BRITISH STANDARD

BS EN 12953-3:2002

Shell boilers —

Part 3: Design and calculation for pressure parts

The European Standard EN 12953-3:2002 has the status of a British Standard

ICS 27.060.30; 27.100

National foreword

This British Standard is the official English language version of EN 12953-3:2002. It partially supersedes BS 2790:1992 which will be withdrawn on publication of BS EN 12953 Parts 1, 2, 3, 4, 5, 6, 8 and 9.

When the reference to this European Standard has been published in the Official Journal of the European Communities (OJ), compliance with it will confer a presumption of conformity with the essential requirements covered by the standard in respect of the Pressure Equipment Directive.

The UK participation in its preparation was entrusted to Technical Committee PVE/16, Shell boilers, which has the responsibility to:

- aid enquirers to understand the text;
- present to the responsible international/European committee any enquiries on the interpretation, or proposals for change, and keep the UK interests informed;
- monitor related international and European developments and promulgate them in the UK.

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

Cross-references

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This British Standard, having been prepared under the direction of the Engineering Sector Policy and Strategy Committee, was published under the authority of the Standards Policy and Strategy Committee on 14 June 2002

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Chaudières à tubes de fumée - Partie 3: Conception et calcul des parties sous pression

Großwasserraumkessel - Teil 3: Konstruktion und Berechnung für drucktragende Teile

This European Standard was approved by CEN on 15 May 2002.

CEN members are bound to comply with the CEN/CENELEC Internal Regulations which stipulate the conditions for giving this European Standard the status of a national standard without any alteration. Up-to-date lists and bibliographical references concerning such national standards may be obtained on application to the Management Centre or to any CEN member.

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EUROPÄISCHES KOMITEE FÜR NORMUNG

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Contents

Foreword

This document (EN 12953-3:2002) 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 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(s), see informative Annex ZA, which is an integral part of this standard.

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 November 2002, and conflicting national standards shall be withdrawn at the latest by November 2002.

This European Standard EN 12953 concerning shell boilers 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.*
— *Part 4: Workmanship and construction of pressure parts of the boiler.*
— *Part 5: Inspection during construction, documentation and marking of pressure parts of the boiler.*
— *Part 6: Requirements for equipment for the boiler.*
— *Part 7: Requirements for firing systems for liquid and gaseous fuels for the boiler.*
— *Part 8: Requirements for safeguards against excessive pressure.*
— *Part 9: Requirements for limiting devices, and of the boiler and accessories.*
— *Part 10: Requirements for boiler feedwater and boiler water quality.*
— *Part 11: Acceptance tests.*
— *Part 12: Requirements for firing systems for solid fuels for the boiler.*
— *Part 13: Operational Instructions.*

CR 12953-14: Guidelines for the involvement of an inspection body independent of the manufacturer.

Although these Parts can be obtained separately, it should be recognized that the Parts are inter-dependent. As such, the design and manufacture of shell boilers requires the application of more than one Part in order for the requirements of the standard to be satisfactorily fulfilled.

The Annex A of this European Standard is informative.

The Annex B of this European Standard is normative.

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

1 Scope

This Part of this European Standard specifies requirements for the design and calculation of pressure parts of shell boilers as defined in EN 12953-1.

2 Normative references

This European Standard incorporates by dated or undated reference, provisions from other publications. These normative references are cited at the appropriate places in the text and the publications are listed hereafter. For dated references, subsequent amendments to or revisions of any of these publications apply to this European Standard only when incorporated in it by amendment or revision. For undated references the latest edition of the publication referred to applies (including amendments).

EN 12953-1:2001, *Shell boilers — Part 1: General.*

EN 12953-2, *Shell boilers — Part 2: Materials for pressure parts of boilers and accessories.*

EN 12953-4, *Shell boilers — Part 4: Workmanship and construction of pressure parts of the boiler.*

EN 12953-5:2002, *Shell boilers — Part 5: Inspection during construction, documentation and marking of pressure parts of the boiler.*

EN 12953-8, *Shell boilers — Part 8: Requirements for safeguards against excessive pressure.*

prEN 12953-10, *Shell boilers — Part 10: Requirements for boiler feedwater and boiler water quality.*

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

3 Terms and definitions

For the purposes of this European Standard the terms and definitions given in EN 12953-1 apply.

4 Symbols and abbreviations

For the purposes of this Part of this European Standard, the symbols given in EN 12953-1:2002, Table 4-1 shall apply. Throughout this standard, additional terminology and symbols have been included where necessary to meet the requirements of the specific text concerned. It should also be noted that in some clauses the same additional symbol is used in different formulae to represent different terms. However, in all such cases, the special meaning of each symbol is indicated for each formula.

5 General

5.1 Boilers

The requirements in this standard shall apply to boilers constructed throughout under the conditions specified herein and which are to be operated under normal operating conditions, with feedwater and boiler water in accordance with prEN 12953-10, and under adequate supervision. Where the risk of abnormal working conditions is foreseen, such as severe cyclic service, the design shall be given special consideration.

5.2 Hot-water generators

For directly fired hot-water generators the difference between the outlet temperature and the inlet temperature should not exceed 50 K. If the difference between these two temperatures is greater than 50 K, either internal or external mixing devices shall be used to limit the differential temperature within the boiler to 50 K.

The difference between the saturation temperature corresponding to the maximum working pressure, and the outlet temperature, should not exceed 80 K. If the difference is greater than 80 K, the distances in accordance with 10.1 shall be increased by 50 %. Furthermore the maximum heat input in accordance with Figure 5.4-1 shall be reduced by 20 %.

The inlet water entering the boiler shall not impinge directly on the furnace tube.

5.3 Design of principal welds

The types of weld employed in the design of the boiler shall be in accordance with EN 12953-4. Welds which are subjected to non-destructive examination (NDE) in accordance with the requirements of EN 12953-5 shall be designed so that the required NDE can be carried out.

The value of the weld factor v used in the calculation for the shell thickness shall be either 0,85 or 1 depending on the extent of NDE to be carried out (see 7.2 and EN 12953-5).

5.4 Thermal design of furnaces tubes

In order to ensure safe burner/boiler combinations, the heat input for a given furnace tube inside diameter shall not exceed the value given in Figure 5.4-1. Burners with a fixed firing rate (also called on/off or single stage burners) shall not be used for heat inputs exceeding 1 MW per furnace tube. Combustion shall be completed in the furnace tube.

NOTE Examples 1 and 2 show how Figure 5.4-1 is used.

EXAMPLE 1 Furnace tube inside diameter required for a given heat input

Oil flow: 0,1 kg/s
Net calorific value: 42,9 MJ/kg
Air flow (with air excess 15 %): 1,76 kg/s
Air temperature (with air preheater): 120 °C
Heat input: 4 290 + 214 = 4 504 kW
Furnace tube steel: P295 GH

a) Minimum plain furnace tube inside diameter: 810 mm
b) Minimum Fox 150 × 50 (Table 13.1-1) corrugated furnace tube inside diameter: 760 mm.

EXAMPLE 2:
Allowable heat input for a given furnace tube inside diameter

a) Plain furnace tube inside diameter: 1 500 mm

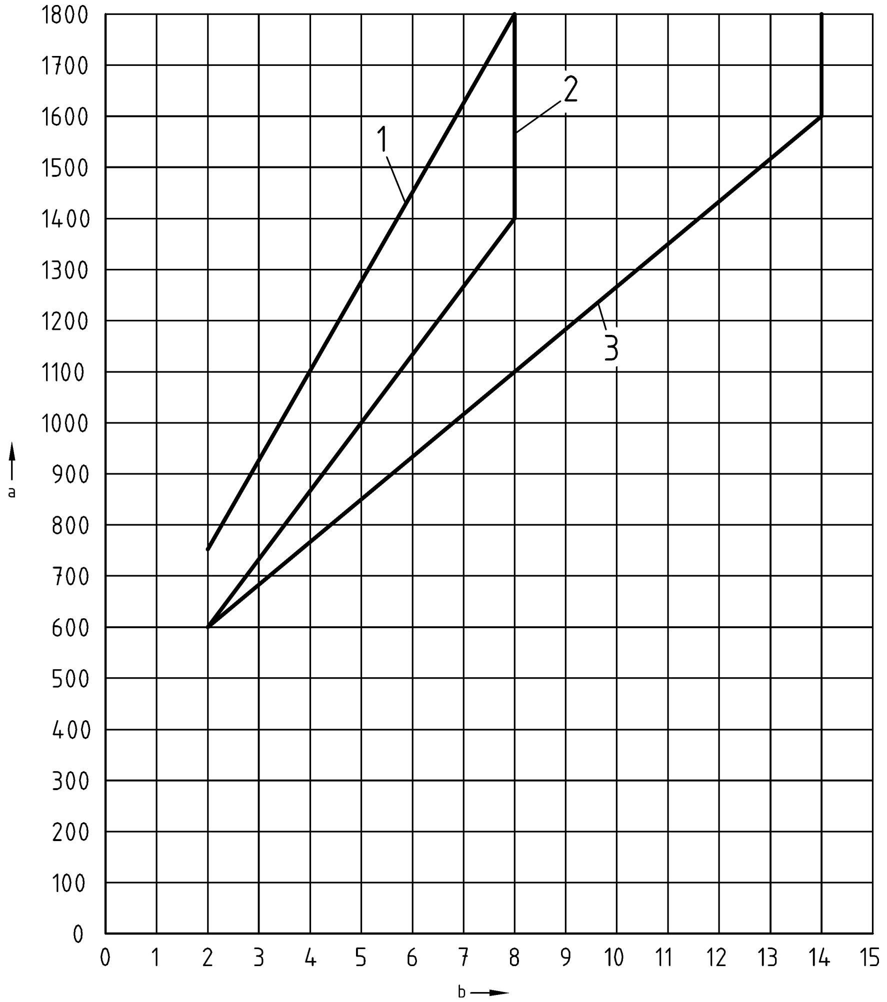
- P295 GH, oil firing: 12,90 MW
- P295 GH, gas firing: 16,77 MW
- P265 GH, oil firing: 8,00 MW
- P265 GH, gas firing: 10,40 MW

b) Fox 150 × 50 (Table 13.1-1) corrugated furnace tube inside diameter 1 000 mm

- P265 GH, gas firing: 7,02 MW.

The length of the refractory should not be larger than one third of the inside furnace tube diameter measured from the end of the burner.

NOTE Attention is drawn to national regulations that for boilers with an inside furnace tube diameter greater than 1 400 mm or a heat input greater than 12 MW a temperature measurement consisting of at least three measurement points in the furnace can be required.



Key

1 Coal firing (grate) P 295 GH
2 Oil firing P 265 GH
3 Oil firing of P 295 GH, P 355 GH

a Minimum furnace tube inside diameter [mm]
b Heat input [MW]

NOTE 1 For corrugated furnaces tubes the minimum inside diameter d_i can be reduced by the depth of the corrugation.

NOTE 2 In the case of gas firing the heat input given for oil firing can be increased by 30 %.

NOTE 3 Heat input is the product of the fuel flow rate and the lower calorific value. Air preheating should be taken into account if the air temperature is greater than 100 °C.

Figure 5.4.1 — Relation between heat input and furnace tube inside diameter

5.5 Dimensions of pressure parts

The wall thickness and other dimensions of pressure parts shall be sufficient to withstand the calculation pressure at calculation temperature and shall be determined in accordance with this Part.

5.6 Establishment of pressures

5.6.1 Maximum allowable pressure

The maximum allowable pressure *PS* is the maximum pressure for which the boiler is designed and shall be measured at the highest point of the shell boiler.

5.6.2 Calculation pressure

The calculation pressure shall be not less than the sum of the maximum allowable pressure and the hydrostatic head. If the latter is less than 3 % of the maximum allowable pressure, the effect of hydrostatic head shall be ignored.

