

BS EN 12952-3:2011



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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 It supersedes BS EN 12952-3:2001 which is withdrawn.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.

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**Water-tube boilers and auxiliary installations - Part 3: Design
and calculation for pressure parts of the boiler**

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of an identical text or by endorsement, at the latest by June 2012, and conflicting national standards shall be withdrawn at the latest by June 2012.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:

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

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;

$\alpha_m = 4$ 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;

$X = 550 \text{ N/mm}^2$;

$Z = 2 \text{ K mm}^4/(\text{Ns})$ for ferritic steels, or

$Z = 1 \text{ K mm}^4/(\text{Ns})$ for austenitic and martensitic steels, or

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

where exact values D_{th} , β_t , E , ν 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 $R_{p0,2 t_c}$ 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 $R_{p0,2 t_c}$ 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:

$$p_t = 1,43 \cdot PS \quad (5.7-1)$$

or

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

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 $R_{p0,2\ 20}/R_{p0,2\ t_c}$ or in the case of austenitic steels, if its elongation after rupture exceeds 30 %, the ratio $R_{p1,0\ 20}/R_{p1,0\ t_c}$ 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 wastage¹⁾

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 be 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 thermostats²⁾, 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

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

where

n is the number of the rows;

P_0 is the longitudinal pitch with $\Phi = 0^\circ$;

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

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

where

P_0 is the longitudinal pitch with $\Phi = 0^\circ$;

P_{90} is the circumferential tube pitch on the external surface with $\Phi = 90^\circ$;

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 $> 950^\circ\text{C}$.

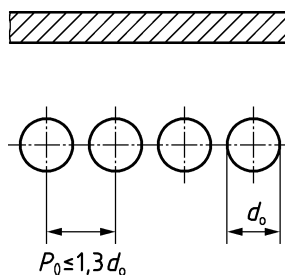


Figure 6.1-1 — Components protected by one row of tubes

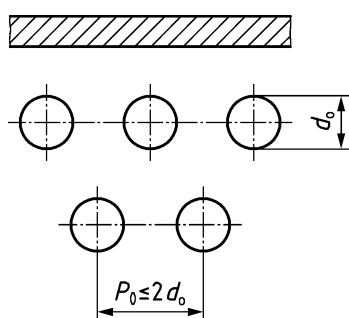


Figure 6.1-2 — Components protected by two rows of tubes

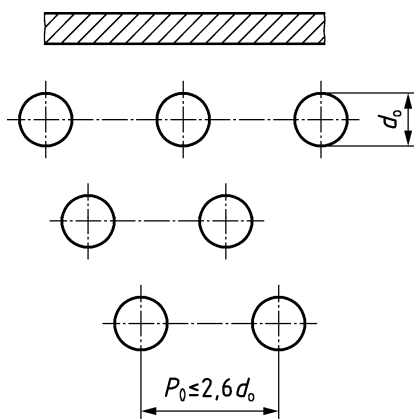
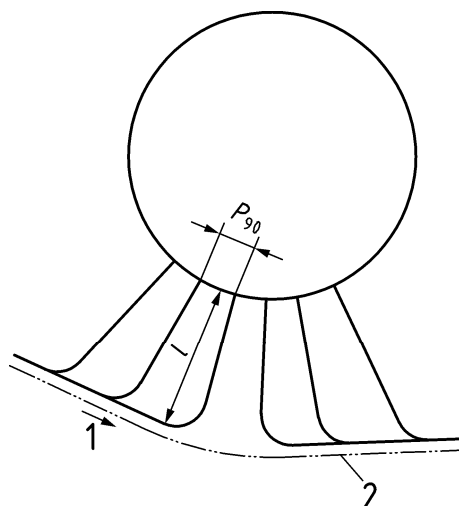


Figure 6.1-3 — Components protected by three rows of tubes



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/m² 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

$$f = \frac{K}{S} \quad (6.3-1)$$

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

$$f_t = \frac{K_t}{S_t} \quad (6.3-2)$$

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

$$f = \min \left\{ \frac{R_{m20}}{2,4}; \frac{R_{eHtc}}{1,5} \text{ and/or } \frac{R_{p0,2tc}}{1,5}; \frac{R_{mTtc}}{1,25} \right\} \quad (6.3-3)$$

where

R_{m20} is the tensile strength at room temperature (20 °C);

R_{eHtc} is the upper yield strength at calculation temperature t_c ;

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

$R_{mTtc}^{3,4}$ 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

$$f = \min \left\{ \frac{R_{p0,2tc}}{1,5}; \frac{R_{mTtc}}{1,25} \right\} \quad (6.3-4)$$

where

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

$R_{mTtc}^{3,4}$ is the creep rupture strength for the specified lifetime T at calculation temperature t_c .

- for steels with a specified minimum transverse elongation $A \geq 30\%$, 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.

$$f = \min \left\{ \frac{R_{p1,0 \text{ } t_c}}{1,5}; \frac{R_{mT \text{ } t_c}}{1,25} \right\} \quad (6.3-5)$$

where

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

$R_{mT \text{ } t_c}^{5, 6}$ is the creep rupture strength for the specified lifetime T at calculation temperature t_c .

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

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

where

$R_{m \text{ } t_c}$ is the tensile strength at calculation temperature t_c ;

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

$R_{mT \text{ } t_c}^{5, 6}$ 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

$$f = \min \left\{ \frac{R_{m20}}{3,2}; \frac{R_{p0,2 \text{ } t_c}}{2,0}; \frac{R_{mT \text{ } t_c}}{2,0} \right\} \quad (6.3-7)$$

where

R_{m20} is the tensile strength at room temperature (20 °C);

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

$R_{mT \text{ } t_c}^{7,8}$ 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

$$f = \min \left\{ \frac{R_{m20}}{4,8}; \frac{R_{p0,2 \text{ } t_c}}{3,0} \right\} \quad (6.3-8)$$

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

$$f = \min \left\{ \frac{R_{m20}}{5,8}; \frac{R_{p0,2 \text{ } t_c}}{4,0} \right\} \quad (6.3-9)$$

where

R_{m20} is the tensile strength at room temperature (20 °C);

$R_{p0,2 \text{ } t_c}$ 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

$$f_t = \frac{R_{p0,2\ 20}}{S_t} \quad (6.3-10)$$

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 $A \geq 30\%$, the design stress f_t shall be

$$f_t = \frac{R_{p1,0\ 20}}{1,05} \quad (6.3-11)$$

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 $A \geq 30\%$.

