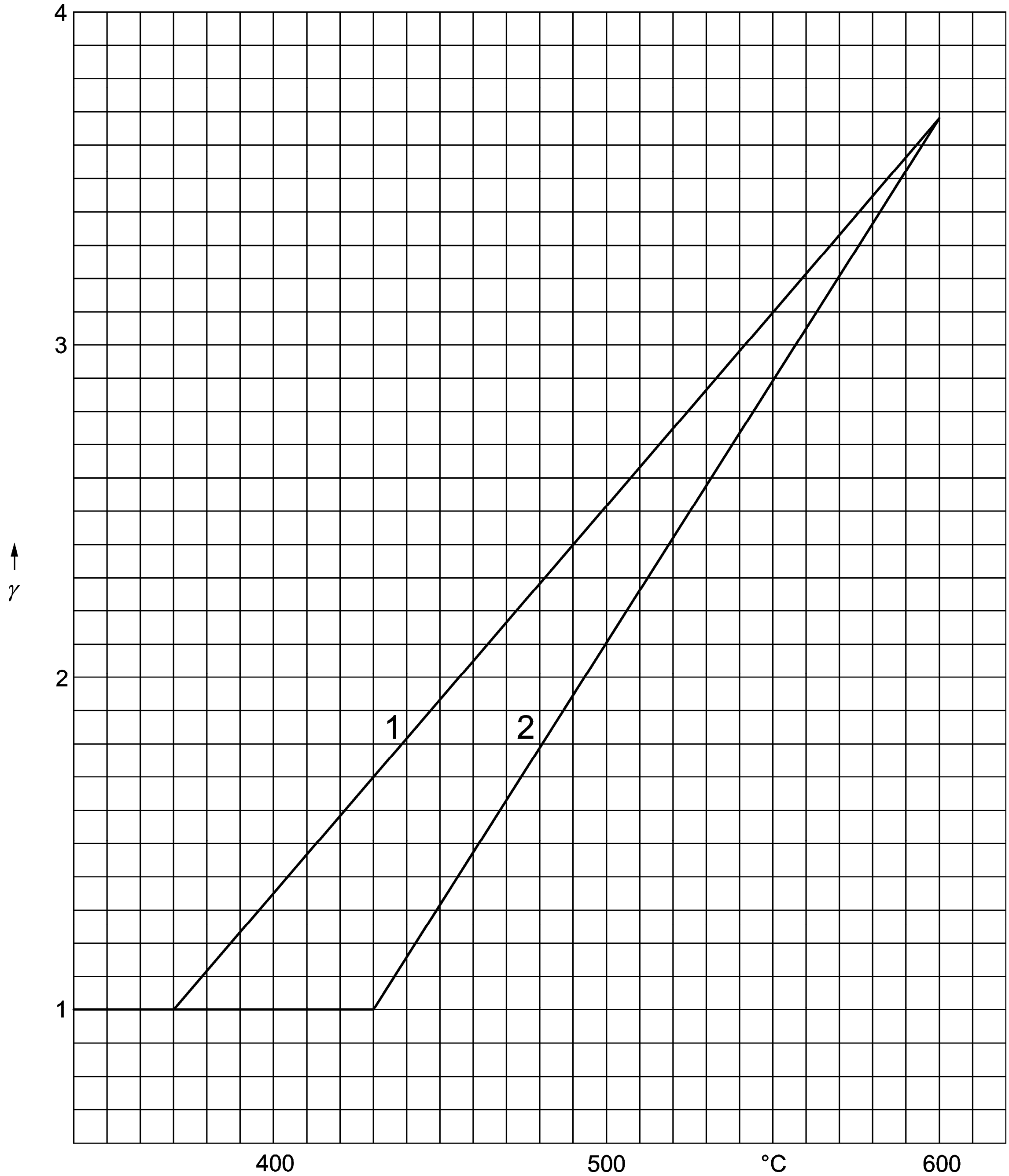
a) full penetration weld

Figure 13.3-3 — Values of *B* for austenitic stainless steels



b) partial penetration weld

Figure 13.3-3 — Values of B for austenitic stainless steels *(continued)*



Key
1 carbon or ferritic alloy steels
2 austenitic stainless steels

Figure 13.3-4 — Temperature dependent factor γ
(γ = 1 for temperatures less than 370 °C)

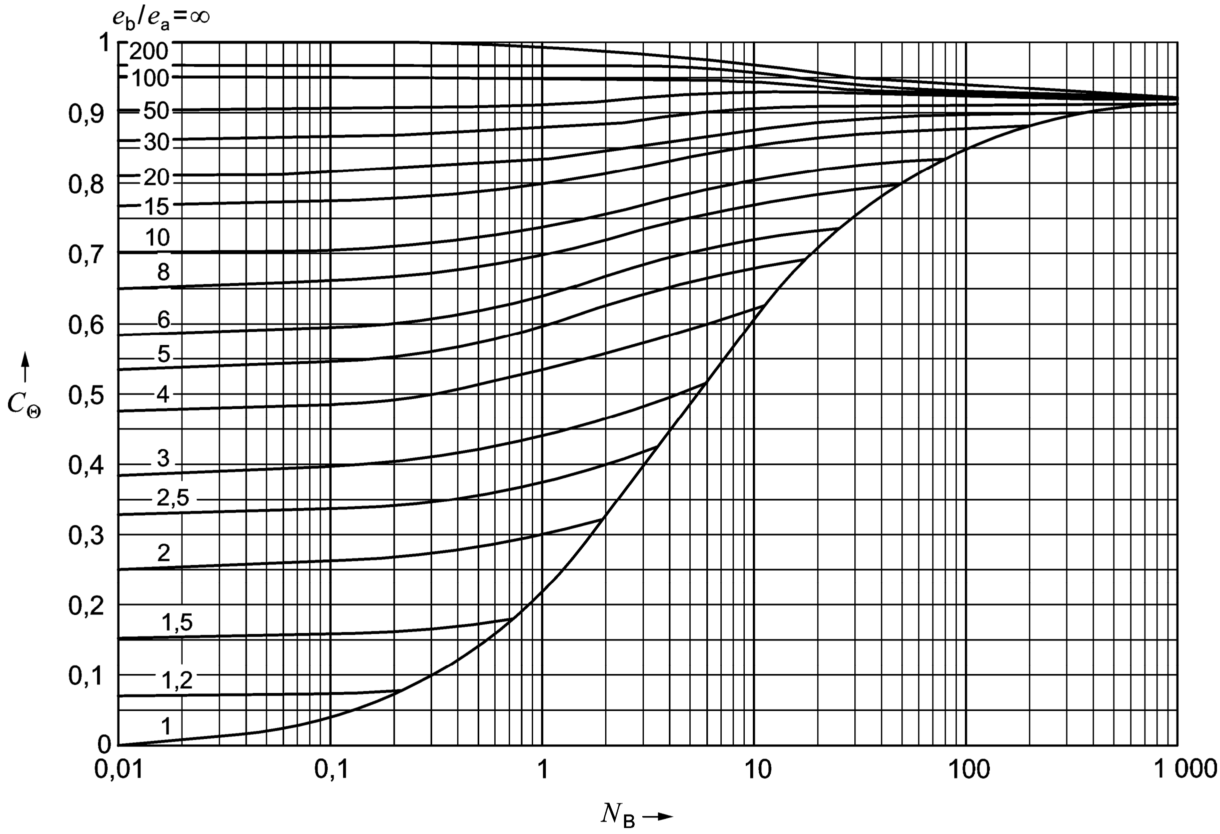


Figure 13.3-5 — Coefficient C_Θ

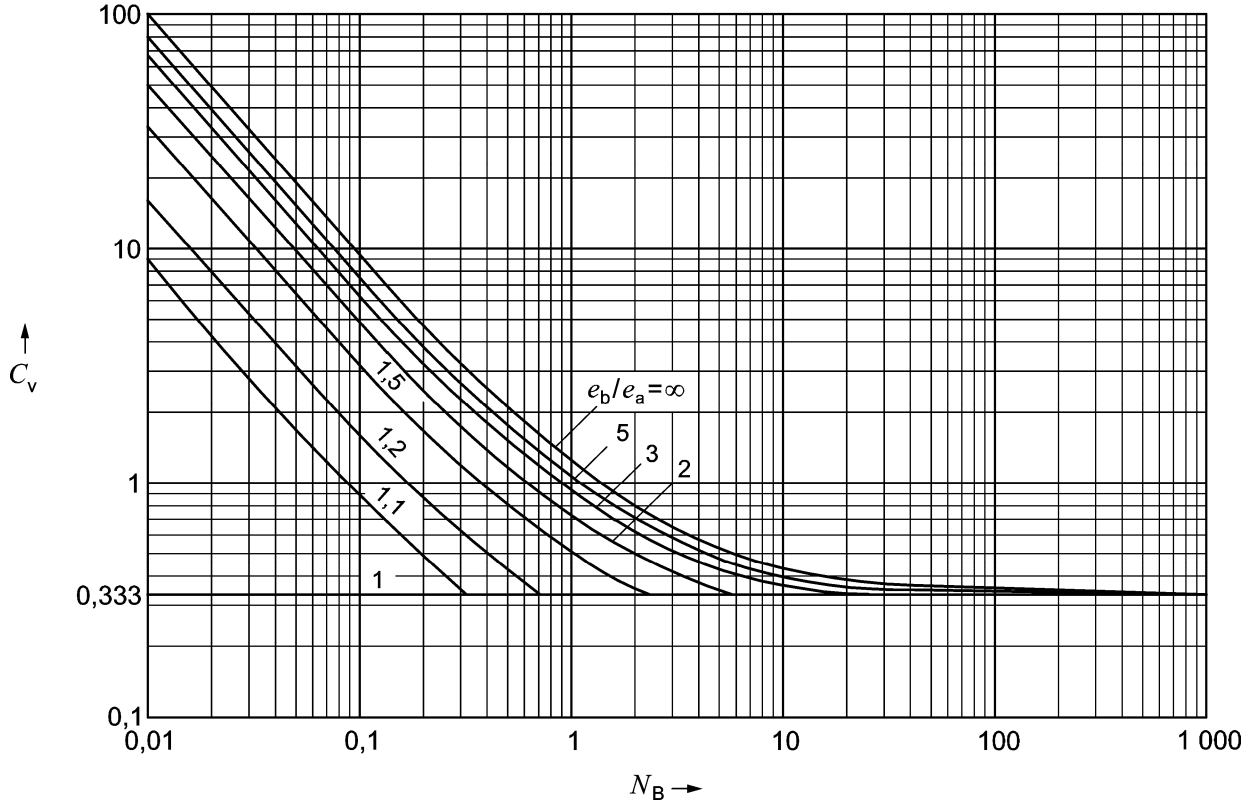


Figure 13.3-6 — Coefficient C_v

13.4 Stress analysis for fatigue calculation

13.4.1 Principal and equivalent stresses

The stresses at the crotch corner at the inner surface of a cylinder to cylinder or cylinder to sphere intersection are tri-directional. The cyclic stresses considered at this location shall be those due to local temperature difference and pressure. When external forces and moments are significant, the additional stresses shall be calculated in accordance with EN 13445-3:2009. The three principal stresses at the bore to be used in Equations (B-4) to (B-6) in accordance with:

$$f_1 = f_{tang} = f_{tang\ p} + f_{tang\ t} \qquad (13.4\text{-}1a)$$

$$f_2 = f_{rad} = -p \qquad (13.4\text{-}1b)$$

$$f_3 = f_{ax} = -p \qquad (13.4\text{-}1c)$$

where

f_1 is the stress tangential to the main body and tangential to the opening, caused by pressure and by temperature difference through the wall;

f_2 is the stress radial to the main body; compensating for the fluid pressure at the inside surface of the main body;

f_3 is the stress axial to the main body; compensating for the fluid pressure at the inside surface of the opening or branch.

Thus, the three differences of the principal stresses to be used in Equations (B-4) to (B-6) shall be:

$$\Delta f_{12} = f_{tang} + p \tag{13.4-2a}$$

$$\Delta f_{23} = 0 \tag{13.4-2b}$$

$$\Delta f_{31} = - (f_{tang} + p) \tag{13.4-2c}$$

Considering a cycling boiler operation with a variation of the pressure and the temperature during time which will result in a stress at the bore with the maximum $\Delta \hat{f}_{12}$ and the minimum $\Delta \check{f}_{12}$, the stress range shall derive to:

$$2 f_v = \Delta \hat{f}_{12} - \Delta \check{f}_{12} \tag{13.4-3}$$

If values for Δf_{12} are calculated using the stress concentration factors α_m or α_{sp} given in 13.4.6 or are derived from finite elements calculations, no further consideration of the notch factor as stated in Annex B shall be necessary, so that $2 f_{va}^{*} = 2 f_v$.

13.4.2 Temperature of a load cycle

The calculation of the allowable stress range in accordance with Annex B and the determination of the physical properties of the materials in accordance with Annex D shall be based on the temperature t^* of the load cycle:

$$t^* = 0{,}75 \times t_{max} + 0{,}25 \times t_{min} \tag{13.4-4}$$

where t_{max} is the metal temperature at the moment, when the highest stress of the load cycle prevails and t_{min} the one during the moment of the lowest stress.

13.4.3 Protection of the magnetite layer

For boiler components made from ferritic or martensitic steel, which may always or sometimes contain water or water and steam mixtures under normal operating conditions, the magnetite layer on the inside of the components shall be protected by the following additional restrictions:

$$f_{tang} \leq f_{tang\, p_o} + 200 \text{ MPa} \tag{13.4-5a}$$

$$f_{tang} \geq f_{tang\, p_o} - 600 \text{ MPa} \tag{13.4-5b}$$

NOTE It is assumed that the magnetite layer forms at operating conditions t_o, p_o so that there is no stress in the layer at operating conditions. Thus there will be compressive stress in the layer after shut-down.

13.4.4 Allowable circumferential stress range at the inside corner of a bore

The operational cyclic stress derived by Equation (13.4-3) shall meet the allowable cyclic stress range $2f_{va}$.

$$\Delta f_v \leq 2 f_{va} \tag{13.4-6}$$

The allowable cyclic stress range $2f_{va}$, determined in accordance with Annex B, shall be a stress intensity range using the shear stress theory. The allowable range of the circumferential stress at the inside corner of a bore in consequence of a load cycle with pressure levels p_{max} and p_{min} shall result from

$$\Delta f_{tang} = 2 f_{va} - (p_{max} - p_{min}) \tag{13.4-7}$$

at the second principle stress in - p_{max} and - p_{min} respectively.

13.4.5 Circumferential stress caused by pressure at the inside corner of a bore

One part of the allowable range of the circumferential stress shall be used up for the stress range $\Delta f_{tang,\,p}$ caused by the pressure range. The stress concentration factors α_m or α_{sp} in Figures 13.4-5 and 13.4-7 however are related to the equivalent stresses in the middle of the wall. Therefore the circumferential stresses at the inside corner of the bore caused by pressure p shall be:

$$f_{tang\,p} = \begin{cases} \alpha_m \, d_{ms} / (2 e_{ms}) \, . \, p & \text{for cylindrical shells} \\ \alpha_{sp} \, d_{ms} / (4 e_{ms}) . p & \text{for spherical shells} \end{cases} \tag{13.4-8}$$

If the stress concentration factor α_m or α_{sp} can neither be determined by measuring nor by calculation, the value shall be taken from Figure 13.4-5 for cylindrical components or from Figure 13.4-7 for spherical components. Figure 13.4-5 gives the values α_m for welded-on branches as shown in Figure 13.4-3, where the weld root is to be made by the Tungsten Inert Gas method (TIG) as shown in Figure 13.4-3 a) or the root shall be machined out or ground over to leave no residual gap as shown in Figure 13.4-3 b).

In the case of deviation the stress concentration factor shall be corrected as follows:

a) α_m may be reduced by 10 %, if the branch is set-through and welded by a full penetration weld as in Figure 13.4-1 or if it is a forged branch as in Figure 13.4-2; in any case being without a residual gap;

b) α_m or α_{sp} shall be increased by 10 % for extruded main bodies with the branch welded to the flange as in Figure 13.4-4; root machined out or ground over, to leave no residual gap;

c) when the root cannot be machined out or ground over, the wall thickness e_{mb} of the branch as in Figure 13.4-5 or 13.4-7 respectively shall be reduced by the residual gap. In addition, the thereby determined value for α_m or α_{sp} shall be increased by 60 % (see Figure 13.4-3 c));

d) expanded tube joints shall be treated like openings without branches (e_{mb} = 0 in Figures 13.4-5 and 13.4-7). If there is a seal weld, α_m or α_{sp} shall be increased by 10 %.

For diameter ratios d_{ib}/d_i > 0,5 and d_i > 300 mm or for welded-on stubs or nozzles with d_{ib} > 120 mm and a yield strength of the main body of more than 355 MPa at 20 °C, a full penetration weld to the branch connection shall be required, with the root machined or ground over, to leave no residual gap.

13.4.6 Stresses on the branch caused by external forces and moments

Another portion of the allowable stress range Δf_{tang} shall be used up by stress variations $\Delta f_{tang\,f}$ caused by fluctuating external forces and moments on the branch. In general, these stresses occur on the outside of the branch connection and are usually insignificant at the inside of a bore.

13.4.7 Thermal stresses

Circumferential (principal) stresses at the inside of the bore caused by through-the wall temperature differences Δt shall be calculated by:

$$f_{tang\,t} = \alpha_t \frac{\beta_{Lt^*} \, E_{t^*}}{1 - \upsilon} \Delta t \tag{13.4-9}$$

where

υ = 0,3 is the Poisson's ratio and
α_t may be taken from Figure 13.4-8 or may be calculated.

The coefficient of linear thermal expansion β_{Lt^*}, and the modulus of elasticity E_{t^*} shall be taken from Annex D.

The through-the-wall temperature difference Δt in Equation (13.4-9) shall be given by

$$\Delta t = t_m - t_i \tag{13.4-10}$$

where

t_m is the integral mean wall temperature and

t_i is the temperature at the inside surface of the wall.

When the temperature is increased, the through-the-wall temperature difference shall be $\Delta t_1 < 0$, when the temperature is decreased it shall be $\Delta t_2 > 0$. Thus, in addition to the stress range caused by the pressure range, there arises a thermal stress range:

$$f_{\mathrm{tang\,t}} = \alpha_t \frac{\beta_{Lt^*} E_{t^*}}{1-\upsilon}\left(\Delta t_2 - \Delta t_1\right) \tag{13.4-9a}$$

13.4.8 Upper and lower limit of the circumferential stress at the inside corner of a bore

The total allowable range of circumferential stress in accordance with Equation (13.4-7) with the restrictions in accordance with Equation (13.4-5) shall be greater than the cyclic range required for the pressure. The difference shall be used for thermal stresses:

below $f_{\mathrm{tang}\,p\,\mathrm{min}}$ for temperature increase at the low pressure level p_{min} and

over $f_{\mathrm{tang}\,p\,\mathrm{max}}$ for temperature reduction at the high pressure level p_{max}.