5.6.3 Safety valves set pressure

The safety valve(s) set pressure shall not exceed the maximum allowable pressure (see also EN 12953-8).

5.6.4 Hydrostatic test pressure

The standard hydrostatic test pressure shall be not less than that obtained from the following:

$$p_t = 1{,}25\ p_d\ \frac{R_{p0,2\ 20}}{R_{p0,2\ tc}} \tag{5.6-1}$$

or

$$p_t = 1{,}43\ p_d \tag{5.6-2}$$

which ever is the higher;

where

$R_{p0,2\ 20}$ is the minimum value of the yield point at 20 °C.

The value of $R_{p0,2\ tc}$ shall be taken for the boiler shell or ends at their calculation temperatures.

In the case of boilers with expanded tubes, the values of p_t = 1,43 p_d may be taken.

5.7 Allowances

5.7.1 Allowances for fabrication tolerances

The allowance c_1 is to compensate for negative tolerances, and also any reduction in thickness as a result of the forming process.

5.7.2 Corrosion allowance

For the purpose of design, corrosion allowance c_2 shall also include for erosion and abrasion if these effects are expected to occur.

For components with wall thickness:

$>$ 30 mm and for all flat components, a corrosion allowance of 0 mm may be used;

$\leq$ 30 mm a minimum corrosion allowance of 0,75 mm shall be taken.

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

6 Calculation temperature and nominal design stress

6.1 Calculation temperature

The calculation temperature t_c shall be the mean metal temperature and shall be determined as specified in a) to e).

a) For shells, drums and other components not subject to heat transfer, the calculation temperature shall be not less than the saturation temperature corresponding to the maximum allowable pressure or the maximum allowable temperature.

b) For smoke tubes, the calculation temperature shall be determined in accordance with the following equations:

$$t_c = t_s + 2e_t \tag{6.1-1}$$

or

$$t_c = t_s + 25 \tag{6.1-2}$$

whichever is the greater.

c) The calculation temperature for areas of plate subject to heat transfer but not swept by flame, or for tube nest areas where the gas entry temperature is not greater than 800 °C, shall be determined in accordance with the following equations:

$$t_c = t_s + 2e_h \tag{6.1-3}$$

or

$$t_c = t_s + 50 \tag{6.1-4}$$

whichever is the greater.

The calculation temperature for flat walls in the flue gas pass with a flue gas temperature $t_G < 400$ °C shall be:

$$t_c = t_s + 20 \tag{6.1-5}$$

d) For tube plates subject to gas entry temperatures exceeding 800 °C in fired boilers using fossil fuels, including natural gas, the calculation temperature shall be determined in accordance with Annex A, using the true gas entry temperature t_G determined from the following equation:

$$t_G = 51 \left(\frac{H}{A}\right)^{0,25} \tag{6.1-6}$$

For fuels where the true gas entry temperature is higher than that obtained with natural gas, and for waste-heat boilers, the calculation temperature shall be determined in accordance with Annex A.

The maximum metal temperature as determined in accordance with Annex A shall not exceed 420 °C.

The condition for the calculation in accordance with Annex A is good contact between smoke tubes and tube sheet. Where this cannot be ensured, the method of attachment shown in Figure 12.4-1f) may be employed with the following limitations:

1) The depth of the connecting weld between smoke tubes and tube sheet shall be greater than or equal to the wall thickness of the smoke tube plus 2 mm;

2) The length of the gap, measured from the root of the weld, shall be less than or equal to four times the wall thickness of the smoke tube. If this dimension is exceeded, a cooling groove shall be provided.

If the above requirements are adhered to, the calculated temperature shall be determined in accordance with equation (6.1-4).

For LPB, the calculation temperature for flat walls subject to a flue gas temperature > 800 °C shall be given by equation (6.1-4).

e) The calculation temperature for furnaces tubes shall be determined by the following equations:

1) In the case of furnaces tubes subjected to the flame and a heat input ≤ 12 MW

$$t_c = t_s + 4e + 15 \quad (6.1\text{-}7)$$

2) In the case of furnaces tubes subjected to flame and a heat input > 12 MW

for oil firing

$$t_c = t_s + 3{,}5e + 80 \quad (6.1\text{-}8)$$

for gas firing

$$t_c = t_s + 3e + 65 \quad (6.1\text{-}9)$$

NOTE Equations (6.1-8) and (6.1-9) are based on a maximum heat flux of 0,3 W/mm^2 for oil firing and 0,24 W/mm^2 for gas firing, taking into account an allowance of 0,25 mm for the thickness of scale with a conductivity of 1,2 W/m K.

Where it can be shown that the heat flux is lower, e.g. low NOx firing, the calculations can be adjusted accordingly, but the value of t_c should be not less than that given by equation (6.1-7).

3) In the case of furnace tubes without flame the calculation temperature t_c shall taken as being equal to the highest of the two values given by the following equations:

$$t_c = t_s + 25 \quad (6.1\text{-}10)$$

$$t_c = t_s + 2e \quad (6.1\text{-}11)$$

6.2 Nominal design stress

Unless otherwise stated in this Part of this European Standard, the nominal design stress f shall be the lower of the values obtained from the following ratios:

$$f = \min\left\{\frac{R_{p0,2\,tc}}{1{,}5}; \frac{R_m}{2{,}4}\right\} \quad (6.1\text{-}12)$$

NOTE The term "nominal design stress", designated by the symbol f, is the stress to be used in the equations herein for the design of pressure parts. The detailed design rules in this Part will maintain the actual maximum stresses within acceptable limits for the type of loading considered.

7 Cylindrical shells under internal pressure

7.1 Shell thickness

7.1.1 Requirements

The shell thickness after deduction of allowances

$$e_{rs} = e_s - c_1 - c_2 \quad (7.1\text{-}1)$$

of cylindrical shell shall be at least the greatest of those required by the following:

a) a minimum of 6 mm for cylindrical shells of outside diameter ≥ 1 000 mm except LPB. For outside diameter < 1 000 mm and LPB a minimum of 4 mm shall be required;

b) the requirements of 7.2;

c) the requirements of 7.2 by applying 8.2 or 8.3.3 and 8.3.4.

7.1.2 Required wall thickness including allowances

The required wall thickness including allowances shall be derived from:

$$e_{sa} = e_{cs} + c_1 + c_2 \quad (7.1\text{-}2)$$

7.2 Basic calculation

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

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

if d_{is} is given or

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

if d_{os} is given.

The equivalent value of the stress in the shell can be calculated by modifying equations (7.2-1) or (7.2-2).

7.3 Boiler support

Experience has shown that it shall not be necessary to carry out strength calculations in regard to boiler supports as fatigue is not normally encountered in this area.

7.4 Reinforcing pads

Reinforcing pads may be used for the reinforcement of openings and branches and for the distribution of load at supports and attachments. Such reinforcing pads shall be designed analogous to the requirements of 8.1.5.1.

8 Openings and branches in cylindrical shells

NOTE This clause specifies the design rules for openings and branches in cylindrical shells. All dimensions exclude allowances c_1 and c_2 for wall thickness.

8.1 General

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

8.1.1.1 For cylindrical shells with openings the 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; and for 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 $l_{so} \le e_{rs}$ (see Figures 8.1-2 and 8.1-3).

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 and -3). The welded joint between the main body and branch shall be a full-strength weld where in the case of branches in accordance with Figure 8.1-3 a residual gap ≤ 1,5 mm may be present. A wall thickness ratio of e_{rb}/e_{rs} up to and including 2 shall be permitted for $d_{ib} > 50$ 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-in and seal-welded-only branches (see Figure 8.1-1) or branches attached to the main body by fillet welds with a residual root gap > 1,5 mm 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 $l_{bo} \geq e_{rb}$ (see Figures 8.1-2 and 8.1-3).

   For branches with $d_{ib}/d_{is} \geq 0{,}7$ reference shall be made to 8.3.3.4.

   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) reinforcement by reinforcing pads analogous to increasing the wall thickness as in a) (see Figures 8.1-4 to 8.1-5).

8.1.1.2 Where there are elliptical 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 notch-free surface[1)]. Edges shall be rounded.

Openings shall be located at an adequate distance from the 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 2 e_{rs} and on a main with a body thickness $e_{rs} > 25$ mm the distance is at least 50 mm from the weld edge.

8.1.2 Effective lengths l_{rs} for calculation of efficiencies and of compensations

For the calculation of efficiencies by way of approximation as described in 8.2 and the calculation of isolated and adjacent branches described in 8.3, effective lengths l_{rs} are required which shall be used for the main body.

$$l_{rs} = \min\left[\sqrt{(d_{is} + e_{rs})e_{rs}}\,; l_{s1}\right] \quad (8.1\text{-}1)$$

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

and for the nozzle with $\Psi \geq 45°$

for external projection

$$l_{rb} = \min\left[\sqrt{(d_{ib} + e_{rb})e_{rb}}\,; l_{b1}\right] \quad (8.1\text{-}2)$$

for internal projection

$$l_{rbi} = \min\left[0{,}5\sqrt{(d_{ib} + e_{rb})e_{rb}}\,; l_{b2}\right] \quad (8.1\text{-}3)$$

1) Welded joints are considered to be notch-free if they are within the limits given in EN 12952-5.

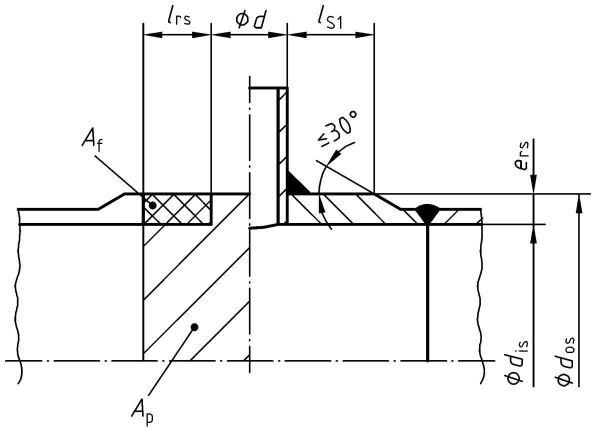


Figure 8.1.1 — Reinforcement by increasing the wall thickness of the main body with opening