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

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

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:

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

or

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

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

- ν is the minimum efficiency ν_b for isolated or ν_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 stress between holes on a diagonal line shall be calculated in accordance with 7.3.4.

7.3.2 Stress from longitudinal loads

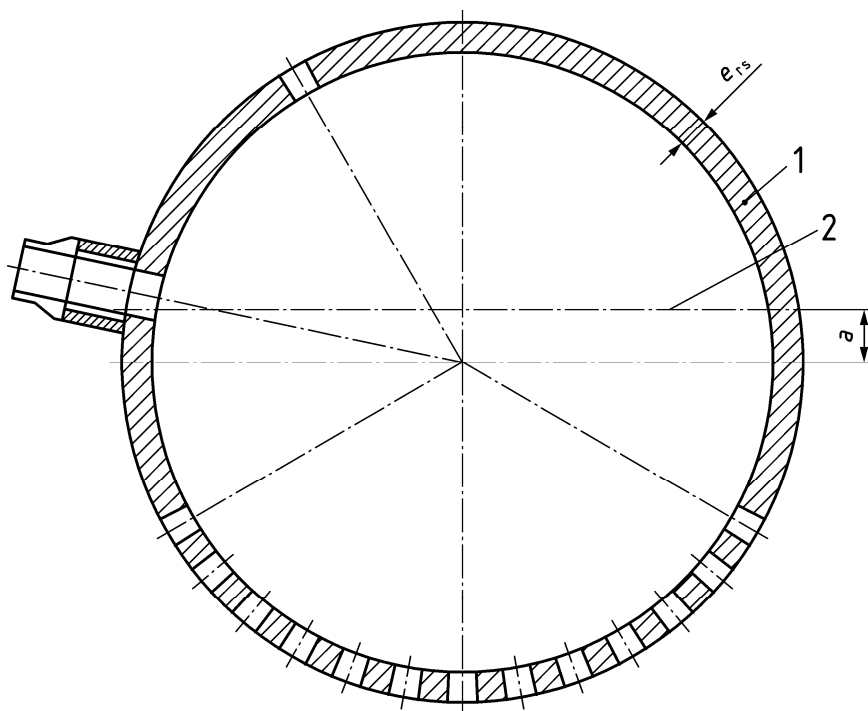
The longitudinal stress f_d averaged over the shell cross-section, caused by internal pressure and the longitudinal component of externally applied loads W , shall be calculated as follows:

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

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:

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



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 .

$$M_R = M_E + M_W \quad (7.3-3)$$

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

$$M_E = \frac{p_c d_{is}^2 \pi a}{4} \quad (7.3-4)$$

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:

$$f_M = \frac{M_R Y}{I_A} \quad (7.3-5)$$

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:

$$f_d + f_M + \frac{p_c}{2} \leq f_s \quad (7.3-6)$$

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:

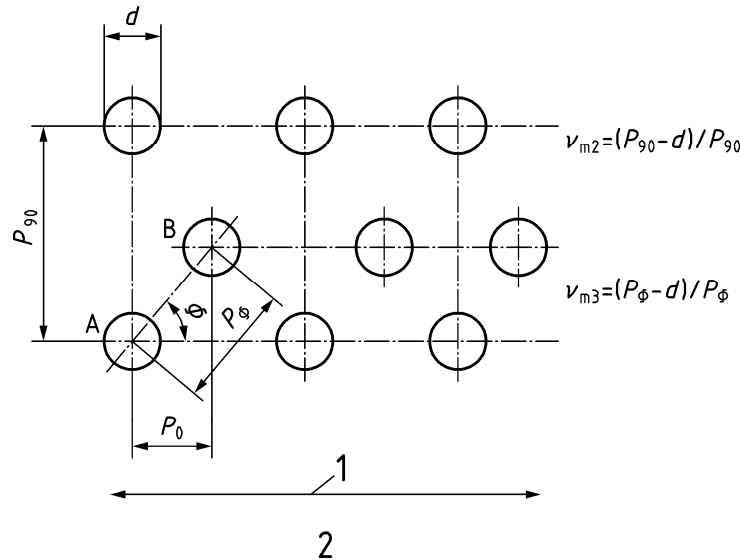
Let the circumferential stress f_1 on cross sectional area of longitudinal section be:

$$f_1 = \frac{p_c d_{is}}{2e_{ts}} \quad (7.3-7)$$

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

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

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:

$$v_{m3} = 1 - \frac{d}{P_{\Phi}} \quad (7.3-9)$$

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

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

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

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

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

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

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

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

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

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

$$f_3 + \frac{p_c}{2} \leq f_s \quad (7.3-15)$$

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:

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

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 tubes⁹⁾

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:

- 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 $\leq 1,5$ 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 $\leq 30^\circ$ (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 $\leq 1,5$ 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 surface¹⁰⁾. 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 ν_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.