The maximum allowable circumferential stress at the inside corner of a bore shall be derived.

a) for austenitic materials in contact with water or steam or other materials only in contact with steam

$$f_{\mathrm{tang\,max}} = f_{\mathrm{tang}\,p\,\mathrm{max}} + g_s \cdot \Delta f_{\mathrm{tang}\,t} \tag{13.4.11 a}$$

b) for ferritic and martensitic materials, in contact with water

$$f_{\mathrm{tang\,max}} = \min\begin{Bmatrix} f_{\mathrm{tang\,p\,max}} + g_s \cdot \Delta f_{\mathrm{tang\,t}} \\ f_{\mathrm{tang\,p_0}} + 200\,\mathrm{MPa} \end{Bmatrix} \tag{13.4.11 b}$$

and the minimum allowable circumferential stress at the inside corner of a bore

c) for austenitic materials in contact with water or steam or other materials only in contact with steam

$$f_{\mathrm{tang\,min}} = f_{\mathrm{tang\,max}} - \Delta f_{\mathrm{tang}\,t} \tag{13.4.11 c}$$

d) for ferritic and martensitic materials, in contact with water

$$f_{\text{tang min}} = \max\begin{Bmatrix} f_{\text{tang p max}} - \Delta f_{\text{tang t}} \\ f_{\text{tang p}_0} - 600\,\text{MPa} \end{Bmatrix} \tag{13.4.11 d}$$

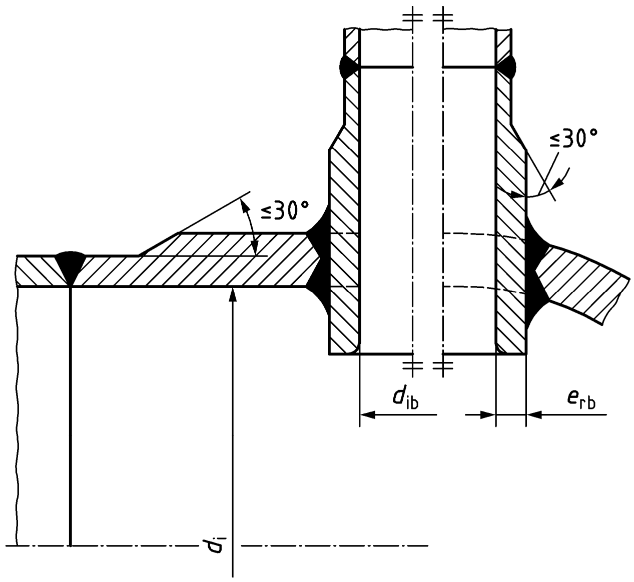
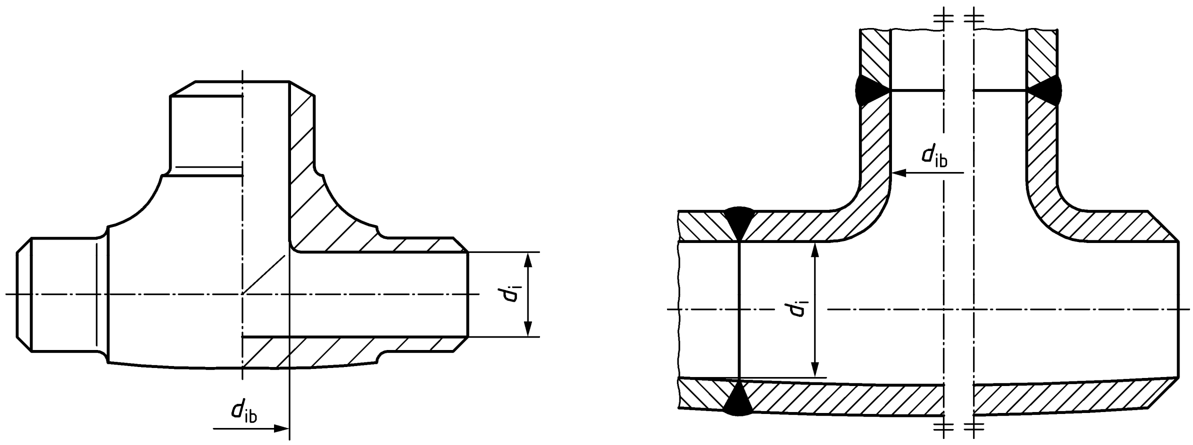


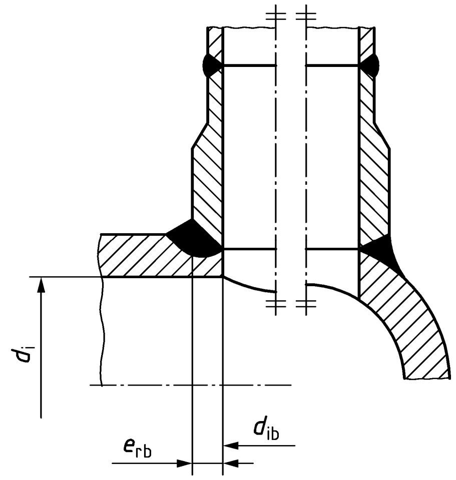
Figure 13.4-1 — Reinforcement by means of set through and full penetration welded branch



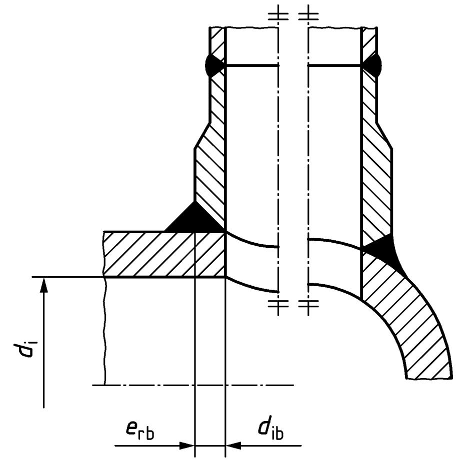
a) branch forged from solid material and subsequently machined

b) die forged branch

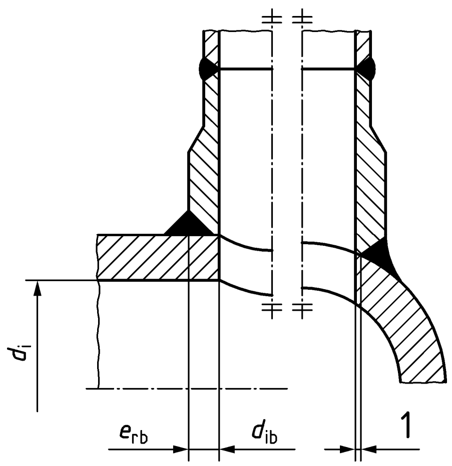
Figure 13.4-2 — Forged branch



a) weld root made from the inside by TIG method



b) root machined out or ground over method, without residual gap



c) with residual gap (1)

Figure 13.4-3 — Reinforcement by means of welded-on branch

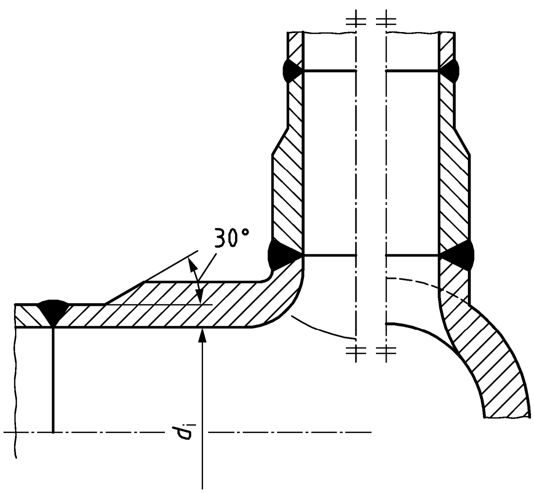
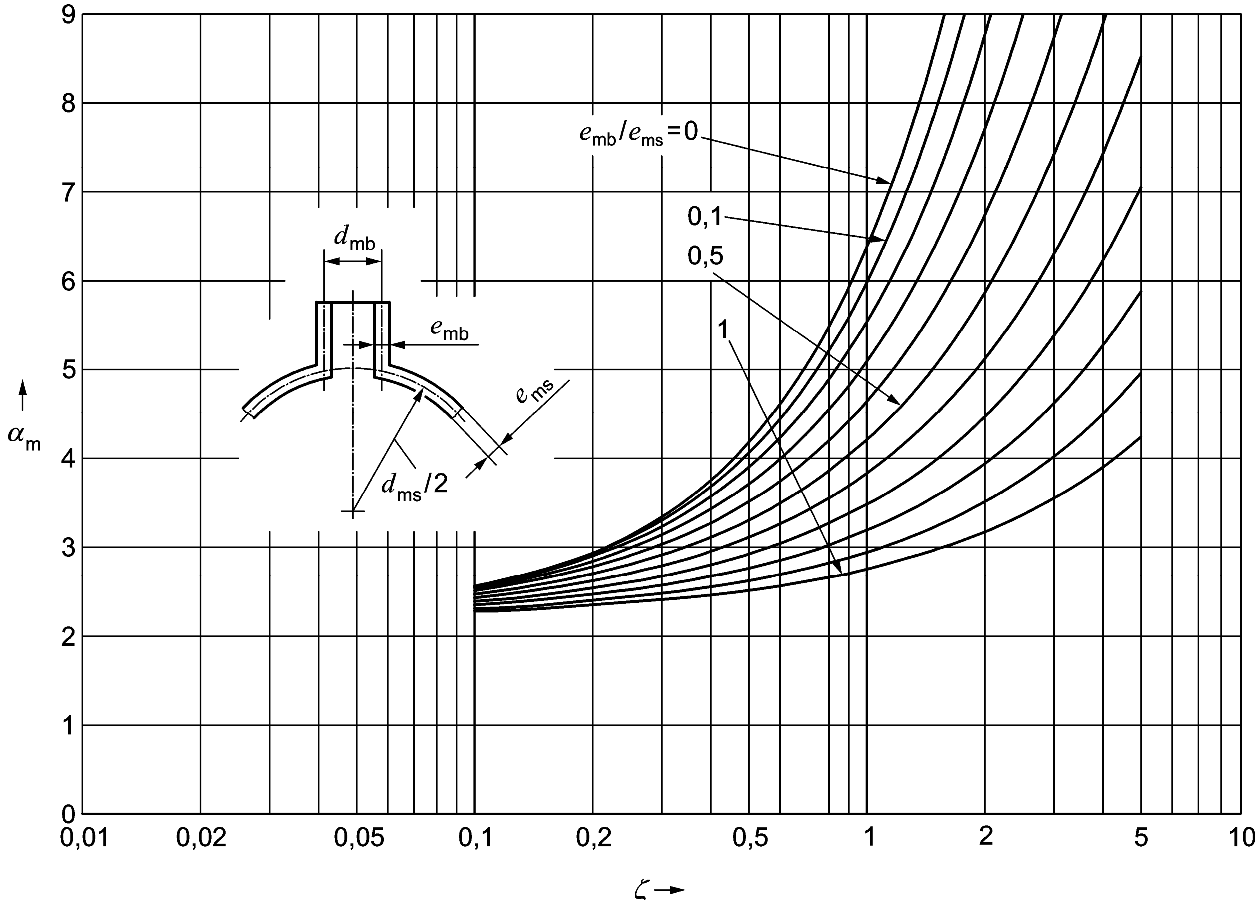


Figure 13.4-4 — Reinforcement by means of branch welded to extruded main body

$$\alpha_m = \frac{f_{max}}{p_o\, d_{ms}/(2e_{ms})}$$



$\alpha_m = 2{,}2 + e^A \cdot \zeta^B$

with

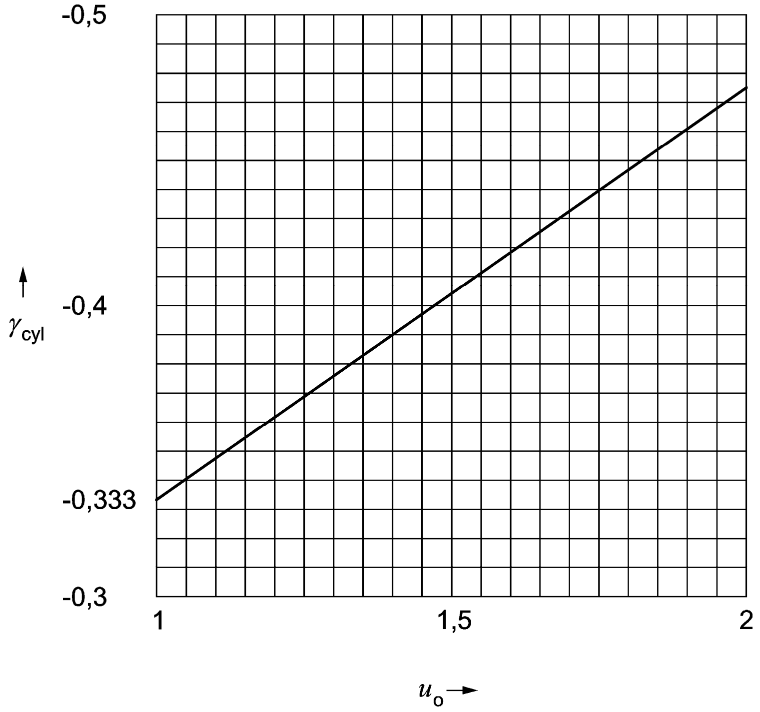
e = 2,718 281 828 5

$$\zeta = \frac{d_{mb}}{d_{ms}}\sqrt{\frac{d_{ms}}{2e_{ms}}}$$

A = -1,14 $(e_{mb}/e_{ms})^2$ - 0,89 (e_{mb}/e_{ms}) + 1,43

B = 0,326 $(e_{mb}/e_{ms})^2$ - 0,59 (e_{mb}/e_{ms}) + 1,08

Figure 13.4-5 — Stress concentration factor α_m for cylindrical shells



$$u_0 = \frac{d_o}{d_i} \quad \text{and} \quad \gamma_{cyl} = \frac{\left(u_0^2 - 1\right)\left(3u_0^2 - 1\right) - 4u_0^4 \ln u_0}{8\left(u_0^2 - 1\right)\left(u_0 - 1\right)^2}$$

Figure 13.4-6 — Shape factor γ_{cyl} for cylindrical shells

where

$0 \le g_s \le 1$ is a factor by which the designer can specify which amount of the stress range $\Delta f_{tang,\,t}$, altogether available for thermal stresses, shall be available at the beginning of a shut-down, when the pressure is at its high level p_{max}.

- $g_s = 0$ uses the total thermal stress range $\Delta f_{tang,\,t}$ for thermal stress caused by temperature increase at the lower pressure level p_{min};
- $g_s = 0{,}2$ distributes the range $\Delta f_{tang,\,t}$, so that 20 % of it are used above $\Delta f_{tang,\,pmax}$ for shut-down at the upper pressure level $p_{max.}$ and 80 % of it for start-up at the lower pressure level p_{min};
- $g_s = 0{,}5$ distributes the range $\Delta f_{tang,\,t}$, into equal parts above $\Delta f_{tang,\,pmax}$ and below $\Delta f_{tang,\,pmin}$, thus creating a symmetry for the thermal stresses at start-up and shut-down.

13.4.9 Admissible through-the-wall temperature differences

In order not to exceed the allowable total stress range in accordance with 13.4.8, the through-the-wall temperature difference Δt shall not exceed the following limits, which are dependant on the actual pressure p:

$$\left(f_{tang\,min} - f_{tang\,p}\right)/W \le \Delta t \le \left(f_{tang\,max} - f_{tang\,p}\right)/W \qquad (13.4\text{-}12)$$

and $W = \alpha_t\, \beta_{L1}\, E_t/(1 - \upsilon)$

Thus the limits are at the

beginning of the start-up $(p = p_{\text{min}}) : \Delta t_1 \geq (f_{\text{tang, min}} - \Delta f_{\text{tang,} p\text{min}})/W$ (13.4-12 a)

end of start-up $(p = p_{\text{max}}) : \Delta t_{1'} \geq (f_{\text{tang, min}} - \Delta f_{\text{tang,} p\text{max}})/W$ (13.4-12 b)

beginning of shut-down $(p = p_{\text{max}}) : \Delta t_2 \geq (f_{\text{tang, max}} - \Delta f_{\text{tang,} p\text{max}})/W$ (13.4-12 c)

end of shut down $(p = p_{\text{min}}) : \Delta t_{2'} \geq (f_{\text{tang, max}} - \Delta f_{\text{tang,} p\text{min}})/W$ (13.4-12 d)

NOTE The temperature difference Δt is negative, when the temperature is increased.

13.4.10 Allowable temperature transients

An exact calculation of the allowable temperature transient requires observation of pressure and temperature fluctuations during operation as described in B.4. Using the simplified assumption of quasi-steady-state conditions, the allowable temperature transients for cylindrical components shall be calculated from the respective allowable through-the-wall temperature differences by:

$$v_{\text{t}} = \Delta t \cdot \frac{D_{\text{th}}}{\gamma_{\text{cyl}}\, e_{\text{ms}}^2} \tag{13.4-13}$$

Here Δt is the allowable through-the-wall temperature difference, dependant on the actual pressure in accordance with 13.4.9. The shape factor γ_{cyl} shall be taken from Figure 13.4-6 or may be calculated.

For spherical components γ_{cyl} shall be replaced by γ_{sp} from Figure 13.4-9 or may be calculated.

13.4.11 Components with oblique and/or non-radial branches

Component with oblique and/or non-radial branches shall be treated as components with radial branches (see above).

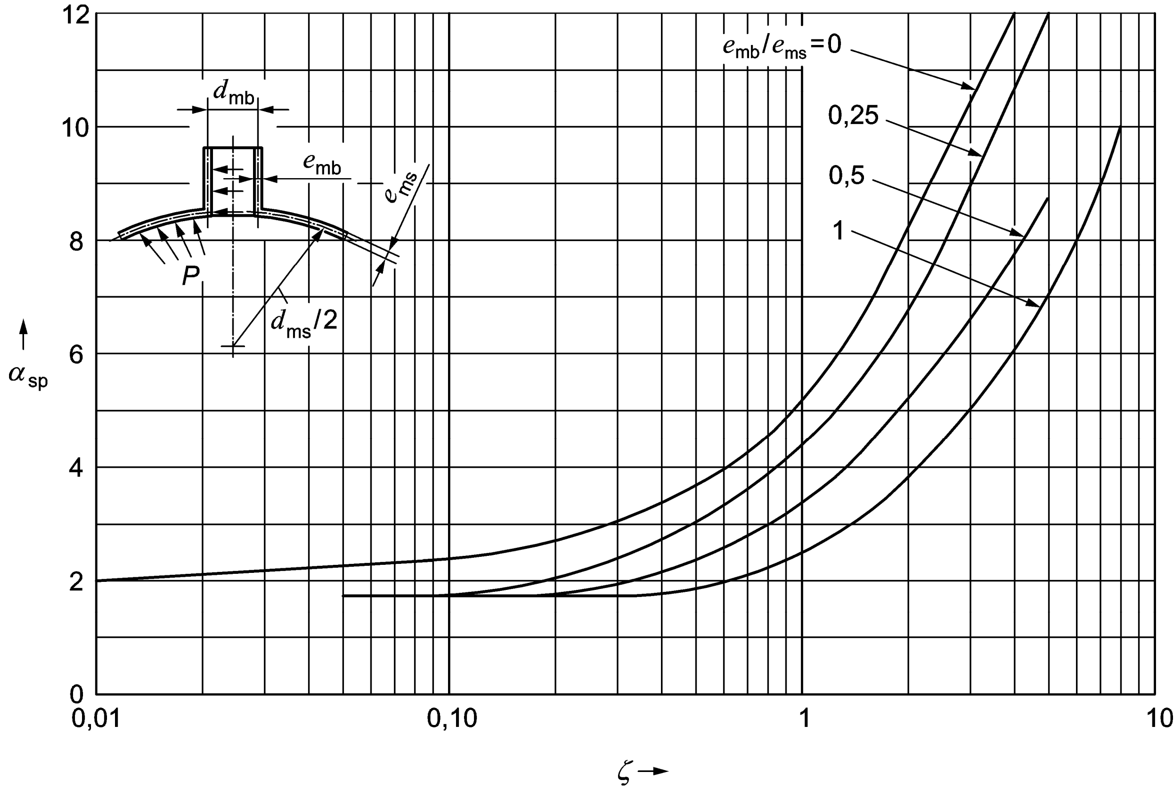
13.5 Example calculations

Annex C provides informative worked examples of the fatigue analysis in accordance with the design requirements of Clause 13 and Annex B.

Worked examples are provided for calculation procedures with respect to:

a) calculation of admissible number of load cycles;

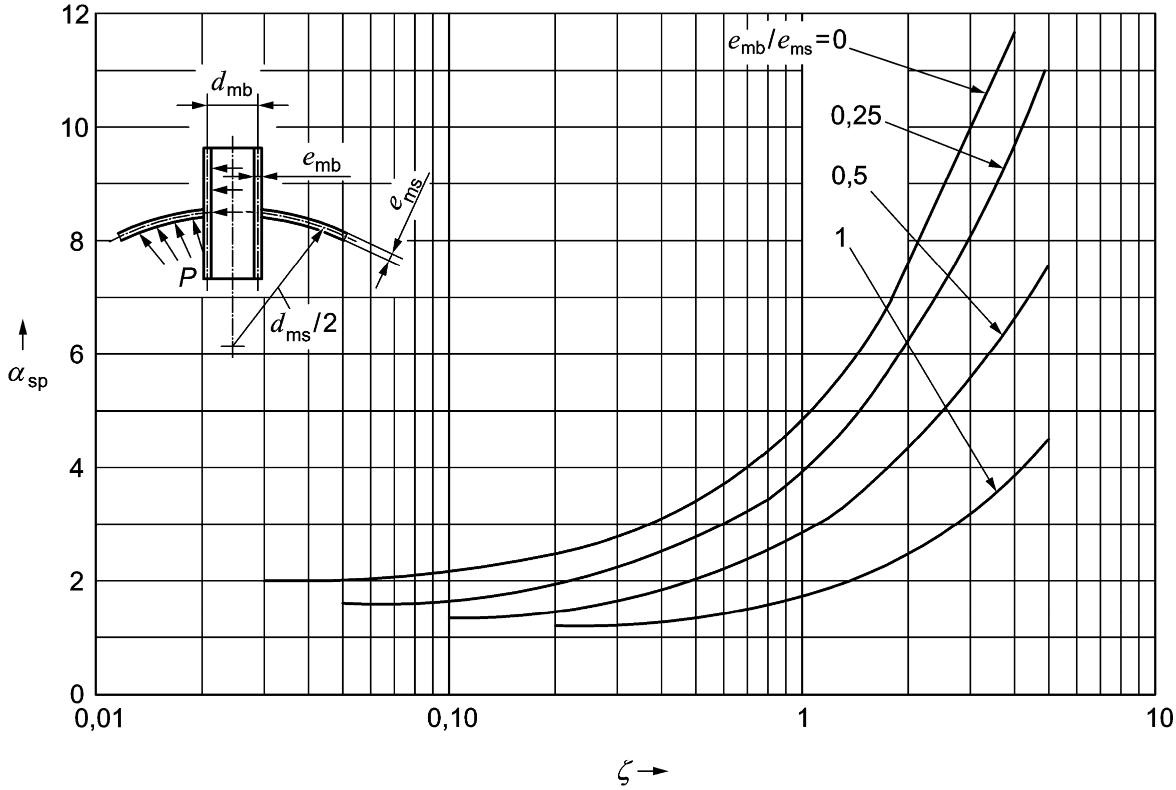
b) calculation of admissible temperature gradient.



a) Maximum stress in sphere for internal pressure (set on nozzles)

$$\varsigma = \frac{d_{mb}}{d_{ms}}\sqrt{\frac{d_{ms}}{2e_{ms}}}$$

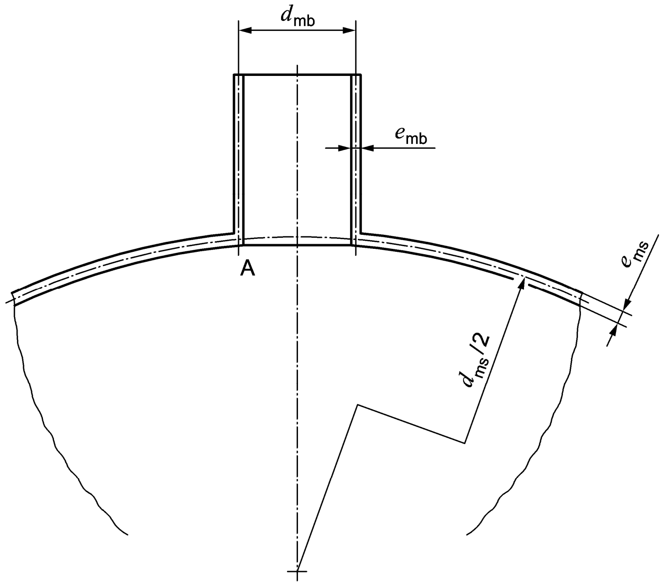
Figure 13.4-7 — Stress concentration factors α for spherical shells