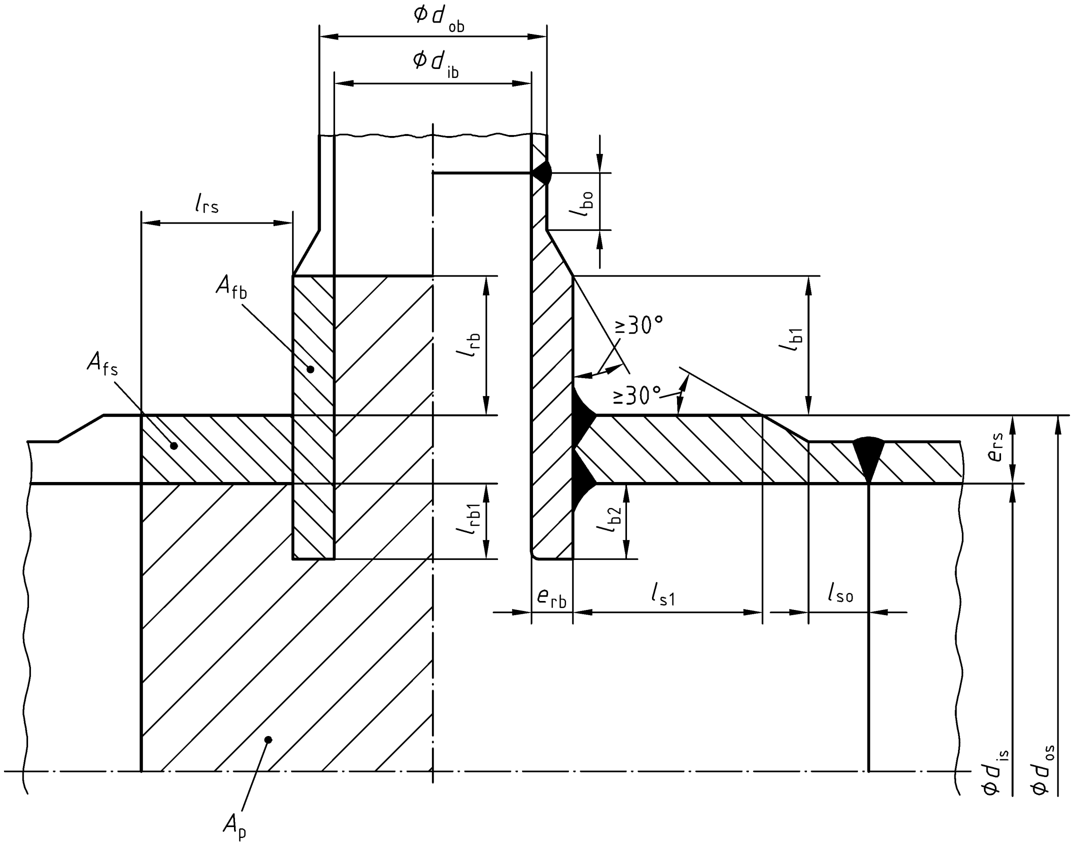


Figure 8.1.2 — Reinforcement by set-through and full penetration welded branch

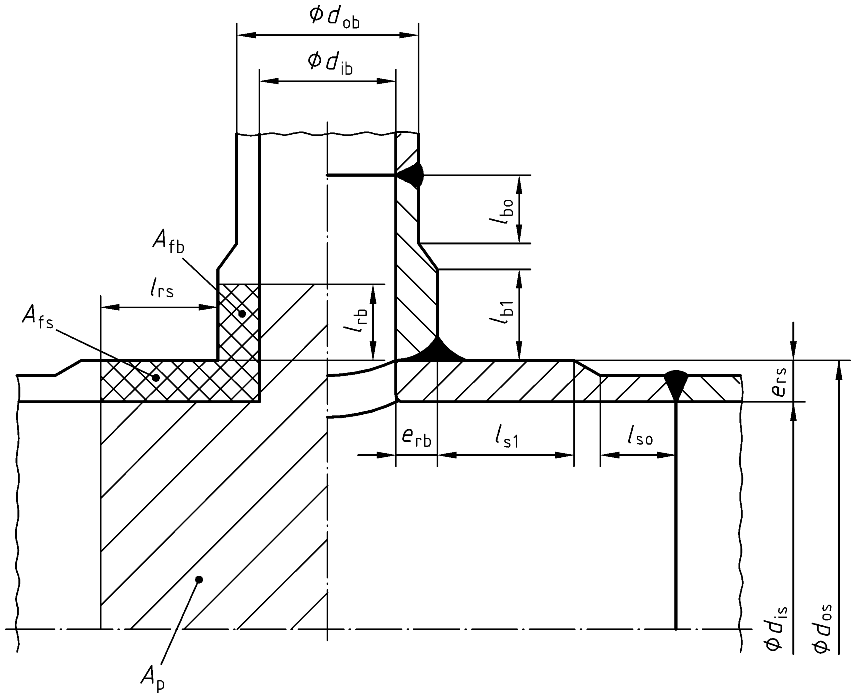


Figure 8.1.3 — Reinforcement by welded-on branch

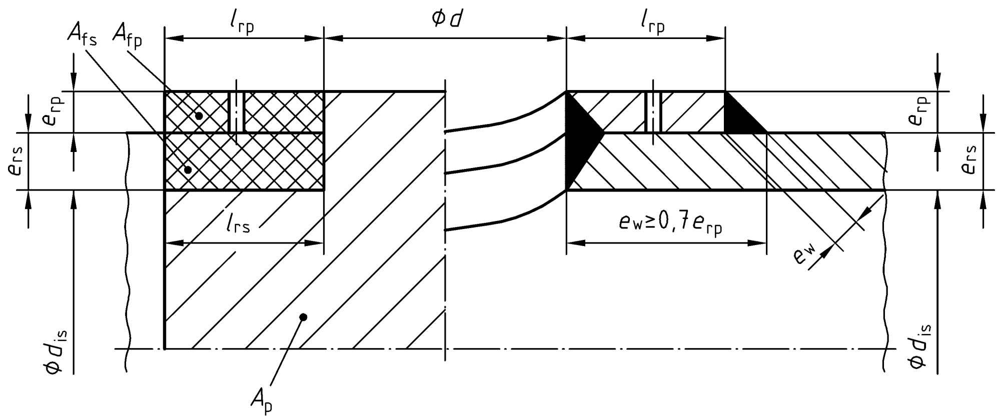
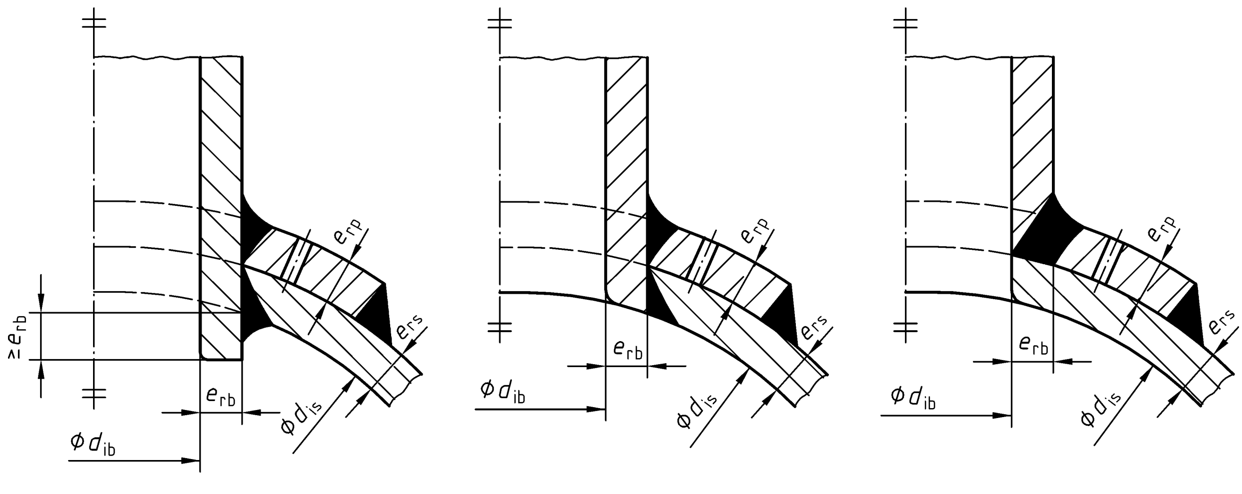


Figure 8.1.4 — Opening with reinforcing pad



a) set through welded branch b) set in welded branch c) set on welded branch

Figure 8.1.5 — Opening with reinforcing pad and full penetration branch

8.1.3 Condition of isolated openings

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

$$P_\phi \geq \frac{\left(\frac{d_{ib1}}{2} + e_{rb1}\right)}{\cos \Psi_1} + \frac{\left(\frac{d_{ib2}}{2} + e_{rb2}\right)}{\cos \Psi} + 2\sqrt{(d_{is} + e_{rs})\, e_{rs}} \qquad (8.1\text{-}4)$$

For openings without a branch, $e_{rb} = 0$ and $\Psi = 0°$(for Ψ see Figure 8.3-1).

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 pads 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 Special case

The nozzle inside diameter d_{ib} shall be used in the calculation even if the hole diameter d in the main body is less than d_{ib}.

8.1.5 Requirements for design of reinforcing pads

8.1.5.1 General

Reinforcing pads shall be designed in accordance with 8.1.1.1.

Where reinforcing pads are fitted (see Figures 8.1-4 and 8.1-5) they shall have close contact with the main body and shall be provided with tell-tale holes to avoid the trapping of gases, unless experience has shown that it is not necessary.

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 not be 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.

a) the ratio d_{ib}/d_{is} of the branch diameter to the cylinder diameter shall be not greater than 1/4 unless the adequacy of the design is demonstrated by experience or by a hydrostatic proof test.

b) the sizes L_i of the inner fillet welds by which reinforcing pads are attached to shell plates (see Figure 8.1-6) shall comply with the following relationship:

$$L_i \geq 0{,}7\, e_{cp} \qquad (8.1\text{-}5)$$

The sizes L_o of the outer peripheral fillet welds by which reinforcing pads (see Figure 8.1-6) are attached to shell plates shall be determined from the following equation, but shall in no case be less than the size of the inner fillet weld:

$$L_o = \max \left[\frac{d_{ip} \left(2\, e_{rs} - L_i \right)}{d_{op}} ; L_i \right] \tag{8.1-6}$$

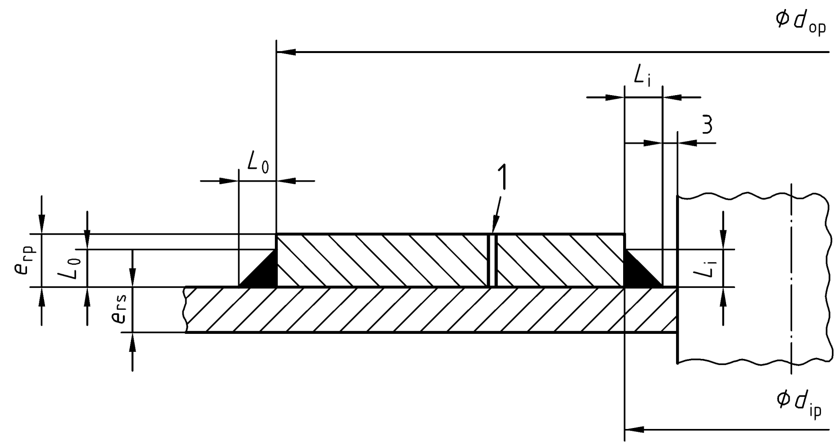
For elliptical reinforcing pads

$$d_{op} = \frac{a_o + b_o}{2} \tag{8.1-7}$$

$$d_{ip} = \frac{a_i + b_i}{2} \tag{8.1-8}$$

c) for calculation purposes the thickness of the reinforcing pad shall be not taken greater than the thickness of the shell.

Dimensions in millimetres

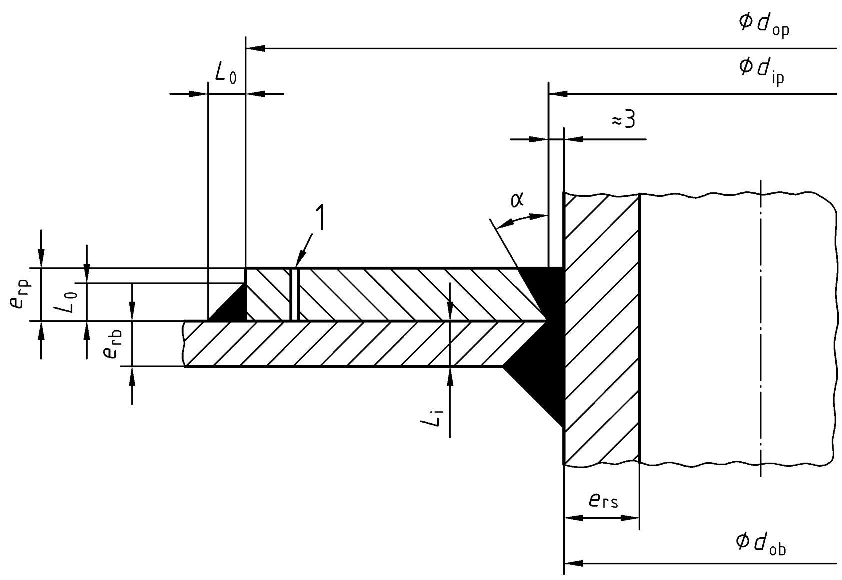


Key

1 Tell-tale hole

a Opening in shell

a)



Key

1 Tell-tale hole

α is 30° to 45°

b)

Figure 8.1.6 — Welding of reinforcing pads

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

8.2.1 General

Exact solutions shall be 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 an unreinforced opening

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

$$v_a = \frac{p_c\, d_{is}}{\left(2\ f_s - p_c\right) e_{rs}} \tag{8.2-1}$$

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

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

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 8.2.3.