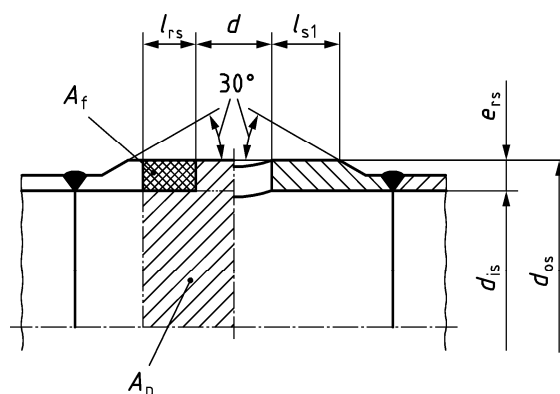


Figure 8.1-1 — Reinforcement by increased wall thickness of the main body with opening

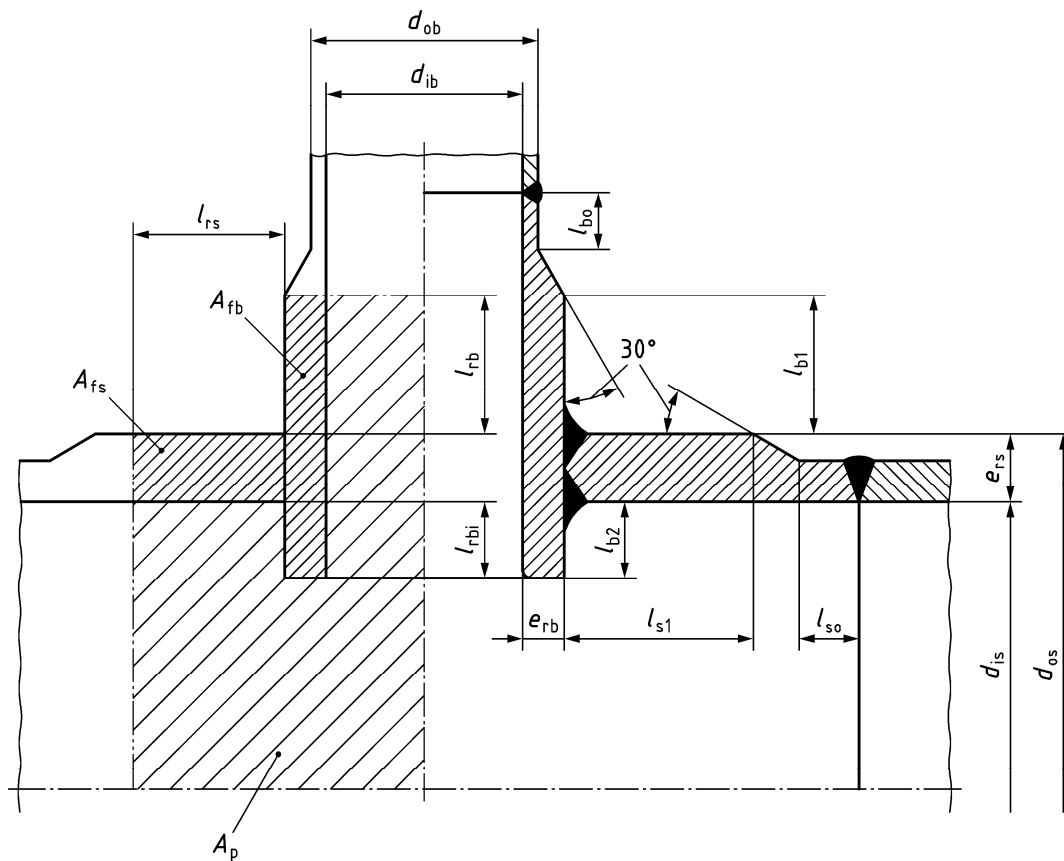
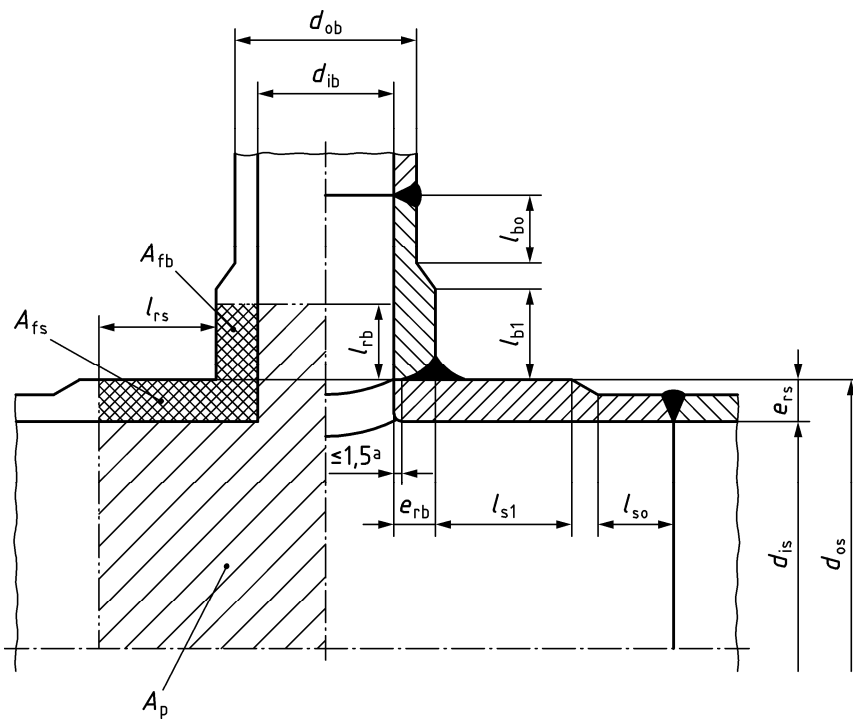