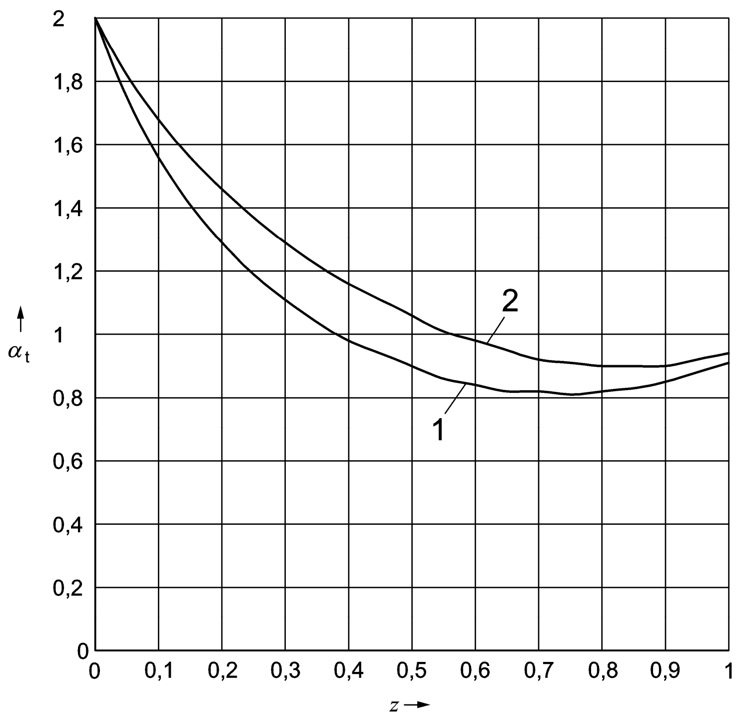


b) Maximum stress in sphere for internal pressure (set through nozzles)

$$\varsigma = \frac{d_{mb}}{d_m}\sqrt{\frac{d_m}{2e_{ms}}}$$

Figure 13.4-7 — Stress concentration factors α for spherical shells *(continued)*



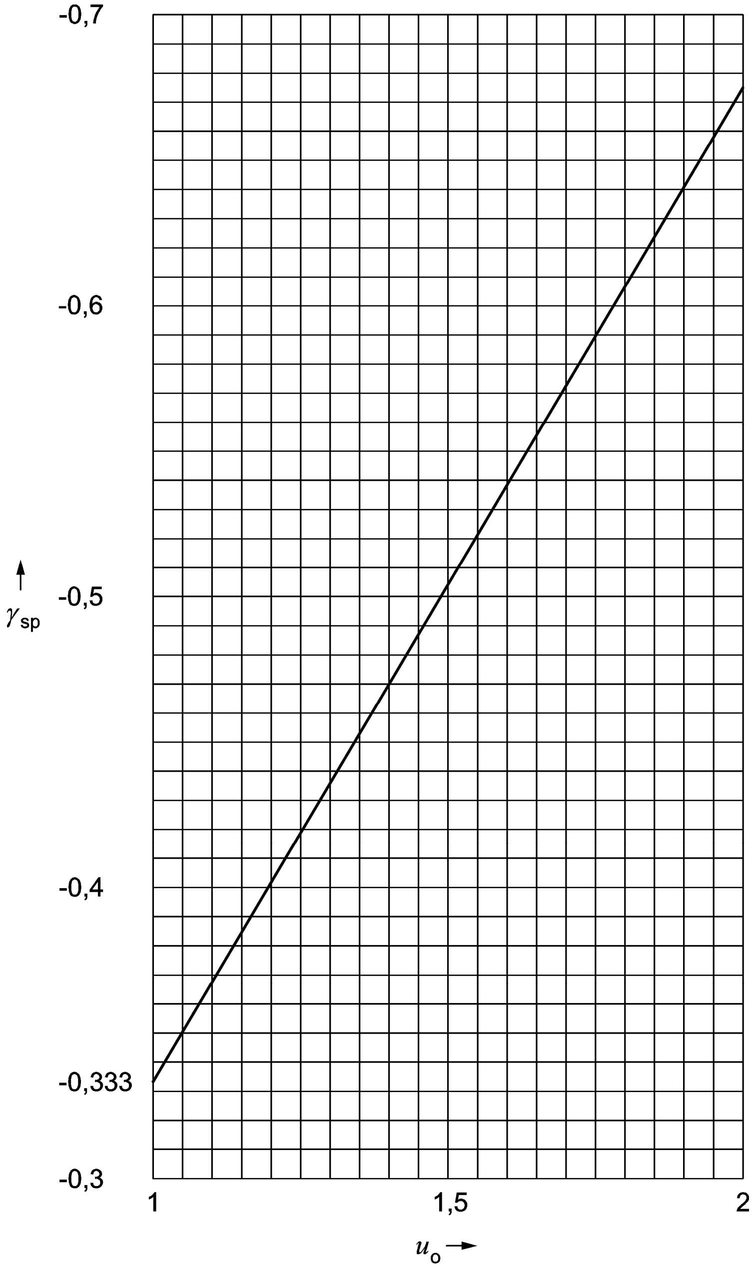


$$\alpha_t = \left\{ \left[2 - \frac{h + 2700}{h + 1700} z + \frac{h}{h + 1700} \left(\exp(-7z) - 1 \right) \right]^2 + 0{,}81 z^2 \right\}^{1/2}$$ and $z = d_{mb}/d_{ms}$

heat-transfer coefficient: $h = \left\{ \begin{array}{ll} 1000\ \mathrm{W/m^2K} & \text{for steam} \\ 3\,000\ \mathrm{W/m^2} & \text{for water} \end{array} \right\}$

Key
1 water
2 steam

Figure 13.4-8 — Stress concentration factor due to thermal stress α_t for cylindrical and spherical shells



$$u_0 = \frac{d_o}{d_i} \quad \text{and} \quad \gamma_{sp} = -\frac{1}{3}\left[u_0 + \frac{(u_0 - 1)^3}{5\left(u_0^3 - 1\right)}\right]$$

Figure 13.4-9 — Shape factor γ_{sp} for spherical shells

Annex A
(normative)

Calculation of tube bends and elbows

A.1 General

Annex A is a supplement to, and shall be used only in conjunction with the design rules for tubes in accordance with Clause 11.

These design rules shall apply to elbows, e.g. to EN 13480-3:2002, and to bent tubes.

These design rules take into account that when an internal pressure is applied to a tube bend, higher stresses occur on the intrados of the bend (and lower stresses on the extrados of the bend) than on a straight tube with identical wall thickness [2].

A.2 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table A.1 shall be used.

Table A.1 — Symbols

Symbol	Description	Unit
B	Design coefficient for the determination of the wall thickness of elbows with uniform wall thickness	—
B_i	Design coefficient for the determination of the wall thickness on the intrados of the tube bend	—
B_o	Design coefficient for the determination of the wall thickness on the extrados of the tube bend	—
e_{cti}	Required wall thickness on the intrados of the tube bend, without allowances, in accordance with Figure A.1	mm
e_{cto}	Required wall thickness on the extrados of the tube bend, without allowances, in accordance with Figure A.1	mm
e_{rti}	Actual wall thickness on the intrados of the tube bend, without allowances	mm
e_{rto}	Actual wall thickness on the extrados of the tube bend, without allowances	mm
e_{ti}	Actual wall thickness on the intrados of the tube bend, with allowances	mm
$e_{ti'}$	Required wall thickness on the intrados of the tube bend, with allowances	mm
e_{to}	Actual wall thickness of the extrados of the tube bend, with allowances	mm
$e_{to'}$	Required wall thickness on the extrados of the tube bend, with allowances	mm
f_{ai}	Mean stress on the intrados of the bend	MPa
f_{ao}	Mean stress on the extrados of the bend	MPa
r, r_b	Curvature radii of the tube bend, in accordance with Figure A.1	Mm

A.2.1 Required wall thickness

The required wall thickness shall be on the intrados of the bend

$$e_{ti'} = e_{cti'} + c_1 + c_2 \tag{A-1}$$

on the extrados of the bend

$$e_{to'} = e_{cto} + c_1 + c_2 \tag{A-2}$$

For the stress calculation of finished tube bends with the wall thickness e_{ti} or e_{to} respectively,

$$e_{rti} = e_{ti} - c_1 - c_2 \tag{A-3}$$

shall be used for the intrados of the bend, and

$$e_{rto} = e_{to} - c_1 - c_2 \tag{A-4}$$

for the extrados of the bend.

NOTE If tubes with inside diameter = nominal diameter are bent, and the outside diameter is maintained, the bend will remain the outside diameter $d_o = d_i + 2\,e_{ct}$ of the straight tube.

To avoid abrupt wall thickness transitions or misalignment, bevels at the transition from bends prepared for welding to the straight run of the tube shall not be considered in the calculation.

A.3 Calculation

A.3.1 Calculation of the wall thickness

A.3.1.1 The wall thickness of the intrados of the bend, without allowances, and with the least required thickness increase shall be calculated as

$$e_{cti} = e_{ct} \cdot B_i \tag{A-5}$$

a) for tube bends with a specified inside diameter using e_{ct} in accordance with Equation (11.2-5) and the coefficient

$$B_i = \frac{e_{cti}}{e_{ct}} = \frac{r}{e_{ct}} - \frac{d_i}{2e_{ct}} - \sqrt{\left(\frac{r}{e_{ct}} - \frac{d_i}{2e_{ct}}\right)^2 - 2\frac{r}{e_{ct}} + \frac{d_i}{2e_{ct}}} \tag{A-6}$$

The coefficient B_i as a function of r/d_i shall be taken from Figure A.2,

b) for tube bends with a specified outside diameter using e_{ct} in accordance with Equation (11.2-3) and the coefficient

$$B_i = \frac{e_{cti}}{e_{ct}} = \frac{d_0}{2e_{ct}} + \frac{r}{e_{ct}} - \left(\frac{d_o}{2e_{ct}} + \frac{\mathrm{r}}{e_{ct}} - 1\right)\sqrt{\frac{\left(\frac{r}{e_{ct}}\right)^2 - \left(\frac{d_o}{2e_{ct}}\right)^2}{\left(\frac{r}{e_{ct}}\right)^2 - \frac{d_o}{2e_{ct}}\left(\frac{d_o}{2e_{ct}} - 1\right)}} \tag{A-7}$$

As normally the curvature radius r_b is specified together with d_o;

$$\frac{r}{e_{ct}} = \sqrt{\frac{1}{2}\left[\left(\frac{d_o}{2e_{ct}}\right)^2 + \left(\frac{r_b}{e_{ct}}\right)^2\right] + \sqrt{\frac{1}{4}\left[\left(\frac{d_o}{2e_{ct}}\right)^2 + \left(\frac{r_b}{e_{ct}}\right)^2\right]^2 - \frac{d_o}{2e_{ct}}\left(\frac{d_o}{2e_{ct}} - 1\right)\left(\frac{r_b}{e_{ct}}\right)^2}} \qquad \text{(A-8)}$$

shall be used in this case.

The coefficient B_i as a function of r_b/d_o shall be taken from Figure A.3.

Equations (A-6) and (A-7) will only produce identical results when using

$$d_o = d_i + e_{cti} + e_{cto} \qquad \text{(A-9)}$$

and

$$r_b = r - ½ \cdot (e_{cti} - e_{cto}) \qquad \text{(A-10)}$$

A.3.1.2 The wall thickness of the extrados of the bend, without allowances, and with the allowable maximum efficiency, shall be calculated as

$$e_{cto} = e_{ct} \cdot B_o \qquad \text{(A-11)}$$

a) for tube bends with a specified inside diameter using e_{ct} in accordance with Equation (11.2-5) and the coefficient

$$B_o = \frac{e_{cto}}{e_{ct}} = \sqrt{\left(\frac{r}{e_{ct}} + \frac{d_i}{2e_{ct}}\right)^2 + 2\frac{r}{e_{ct}} + \frac{d_i}{2e_{ct}}} - \frac{d_i}{2e_{ct}} - \frac{r}{e_{ct}} \qquad \text{(A-12)}$$

The coefficient B_o, as a function of r/d_i, shall be taken from Figure A.4.

b) for tube bends with a specified outside diameter using e_{ct} in accordance with Equation (11.2-3) and the coefficient

$$B_o = \frac{e_{cto}}{e_{ct}} = \frac{d_o}{2e_{ct}} - \frac{r}{e_{ct}} - \left(\frac{d_o}{2e_{ct}} - \frac{r}{e_{ct}} - 1\right)\sqrt{\frac{\left(\frac{r}{e_{ct}}\right)^2 - \left(\frac{d_o}{2e_{ct}}\right)^2}{\left(\frac{r}{e_{ct}}\right)^2 - \frac{d_o}{2e_{ct}}\left(\frac{d_o}{2e_{ct}} - 1\right)}} \qquad \text{(A-13)}$$

Herein, r/e_{ct} shall be used in accordance with Equation (A-8).

The coefficient B_o, as a function of r/d_o, shall be taken from Figure A.5.

Equations (A-12) and (A-13) shall only produce identical results, if the relationships in Equations (A-9) and (A-10) exist between d_i, d_o, r and r_b.

A.3.1.3 For elbows with uniform wall thickness, the required wall thickness shall be determined as follows

$$e_{cti} = e_{cto} = e_{ct} \cdot B \qquad \text{(A-14)}$$

a) for tube bends with a specified internal diameter using e_{ct} in accordance with Equation (A-5) and the coefficient $B = B_i$ in accordance with Equation (A-6) or Figure A.2.

b) for elbows with a specified outside diameter using e_{ct} in accordance with Equation (A-6) and the coefficient

$$B = \frac{e_{cti}}{e_{ct}} = \frac{e_{cto}}{e_{ct}} = \frac{d_o}{2e_{ct}} - \frac{r_b}{e_{ct}} + \sqrt{\left(\frac{d_o}{2e_{ct}} - \frac{r_b}{e_{ct}}\right)^2 + 2\frac{r_b}{e_{ct}} - \frac{d_o}{2e_{ct}}} \qquad \text{(A-15)}$$

The coefficient B as a function of r_b/d_o shall be taken from Figure A.6.

Equation (A-6), in conjunction with Equation (A-14) shall only produce results identical to Equation (A-15) upon using

$$d_o = d_i + 2\, e_{cti} \qquad \text{(A-16)}$$

and

$$r_b = r \qquad \text{(A-17)}$$

A.3.2 Calculation of stress

A.3.2.1 The strength conditions for the intrados of the bend shall be:

a) for tube bends with a specified inside diameter:

$$f_{ai} = \frac{p_c d_i}{2e_{rti}} \cdot \frac{2r - 0{,}5d_i}{2r - d_i - e_{rti}} + \frac{p_c}{2} \le f \qquad \text{(A-18)}$$

b) for tube bends with a specified outside diameter:

$$f_{ai} = \frac{p_c\left(d_0 - e_{rti} - e_{rto}\right)}{2e_{rti}} \cdot \frac{2r_b - 0{,}5d_o + 1{,}5e_{rti} - 0{,}5e_{rto}}{2r - d_o + e_{rti}} + \frac{p_c}{2} \le f \qquad \text{(A-19)}$$

A.3.2.2 The strength conditions for the extrados of the bend shall be:

a) for tube bends with a specified inside diameter:

$$f_{ao} = \frac{p_c d_i}{2e_{rto}} \cdot \frac{2r + 0{,}5d_i}{2r + d_i + e_{rto}} + \frac{p_c}{2} \le f \qquad \text{(A-20)}$$

b) for tube bends with a specified outside diameter:

$$f_{ao} = \frac{p_c\left(d_o - e_{rti} - e_{rto}\right)}{2e_{rto}} \cdot \frac{2r_b + 0{,}5d_o + 0{,}5e_{rti} - 1{,}5e_{rto}}{2r_b + d_o - e_{rto}} + \frac{p_c}{2} \le f \qquad \text{(A-21)}$$

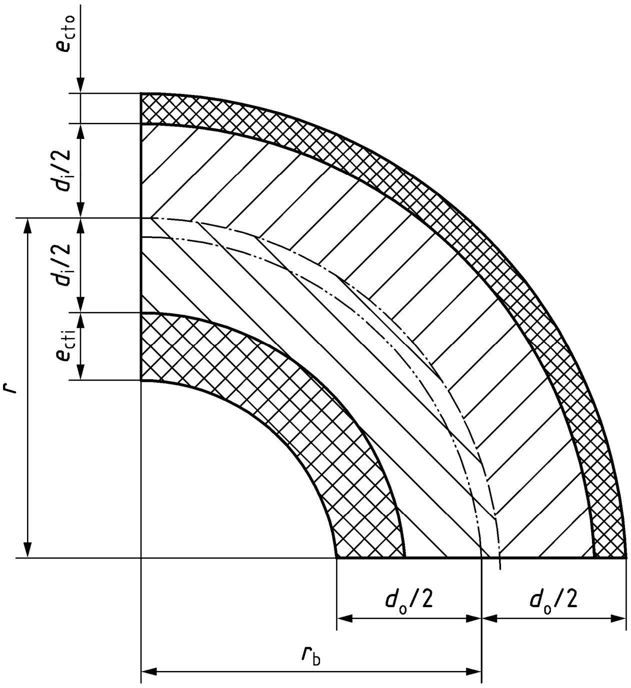


Figure A.1 — Notations for tube bends

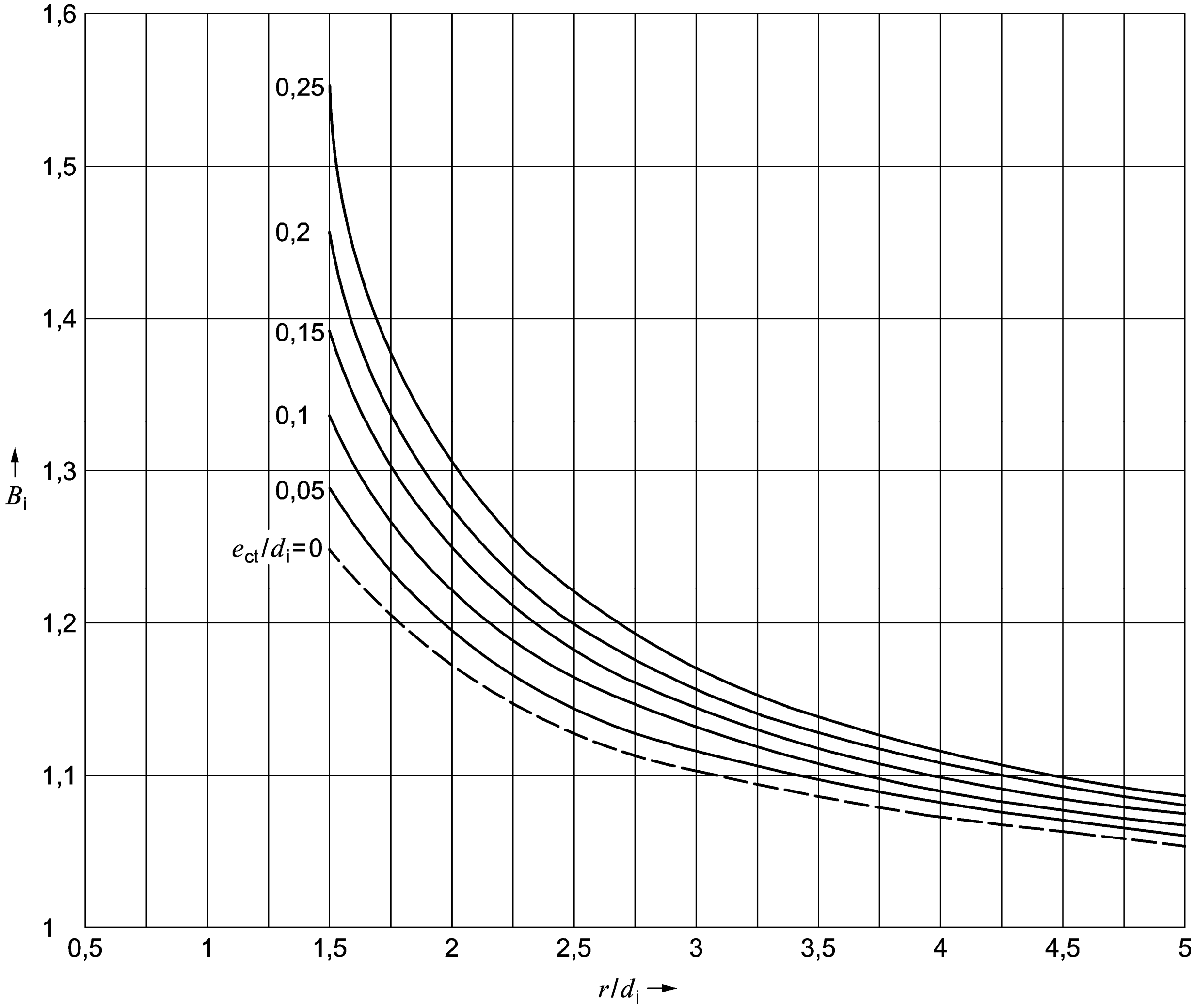


Figure A.2 — Design coefficient B_i for the intrados of tube bends with inside diameter equal to nominal diameter