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

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

and

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

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

This shall also be valid for counter borings and for partial penetration holes even if the equations (8.2-6) or (8.2-9) or the more exact calculations in accordance with 8.3.3 recommend a smaller diameter d as given in equation (8.2-3).

Where a tube with an outside diameter d_{ob} is attached to an opening, as shown in 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

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

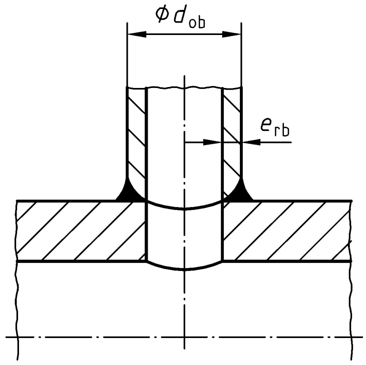


Figure 8.2-1 — Tube attached to main body

If the diameter d_{is} and the wall thickness e_{rs} of the main body have been determined, an isolated hole for a tube with a maximum outside diameter

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

shall be permitted.

8.2.4 Adjacent openings

Where the condition for the centre distance P_Φ of adjacent openings given in 8.1.3 is not met, and tubes with an outside diameter d_{ob} are connected to the opening, with the tubes only able to withstand the internal pressure on account of their wall thickness e_{rb}, the efficiency factor of adjacent openings shall be derived as follows:

$$v_m = \frac{2\left(P_\Phi - d_{ob}\right)}{\left(1 + \cos^2 \Phi\right) P_\Phi} \leq 1 \quad (8.2\text{-}7)$$

i.e. for longitudinal pitch P_Φ with $\Phi = 0$:

$$v_m = \frac{P_\Phi - d_{ob}}{P_\Phi} \quad (8.2\text{-}8)$$

Where the outside diameters of adjacent openings differ from each other, the following shall apply

$$d_{ob} = \frac{d_{ob1} + d_{ob2}}{2} \quad (8.2\text{-}9)$$

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

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 12953-1:2002, Table 4-1, those shown in Figures 8.3-1 to 8.3-4 shall be used.

8.3.2 Requirements for application

8.3.2.1 Openings

The rules specified in 8.3.3 to 8.3.4 shall apply to circular, elliptical and obround openings and nozzles (including oblique nozzles) arranged singly or in groups, in cylindrical shells, provided that the following conditions are satisfied:

a) Openings and nozzles non radial to the shell (Figure 8.3-2)

The ratio of the major to minor axes of the opening shall not exceed 2;

b) Oblique nozzles (Figure 8.3-1)

The nozzle is of circular cross section and the angle between the axis of the nozzle and a line normal to the shell surface shall not exceed 45°;

c) All nozzles

No significant external forces and moments shall be applied to the nozzle. If this is not the case, EN 13445-3 shall be used to calculate the resulting stresses.

The nozzle inside diameter d_{ib} shall be used in the calculation even if the hole diameter *d* in the main body is less than d_{ib}.

Nozzle connections with a residual gap greater than 1,5 mm or set-through and seal-welded-only branches, shall be considered 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 Minimum thickness of nozzles and branch connections

The thickness of nozzles and branches shall be in accordance with 8.1.1.1b), but the thickness shall be not less than that given by the following equation:

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

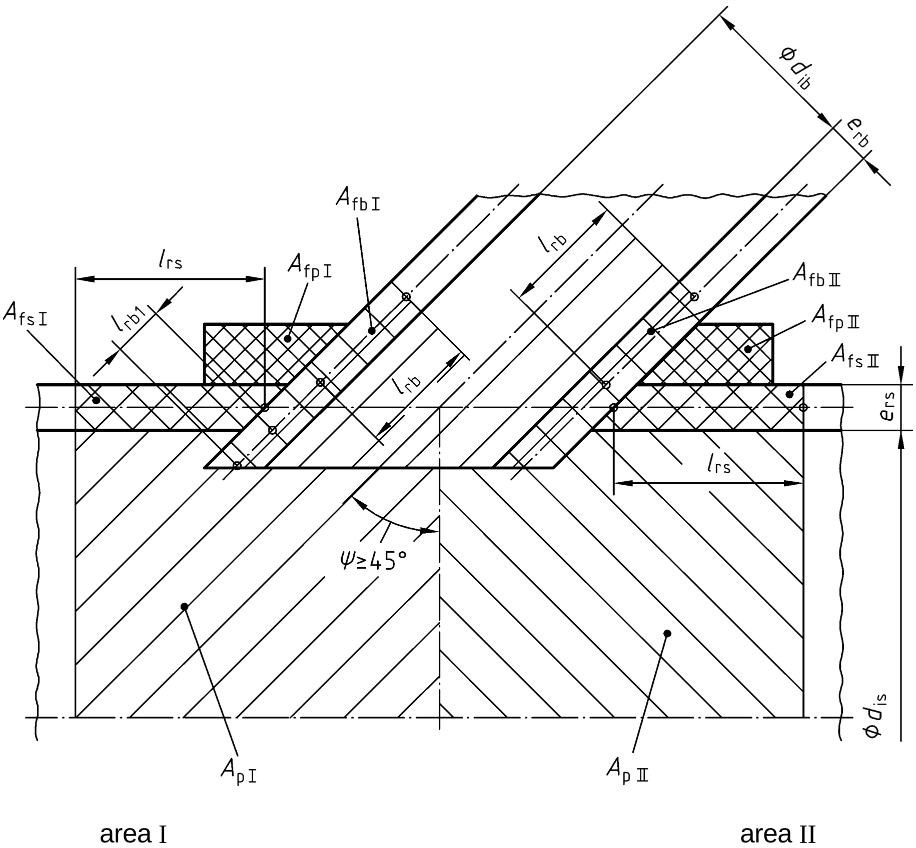


Figure 8.3-1 — Load diagram for cylindrical shell with oblique branch and reinforcing pad

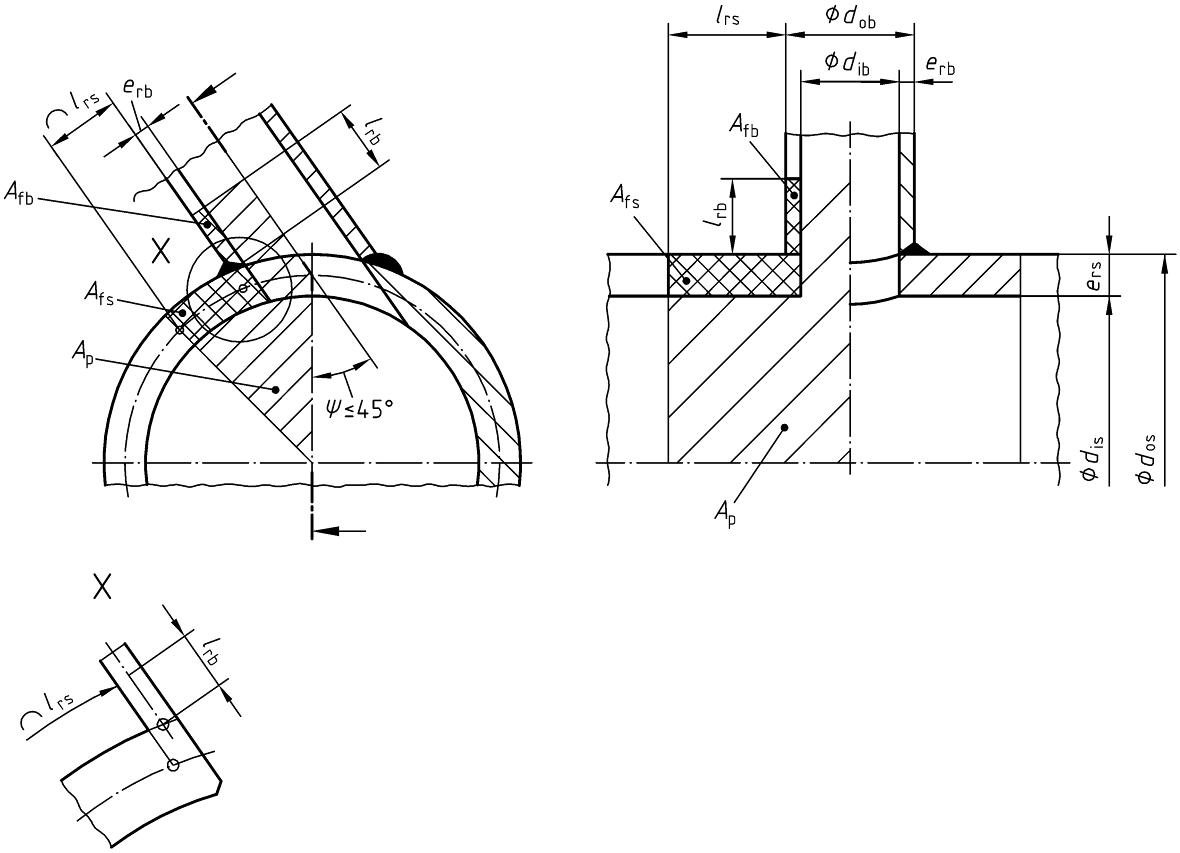
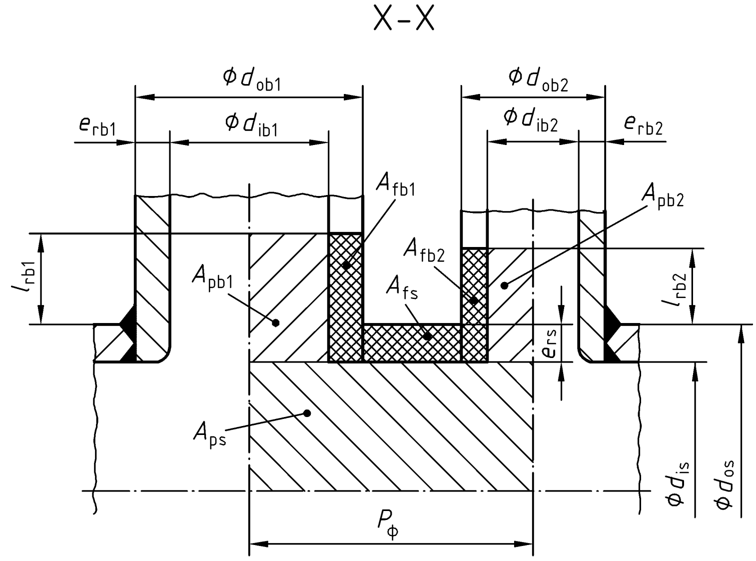
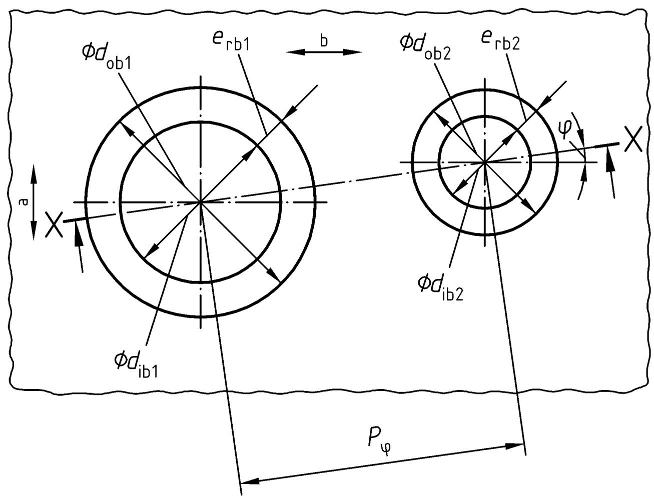