Figure 8.1-2 — Reinforcement by set through and full penetration welded branch



Key
a residual gap

Figure 8.1-3 — Reinforcement by welded on branch

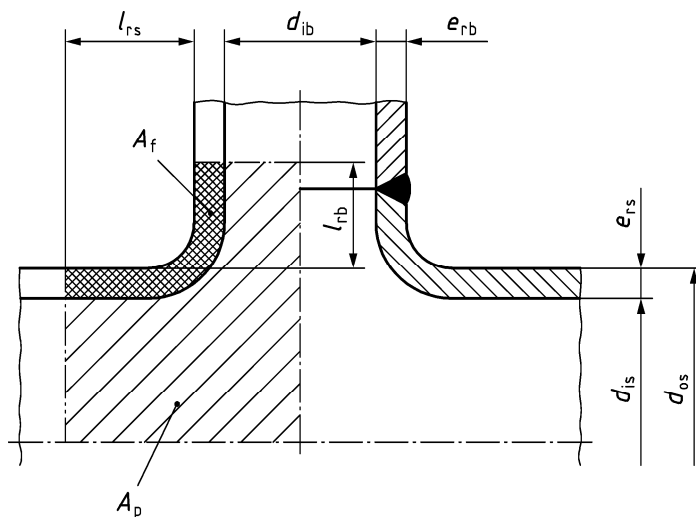


Figure 8.1-4 — Reinforcement by branch welded to extruded main body

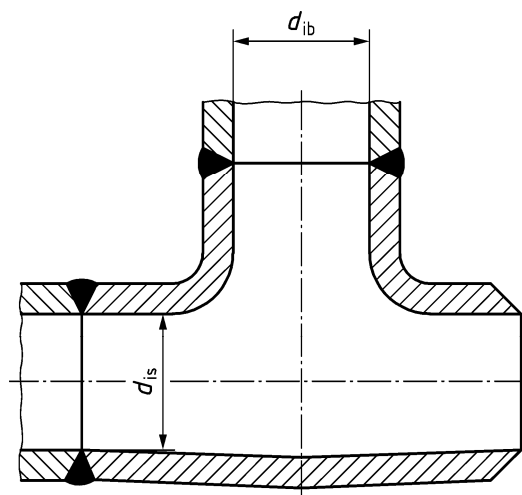


Figure 8.1-5 — Die forged branch

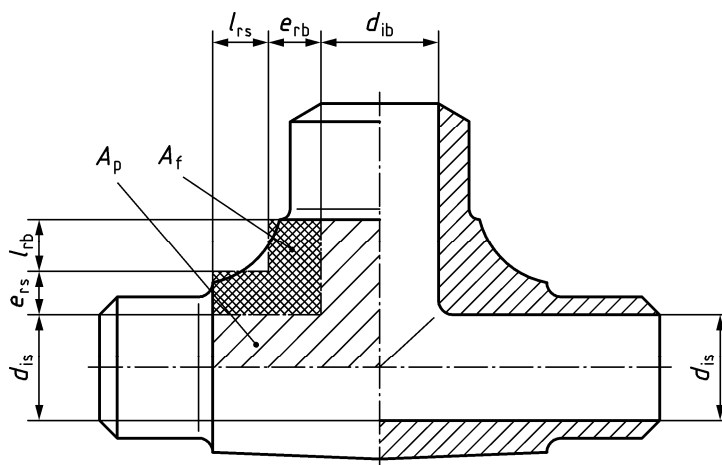


Figure 8.1-6 — Branch forged from solid material, subsequently bored and turned

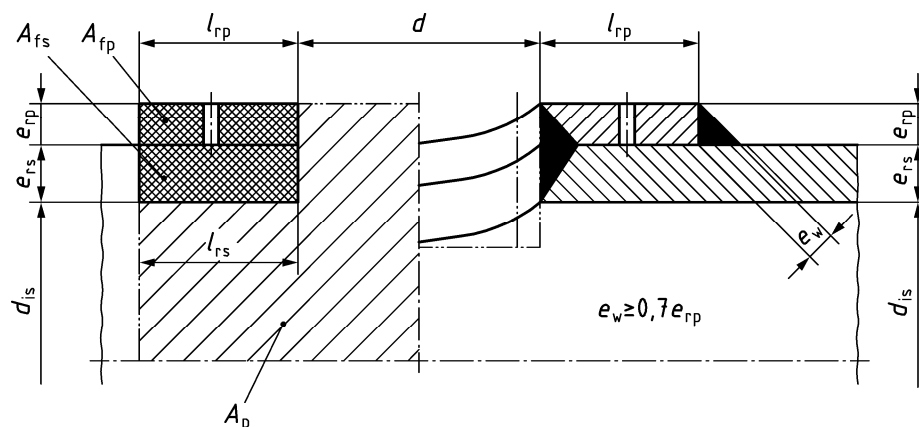


Figure 8.1-7 — Opening with reinforcing pad (allowable for $t_c \leq 250\text{ }^{\circ}\text{C}$)

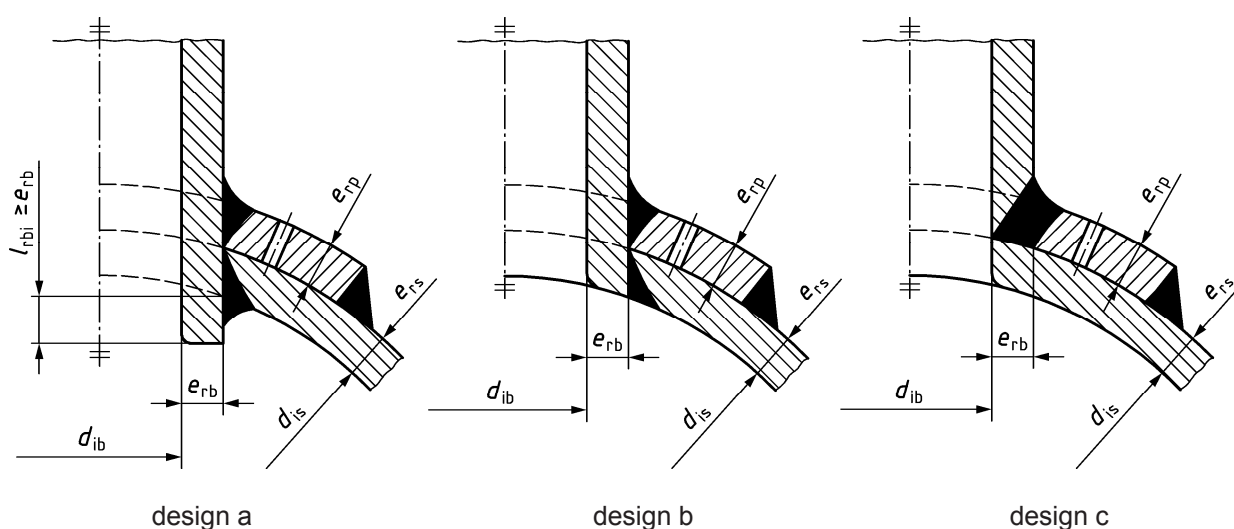


Figure 8.1-8 — Opening with reinforcing pad and full penetration set-through and set on welded branch (welds shown for cylindrical shells accessible from the inside)