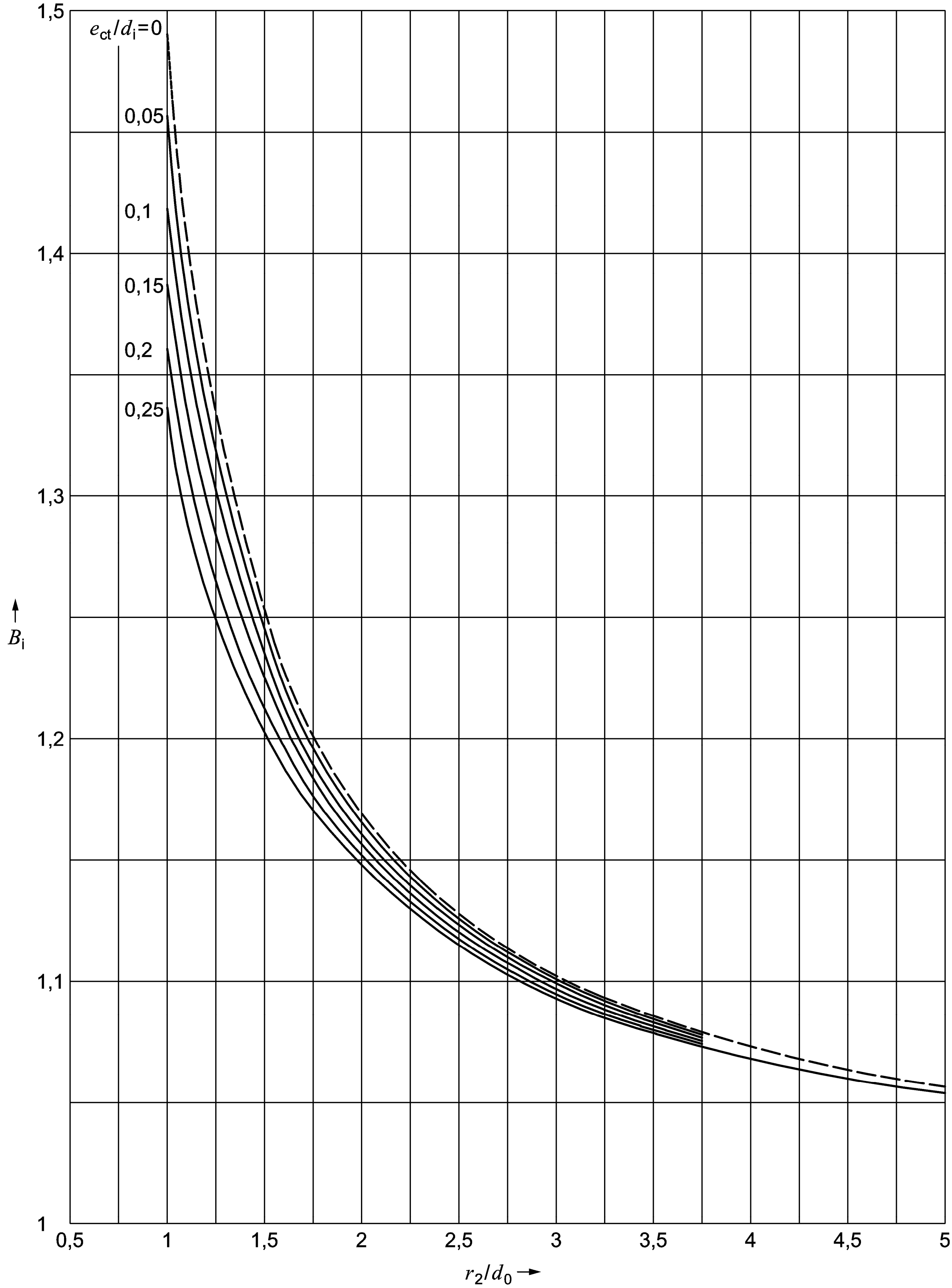


Figure A.3 — Design coefficient B_i for the intrados of tube bends with outside diameter equal to nominal diameter

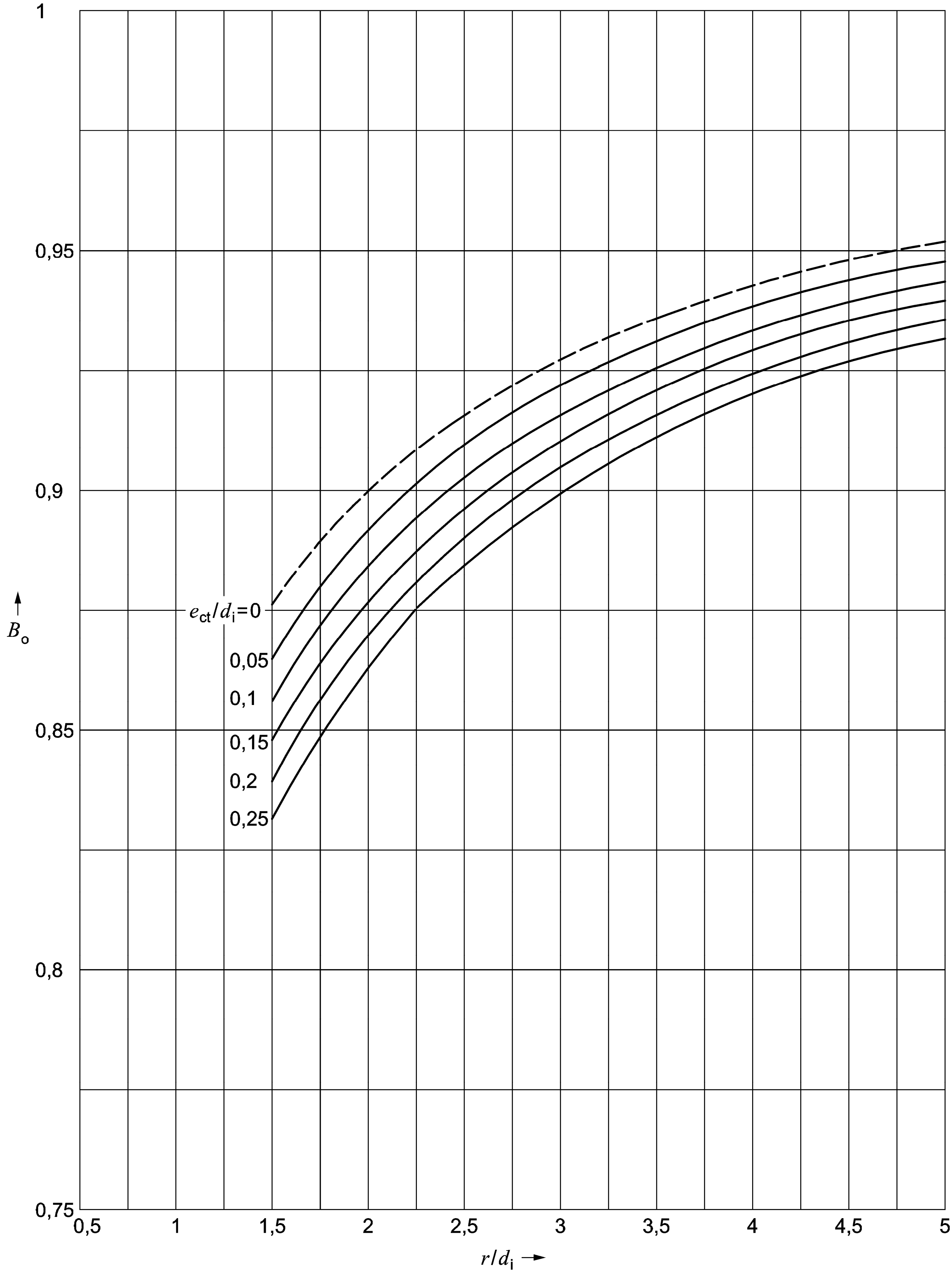


Figure A.4 — Design coefficients B_0 for the extrados of tube bends with inside diameter

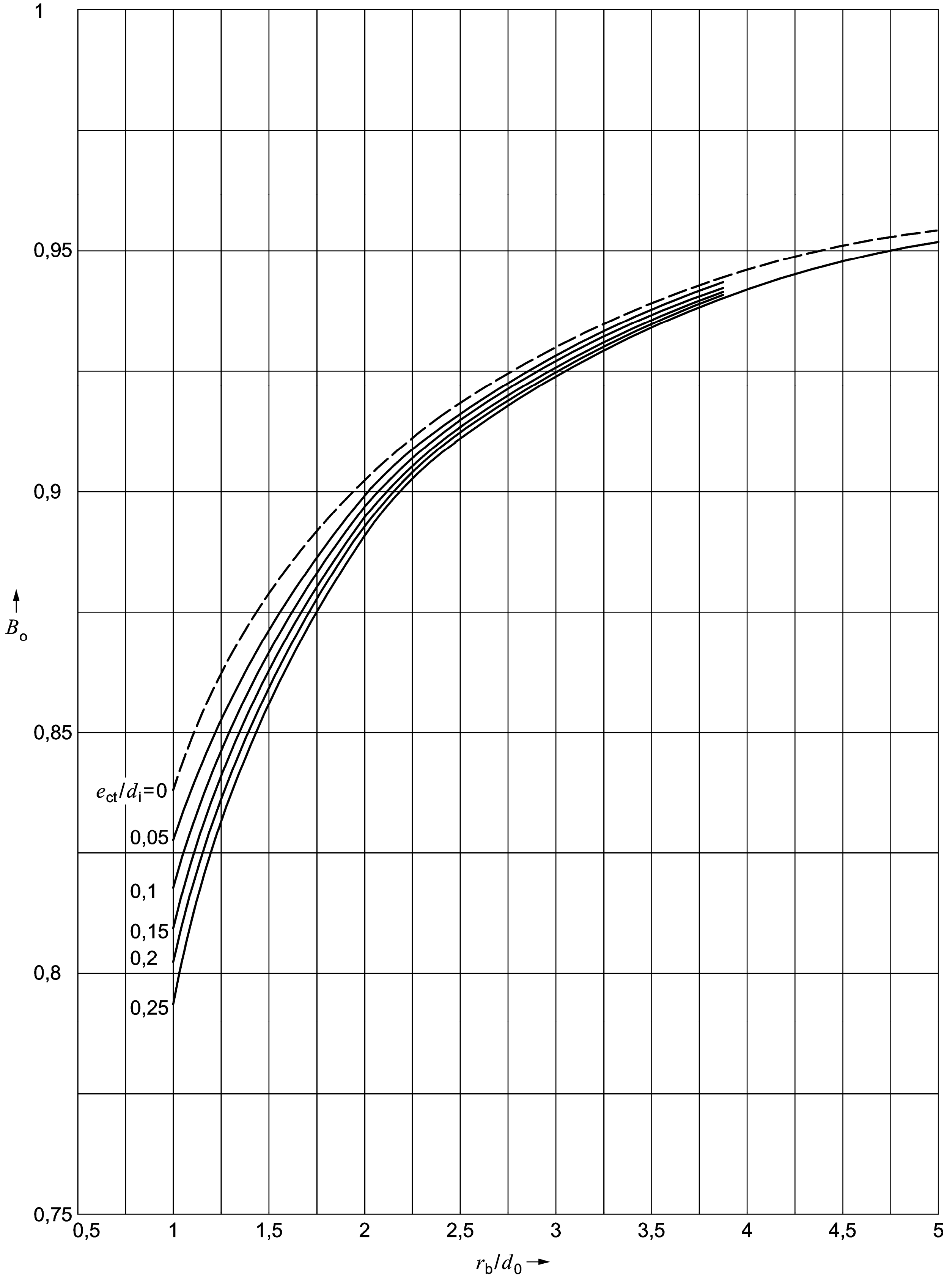


Figure A.5 — Design coefficient B_0 for the extrados of tube bends with outside diameter equal to nominal diameter

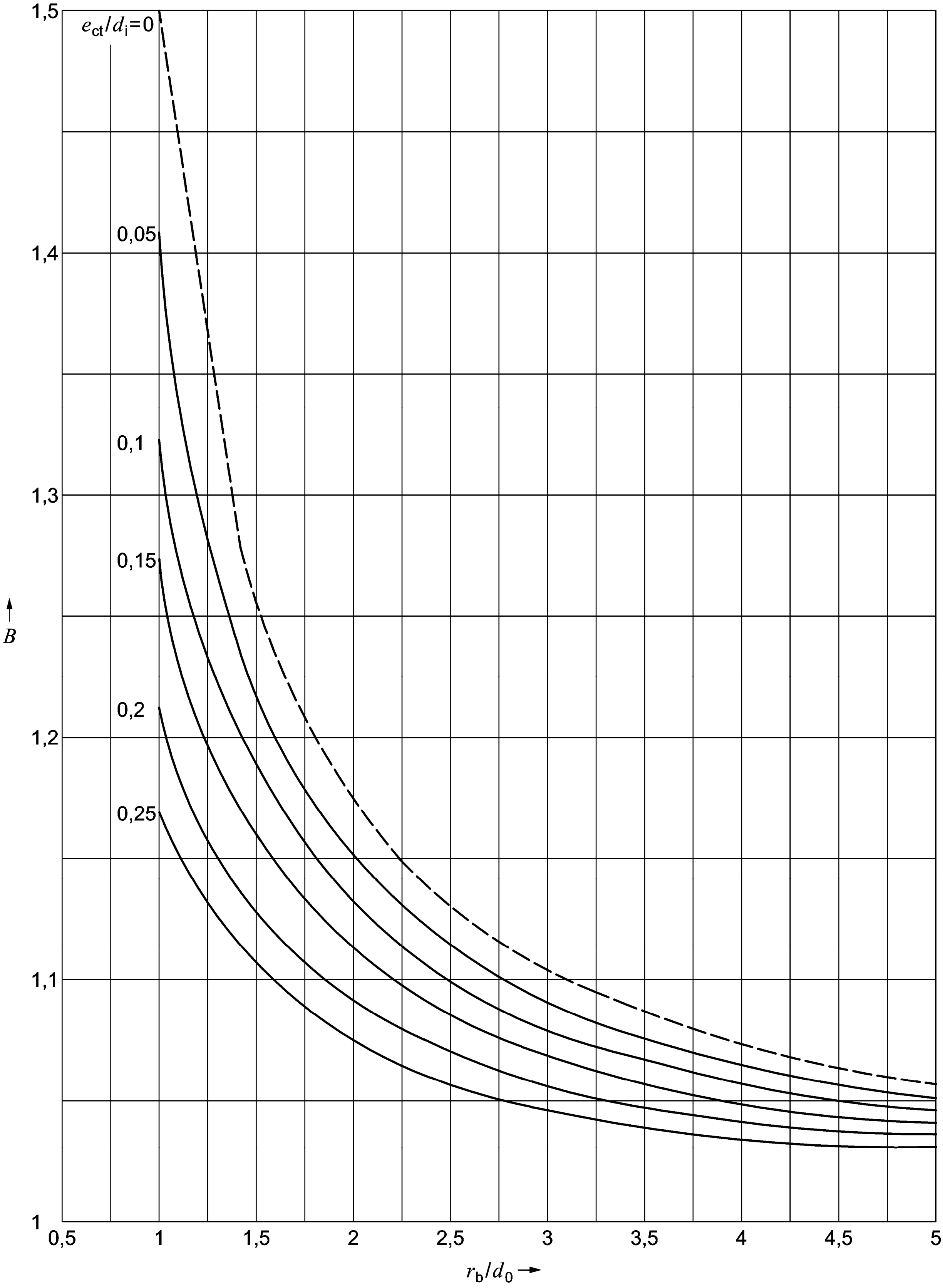


Figure A.6 — Design coefficient B for elbows of equal wall thickness (e_{cti} = e_{cto}) with outside diameter equal to nominal diameter

Annex B
(normative)

Fatigue cracking – Design to allow for fluctuating stress

B.1 General

B.1.1 The design rules presented below apply to the design of pressurized components of boilers made from ferritic and austenitic rolled or forged steels. These rules allow for the fluctuating stresses[16)] occurring at the most highly stressed points as a result of internal pressure and differences in temperature and/or the addition of external forces and moments.

B.1.2 This annex need not be applied if the conditions in accordance with 13.3 shall be satisfied.

B.1.3 Due to the simplicity of this analysis, the results may be conservative with respect to life prediction. More complex methods, e.g. finite element analysis may be applied to obtain more exact life predictions.

B.2 Conditions

B.2.1 The fluctuating stress calculations require a thorough knowledge of the stress and load conditions; see 13.3.4. The loading conditions of the component shall be determined by the mode of operation of the boiler.

B.2.2 The stresses or strains used in this analysis shall be determined either experimentally or mathematically. They shall be assessed in a fatigue analysis, with distinction being made between uni-axial, bi-axial or tri-axial states of stress.

B.2.3 The term "Incipient crack"[17)] shall be used as a failure criterion attributed to fluctuating stresses.

B.2.4 The most critical feature of the most critical component shall be identified and analysed. The dimensions of the component will be subject to manufacturing tolerances. If these dimensions have been measured the actual dimensions shall be used. In the absence of measured dimensions mean dimensions shall be used.

B.2.5 For the purpose of calculation the cyclic temperature to be used during the load cycle under consideration shall be:

$$t^* = 0{,}75\,\hat{t} + 0{,}25\,\check{t} \tag{B-1}$$

where

$$\hat{t} = \max\left\{ t_{\hat{f}}, t_{\check{f}} \right\};$$

[16)] The term "fluctuating stress" used here in a general sense denotes the variation of stress with respect to time independent of the magnitude and the sign of the mean value.

[17)] An "incipient crack" is a material separation which can be detected with optical aids or non-destructive testing methods.

$$\check{t} = \min\left\{ t_{\hat{f}}, t_{\check{f}} \right\}.$$

All temperature-dependent variables shall be referred to this cyclic temperature t^* for the load cycle in question.

B.2.6 The maximum shear stress theory shall be used in the determination of the decisive cyclic stress amplitude and the mean cyclic stress in accordance with B.4.2.

B.2.7 Procedure for determining the permissible stress amplitude or the permissible number of load cycles during operation from fatigue tests on test specimens shall, in individual cases, be selected by the manufacturer as regards the nature of the procedure, the boundary conditions, the number of test specimens and the factors of safety (see [1]).

B.3 Symbols and abbreviations

In addition to the symbols shown in EN 12952-1:2001, Table 4-1, the symbols given in Table B.1 shall be used.

Table B.1 — Symbols

Symbol	Description	Unit
C_k	correction factor for taking into account the notch effect associated with surface roughness or welds,	—
C_{t^*}	correction factor for temperature influence,	—
$\overline{f}_\mathrm{v}$	mean cyclic stress,	MPa
$\overline{f}_\mathrm{v}^*$	corrected mean cyclic stress,	MPa
$2f_\mathrm{a}$	stress range in an unnotched bar specimen which for a certain number of load cycles $n = N_\mathrm{A}$ produces an incipient crack (see B.2.3), and which in connection with the procedure for verifying whether a fluctuating stress is permissible in accordance with this annex, is also used as the permissible stress range,	MPa
$2f_{\mathrm{a\ t^*}}$	virtual controlling stress range at reference temperature t^* > 100 °C,	MPa
$2f_\mathrm{a}^*$	controlling stress range	MPa
$2f_\mathrm{va}$	cyclic stress range	MPa
$2f_\mathrm{va}^*$	corrected cyclic stress range	MPa
N	Design number of load cycles to be expected during operation	—
N_A	number of load cycles for crack initiation	—
$R_{\mathrm{m\ T\ t^*}}$	mean creep strength value for T hours at reference temperature t^*	MPa
$R_{\mathrm{p0,2\ t^*}}$	high-temperature yield point or 0,2 % elastic limit at reference temperature t^*	MPa
S_s	stress safety factor	—

Table B.1 *(continued)*

Symbol	Description	Unit
S_L	load-cycle safety factor	—
t^*	reference temperature for fluctuating stress	° C
Superscripts- and subscripts:		
Superscript $\wedge$	= maximum value, e.g. $\hat{f}$	
Superscript $\vee$	= minimum value, e.g. $\check{f}$	
Superscript $^{-}$	= mean value, e.g. $\bar{f}$	
Subscript k, j	= number index, e.g. n_k, T_j	

B.4 Cyclic stress range and mean cyclic stress in the case of uniaxial and multiaxial fluctuating stress

B.4.1 General

For the reference analysis, the cyclic stress range and the corresponding mean cyclic stress shall be determined for each load cycle occurring during operation. To this end, for every significant point of time during a load cycle, the local stress state shall be known.

B.4.2 Uniaxial stress state

In the case of uniaxial stress as depicted in Figure B.1, the cyclic stress range $2f_{va}$ shall be determined using:

$$2f_{va} = \left(\hat{f} - \check{f}\right) \tag{B-2}$$

and the mean cyclic stress shall be determined using:

$$\bar{f}_v = \tfrac{1}{2}\left(\hat{f} + \check{f}\right). \tag{B-3}$$

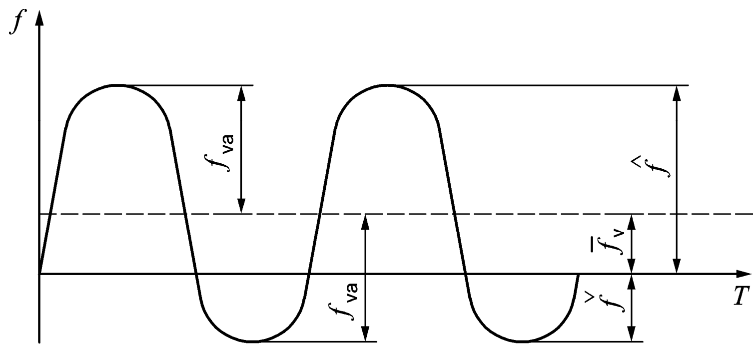


Figure B.1 — Stress variation in the case of uniaxial stress (schematic)

B.4.3 Multiaxial stress state with principal stress directions constant

For a triaxial stress state as shown diagrammatically in Figure B.2, initially the variations of the principal stresses f_1, f_2, and f_3 shall be determined.