Figure 8.3-2 — Load diagram for cylindrical shell with non-radial branch

a) section view X — X



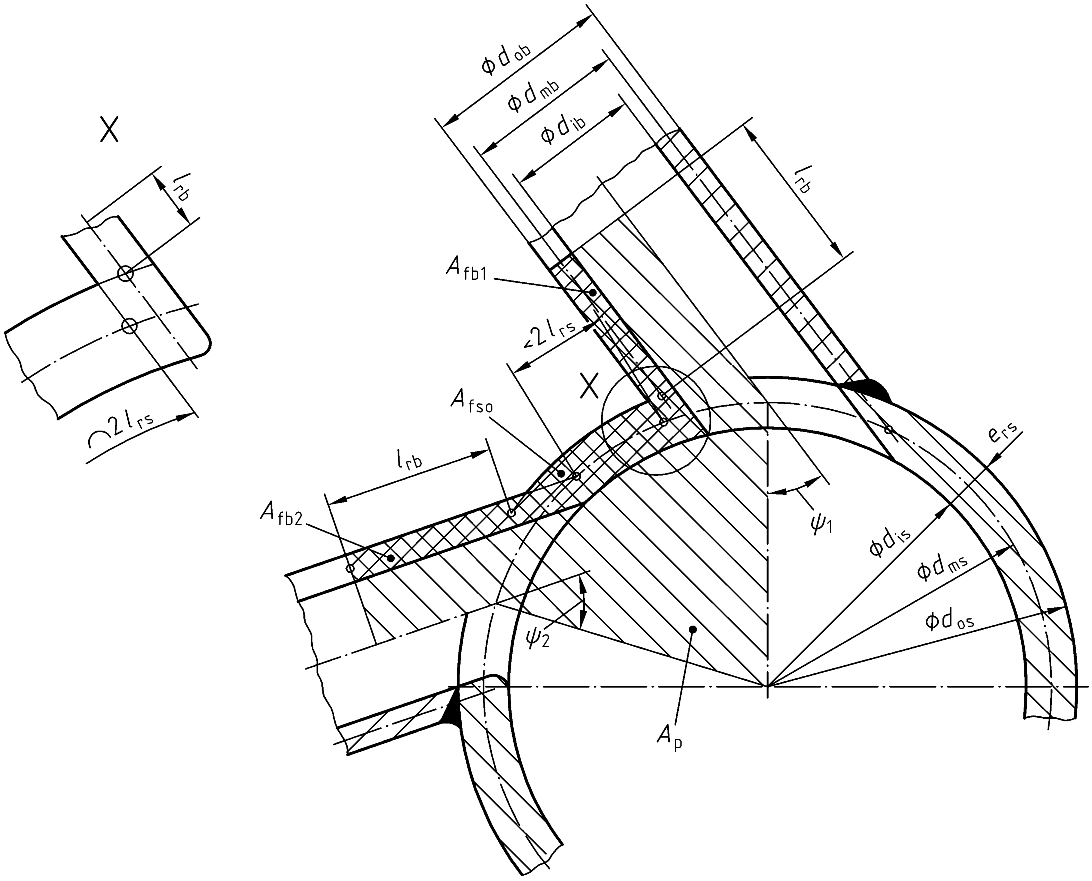
b) plan view



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



with

$\psi_1 \leq 45°$

$\psi_2 \leq 45°$

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

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 be not less than that calculated for $v = 1$ in accordance with 7.2.

8.3.3.2 Isolated opening with a radial branch

8.3.3.2.1 For isolated openings fitted with a radial branch without additional reinforcement 8.3.3.4 and 8.3.3.5 (with $\psi_1 = 0$) shall be additionally taken into account.

8.3.3.2.2 If the design stress of the branch is equal to or greater than that for the main body the following strength condition shall apply:

$$f_a = \min\left[p_c\left(\frac{A_p}{A_{fs} + A_{fb}} + \frac{1}{2}\right);\ f_s\right] \quad (8.3\text{-}2)$$

The efficiency [2)] shall be:

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

8.3.3.2.3 If the design stress of the branch is less than that of the main body, the following strength condition shall apply:

$$f_a = \frac{p_c\left(2\, A\text{p} + A_{fs} + A_{fb}\right)}{2\left(A_{fs} + \dfrac{f_b}{f_s}\, A_{fb}\right)} \le f_s \qquad (8.3\text{-}4)$$

In this case the efficiency[2)] shall be:

$$v_b = \frac{d_{is}\left(A_{fs} + \dfrac{f_b}{f_s}\, A_{fb}\right)}{e_{rs}\left(2\, A_p + A_{fb} - \dfrac{f_b}{f_s}\, A_{fb}\right)} \le 1 \qquad (8.3\text{-}5)$$

8.3.3.3 Isolated opening with an oblique branch and additional reinforcing pad

8.3.3.3.1 For isolated openings fitted with an oblique branch and additional reinforcing pad 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 If the design stress of the branch is equal to or greater than that for the main body, the strength condition for area I shall be:

$$f_{a\,\mathrm{I}} = p_c\left(\frac{A_{p\,\mathrm{I}}}{A_{fs\,\mathrm{I}} + A_{fb\,\mathrm{I}} + 0{,}7\, A_{fp\,\mathrm{I}}} + \frac{1}{2}\right) \le f_s \qquad (8.3\text{-}6)$$

and for area II

$$f_{a\,\mathrm{II}} = p_c\left(\frac{A_{p\,\mathrm{II}}}{A_{fs\,\mathrm{II}} + A_{fb\,\mathrm{II}} + 0{,}7\, A_{fp\,\mathrm{II}}} + \frac{1}{2}\right) \le f_s \qquad (8.3\text{-}7)$$

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

$$\left(f_s - \frac{p_c}{2}\right) A_{fs\,\mathrm{I}} + \left(f_b - \frac{p_c}{2}\right) A_{fb\,\mathrm{I}} + \left(f_p - \frac{p_c}{2}\right) 0{,}7\, A_{fp\,\mathrm{I}} \ge p_c\, A_{p\,\mathrm{I}} \qquad (8.3\text{-}8)$$

and for area II.

$$\left(f_s - \frac{p_c}{2}\right) A_{fs\,\mathrm{II}} + \left(f_b - \frac{p_c}{2}\right) A_{fb\,\mathrm{II}} + \left(f_p - \frac{p_c}{2}\right) 0{,}7\, A_{fp\,\mathrm{II}} \ge p_c\, A_{p\,\mathrm{II}} \qquad (8.3\text{-}9)$$

8.3.3.4 Cross-section vertical to the main body axis

For branch connections where $d_{ib}/d_{is} \ge 0{,}7$ 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:

2) The approximate calculation in accordance with 8.2 can be used instead of this calculation, in which case the reinforcement effect of the nozzles should not be considered.

$$f_{ab} = \frac{p_c}{1,5}\left(\frac{d_{is} + e_{rs}}{2\, e_{rs}} + 0,2\,\frac{d_{ib} + e_{rb}}{e_{rb}}\sqrt{\frac{d_{is} + e_{rs}}{e_{rs}}}\right) \le \min\left(f_s;\, f_b\right) \qquad (8.3\text{-}10)$$

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, the higher loading can 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) shall apply, 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 reinforcing pad (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 according to 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}.

8.3.4 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 Figures 8.3-3. The following strength condition shall apply:

$$f_{a\Phi} = \frac{p_c}{2}\,\frac{2\,A_{p0}\,\frac{1+\cos^2\Phi}{2} + 2\,A_{p1} + 2\,A_{p2}}{A_{fs0} + A_{fb1} + A_{fb2}} + \frac{p_c}{2} \le f_s \qquad (8.3\text{-}11)$$

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

The efficiency [2)] shall be:

$$v_m = \frac{d_{is}}{e_{rs}}\,\frac{A_{fs0} + A_{fb1} + A_{fb2}}{2\,A_{p0}\,\frac{1+\cos^2\Phi}{2} + 2\,A_{p1} + 2\,A_{p2}} \le 1 \qquad (8.3\text{-}12)$$

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:

$$f_{a\Phi} = \frac{p_c}{2}\,\frac{2\,A_{p0}\,\frac{1+\cos^2\Phi}{2} + 2\,A_{p1} + 2\,A_{p2} + A_{fs0} + A_{fb1} + A_{fb2}}{A_{fs0} + \frac{f_{b1}}{f_s}\,A_{fb1} + \frac{f_{b2}}{f_s}\,A_{fb2}} \le f_s \qquad (8.3\text{-}13)$$

In this case, the efficiency 2) shall be:

$$v_{\mathrm{m}} = \frac{d_{\mathrm{is}}}{e_{\mathrm{rs}}} \frac{A_{\mathrm{fs0}} + \frac{f_{\mathrm{b1}}}{f_{\mathrm{s}}} A_{\mathrm{fb1}} + \frac{f_{\mathrm{b2}}}{f_{\mathrm{s}}} A_{\mathrm{fb2}}}{2\,A_{\mathrm{p0}} \frac{1+\cos^2 \Phi}{2} + 2\,A_{\mathrm{p1}} + 2\,A_{\mathrm{p2}} + A_{\mathrm{fb1}} + A_{\mathrm{fb2}} - \frac{f_{\mathrm{b1}}}{f_{\mathrm{s}}} A_{\mathrm{fb1}} - \frac{f_{\mathrm{b2}}}{f_{\mathrm{s}}} A_{\mathrm{fb2}}} \leq 1 \qquad (8.3\text{-}14)$$

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.

9 Ends

9.1 Unstayed dished heads without openings

9.1.1 Unstayed dished heads under internal pressure

The minimum thickness of unstayed dished heads without openings shall be in accordance with equation

$$e_{\mathrm{s}} = e_{\mathrm{cs}} + c + c_1 + c_2 \qquad (9.1\text{-}1)$$

and the following equation:

$$e_{\mathrm{cs}} = \frac{p_{\mathrm{e}}\, d_{\mathrm{o}}\, C}{2\, f} \qquad (9.1\text{-}2)$$

In addition, the thickness of a torispherical head shall be not less than that given by equation (9.1-1) and the following equation:

$$e_{\mathrm{cs}} = \frac{p_{\mathrm{e}}\, r_{\mathrm{is}}}{2\, f - 0{,}5\, p_{\mathrm{e}}} \qquad (9.1\text{-}3)$$

The shape factor C for unstayed dished heads without openings shall be as given in Figure 9.1-1. However, the limiting conditions given in 9.1.2 shall apply.