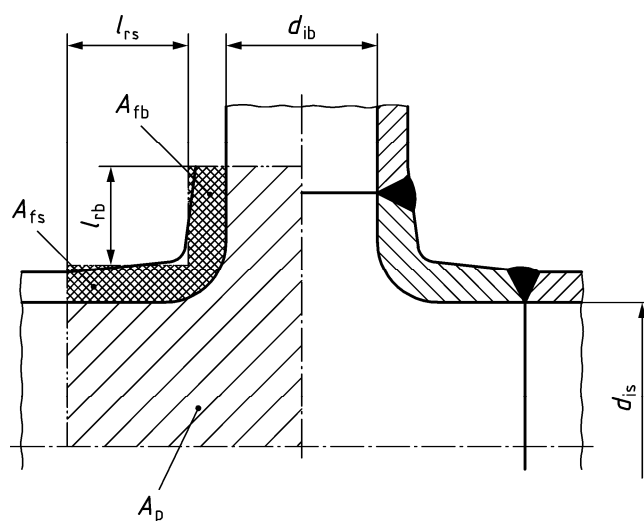


Figure 8.1-9 — Branch with conical transitions and fillets

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

$$l_{rs} = \min \left\{ \sqrt{(d_{is} + e_{rs}) e_{rs}} \right. \\ \left. l_{s1} \right\} \quad (8.1-1)$$

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

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

$$l_{rb} = \min \left\{ \sqrt{(d_{ib} + e_{rb}) e_{rb}} \right. \\ \left. l_{b1} \right\} \quad \text{for external projection;} \quad (8.1-2)$$

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

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:

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

For openings without a branch $e_{rb} = 0$ and $\psi = 90^\circ$.

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 ≤ 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 $\frac{1}{4}$ 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

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

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

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

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:

$$d \leq 0,14 l_{rs} \quad (8.2-3)$$

and

$$e_{rs} \leq 0,1 d_{os} \quad (8.2-4)$$

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

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

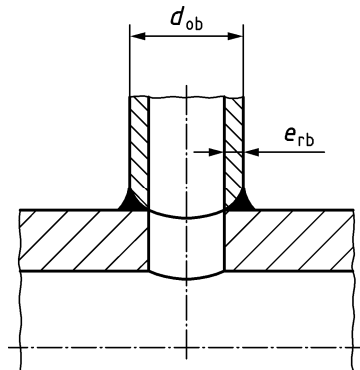


Figure 8.2-1 — Tube attached to main body

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.

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

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} .

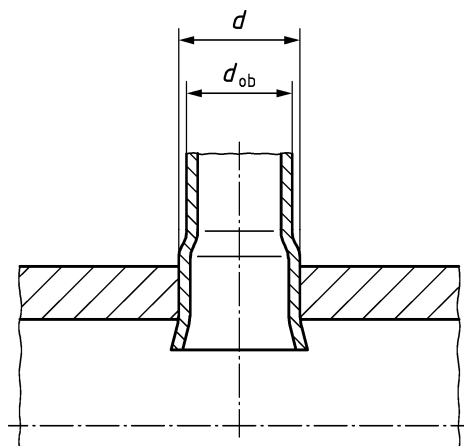
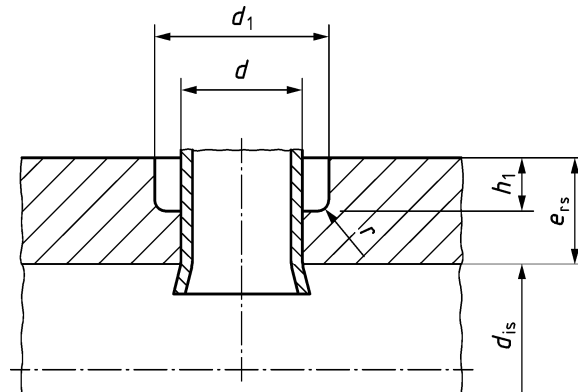


Figure 8.2-2 — Expanded tube

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

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



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

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

and

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

- 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.

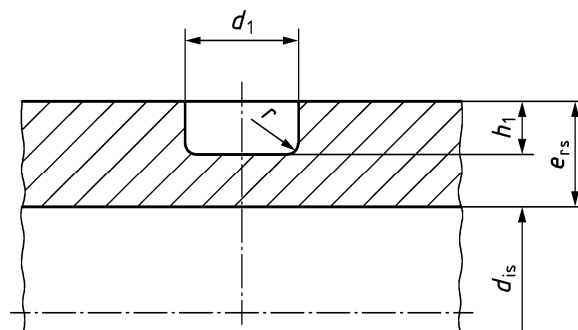


Figure 8.2-4 — Partial penetrating hole

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

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

Thus the allowable diameter of the hole is:

$$d_1 = \frac{2 l_{rs} e_{rs}}{h_1} \left[\frac{2 e_{rs}}{d_{is}} \left(\frac{f_s}{p_c} - \frac{1}{2} \right) - 1 \right] \quad (8.2-11)$$

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:

$$v_m = \frac{2(P_\phi - d_{ob})}{(1 + \cos^2 \Phi) P_\phi} \leq 1 \quad (8.2-12)$$

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

$$v_m = \frac{P_0 - d_{ob}}{P_0} \quad (8.2-13)$$

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

$$d_{ob} = \frac{d_{ob1} + d_{ob2}}{2} \quad (8.2-14)$$

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 obround¹¹⁾ 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:

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

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.