Then, the variations with respect to time of the three principal stress differences Δf_{12}, Δf_{23}, Δf_{31} shall be determined using Equations (B-4, B-5 and B-6):

$$\Delta f_{12} = f_1 - f_2 \tag{B-4}$$

$$\Delta f_{23} = f_2 - f_3 \tag{B-5}$$

$$\Delta f_{31} = f_3 - f_1 \tag{B-6}$$

For each of these three variations of the principal stress differences, noting the signs the maximum and minimum values shall be found. The cyclic stress range $2f_{va}$ shall be obtained from Equation (B-7) as follows (see also Figure B.3):

$$2f_{va} = \max \left\{ \begin{matrix} \Delta \hat{f}_{12} - \Delta \check{f}_{12} \\ \Delta \hat{f}_{23} - \Delta \check{f}_{23} \\ \Delta \hat{f}_{31} - \Delta \check{f}_{31} \end{matrix} \right\}. \tag{B-7}$$

The mean cyclic stress $\overline{f}_v$, corresponding to the cyclic stress range $2f_a$, is the mean value from those principal stress differences from which the cyclic stress range shall also be obtained.

$$f_v = \frac{1}{2}\left(\hat{f} - \check{f} \right). \tag{B-8}$$

For cases where the location of the principal stress directions varies with respect to time, attention is drawn to B.6.

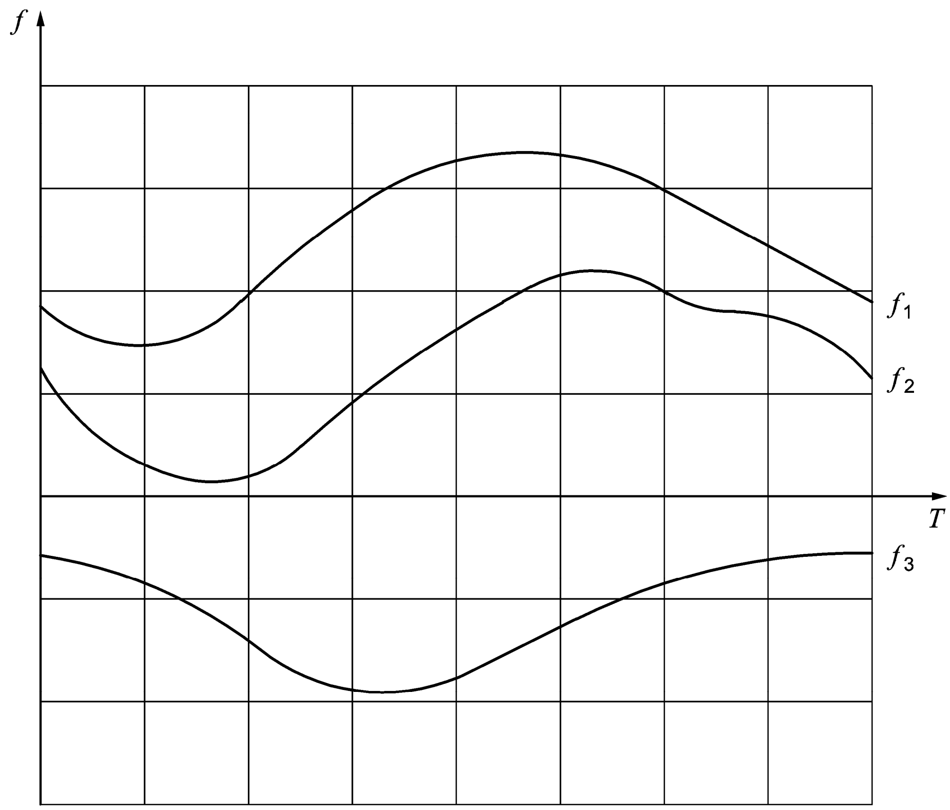
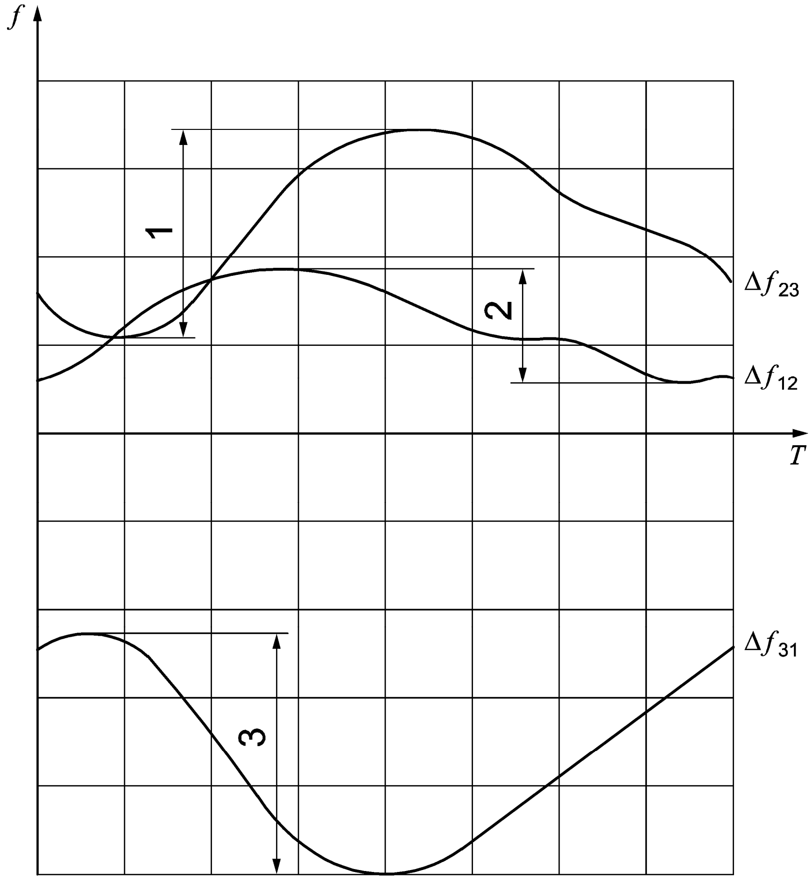


Figure B.2 — Stress variation with triaxial stress, with principal stresses out-of-phase (schematic)



Key

1 $\Delta\hat{f}_{23} - \Delta\check{f}_{23}$

2 $\Delta\hat{f}_{12} - \Delta\check{f}_{12}$

3 $2f_{\text{va}} = \left|\hat{f}_{31} - \Delta\check{f}_{31}\right|$

Figure B.3 — Variation of the differences fort he principal stresses shown in Figure B.2 and cyclic-stress amplitude 2f_{va}

B.5 Correction factors for taking into account the influences of surface finish and weldments

B.5.1 Cyclic stress range and mean cyclic stress in accordance with B.4 shall be increased to account for the notch effect (micro notch effect) associated with surface and weld influences. Here, the governing factor in each case is the final state following manufacture. The corrected cyclic stress range shall be determined using:

$$2f_{\text{va}}^{*} = 2f_{\text{va}} \cdot C_{\text{k}} \tag{B-9}$$

and the corrected mean cyclic stress using:

$$\bar{f}_{\text{v}}^{*} = \bar{f}_{\text{v}} \cdot C_{\text{k}} \tag{B-10}$$

The correction factor C_k shall be determined by means of fatigue tests or be determined as the appropriate (C_{k0}, C_{k1}, C_{k2} or C_{k3}) from the following subclauses.

B.5.2 For smooth rolled surfaces relevant to quality specifications and technical supply conditions, e.g. in accordance with EN 10028-1:2007, the mill-scale correction factor C_{k0} shall be in accordance with Figure B.4. The correction factor C_{k0} can also be described by the Equation (B-11):

$$C_{ko} = \min\left\{1 - 4{,}6 \cdot 10^{-4} \cdot R_m + 2{,}3 \cdot 10^{-4} \cdot R_m \cdot \lg N_A ; 1 + 10^{-3} \cdot R_m\right\} \quad \text{(B-11)}$$

B.5.3 The value C_{k0} = 1 shall be used for calculating the influence of the surfaces in the non-welded region of a component having roughness depths[18] R_z < 6 µm, such as can be achieved by grinding or machining.

Surfaces in the non-welded region with peak-to-valley heights R_z of 6 to 50 µm can be assigned values[19] of C_{k0} = 1,3 for R_m = 1 000 MPa, and C_{k0} = 1,1 for R_m = 400 MPa.

In Tables B.1, B.2 and B.3, welded joints of the kind customary for pressure vessels are depicted and classified regarding their notch-effect into three configuration groups, K 1, K 2 and K 3.

[18] Guidance values and empirical values for attainable, averaged roughness depth R_z in accordance with ISO 4287 for different production methods are specified in EN ISO 4287.

[19] Intermediate values may be obtained by means of linear interpolation.

Table B.2 — Configuration examples of welded joints with a slight notch-effect (Configuration Group K 1)

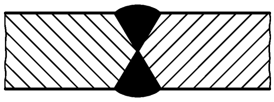
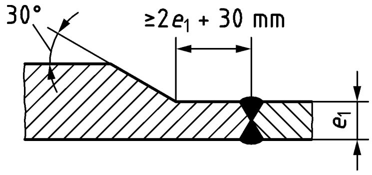
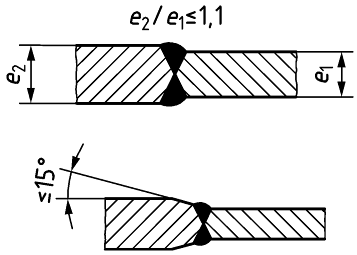
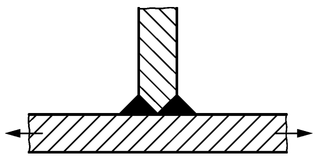
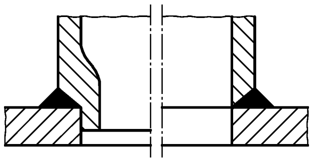
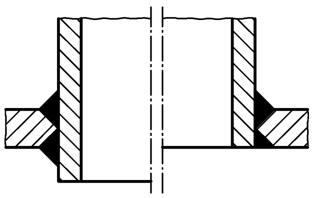
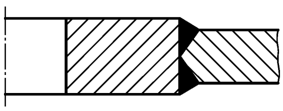
No	Illustration	Description	Requirements
1		Longitudinal or circum-ferential weld in walls of equal thickness	Welded from both sides
2		Longitudinal or circum-ferential weld in walls of unequal thickness	Welded from both sides
			Welded from both sides; edge-offset the same inside and outside
3		Welded attachments with no additional cyclic forces or moments	Full penetration weld from one or both sides; external inspection in accordance with, EN 12952-6, excluding the features weld convexity and concavity, and asymmetry
4		Set-on nozzle	Fully load-bearing weld (no residual gap); nozzle machined out or root ground flush
5		Set-in or set-through nozzle	Full penetration welded from one or both sides
6		Butt welded-in ring flange	Full penetration welded from one or both sides

Table B.3 — Configuration examples of welded joints with a moderate notch-effect (Configuration Group K 2)

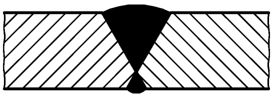
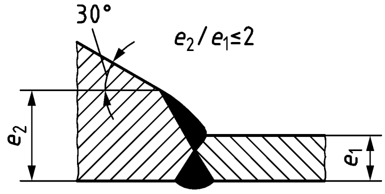
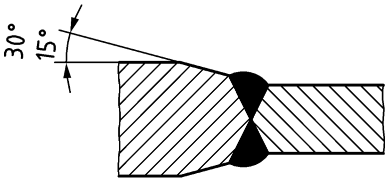
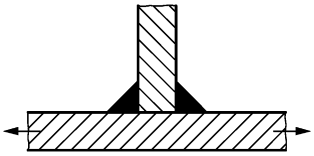
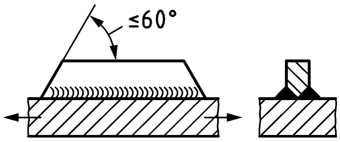
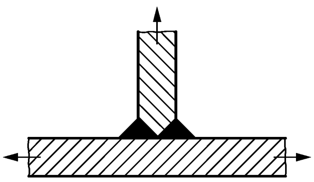
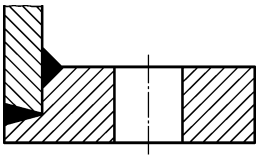
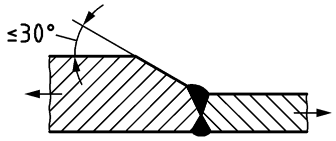
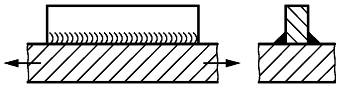
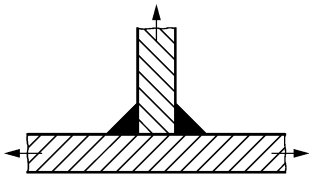
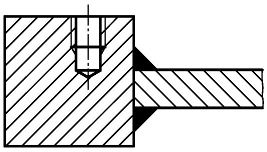
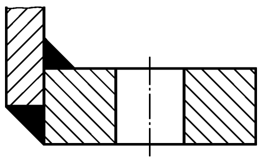
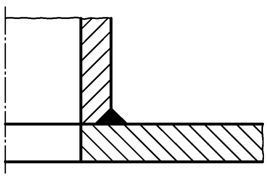
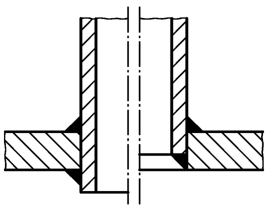
No	Illustration	Description	Requirements
1		Longitudinal or circumferential weld in walls of equal thickness	Welded from both sides; if equivalent to joint welded from both sides, joint can be assigned to Configuration Group K 1
2		Longitudinal or circumferential weld in walls of unequal thickness	Welded from both sides
			Welded from both sides; edge-offset the same inside and outside
3		Welded attachments with no additional cyclic forces or moments	Welded from both sides (not full penetration); external inspection in accordance with EN 12952-6:2011, excluding the features weld convexity and concavity, and asym-metry
4			Full penetration welded from both sides; external inspection in accordance with EN 12952-6:2011, excluding features weld convexity and concavity, and asymmetry
5		Welded-on attachments with additional cyclic forces or moments; stiffening ribs	Full penetration welded from both sides; external inspection in accordance with EN 12952-6:2011
6		Welded-on flange	Welded from both sides, in accordance with EN 12952-5:2011, Annex B

Table B.4 — Configuration examples of welded joints with a pronounced notch effect (Configuration Class K 3)

No	Illustration	Description	Requirements
1		Longitudinal or circumferential weld in walls of unequal thickness	Welded from one or both sides
2		Welded-on attachments with no additional cyclic forces or moments	Welded from one or both sides(not full penetration)
3		Welded-on attachments with no additional cyclic forces or moments, stiffening ribs	Welded from both sides (not full penetration)
4		Welded-in pad	Welded from both sides (not full penetration)
5		Welded-on flange	Welded from both sides; in accordance with EN 12952-5:2011, Annex B (not full penetration)
6		Set-on nozzle	Weld not fully load-bearing
7		Set-in or set through nozzle	Weld not fully load-bearing (not full penetration)

The corresponding weld correction factors C_{k1}, C_{k2} and C_{k3} shall be obtained from Figures B.5 to B.7, or calculated using the equations shown in Table B.4. They embrace the influences of both internal and also external notches and there shall be no need therefore for them to be weighted with C_{k0} for the calculation of the corrected cyclic stress range $2\,f^{*}_{va}$ and mean cyclic stress $\overline{f}^{*}_{v}$ in the weld zone.

The correction factors C_{k1} for nozzle welds are not applicable for taking account of stress concentrations at nozzle hole edges.

B.5.4 For longitudinal or circumferential welds in pressurized walls as shown in Table B.1, No. 1 and No. 2, which are machined back flush with the plate on both sides in order to increase the service life, the correction factor C_{k1} can be reduced to the value for C_{k0} using Equation (B-11) provided that the extent of the non-destructive examination shall be 100 %.

B.5.5 The correction factors for other surface finishes and welded joints which are not covered here shall be selected in accordance with the manufacturer's own proven practice which shall ensure the safety of the boiler is not impaired.

Table B.5 — Correction factors Ck for taking account of the notch–effect associated with the influence of the weldments

Tensile strength R_m	Welded joints		
	Configuration group K 1 (slight notch–effect) Table B.1	Configuration group K 2 (moderate notch–effect) Table B.2	Configuration Group K 3 (pronounced notch–effect) Table B.3
MPa	C_{k1}	C_{k2}	C_{k3}
400	$1{,}5 \leq 0{,}19 \lg N_a + 0{,}62 \leq 1{,}8$	$1{,}6 \leq 0{,}21 \lg N_a + 0{,}79 \leq 2{,}1$	$1{,}8 \leq 0{,}34 \lg N_a + 0{,}66 \leq 2{,}8$
600	$1{,}7 \leq 0{,}40 \lg N_a + 0{,}20 \leq 2{,}7$	$1{,}9 \leq 0{,}40 \lg N_a + 0{,}60 \leq 3{,}1$	$2{,}1 \leq 0{,}56 \lg N_a + 0{,}40 \leq 4{,}0$
800	$1{,}8 \leq 0{,}56 \lg N_a + 0{,}12 \leq 3{,}4$	$2{,}1 \leq 0{,}56 \lg N_a + 0{,}44 \leq 4{,}0$	
1 000	$1{,}9 \leq 0{,}70 \lg N_a + 0{,}40 \leq 4{,}0$	$2{,}5 \leq 0{,}75 \lg N_a + 0{,}25 \leq 5{,}0$	

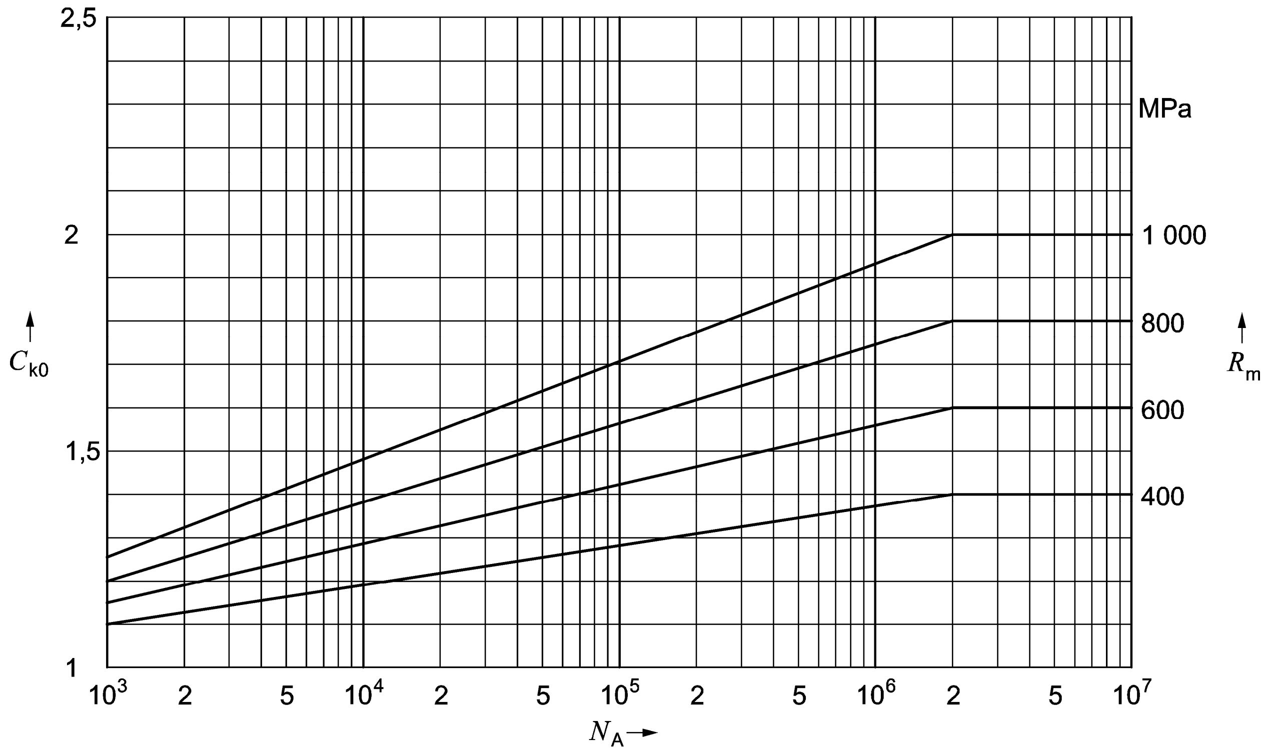


Figure B.4 — Correction factor C_{k0} for taking account of the surface notch effect due to mill scale