9.1.2 Limiting conditions

The limiting conditions shall be for:

a) Hemispherical heads

$0{,}005\, d_{\mathrm{o}} \leq e_{\mathrm{cs}} \leq 0{,}16\, d_{\mathrm{o}}$;

b) Ellipsoidal heads

$0{,}005\, d_{\mathrm{o}} \leq e_{\mathrm{cs}} \leq 0{,}08\, d_{\mathrm{o}}$ and

$h_{\mathrm{c}} \geq 0{,}18\, d_{\mathrm{o}}$;

c) Torispherical heads

$0{,}005\, d_{\mathrm{o}} \leq e_{\mathrm{cs}} \leq 0{,}08\, d_{\mathrm{o}}$ and

$r_{\mathrm{ik}} \geq 0{,}1\, d_{\mathrm{o}}$ and

$r_{ik} \geq 2\ e_{cs}$ and

$r_{is} \leq d_o$ and

$h_c \geq 0{,}18\ d_o$;

or

$0{,}01\ d_o \leq e_{cs} \leq 0{,}03\ d_o$ and

$r_{ik} \geq 0{,}1\ d_o$ and

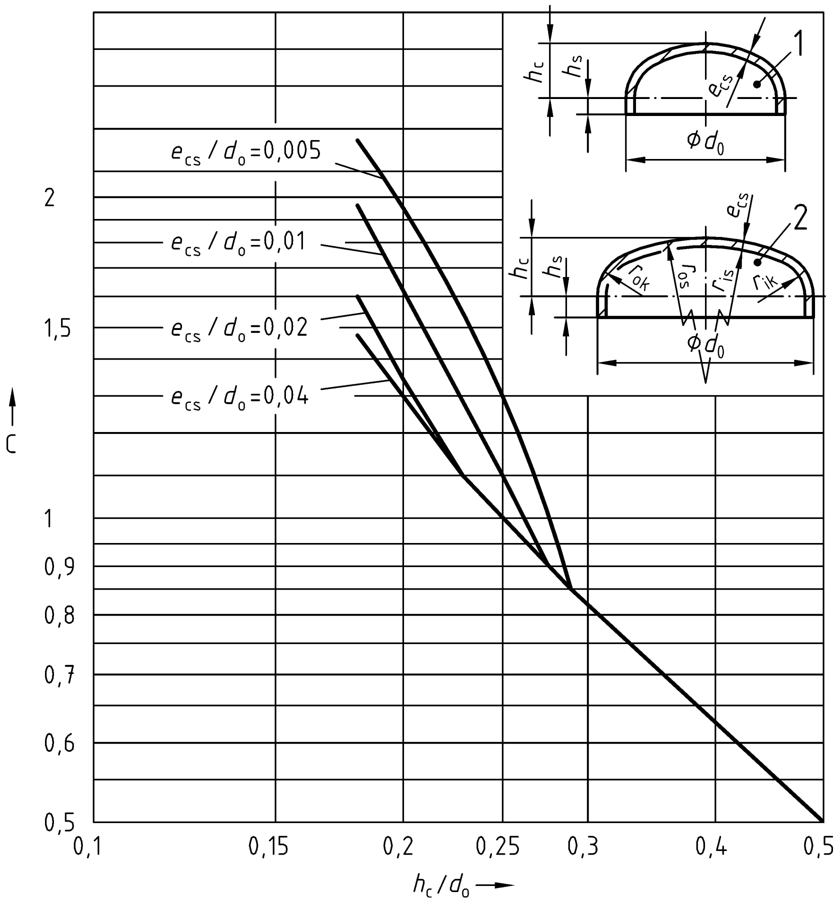
$h_c \geq 0{,}18\ d_o$;

or

$0{,}02\ d_o \leq e_{cs} \leq 0{,}03\ d_o$ and

$r_{ik} \geq 0{,}1\ d_o$ and

$0{,}18\ d_o \leq h_c \leq 0{,}22\ d_o$.



Key
1 Ellipsoidal head
2 Torispherical head

Figure 9.1-1 - Shape factor *C* for unstayed dished heads without openings

9.1.3 Unstayed dished heads under external pressure

The calculation pressure p_c shall be the lower of the values obtained from the following equations:

$$p_c = \frac{e_{cs}\ R_{p\,0,2}}{1{,}2\ r_{os}} \quad (9.1\text{-}4)$$

$$p_c = \frac{0{,}8\ E}{9 + 0{,}006\left(\frac{r_{os}}{e_{cs}}\right)} \left(\frac{e_{cs}}{r_{os}}\right)^2 \quad (9.1\text{-}5)$$

In addition, the thickness of torispherical and ellipsoidal heads under external pressure shall be not less than 1,2 times the thickness required for a head of the same shape subject to internal pressure (see 9.1.1).

9.2 Flat unstayed removable closures

For calculation pressures exceeding 2 N/mm^2, only internally fitted doors shall be used. Otherwise, external closures of the blanked flange type shall be used.

The inside diameter of the opening shall be limited to a maximum of DN 500.

The thickness shall be determined in accordance with equation (9.2-1) with d_i taken as indicated in Figure 9.2-1.

$$e_n = C_1\ d_i\ \sqrt{p_c / f} \quad (9.2\text{-}1)$$

When the closure is external, C_1 shall be taken as 0,41, except where closures of the type shown in Figure 9.2-1e) are used, where the bolting adds to the bending moment in the plate. In such cases the following shall apply:

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 9.2-1a) and b) are used, account shall be taken of the additional bending moment in the plate caused by the bolting, by replacing p_c with the following:

$$p_c + 7{,}5\ W_b / d_i^2$$

where

W_b is the total load in one bolt in Newton, calculated from allowable design stress in the bolts at design temperature multiplied by the total effective cross-sectional area of bolts.

NOTE A design stress value of 550 N/mm^2 can be used for carbon steel bolts of grade 4.6 or equivalent for design temperatures not exceeding 300 °C. For other bolting materials and greater temperatures, see EN 13445-2 and EN 13445-3.

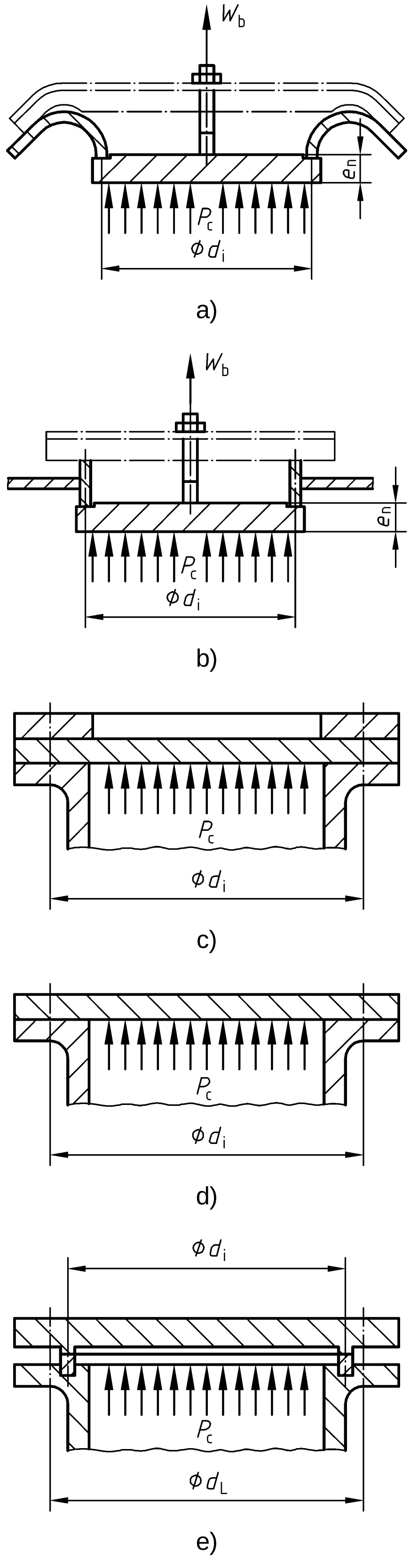


Figure 9.2-1 — Flat unstayed removable closures

9.3 Flange connections

Where used, flange connections shall comply with applicable European Standards (e.g. EN 1092), and they shall be suitable for the maximal allowable pressure and temperature.

10 Supported flat plates, stays and stiffeners

10.1 Breathing space for flat plates

A shell boiler incorporates items in its construction (e.g. furnace tube, tube nests) that operate at different temperatures to each other and to the shell and, because of this, differential expansion will occur.

This differential expansion is normally transferred to the boiler end plate and tube plates, which results in displacement.

Under these conditions, the boiler is said to "breathe", and in order to provide the necessary flexibility, breathing spaces are required.

Stays shall give breathing space around the furnace tube connections and tube nests (see Figure 10.1-1) and equally divide the unstayed areas. Breathing space between furnace tube and tube nests shall be a minimum of 50 mm or 5 % of the shell outside diameter, whichever is the larger, but need not be more than 100 mm.

Breathing space between furnace tube and shell shall be in accordance with Table 10.1-1 and Table 10.1-2 respectively, but shall be not less than 50 mm or, for bowling hoop furnaces tubes, not less than 75 mm.

Table 10.1-1 — Breathing space between furnace tube and shell when the thickness of the end plate is 25 mm or less

Design	Length between boiler end plates L_b m	Breathing space	
		Nominal percentage of outside diameter %	maximum mm
Inserted flat ends	$L_b \leq 5{,}5$	5	100
	$5{,}5 < L_b \leq 6$	5,5	110
	$6 < L_b \leq 6{,}5$	6	120
	$6{,}5 < L_b \leq 7$	6,5	130
Flanged ends	any length	5	100

When the actual thickness of the end plate exceeds 25 mm, the breathing spaces in accordance with Table 10.1-1 shall be increased in accordance with Table 10.1-2.

In the case of reverse flame boilers, the breathing space at the front end between the furnace tube and tube nests shall be not less than 50 mm. Additionally, the sum of this breathing space and the breathing space formed by the outer annular area of the furnace tube rear plate shall be not less than 50 mm or 5 % of the shell inside diameter, whichever is the larger, with a maximum of 100 mm.

Table 10.1-2 — Breathing space between furnace tube and shell when the thickness of the end plates exceeds 25 mm

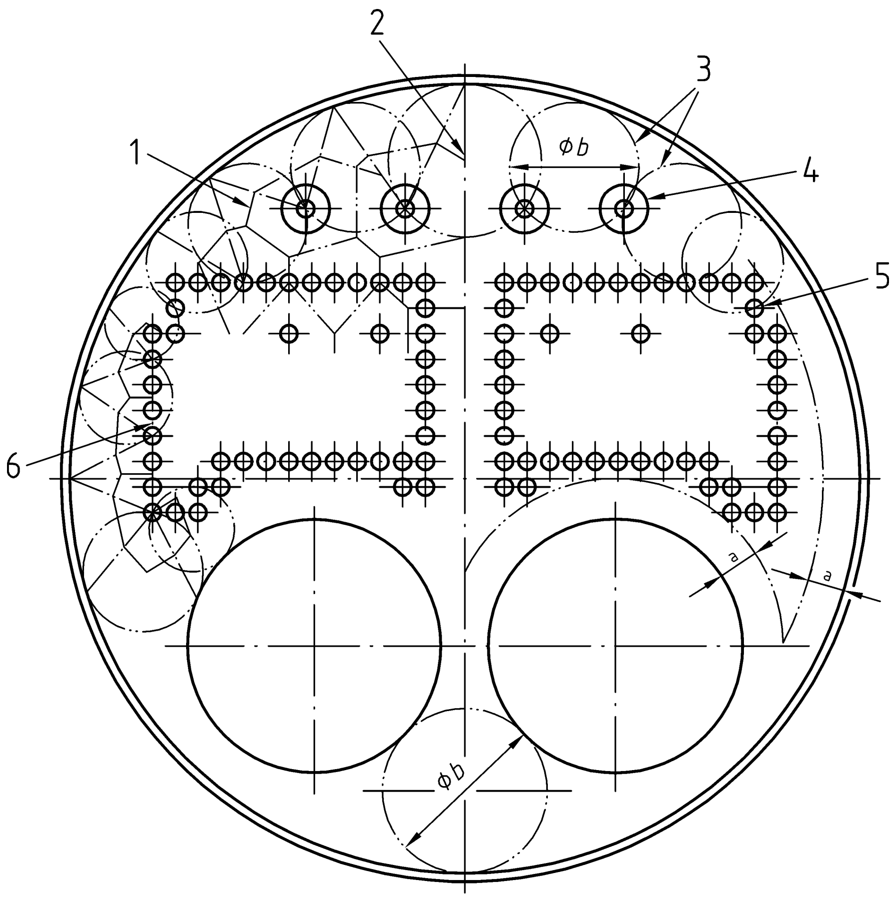
Design	Length between boiler end plates L_b m	Breathing space Nominal percentage of outside diameter %	maximum mm	minimum mm
Inserted flat ends	$L_b \leq 5{,}5$	6,5	130	65
	$5{,}5 < L_b \leq 6$	7	140	70
	$6 < L_b \leq 6{,}5$	8	150	75
	$6{,}5 < L_b \leq 7$	10	160	80
Flanged ends	any length	5	100	50

Breathing space between furnaces tubes shall be not less than 120 mm.