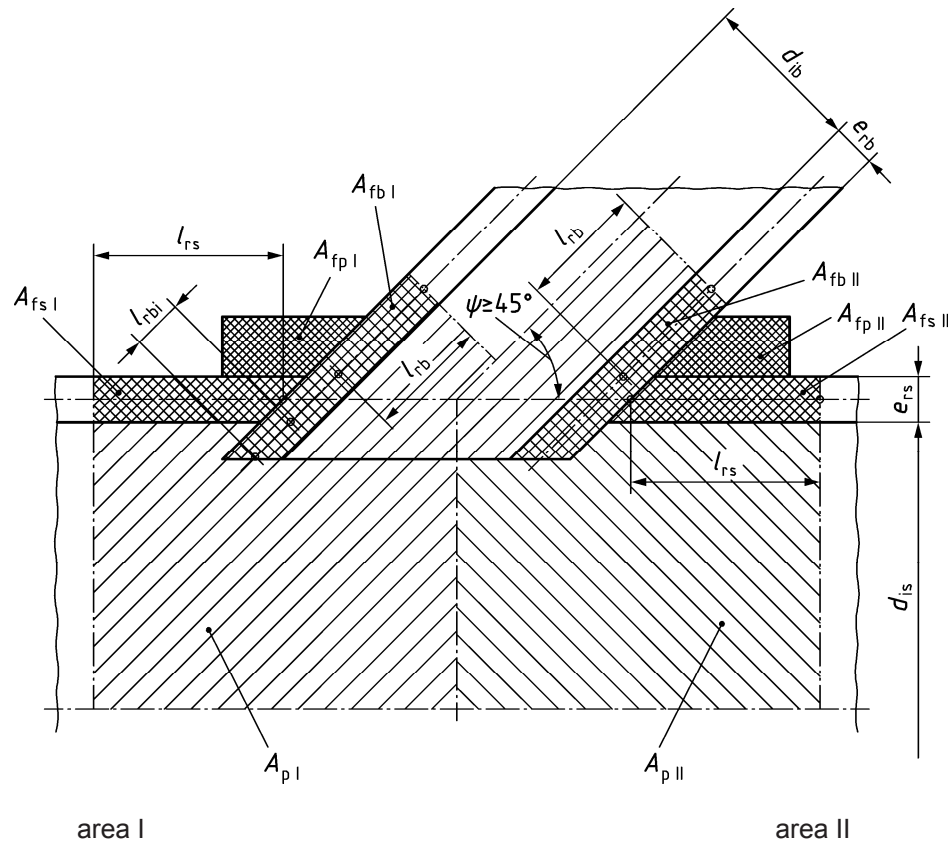
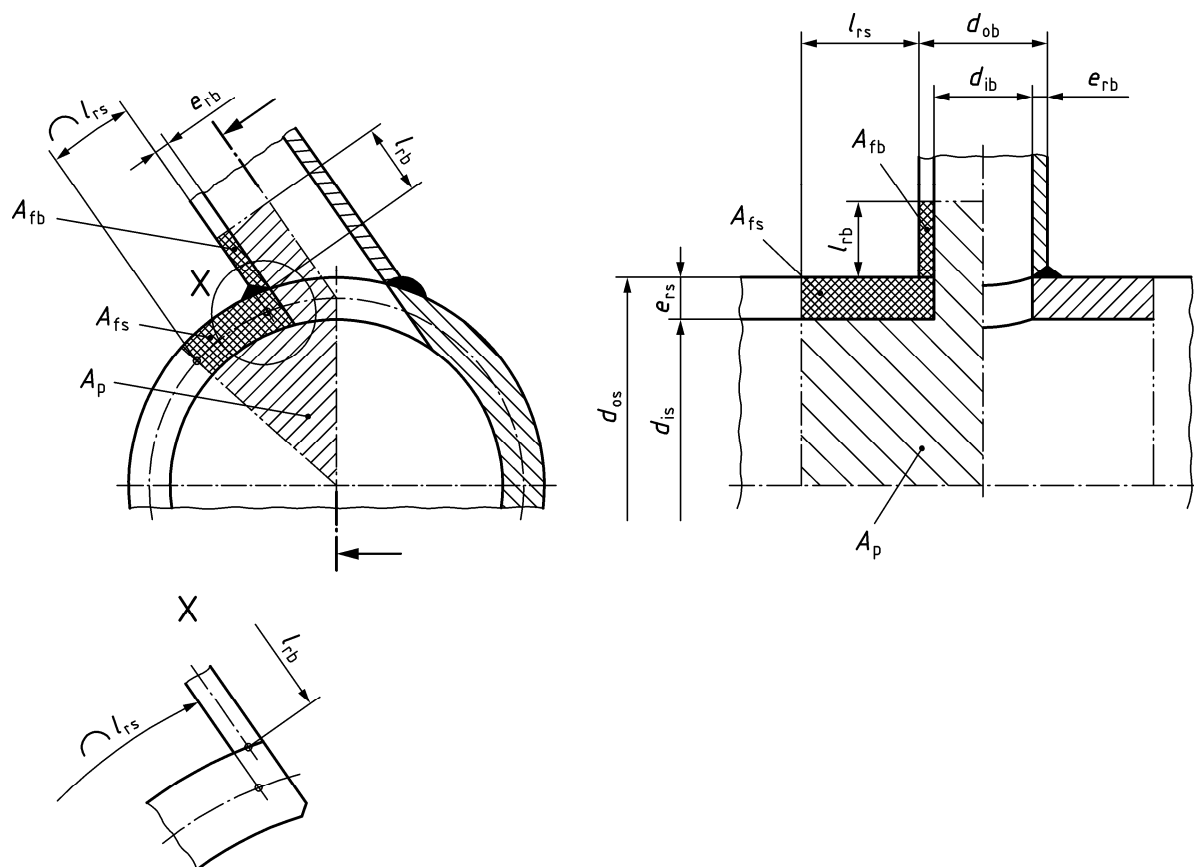
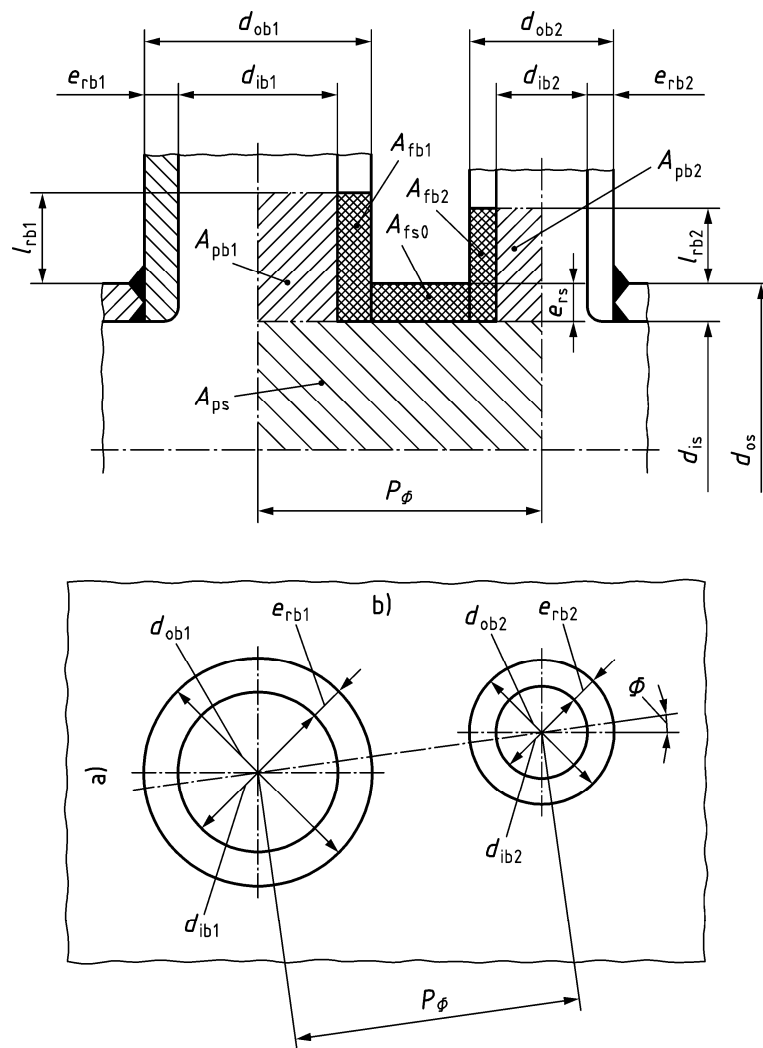