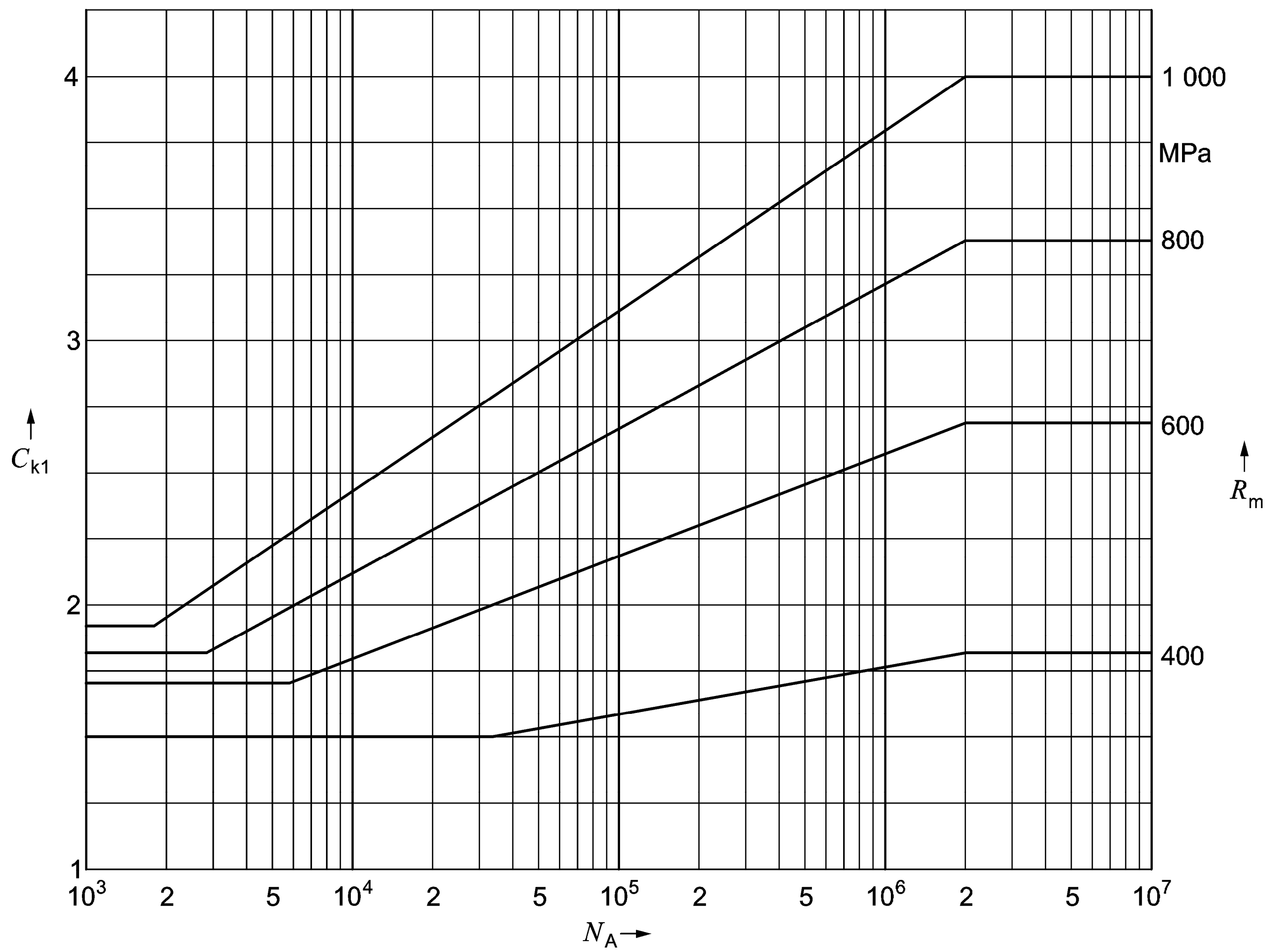


Figure B.5 — Correction factor C_{k1} for taking account of the weld notch effect for configuration group K 1

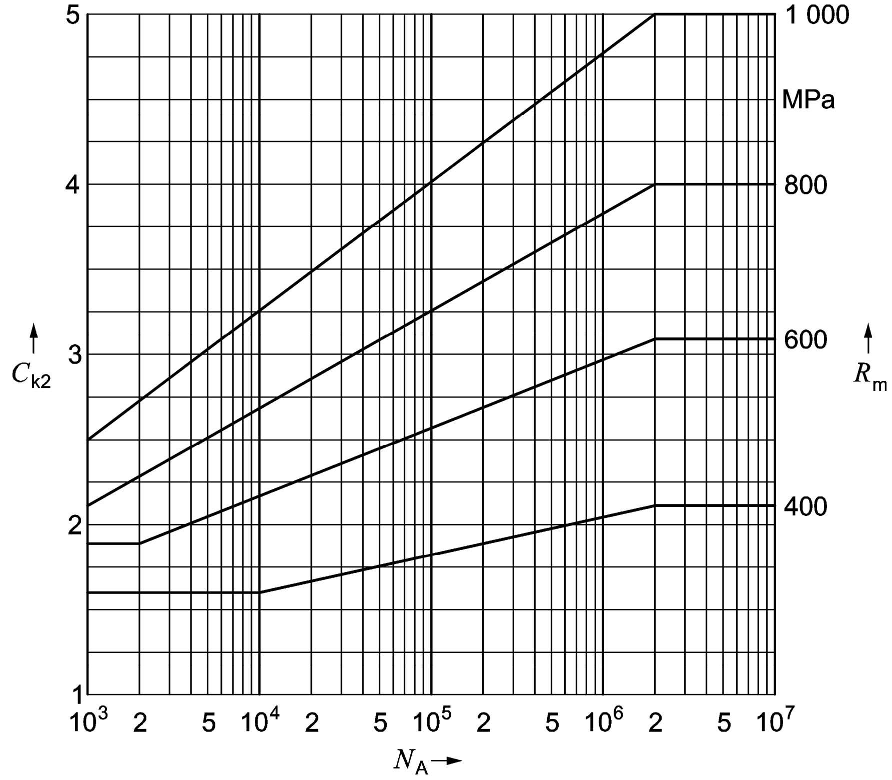


Figure B.6 — Correction factor C_{k2} for taking account of the weld notch effect for configuration group K 2

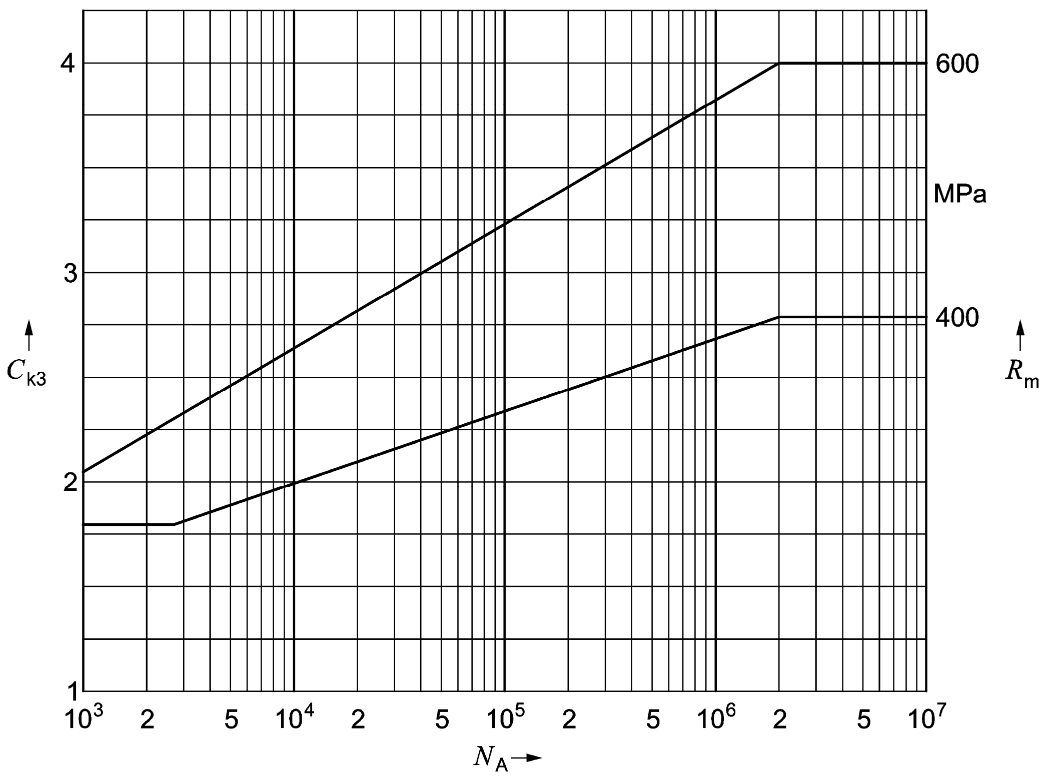


Figure B.7 — Correction factor C_{k3} for taking account of the weld notch effect for configuration group K 3

B.6 Controlling stress range

B.6.1 General

The controlling stress range $2f_a^*$, for comparison with the stress range $2f_a$ in accordance with Figure B.9 shall be obtained as follows:

B.6.1.1 Elastic range

If $\left|f_v^*\right| + 2f_{va}^*/2 \leq R_{p0,2\,t^*}$, the stress range $2f_a$* shall be determined using the corrected range of the equivalent stress $2f_{va}^*$ and the corrected medium value f_v^* of the range of the equivalent stress in Equation (B-12):

$$2f_a^* = \frac{2f_{va}^*}{\left(1-\left(f_v^*/R_m\right)^2\right)} \quad \text{(B-12)}$$

This equation takes into account the modifying influence of the medium stress on the fatigue strength.

B.6.1.2 Partly elastic range

If during a load cycle the maximum stress f_v^*

$$f_v^* = C_k \cdot \max\left(\left|\Delta f_{12}\right|, \left|\Delta f_{23}\right|, \left|\Delta f_{31}\right|\right) \quad \text{(B-13)}$$

exceeds the yield strength $R_{p0,2\ t^*}$, that means if $\left|f_v^*\right| + 2f_{va}^*/2 > R_{p0,2\ t^*}$, but if $2f_{va}^*/2 \leq R_{p0,2\ t^*}$, also Equation (B-12) shall be used for calculation of the decisive stress range $2f_a$*. However, in this case the lowered medium stress

$$f_{vR}^* = R_{p0,2\ t^*} - 2f_{va}^*/2 \quad \text{(B-14)}$$

shall be applied instead of f_v^*.

B.6.1.3 Fully plastic range

If the corrected cyclic stress range $2f_{va}^*$ exceeds twice the yield point $(2f_{va}^* > 2R_{p0,2\ t^*})$, the mean cyclic stress shall be taken as $f_v = 0$ and the controlling stress range $2f_a^*$ shall be determined as a function of the yield point from:

$$2f_a^* = \frac{\left(2f_{va}^*\right)^2}{2R_{p0,2\ t^*}} \quad \text{(B-15)}$$

Equation (B-13) shall not be used if the cyclic stress range has been determined as a virtual stress from the total strain $2\varepsilon_{a\ tot}$ (elastic + plastic) in a theoretical or experimental stress analysis, from $2f_a^* = 2E\varepsilon_{a\ tot}$.

B.6.2 Correction factor

In the case of a load-cycle temperature t* ≥ 100 °C, the reduction in the fatigue strength caused by the temperature shall be taken into account by means of a correction factor C_{t^*}.

With regard to the procedure for the determination of the permissible stress range for a known number of load cycles in accordance with B.7, it shall be advisable in this connection to convert the controlling stress range $2f^*_a$ in accordance with Equations (B-12) and (B-15) with the reciprocal value $1/C_{t^*}$ to a virtual controlling stress range $2f^*_{a\,t^*}$, so that there is no need for service-life diagrams to be given for a temperature t^* = 100 °C.

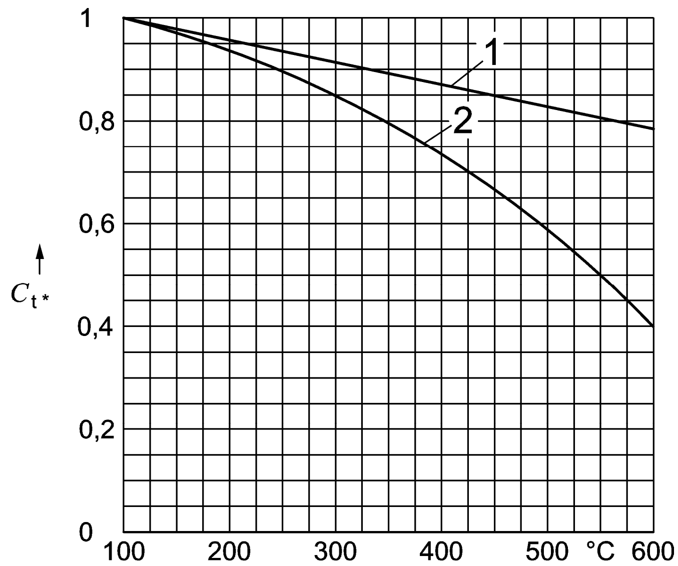
The correction factor C_{t^*} shall be obtained from Figure B.8, or shall be determined over the temperature range 100 °C ≤ t^* ≤ 600 °C using:

$$C_{t^*} = 1{,}03 - 1{,}5 \cdot 10^{-4}\, t^* - 1{,}5 \cdot 10^{-6}\, t^{*2} \quad \text{(ferritic)} \tag{B-16}$$

$$C_{t^*} = 1{,}043 - 4{,}3 \cdot 10^{-4}\, t^* \quad \text{(austenitic)} \tag{B-17}$$

The virtual controlling stress range $2f^*_{a\,t^*}$ shall be obtained as follows:

$$2f^*_{a\,t^*} = \frac{2f^*_a}{C_{t^*}} \tag{B-18}$$



Key
1 austenitic
2 ferritic

Figure B.8 — Correction factor C_{t^*} for taking account of the temperature influence

B.7 Permissible stress range with a known number of load cycles

B.7.1 As the mean values of the stress range shown in Figure B.9 have been derived from experimental data, they do not include safety factors. For the determination of the permissible stress range a stress safety factor of S_s = 1,5 shall be used, and as a load-cycle safety factor S_L = 10 shall be used (see [1]).

The curves shown in Figure B.9 shall be described by the Equation (B-19). In the case of infinite number of load cycles the stress range merge to the value of Equation (B-20):

$$2f_a = 0{,}8R_m + (173150 - 0{,}8R_m)\,N_A^{-0{,}547} \quad \text{(B-19)}$$

For an infinite number of load cycles

$$2f_a = 0{,}8R_m \quad \text{(B-20)}$$

B.7.2 The permissible stress range shall be derived from the minimum of the stress $2f_{as}/S_s$ and $2f_{aL}$:

$$2f^*_{a\,t^*} \leq \min \begin{Bmatrix} 2f_{as}/S_s \\ 2f_{aL} \end{Bmatrix} \quad \text{(B-21)}$$

For the design number of load cycles N the stress range $2f_{as}$ shall be determined from Figure B.9 or Equation (B-19) with

$$N_A = N \quad \text{(B-22)}$$

and the stress range $2f_{aL}$ shall be determined from Figure B.9 or Equation (B-19) with

$$N_A = S_L\,N \quad \text{(B-23)}$$

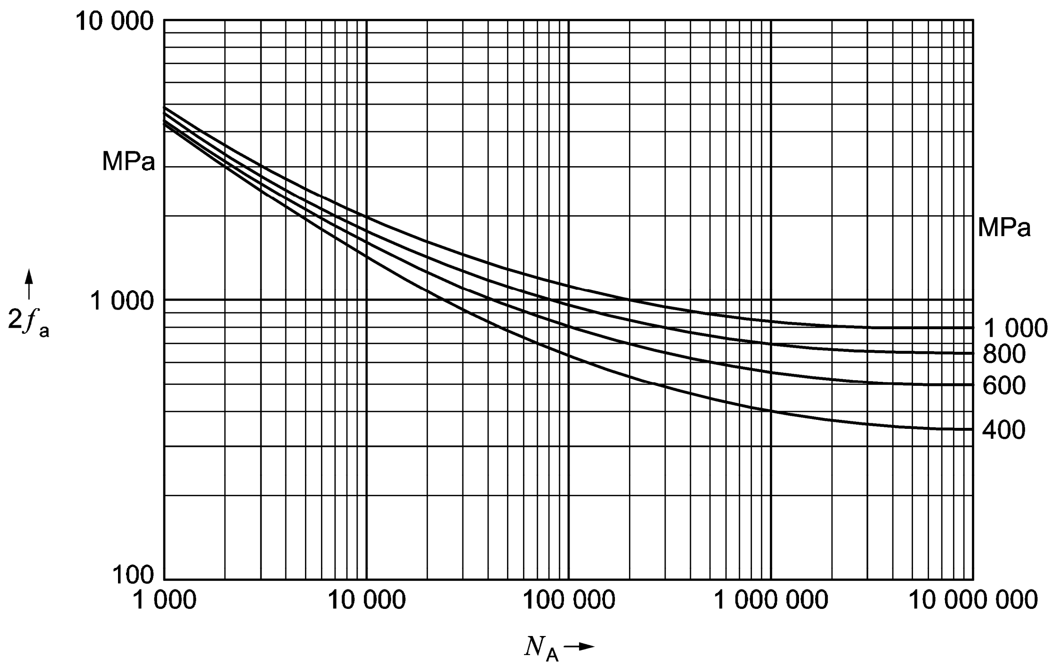


Figure B.9 — Number of load cycles NA for crack initiation (mean value of the scatter band) as a function of stress range for unnotched bar specimens of high-temperature ferritic rolled or forged steels at room temperature and f_v = 0

Annex C
(informative)

Examples of calculating the effects of fatigue

C.1 General

This annex contains examples for calculating the effects of fatigue. They are intended to assist the designer in the understanding of the standard and to show a methodology for calculation.

C.2 Calculation of the admissible number of load cycles

C.2.1 The Table C.1 provides an example of a procedure, using the equations in Clause 13 and Annex B, to calculate the admissible number of load cycles for a given component geometry and required temperature gradient.

Table C.1 — Calculation of the admissible number of load cycles

Example 1: Calculation of the fatigue loading, in accordance with Clause 13 and Annex B: Calculation of the admissible number of load cycles					
This calculation shall be carried out for a different number of cycles such that $\Sigma(n/N) \leq 1$ if the load regime is specified. If the load regime is not specified see 13.1.2					
Component: Drum 1 600 outside diameter ×56 minimum specified wall thickness Drawing No:				Page 1 of 4	
material: xxx EN xxx					
cycle number:			1		
cycle type:			cold start		
calculation pressure	p_c	MPa	8,53		
calculation temperature	t_c	° C	300,00		
operation pressure	p_o	MPa	8,15		
min. cyclic pressure	p_{min}	MPa	0,00		
max. cyclic pressure	p_{max}	MPa	8,15		
min. cyclic temperature	t_{min}	° C	20,00		
max. cyclic temperature	t_{max}	° C	297,00		
reference temperature	t^*	° C	227,75		
material properties: (F = Ferritic, M = Martensitic, A = Austenitic)			F		
tensile strength at room temperature	R_m	MPa	510,00		
yield strength at t^*	$R_{e(t^*)}$	MPa	253,90		
coefficient of linear thermal expansion at t^*	β_L	1/K	1,40E-05		
modules of elasticity at t^*	E	MPa	1,97E+05		

Table C.1 *(continued)*

Example 1 *(continued)*				Page 2 of 4	
thermal diffusivity at t*	D_{th}	mm^2/s	10,19		
Poisson's ratio	υ		0,30		
component dimensions:					
outside diameter of the drum	d_o	mm	1 600,00		
mean wall thickness of the drum	e_{ms}	mm	57,00		
mean diameter of the drum	d_{ms}	mm	1 543,00		
outside diameter of the branch	d_{ob}	mm	108,00	298,50	
mean wall thickness of the branch	e_{mb}	mm	8,80	33,35	
mean diameter of the branch	d_{mb}	mm	99,20	265,15	
factors:					
e_{mb}/e_{ms}			0,154 4	0,585 1	
$\zeta = d_{mb}/d_{ms} \cdot SQRT(d_{ms}/2/e_{ms})$	ζ		0,236 5	0,632 2	
α_m (Figure 13.4-5 / Figure 13.4-7, 13.4)	α_m		3,04	3,34	
water/steam			water	water	
$z = d_{mb}/d_{ms}$	z		0,064 3	0,171 8	
α_t (Figure 13.4-8)	α_t		1,692	1,354	
$u_0 = d_o/(d_{ms} - e_{ms})$	u_0		1,076 7		
γ(Figure 13.4-6 or Figure 13.4-9)	γ		-0,345 9		
$W = \alpha_t \cdot \beta_L \cdot E/(1-\upsilon)$	W	MPa/K	6,665 3	5,334 2	
$V = D_{th}/\gamma/e_{ms}^2$	V	1/s	-9,07E-03		
assumed cycle conditions:					
d_t given:					
d_{t1} (begin of start-up)	d_{t1}	K	-30,80		
v_{t1} (begin of start-up) = $V \cdot d_{t1}$	v_{t1}	K/s	0,28		
v_t given:					
V_{t1} (begin of start-up)	v_{t1}	K/s			
d_{t1} (begin of start-up) = v_{t1}/V	d_{t1}	K			
stresses:					
$S_{p,o} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_o$	$S_{p,o}$	MPa	343,76	376,57	
$S_{p,min} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_{min}$	$S_{p,min}$	MPa	0,00	0,00	
$S_{p,max} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_{max}$	$S_{p,max}$	MPa	343,76	376,57	
$S_{t,min} = W \cdot d_{t1}$	$S_{t,min}$	MPa	-205,29	-164,29	
CASE (ferritic and martensitic materials, in contact with water)					

Table C.1 *(continued)*

Example 1 *(continued)*				Page 3 of 4	
IF $((S_{p,o} - p_o - 600) < (S_{p,min} + S_{t,min}))$					
$f_1 = S_{p,min} + S_{t,min}$	f_1	MPa	-205,29	-164,29	
ELSE Assume new cycle conditions and calculate stresses					
CASE (austenitic materials in contact with water or steam or other materials only in contact with steam)					
$f_1 = S_{p,min} + S_{t,min}$	f_1	MPa			
$f_2 = S_{p,max}$	f_2	MPa	343,76	376,57	
$f_v = (f_1 + f_2)/2$	f_v	MPa	69,23	106,14	
$\Delta f_v = f_2 - f_1$	Δf_v	MPa	549,05	540,87	
Notch factor (as appropriate):					
C_{k0}			1,00	1,00	
C_{k1}					
C_{k2}					
C_{k3}					
C_k			1,00	1,00	
$f_v^* = f_v \cdot C_k$	f_v^*	MPa	69,23	106,14	
$2f_{va}^* = \Delta f_v \cdot C_k$	$2f_{va}^*$	MPa	549,05	540,87	
IF $(2f_{va}^* < R_{e(t^*)})$					
$2f_a^* = 2f_{va}^*/(1 - (f_v^*/R_m)^2)$	$2f_a^*$	MPa			
IF $(R_{e(t^*)} \leq f_v^* \leq 2R_{e(t^*)}$ OR $R_{e(t^*)} \leq 2f_{va}^* \leq 2R_{e(t^*)})$					
$f_v^* = R_{e(t^*)} - 2f_{va}^*/2$	f_v^*	MPa			
$2f_a^* = 2f_{va}^*/(1 - (f_v^*/R_m)^2)$	$2f_a^*$	MPa			
IF $(2f_{va}^* > R_{e(t^*)})$					
$2f_a^* = (2f_{va}^*)^2/(2R_{e(t^*)})$	$2f_a^*$	MPa	593,66	576,09	
C_{t^*} (Annex B, Figure B.8)	C_{t^*}		0,92		
$2f_{a(t^*)}^* = 2f_a^*/C_{t^*}$	$2f_{a(t^*)}^*$	MPa	646,66	627,52	

Table C.1 *(continued)*

Example 1 *(continued)*				Page 4 of 4	
S_s	S_s		1,50	1,50	
$2f_{as} = 2f^*_{a(t^*)} \cdot S_s$	$2f_{as}$	MPa	969,99	941,29	
N_{As} (Annex B, Figure B.9 using $2f_{as}$	N_{As}		35 305	38 857	
$2f_{aL} = 2f^*_{a(t^*)}$	$2f_{aL}$	MPa	646,66	627,52	
N_{AL} (Annex B, Figure B.9, using $2f_{aL}$)	N_{AL}		168 973	196 870	
S_L = 10	S_L		10	10	
N = MIN (N_{As}, N_{AL}/S_L)	N		16 897	19 687	
assumed number of cycles	n		2 000	2 000	
using factor = n/N			0,118 4	0,101 6	

C.3 Calculation of the admissible temperature gradient

C.3.1 Table C.2 provides an example of a procedure, using the equations contained in Clause 13 and Annex B, to calculate the admissible temperature gradient for a given component geometry and required number of load cycles.