Breathing spaces between gusset or link stays and furnaces tubes shall be not less than 200 mm, except for a shell outside diameter less than 1 400 mm or a furnace tube length less than 3 000 mm, in which case the breathing spaces shall be not less than 150 mm.

All other breathing spaces shall be a minimum of 50 mm or 3 % of the shell outside diameter, whichever is the larger, but need not be more than 100 mm.

For LPB, 50 % of the breathing space limits shall apply.



Key

1 Boundaries of areas supported by individual stays (see 10.2.7b))

2 To establish the area supported by bar stays or stay tubes in boundary rows, the boundary of the loaded area should terminate at the centre of the associated main circle

3 Main circles, diameter *b* (see 10.2)

4 Bar stays

5 Stay tubes

6 Termination of boundary areas where stay tubes are situated in the boundary rows only (see 10.2.7)

[a] Breathing spaces (see 10.1)

NOTE See also Figures 10.2-1 and 10.2-2

Figure 10.1-1 — Typical arrangement of end plate in a multitubular boiler

10.2 Stayed flat surfaces

10.2.1 General

Both end plates shall be adequately supported by using plain bar stays, stay tubes, gusset stays, or a combination of these.

NOTE In order to keep the thickness of the end plate as small as possible it is recommended that an appropriate number of stays be provided.

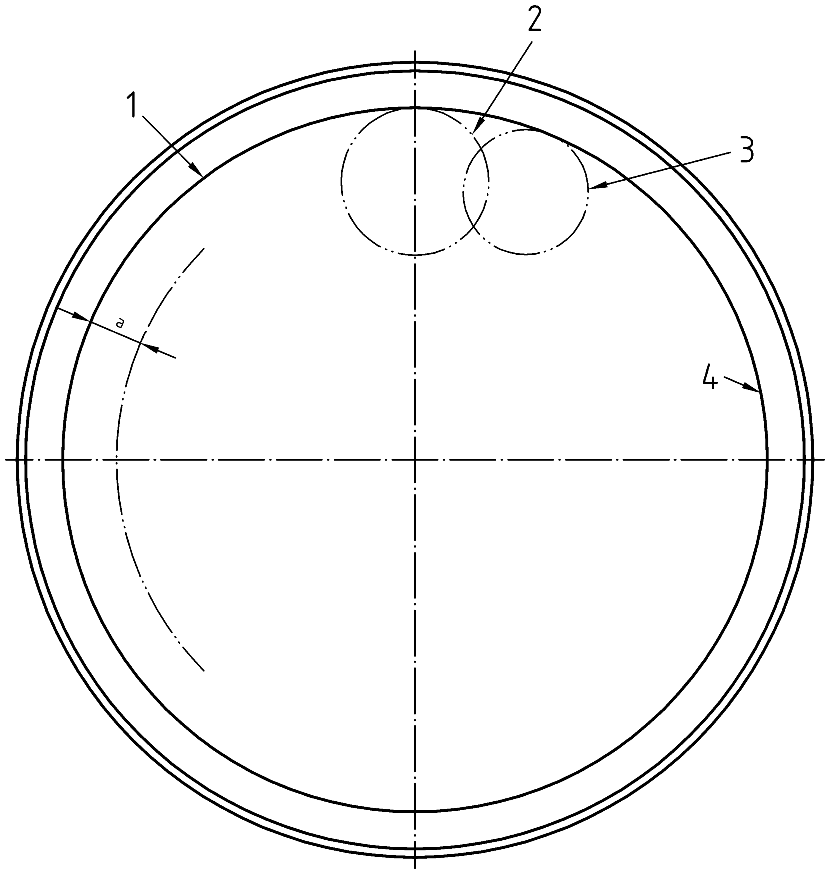
10.2.2 Radius of flange

Where flat ends are flanged, the inside radius of flanging shall be at least 1,3 times the thickness of the plate, but not less than 30 mm.

10.2.3 Point of support

Where the flange curvature is a point of support, the point of support shall be taken at half the distance between the inside of the shell and the commencement of curvature, or at a line 3,5 times the thickness of the plate measured from the outside for the plate, whichever is nearer to the flange (see Figure 10.2-1). Where a flat plate is welded directly to a shell or wrapper plate, the point of support shall be taken at the inside of the shell or wrapper plate.

The point of support shall be taken at half the distance between the inside of the shell and the commencement of curvature, or at a line 3,5 times the thickness of the plate measured from the outside of the plate, whichever is nearer to the flange (see 10.2.2).



Key

1 Point of support
2 Main circle
3 Sub-circle
4 Outer limit

a Breathing space

Figure 10.2-1 — Outer limits for supported areas, breathing spaces, main circles and sub-circles in flanged end plates

10.2.4 Thickness

The thickness of those portions of flat plates supported by stays shall be determined from the following equations:

$$e_{pa}' = e_{ch} + c_1 + c_2 \quad (10.2\text{-}1)$$

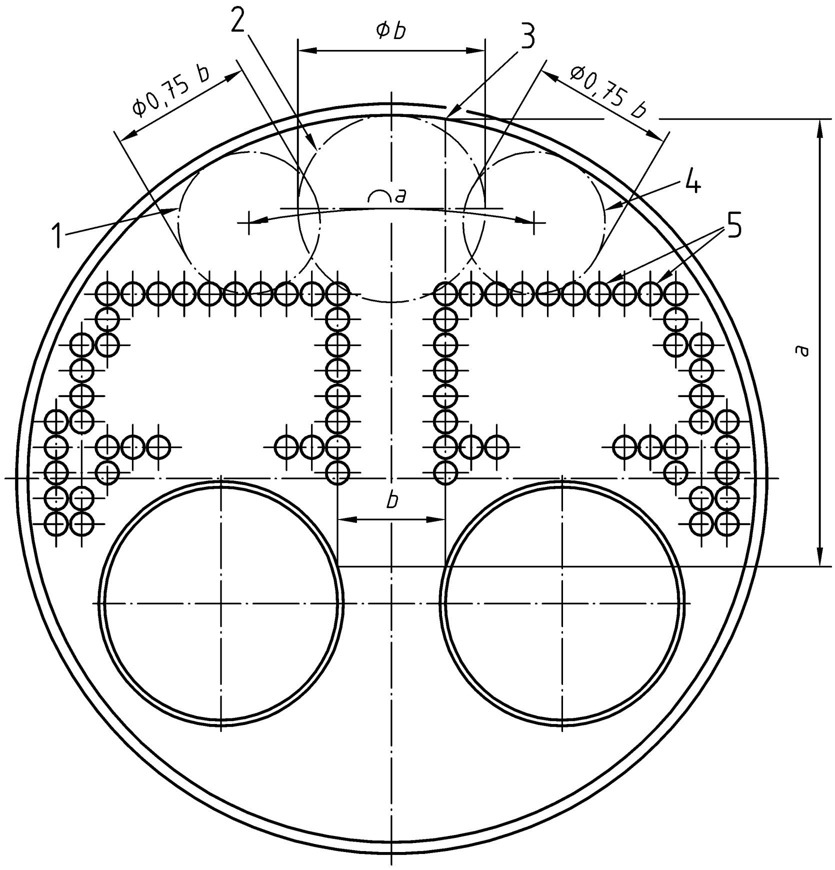
$$e_{ch} = c_4\, b\, y \sqrt{\frac{p_c}{f}} \quad (10.2\text{-}2)$$

For areas enclosed by circles which pass through four or more evenly distributed points of support, *y* shall be taken as 1.

For areas enclosed by circles which pass through three points of support (such that the centre of a sub-circle which passes through at least two points of support in accordance with Figure 10.2-2 or -3 with a diameter equal to 0,75 times the diameter of the main circle lies outside the main circle) y, taken from Figure 10.2-4, shall be determined using dimensions *a* and b as indicated in Figure 10.2-2 or -3. Where the main circle passes through three points of support, not more than two of them shall lie on one side of any diameter. In this case y shall be taken as not less than 1,1. For annular areas, e.g. areas supported only by shell and uptake, y shall be taken as 1,56.

For unstayed areas of rectangular shape, the dimensions *a* and b shall be as indicated in Figure 10.2-2.

Two or three adjacent stay tubes can be considered as one point of support, in which case the circle shall pass through the geometrical centre of the stay tube (see Figure 10.2-3)

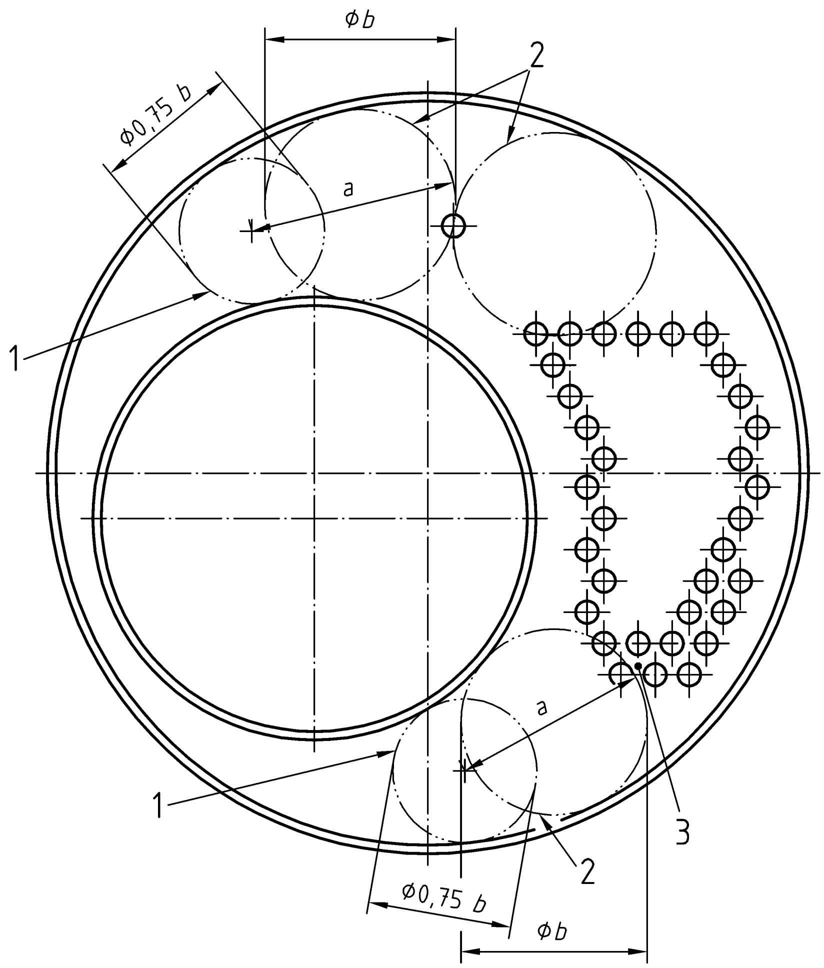


Key

1 Sub-circle
2 Main circle, diameter *b*
3 Unstayed rectangular area (see 10.2.3)
4 Sub-circle
5 Stay tubes

NOTE See also Figure 10.2-1.