Figure 8.3-1 — Load diagram for cylindrical shell with oblique branch and reinforcing pad



NOTE Hatching and weld details in X omitted for clarity.

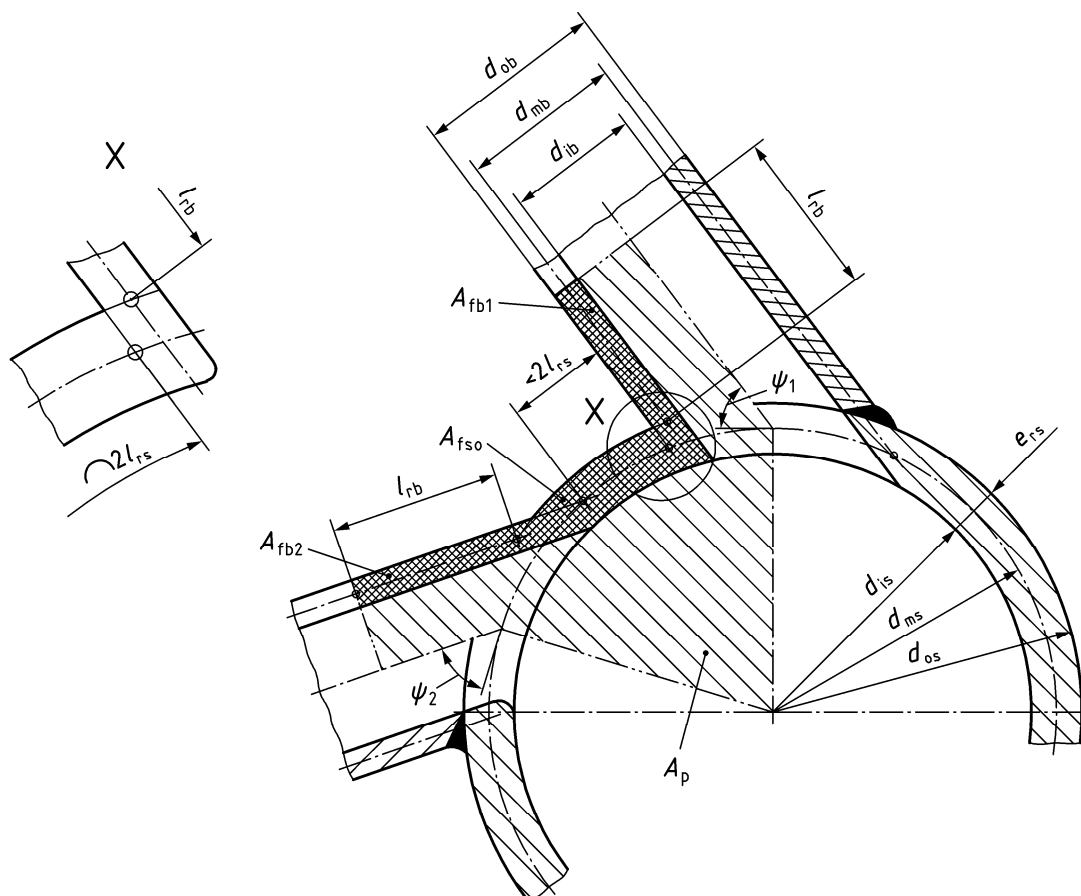
Figure 8.3-2 — Load diagram for cylindrical shell with non-radial branch