Table C.2 — Calculation of admissible temperature gradient

Example 2: Calculation of the fatigue loading, in accordance with Clause 13 and Annex B: Calculation of the admissible temperature gradient					
This calculation shall be carried out for a different number of cycles such that $\Sigma(n/N) \leq 1$ if the load regime is specified. If the load regime is not specified see 13.1.2.					
Component: Drum 1 600 outside diameter × 56 minimum specified wall thickness				Page 1 of 3	
Drawing No.:					
Material: xxx EN xxx					
cycle number:			1		
cycle type:			cold start		
calculation pressure	p_c	MPa	8,53		
calculation temperature	t_c	° C	300,00		
operation pressure	p_o	MPa	8,15		
min. cyclic pressure	p_{min}	MPa	0,00		
max. cyclic pressure	p_{max}	MPa	8,15		
min. cyclic temperature	t_{min}	°C	20,00		
max. cyclic temperature	t_{max}	°C	297,00		
reference temperature	t^*	°C	227,75		
material properties: (F = Ferritic, M = Martensitic, A = Austenitic)			F		
tensile strength at room temperature	R_m	MPa	510,00		
yield strength at t^*	$R_{e(t^*)}$	MPa	253,90		
coefficient of linear thermal expansion at t^*	β_L	1/K	1,40E-05		
modules of elasticity at t^*	E	MPa	1,97E+05		
thermal diffusivity at t^*	D_{th}	mm^2/s	10,19		
Poisson's ratio	υ		0,30		
component dimensions:					
outside diameter of the drum	d_o	mm	1 600,00		
mean wall thickness of the drum	e_{ms}	mm	57,00		
mean diameter of the drum	d_{ms}	mm	1 543,00		
outside diameter of the branch	d_{ob}	mm	108,00	298,50	
mean wall thickness of the branch	e_{mb}	mm	8,80	33,35	
mean diameter of the branch	d_{mb}	mm	99,20	265,15	
factors:					
e_{mb}/e_{ms}			0,154 4	0,585 1	
$\zeta = d_{mb}/d_{ms} \cdot SQRT(d_{ms}/2/e_{ms})$	ζ		0,236 5	0,632 2	
α_m (Figure 13.4-5 / Figure 13.4-7, 13.4)	α_m		3,04	3,34	
water/steam			water	water	

Table C.2 *(continued)*

Example 2 *(continued)*				Page 2 of 3	
$z = d_{mb}/d_{ms}$	z		0,064 3	0,171 8	
α_t (Figure 13.4-8)	α_t		1,692	1,354	
$u_0 = d_o/(d_{ms} - e_{ms})$	u_0		1,076 7		
γ (Figure 13.4-6 or Figure 13.4-9)	γ		-0,345 9		
$W = \alpha_t \cdot \beta_L \cdot E/(1-\upsilon)$	W	MPa/K	6,665 3	5,3342	
$V = D_{th}/\gamma/e_{ms}^2$	V	1/s	-9,07E-03		
number of cycles $N = N_{AS}$ (= 2 000/0,4)	N		5 000		
$2f_{as}$ (Annex B, Figure B.9)	$2f_{as}$	MPa	2 045,05		
S_s = 1,5	S_s		1,5		
S_L = 10	S_L		10		
$N_{AL} = N \cdot S_L$	N_{AL}		50 000		
$2f_{aL}$ (Annex B, Figure B.9)	$2f_{aL}$	MPa	872,58		
$2f^*_{a(t^*)}$ = MIN($2f_{as}/S_s$; $2f_{aL}$)	$2f^*_{a(t^*)}$	MPa	872,58		
C_{t^*} (Annex B, Figure B.8)	C_{t^*}		0,918		
$2f^*_a = 2f^*_{a(t^*)} \cdot C_{t^*}$	$2f^*_a$	MPa	801,06		
IF ($2f^*_a < R_{e(t^*)}$)					
$2f^*_{va} = 2f^*_a$	$2f^*_{va}$	MPa			
IF ($R_{e(t^*)} \le 2f^*_a \le 2R_{e(t^*)}$)					
$Y = R_m/2f^*_a$					
$X = R_{e(t^*)}/R_m$					
$2f^*_{va} = 2R_m \cdot (X - Y + SQRT(1 - 2YX + Y^2))$	$2f^*_{va}$	MPa			
IF ($2R_{e(t^*)} < 2f^*_a$)					
$2f^*_{va} = SQRT(2R_{e(t^*)} \cdot 2f^*_a)$	$2f^*_{va}$	MPa	637,79		
Notch factors (as appropriate):					
C_{k0}			1,00	1,00	
C_{k1}					
C_{k2}					
C_{k3}					
C_k			1,00	1,00	

Table C.2 *(continued)*

Example 2 *(continued)*				Page 3 of 3	
$\Delta f_v = 2 f_{va}^* / C_k$	Δf_v	MPa	637,79	637,79	
Stresses:					
$S_{p,o} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_o$	$S_{p,o}$	MPa	343,76	376,57	
$S_{p,min} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_{min}$	$S_{p,min}$	MPa	0,00	0,00	
$S_{p,max} = (\alpha_m \cdot d_{ms}/2/e_{ms} + 1)p_{max}$	$S_{p,max}$	MPa	343,76	376,57	
$\Delta S_p = S_{p,max} - S_{p,min}$	ΔS_p	MPa	343,76	376,57	
factor g_s $(0 \leq g_s \leq 1)$	g_s		0,00		
$S_1 = (S_{p,min} + (\Delta S_p - \Delta f_v) \cdot (1 - g_s))$		MPa	-294,03	-261,22	
$S_2 = S_1 + \Delta f_v$		MPa	343,67	376,57	
CASE (ferritic and martensitic materials, in contact with water)					
f_1 = MAX(S_1; $S_{p,o} - p_o - 600$)	f_1	MPa	-264,39	-231,58	
f_2 = MIN(S_2; $S_{p,o} - p_o + 200$)	f_2	MPa	343,76	376,57	
CASE (austenitic materials in contact with water or steam or other materials only in contact with steam)					
$f_1 = S_1$	f_1	MPa			
$f_2 = S_2$	f_2	MPa			
operating conditions:					
d_{t1} (begin of start-up) = $(f_1 - S_{p,min})/W$	d_{t1}	K	-39,67	-43,41	
v_{t1} (begin of start-up) = $d_{t1} \cdot V$	v_{t1}	K/s	0,36	0,39	
$d_{t1'}$ (end of start-up) = $(f_1 - S_{p,max})/W$	$d_{t1'}$	K	-91,24	-114,01	
$v_{t1'}$ (end of start-up) = $d_{t1'} \cdot V$	$v_{t1'}$	K/s	0,83	1,03	
d_{t2} (begin of shut down) = $(f_2 - S_{p,max})/W$	d_{t2}	K	0,00	0,00	
v_{t2} (begin of shut down) = $d_{t2} \cdot V$	v_{t2}	K/s	0,00	0,00	
$d_{t2'}$ (end of shut down) = $(f_2 - S_{p,min})/W$	$d_{t2'}$	K	51,57	70,60	
$v_{t2'}$ (end of shut down) = $dt_{2'} \cdot V$	$v_{t2'}$	K/s	-0,47	-0,64	

Annex D
(informative)

Physical properties of steels

D.1 General

Annex D may be used in conjunction with Clause 13, and EN 12952-4:2011, Annex B.

D.2 Symbols and abbreviations

The symbols are explained in the text and/or listed in EN 12952-1:2001, Table 4-1 and EN 12952-3:2011, Tables 13.2-1 and B.3.

D.3 Physical properties

The physical properties of steels are required for stress analysis calculations.

D.3.1 Density

The density ρ depends on the temperature. It may be calculated by

$$\rho_t = \frac{\rho_{20}}{\left(1+\beta_{20,t}\left(t-20\ °C\right)\right)^3} \ . \tag{D.3-1}$$

In this equation the "linear coefficient of thermal expansion from 20 °C to temperature" should be used. This is given by

$$\beta_{20,\mathrm{t}} = \frac{1}{l_{20}} \cdot \frac{l_\mathrm{t} - l_{20}}{t - 20\ °\mathrm{C}} \ . \tag{D.3-2}$$

where

l_t is the length of a specimen at temperature t.

For the calculation of the weight of a component the density ρ_{20} at 20 °C may be used.

D.3.2 Differential coefficient of linear thermal expansion

When the thermal stress caused by a temperature difference $\Delta t = t_2 - t_1$ should be calculated, the "differential coefficient of linear thermal expansion" $\beta_{\mathrm{diff},\ t^*}$ at temperature

$$t^* = 0{,}75 \max\ (t_1;\ t_2) + 0{,}25 \min\ (t_1;\ t_2) \tag{D.3-3}$$

may be used.

The relationship between $\beta_{20,\ \mathrm{t}}$ and $\beta_{\mathrm{diff},\ \mathrm{t}}$ is

$$\beta_{diff,t} = \beta_{20,t} + \frac{\partial \beta_{20,t}}{\partial t}\left(t - 20\ °C\right). \quad \text{(D.3-4)}$$

D.3.3 Heat capacity

The relationship between the "mean heat capacity from 20 °C to temperature" $c_{p,\,20,\,t}$ and the "differential heat capacity" $c_{p,\,diff,\,t}$ (similar to the coefficient of linear thermal expansion) is given by:

$$c_{p,diff,t} = c_{p,20,t} + \frac{\partial c_{p,20,t}}{\partial t}\left(t - 20\ °C\right). \quad \text{(D.3-5)}$$

D.3.4 Thermal diffusivity

The thermal diffusivity D_{th} is given by

$$D_{th} = \frac{\lambda_t}{\rho_t\, c_{p,diff,t}} \quad \text{(D.3-6)}$$

where λ_t is the temperature dependent thermal conductivity as given in D.4.

D.3.5 Poisson's ratio

The Poisson's ratio ν = 0,3 may be chosen independent of the temperature and the material.

D.4 Physical properties of steels

D.4.1 Data sheet (tables)

Table D.1 — Ferritic steels of steel group (St.) 1, 2.1, 4, 5.1 and 5.2 Density at 20 °C: 7 850 kg/m^3

Tempe-rature	Modulus of elasticity	Coefficient of linear thermal expansion $\beta_{xx,\,t}$		Thermal conductivity λ_t						Heat capacity $c_{p\,xx,\,t}$ (also for St. 6)	
t	E_t	mean from 20 °C to tempera-ture	diffe-rential	St.1 C-Mn	St.1 0,3 Mo	St.2.1	St.4	St.5.1	St.5.2	mean from 20 °C to tempera-ture	differ-ential
°C	kN/mm^2	10^{-6}/K	10^{-6}/K	W/(m K)	W/(m K)	W/(m K)	W/(m K)	W/(m K)	W/(m K)	J/(kg K)	J/(kg K)
20	212	—	11,30	54,7	49,3	39,7	46,3	44,6	37,0	—	461
100	206	11,9	12,47	53,8	48,3	40,9	47,1	43,7	37,4	479	496
200	198	12,6	13,71	50,5	46,2	41,0	46,2	42,2	37,2	499	533
300	191	13,1	14,69	47,0	43,7	39,9	44,1	40,1	36,5	517	568
400	183	13,7	15,41	43,5	41,0	38,2	41,3	38,0	35,0	536	611
500	174	14,1	15,88	40,0	38,8	36,1	38,6	35,8	33,2	558	677
600	165	14,4	16,09	36,5	35,3	33,8	35,8	33,5	31,0	587	778

Table D.2 — Ferritic steels of steel group 6 (9Cr1Mo and 12Cr1MoV)
Density at 20 °C: 7 760 kg/m³

Temperature t	Modulus of elasticity E_t	Coefficient of linear thermal expansion $\beta_{xx,t}$		Thermal conductivity λ_t		Heat capacity $c_{p\,xx,t}$ (only St. 6)	
		mean from 20 °C to temperature	differential	St. 6 9Cr1Mo	St. 6 12Cr1MoV	mean from 20 °C to temperature	differential
°C	kN/mm²	10^{-6}/K	10^{-6}/K	W/(m K)	W/(m K)	J/(kg K)	J/(kg K)
20	214	—	10,32	28,1	23,3	—	439
100	211	10,7	11,10	28,3	23,6	474	500
200	205	11,2	11,94	28,0	24,3	498	534
300	197	11,6	12,63	28,0	25,5	517	571
400	188	11,9	13,16	28,0	25,7	538	625
500	178	12,2	13,55	27,8	26,0	564	704
600	168	12,5	13,79	27,0	26,5	598	831

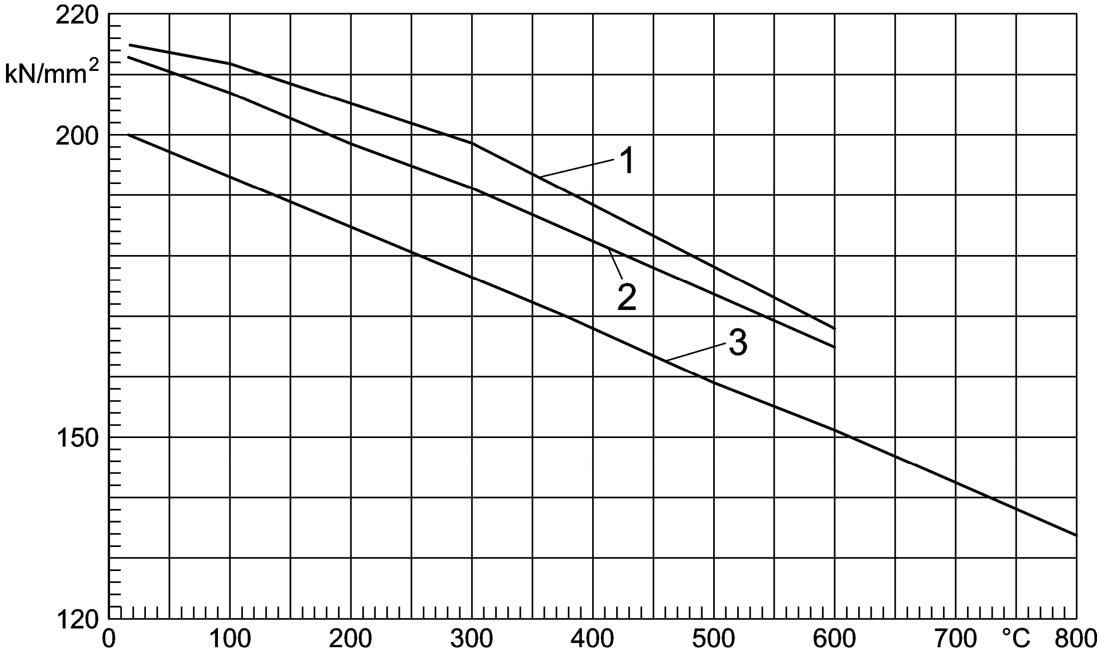
Table D.3 — Austenitic steels of steel group 8.2 and 215S15
Density at 20 °C: 7 900 kg/m³

Temperature t	Modulus of elasticity E_t	Coefficient of linear thermal expansion $\beta_{xx,t}$		Thermal conductivity λ_t		Heat capacity	
						$c_{p20,t}$	$c_{p\,diff,t}$
		mean from 20 °C to temperature	differential	St. 8.2	215S15	mean from 20 °C to temperature	differential
°C	kN/mm²	10^{-6}/K	10^{-6}/K	W/(m K)	W/(m K)	J/(kg K)	J/(kg K)
20	200	—	15,29	14,3	12,6	—	472
100	193	15,9	16,47	15,5	14,0	487	501
200	185	16,6	17,77	17,0	15,5	503	525
300	176	17,2	18,87	18,4	17,0	512	532
400	168	17,7	19,77	20,0	18,6	520	555
500	159	18,3	20,47	21,5	20,1	530	582
600	151	18,7	20,97	23,0	21,7	541	604
700	142	19,1	21,27	24,5	23,0	551	610
800	134	19,3	21,37	26,0	24,2	559	609

Origin of the data:

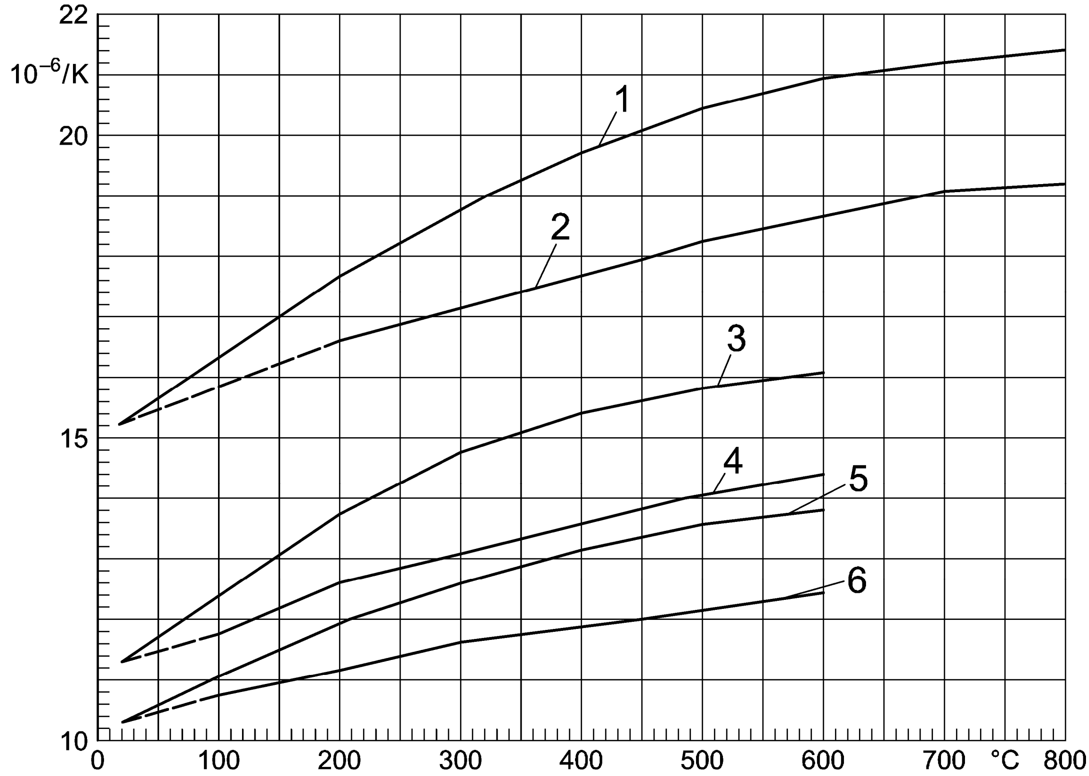
Heat capacities:	SEW 310 (see literature [5])
Modulus of elasticity, mean, thermal expansion, thermal conductivity:	BS 3059 (see literature [6])
Differential linear thermal expansion:	Best-fit-Polynome for $\beta_{20,\,t}$ and Equation (D.3-4)