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 $\psi_1 \geq 45^\circ$,
 $\psi_2 \geq 45^\circ$,
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

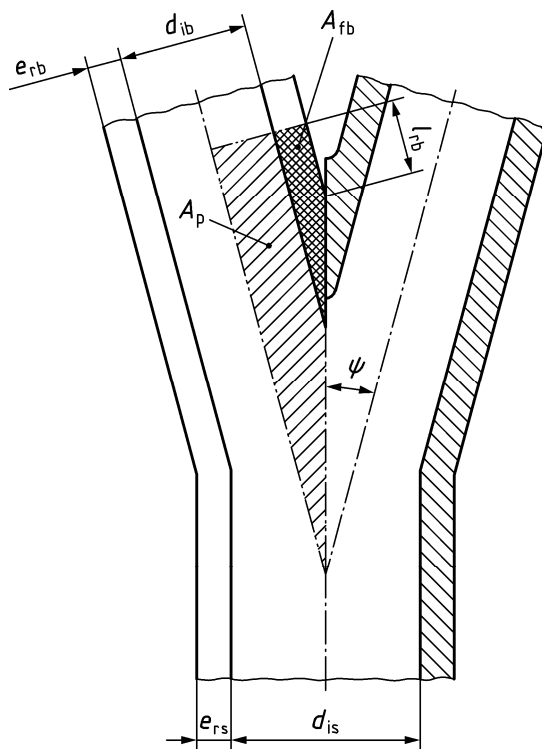


Figure 8.3-5 — Load diagram for Y-shaped branches

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 $\nu = 1$ 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 condition¹²⁾:

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

The efficiency¹²⁾ shall be

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

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 condition¹³⁾ applies:

$$f_a = \frac{p_c (2 A_p + A_{fs} + A_{fb})}{2 \left(A_{fs} + \frac{f_b}{f_s} A_{fb} \right)} \leq f_s \quad (8.3-4)$$

and in this case, the efficiency¹³⁾ shall be

$$v_b = \frac{d_{is} \left(A_{fs} + \frac{f_b}{f_s} A_{fb} \right)}{e_{rs} \left(2 A_p + A_{fb} - \frac{f_b}{f_s} A_{fb} \right)} \leq 1 \quad (8.3-5)$$

$f_b/f_s > 1$ 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

$$f_{aI} = p_c \left(\frac{A_{pI}}{A_{fsI} + A_{fbI} + 0,7 A_{fpI}} + \frac{1}{2} \right) \leq f_s \quad (8.3-6)$$

and for area II

$$f_{aII} = p_c \left(\frac{A_{pII}}{A_{fsII} + A_{fbII} + 0,7 A_{fpII}} + \frac{1}{2} \right) \leq f_s \quad (8.3-7)$$

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:

$$\left(f_s - \frac{p_c}{2} \right) A_{fsI} + \left(f_b - \frac{p_c}{2} \right) A_{fbI} + \left(f_p - \frac{p_c}{2} \right) 0,7 A_{fpI} \geq p_c A_{pI} \quad (8.3-8)$$

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 $d_{ib}/d_{is} \geq 0,7$ and simultaneously $e_{rb}/e_{rs} < d_{ib}/d_{is}$, the following condition shall be met for the main body / branch transition in the cross-section vertical to the main body axis:

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.

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

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 $\psi \geq 15^\circ$. The strength condition for the loaded gusset area shall be

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

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 $\psi > 45^\circ$,

$$y = 1 \quad (8.3-11)$$

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

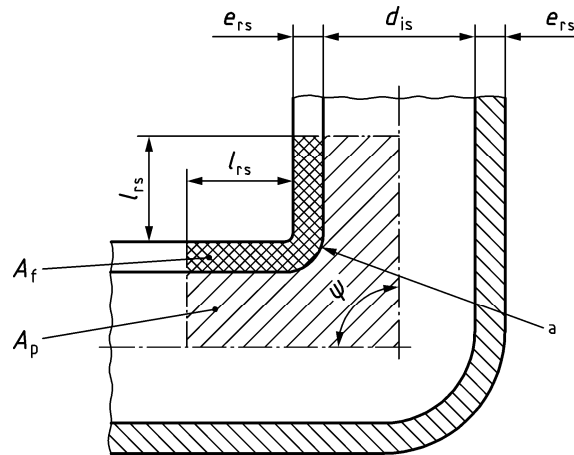
$$y = 1 + 0,005 (45^\circ - \psi) \quad (8.3-12)$$

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 $R_{m \text{ T TC}}$, the following strength condition shall apply

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

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



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 condition¹⁴⁾ shall apply:

$$f_{a\phi} = \frac{p_c}{2} \cdot \frac{2A_{ps} \frac{1+\cos^2\phi}{2} + 2A_{pb1} + 2A_{pb2}}{A_{fs0} + A_{fb1} + A_{fb2}} + \frac{p_c}{2} \leq f_s \quad (8.3-14)$$

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 efficiency¹⁴⁾ shall be

$$v_m = \frac{d_{is}}{e_{rs}} \cdot \frac{A_{fs} + A_{fb1} + A_{fb2}}{2A_{ps} \frac{1+\cos^2\phi}{2} + 2A_{pb1} + 2A_{pb2}} \leq 1 \quad (8.3-15)$$

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

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

In this case, the efficiency¹⁵⁾ shall be

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

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

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.