D.4.2 Graphs



Key
1 steel group 1 to 5.2
2 steel group 6
3 steel group 8.2

Figure D.1 — Modulus of elasticity



Key

1 $\beta_{diff,t}$ steel group 8.2

2 $\beta_{20,t}$ steel group 8.2

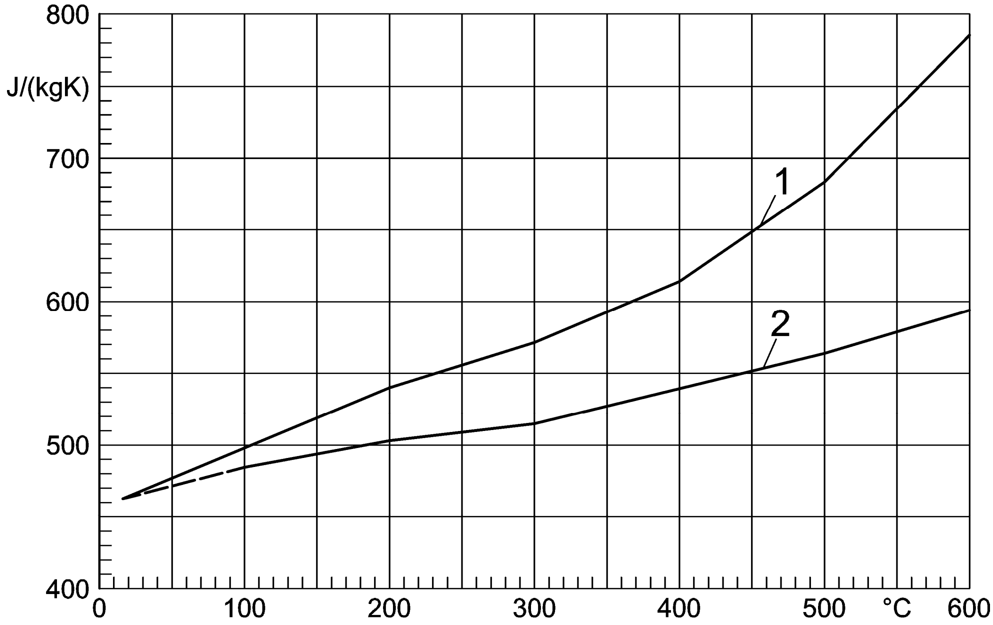
3 $\beta_{diff,t}$ steel group 1 to 5.2

4 $\beta_{20,t}$ steel group 1 to 5.2

5 $\beta_{diff,t}$ steel group 6

6 $\beta_{20,t}$ steel group 6

Figure D.2 — Coefficient of linear thermal expansion

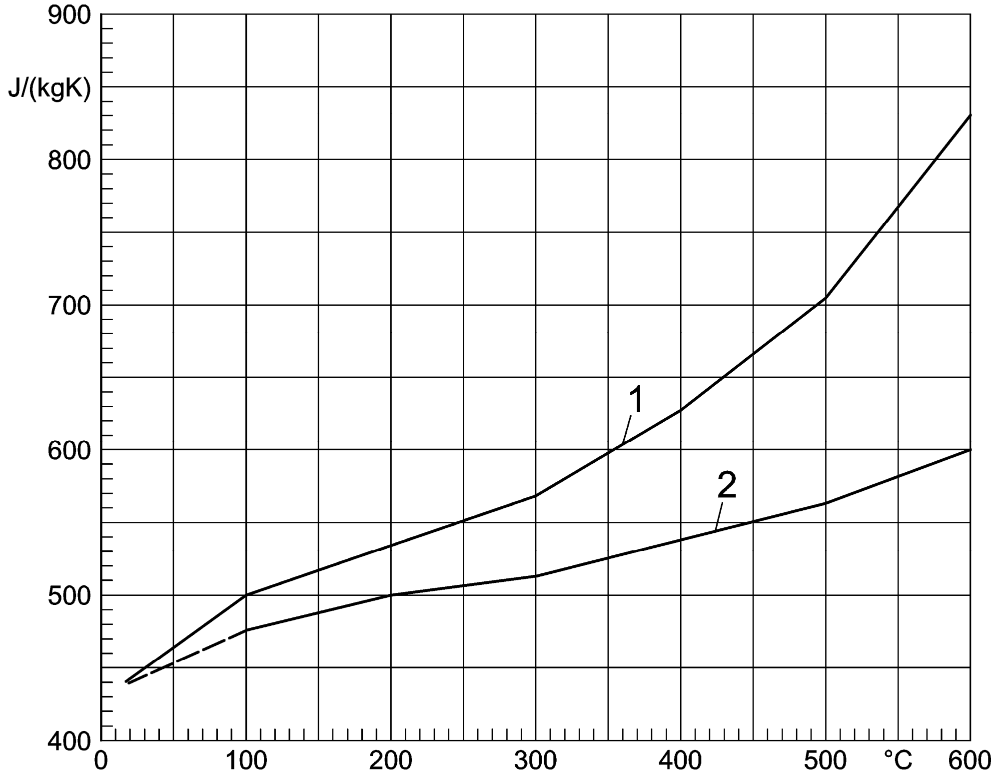


Key

1 $c_{\mathrm{p\ diff,t}}$

2 $c_{\mathrm{p20,t}}$

Figure D.3a — Heat capacity for steel group 1 to 6 (9Cr1Mo)

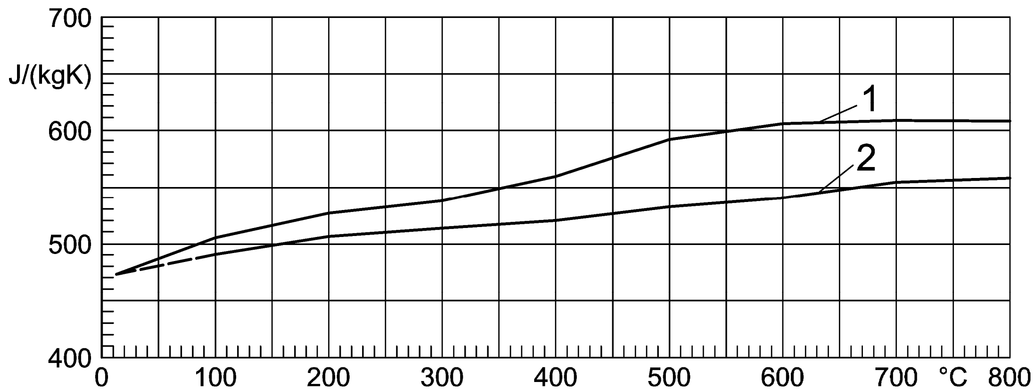


Key

1 $c_{\text{p diff, t}}$

2 $c_{\text{p20, t}}$

Figure D.3b — Heat capacity for steel group 6 (12Cr1MoV)

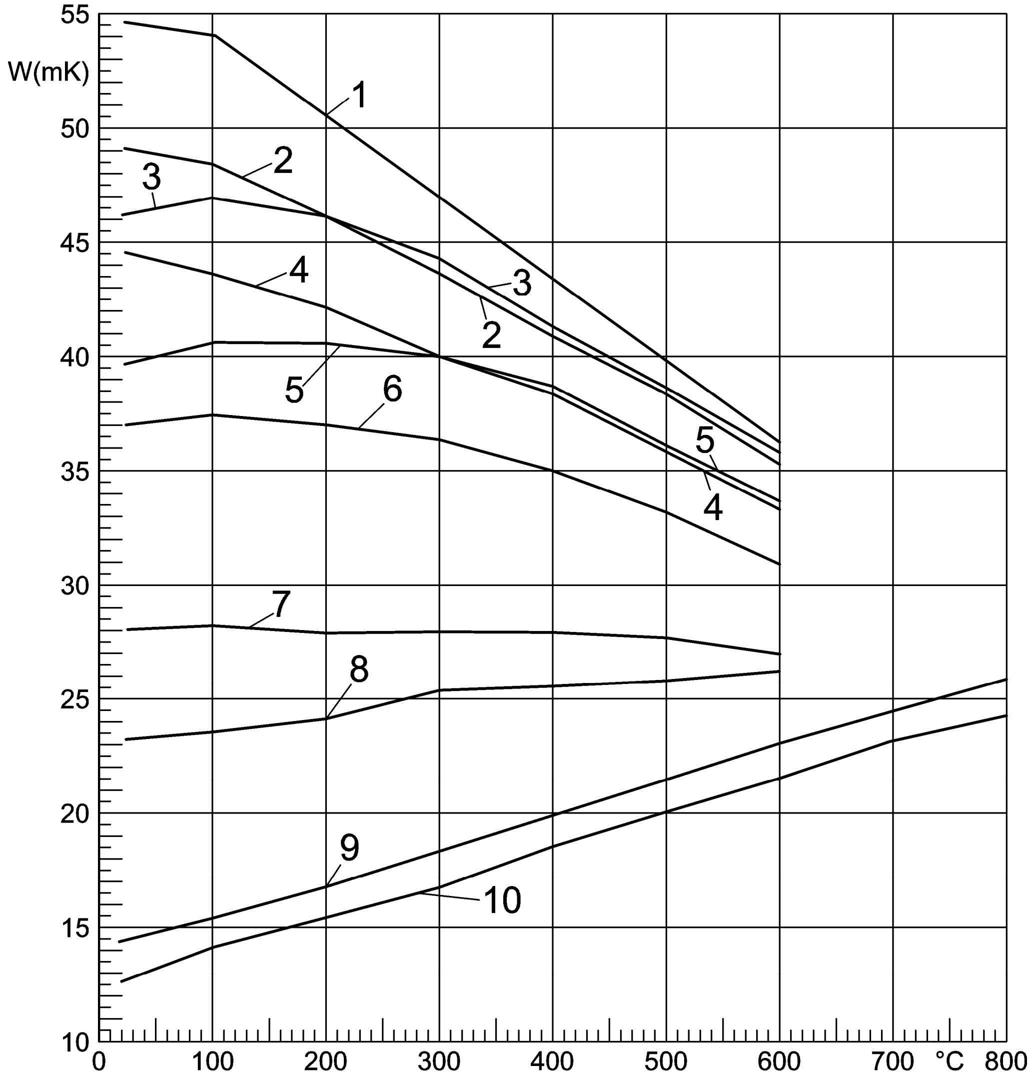


Key

1 $c_{\text{p diff, t}}$

2 $c_{\text{p20, t}}$

Figure D.3c — Heat capacity for steel group 8.2 and 215S15



Key

1	steel group 1 C-Mn	6 steel group 5.2
2	steel group 1 0,3 Mo	7 steel group 6 (9Cr1Mo)
3	steel group 4	8 steel group 6 (12Cr1MoV)
4	steel group 5.1	9 steel group 8.2
5	steel group 2.1	10 215S15

Figure D.4 — Thermal conductivity

D.4.3 Polynomials

The physical properties may also be calculated using Equation (D.4-1) as well. The temperature t is to be used in °C. The calculated property Z will result in units as given in the tables below.

$$Z = c_0 + c_1 t + c_2 t^2 + c_3 t^3 + \ldots \quad \text{(D.4-1)}$$

Table D.4 — Polynomcoefficients for modulus of elasticity

Modulus of Elasticity E_t in kN/mm^2	Coefficients for polynomials for temperature t in °C			Max. deviations to Tables in D.4.1
Steel group	c_0	c_1	c_2	%
1 to 5.2	213,16	-6,91 E-2	-1,824 E-5	0,4
6	215,44	-4,28 E-2	-6,185 E-5	0,3
8.2, 215S15	201,66	-8,48 E-2	0	0,3

Table D.5 — Polynomcoefficients for linear thermal expansion

Linear thermal expansion $\beta_{xx,\,t}$ in µm/(m K) (= 10^{-6} K^{-1})		Coefficients for polynomials for temperature t in °C			Max. deviations to Tables in D.4.1
Steel group		c_0	c_1	c_2	%
1 to 5.2	$\beta_{20,\,t}$	11,14	8,03 E-3	-4,29 E-6	0,5
	$\beta_{diff,\,t}$	10,98	1,623 E-2	-1,287 E-5	0 (per Def.)
6	$\beta_{20,\,t}$	10,22	5,26 E-3	-2,5 E-6	0,3
	$\beta_{diff,\,t}$	10,11	1,062 E-2	-7,5 E-6	0 (per Def.)
8.2, 215S15	$\beta_{20,\,t}$	15,13	7,93 E-3	-3,33 E-6	0,4
	$\beta_{diff,\,t}$	14,97	1,599 E-2	-9,99 E-6	0 (per Def.)

Table D.6 — Polynomcoefficients for linear thermal conductivity

Thermal conductivity λ_t in W/(m K)	Coefficients for polynomials for temperature t in °C			Max. deviations to Tables in D.4.1
Steel group	c_0	c_1	c_2	%
1 C-Mn	55,72	-2,464 E-2	-1,298 E-5	1,3
1 0.3Mo	49,83	-1,613 E-2	-1,372 E-5	0,5
2.1	39,85	1,111 E-2	-3,611 E-5	1,0
4	46,85	7,2 E-4	-3,305 E-5	1,2
5.1	45,0	-1,287 E-2	-1,075 E-5	0,5
5.2	39,97	6,40 E-3	-2,749 E-5	0,4
6 (9Cr1Mo)	28,05	1,85 E-3	-5,58 E-6	0,8
6 (12Cr1MoV)	22,97	8,73 E-3	-4,82 E-6	1,4
8.2	13,98	1,502 E-2	0	0,5
215S15	12,48	1,501 E-2	0	1,2

Table D.7— Polynomcoefficients for heat capacity

Heat capacity $c_{\text{p xx, t}}$ in J/(kg K)		Coefficients for polynomials for temperature t in °C					Max. deviations to Tables in D.4.1
Steel group		c_0	c_1	c_2	c_3	c_4	%
1 to 6 (9Cr1Mo)	$\beta_{20,t}$	454,93	0,28139	-3,8815 E-4	4,7542 E-7	0	0,2
	$\beta_{\text{diff, t}}$	449,30	0,57830	-1,1930 E-3	1,9017 E-6	0	0,3
6 (12Cr1MoV)	$\beta_{20,t}$	433,33	0,4334	-7,4702 E-4	8,0289 E-7	0	0,9
	$\beta_{\text{diff, t}}$	424,66	0,89672	-2,2892 E-3	3,2116 E-6	0	1,1
8.2, 215S15	$\beta_{20, t}$	467,77	0,2490	-5,5393 E-4	8,3266 E-7	-4,3916 E-10	0,4
	$\beta_{\text{diff, t}}$	462,69	0,52026	-1,7117 E-3	3,3658 E-6	-2,1958 E-9	1,2

Range of validity of the polynominals:

Ferritic steels (steel group 1 to 6): temperatures from 0 °C to 600 °C,

Austenitic steels (steel group 8.2 and 215S15): temperatures from 0 °C to 800 °C

Origin of the coefficients:

Best-fit-polynomials for the data in the tables in D.4

Annex E
(informative)

Significant technical changes between this European Standard and the previous edition

Clause/paragraph/table/figure	Change
General	The standard was completely revised in terms of: — correction of formula and calculations; — correction of symbols; — correction of figures; — modification of concept for pressure testing.
2 / Normative references	References updated.
6.3 / Design stress	Subclause was completely modified.
NOTE The technical changes referred include the significant technical changes from the EN revised but is not an exhaustive list of all modifications from the previous version.	

Annex ZA
(informative)

Clauses of this European Standard addressing essential safety requirements of the Pressure Equipment Directive 97/23/EC

This European standard has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association to provide a means of conforming to Essential Requirements of the New Approach Directive 97/23/EC.

Once this standard is cited in the Official Journal of the European Union under that Directive and has been implemented as a national standard in at least one Member State, compliance with the clauses of this standard given in Table ZA.1 confers, within the limits of the scope of this standard, a presumption of conformity with the corresponding Essential Requirements of that Directive and associated EFTA regulations.

Table ZA.1 — Comparison between the Pressure Equipment Directive 97/23/EC and EN 12952-3 with respect to design and calculation for pressure parts for water-tube boilers

EN 12952-3 harmonized clauses	Content	Pressure Equipment Directive 97/23/EC Annex I
5	Proper design using relevant factors	2.1 -1st paragraph
6, 8.4	Safety coefficients and margins	2.1 -2nd paragraph
5.7.1, 5.7.3	Internal/external pressure	2.2.1 -1st indent
6.1, 6.2	Ambient and operating temperatures	-2nd indent
5.7.2(a)	Static pressure	-3rd indent
7.3	Mass of contents	-3rd indent
5.3	Wind and earthquake loading	-4th indent
5.3, 5.9, 7.4, 11.5	Reaction forces and moments from attachments, etc.	-5th indent
5.8.1, 5.8.2, 7.1.2, 10.2.2.1, 13, Annex A-A.3, Annex B	Corrosion, erosion, fatigue, etc.	-6th indent
5.2, 5.3, 6.2, 7.3, 7.4, 7.5	Simultaneous occurrence of loading	2.2.1 -2nd paragraph
7 to 11, Annex A, Annex B	**Design for adequate strength shall be based on:** Design by calculation methods	2.2.2 -1st indent
12.1	Design by experimental methods	2.2.2 -2nd indent
6.1, 6.2, 6.3, 8.4	Failure modes and safety factors	2.2.3a -1st paragraph
6.1, 6.2, 6.3, 8.4	Safety margins	2.2.3a -2nd paragraph
7 to 11, 13, Annex A, Annex B	**Applicable methods** Design by formula	2.2.3a -3rd paragraph -1st indent
5.4	Design by analysis	-2nd indent
5.7	Calculation pressure	2.2.3b -1st indent
6.1, 6.2	Calculation temperature and margins	2.2.3b -2nd indent
5.2	Combinations of pressure and temperature	2.2.3b -3rd indent
13, Annex B	Maximum stresses and stress concentrations	2.2.3b -4th indent

Table ZA.1 *(continued)*

EN 12952-3 harmonized clauses	Content	Pressure Equipment Directive 97/23/EC Annex I
6.3	Yield/proof strength	2.2.3b -5th indent -1st sub-indent
6.3	Tensile strength	-2nd sub-indent
6.3	Creep strength	-3rd sub-indent
13, Annex B	Fatigue	-4th sub-indent
5.6.1 (g), 11.2.4	Joint factors	2.2.3b -6th indent
5.6, 5.8, 6.3, 13, Annex B	Foreseeable degradation mechanism (e.g. creep, fatigue, corrosion)	-7th indent
5.2, 7.4, 7.5, 11.4, 11.5	Stability aspects	2.2.3c
12.1.2, 12.1.3	Pressure strength test	2.2.4a
5.6.2	Design for manual examination	2.4a
5.6.2	Means of internal manual examination	2.4b
5.6.2	Physical access is too small	2.4c -1st indent
5.6.2	Substances contained not harmful to press. equip.	2.4c -3rd indent
5.6.3	To avoid harmful effects i.e. water hammer	2.5 -1st indent
5.6.3	To permit cleaning, inspection, etc.	2.5 -2nd indent
5.8	Corrosion or other chemical attack	2.6
5.8.5	Additional thickness, pads, liners, etc.	2.7 -1st indent
5.7.4.3	Proof test	3.2.2
5	Calculation and design	5
6.3	Ferritic steels — minimum of 2/3 of $R_{p0,2\ t}$ or 5/12 of $R_{m,\ 20}$	7.1.2 -1st indent
6.3	Austenitic steels — 2/3 of $R_{p1,0\ t}$ where elongation exceeds 30 %	7.1.2 -2nd indent -1st sub-indent
6.3	Austenitic steels — 5/6 of $R_{p1,0\ t}$ or 1/3 of $R_{m\ t}$ where elongation exceeds 35 %	7.1.2 -2nd indent -2nd sub-indent
6.3	Non-alloy or low alloy cast steel — minimum of 10/19 of $R_{p0,2\ t}$ 1/3 of $R_{m\ 20}$	7.1.2 -3rd indent
11.2.4	Joint coefficients 1,00 for joints subject to 100% NDE 0,85 for joints subject to random NDE	7.2 -1st sub-indent -1st indent -2nd indent
5.7.4.3	Hydrostatic test pressure	7.4

WARNING —Other requirements and other EU Directives may be applicable to the product(s) falling within the scope of this standard.

Bibliography

[1] BS 3799:1974, *Specification for steel pipe fittings, screwed and socket-welding for the petroleum industry*

[2] Schwaigerer, S.: *Festigkeitsberechnung von Bauteilen des Dampfkessel-, Behälter- und Rohrleitungsbaus*; 2nd revised edition, Springer, Berlin/Heidelberg/New York (1970)

[3] Makkinejad, N.: *Festigkeitsberechnung von Bauteilen des Dampfkessel-, Behälter- und Rohrleitungsbaus*; 2nd revised edition, Springer, Berlin/Heidelberg/New York (1970)

[4] Appendix to AD-Merkblatt S 2, Edition March 1990

[5] SEW 310 (Stahl-Eisen-Werkstoffblätter des Vereins Deutscher Eisenhüttenwerke) "*Physikalische Eigenschaften von Stählen*", 1. Edition August 1992, Verlag Stahleisen GmbH, Postfach 105164; 40042 Düsseldorf

[6] British Standard BS 3059, Part 2: "*Steel Boiler and Superheater Tubes*." Edition 1990

[7] Directive 97/23/EC of the European Parliament and of the Council of 29 May 1997 on the approximation of the laws of the Member States concerning pressure equipment; OJEC L181

[8] Pressure Equipment Directive (PED): *Guidelines*

EN 1591-1, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*

EN 12952-11:2007, *Water-tube boilers and auxiliary installations — Part 11: Requirements for limiting devices of the boiler and accessories*

EN ISO 4287, *Geometrical Product Specifications (GPS) — Surface texture: Profile method — Terms, definitions and surface texture parameters (ISO 4287:1997 + Cor 1:1998 + Cor 2:2005 + Amd 1:2009)*

ISO 7-1, *Pipe threads where pressure-tight joints are made on the threads — Part 1: Dimensions, tolerances and designation*

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