Figure 10.2-2 — Use of sub-circles (twin furnace tube boiler)

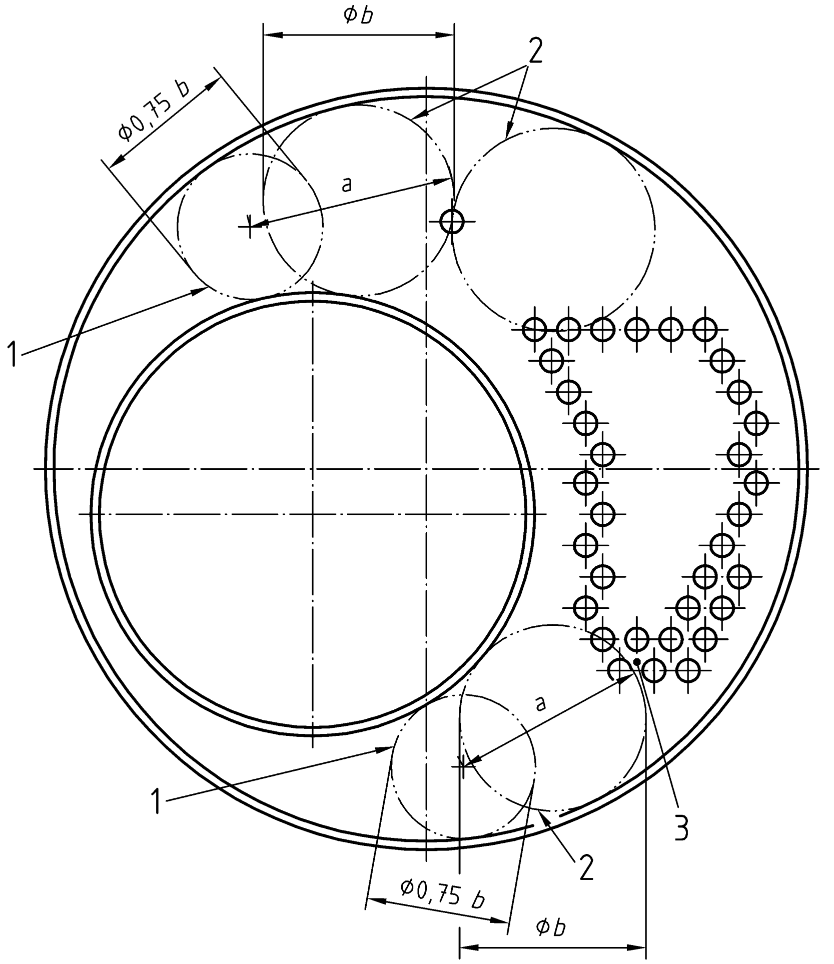


Key

1 Sub-circle
2 Main circles
3 Geometrical centre of the stay tube

NOTE See also Figure 10.2-1.

Figure 10.2-3 — Use of sub-circles (single furnace tube boiler)



Key

1 Rectangular areas

2 Elliptical areas

NOTE See 10.2.4

Figure 10.2-4 - Determination of factor y

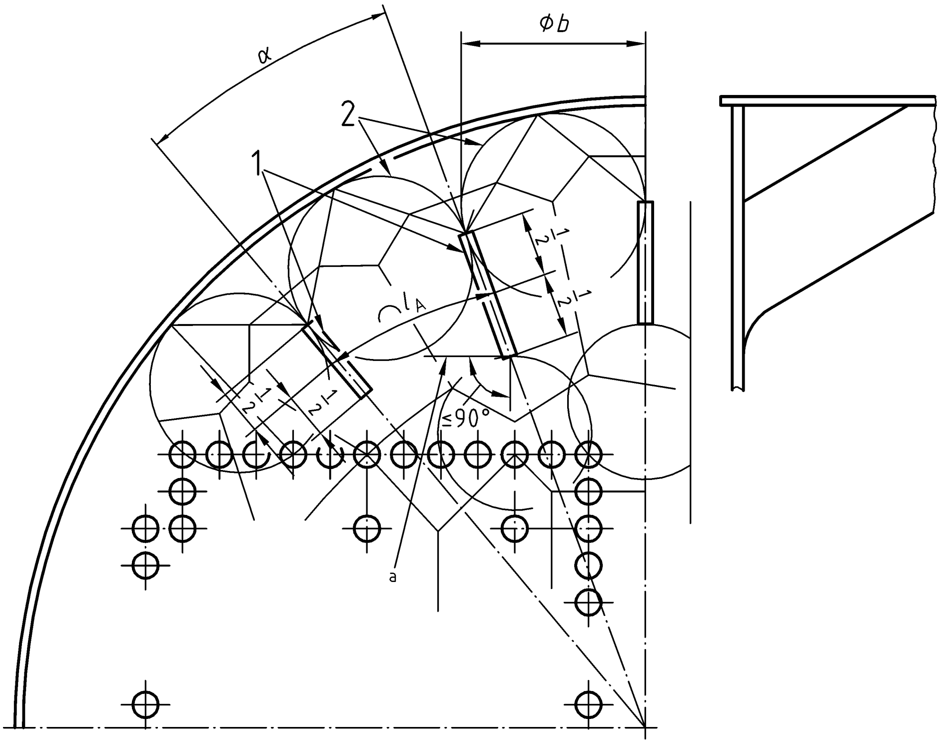
10.2.5 Values of constant C_4

Where various forms of support apply to the portion of flat plate under consideration, the constant C_4 shall be the mean of the values for the respective methods adopted. End plates, set on to the shell, shall not be permitted.

The values of constant C_4 in equation (10.2-2) shall be in accordance with Table 10.2-1.

Table 10.2-1 — Values for constant C_4

Component	Values for C_4
Freely supported plates with locking devices, e.g. manhole covers	0,45
Inset flat end plates, full penetration welded from one side	0,45
Plates which are bolted along their circumference:	
when the ratio $D_L/D_b = 1$	0,45
when the ratio $D_L/D_b = 1,3$	0,6
When the ratio $D_L/D_b = 1$ and 1,3, the values of the constant shall be determined by linear interpolation.	
Flanged end plate	0,32
Gusset stays or bar stays	0,3
Gusset stays in which the angle Θ is greater than 30° (see Figure 10.2-5)	0,45
Unstayed tube nest with plain tubes welded at both ends	0,3
Plain furnaces tubes less than 6 m long	0,3
Plain furnaces tubes 6 m to 8 m long	0,32
Corrugated furnaces tubes with corrugation depths $\leq$ 50 mm	0,32
Corrugated furnaces tubes with corrugation depths $>$ 50 mm:	
with a length $\leq$ 6 m	0,35
with a length $>$ 6 m	0,37
Bowling hoop furnaces tubes	0,35
Isolated plain bar stays (see Figure 10.2-6) or isolated stay tube (see Figure 10.2-7)	0,45
Non-isolated plain bar stays (see Figure 10.2-6) or non-isolated stay tubes (see Figure 10.2-7)	0,39
Bar stays or stay tubes shall be considered isolated if there are less than three in a group outside the tube nest.	
Bar stays with washers (see Figure 10.2-8a) and –8b))	0,35
Bar stays with washers (see Figure 10.2-8c) and –8d))	0,33
Reversal chamber bar stays and stay tubes (see Figure 10.2-9)	0,39
Reversal chamber access openings welded from both sides	0,3
Reversal chamber access opening (where it is not possible to effect back weld) (see Figure 10.2-10)	0,45
Flat unflanged end plates welded to the shell from both sides with the following end plate thickness to shell plate thickness ratios e_h/e_s:	
$\leq$ 1,4	0,33
$>$ 1,4 $\leq$ 1,6	0,36
$>$ 1,6 $\leq$ 1,8	0,39
$>$ 1,8 $\leq$ 2,0	0,42
For set-on plates, the thickness of the welding to the end shall be not less than 1,5 times the required minimum wall thickness e_{cs} of the shell using a weld coefficient of 1,0 (see figures in EN 12953-4).	
Flat surface of reversal chamber reinforced by stiffeners continuously welded above or flush stiffeners equipped with waterways (see Figure 10.2-11a) to 10.2-11e)) and with a stiffener height of between six and eight times its thickness.	0,4
Portion of unheated end plates containing a manhole with reinforcing ring (see Figure 11.3-1), when the distance from the edge of the manhole ring to the edges of the furnace tube, smoke tubes or shell is not more than four times the end plate thickness (see Figure 10.2-12)); the following single value shall be used.	0,27
If the distance exceeds four times the end plate thickness, the manhole is ignored, and the constant C_4 shall be determined in the normal manner, from the mean of the values for the respective methods of attachment given above.	



Key
1 Gusset stays
2 Main circles

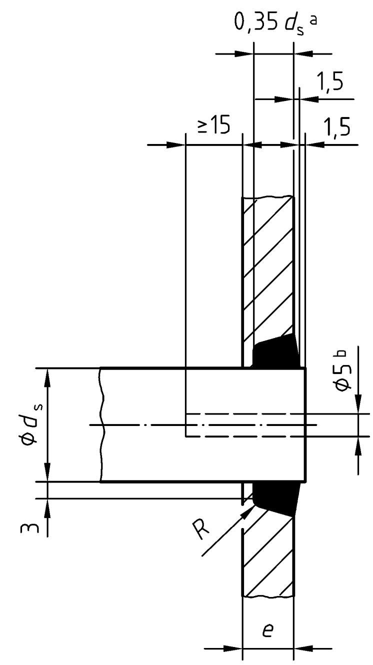
l_A Distance between centre lines of stays

[a] Tangent to inner or lower circle (see 10.2.4)

NOTE See 10.2.9

Figure 10.2-5 — Example of gusset stays

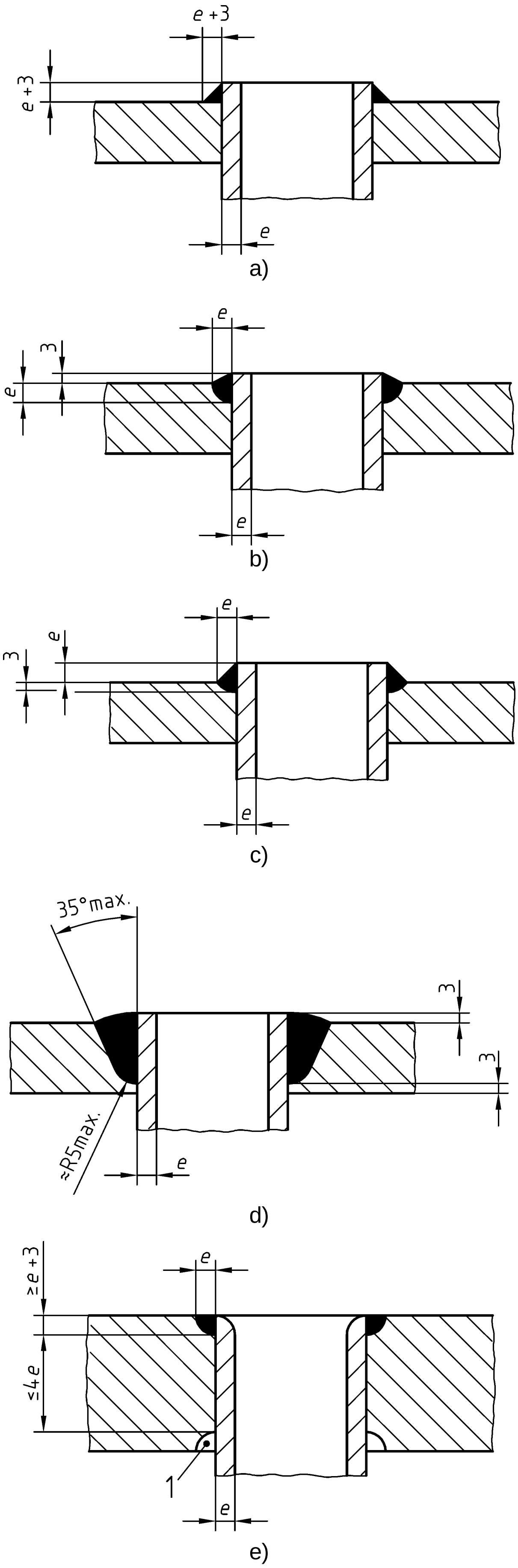
Dimensions in millimetres



[a] with a minimum of 8 mm
[b] nominal

Figure 10.2-6 — Permitted weld details of plain bar stay

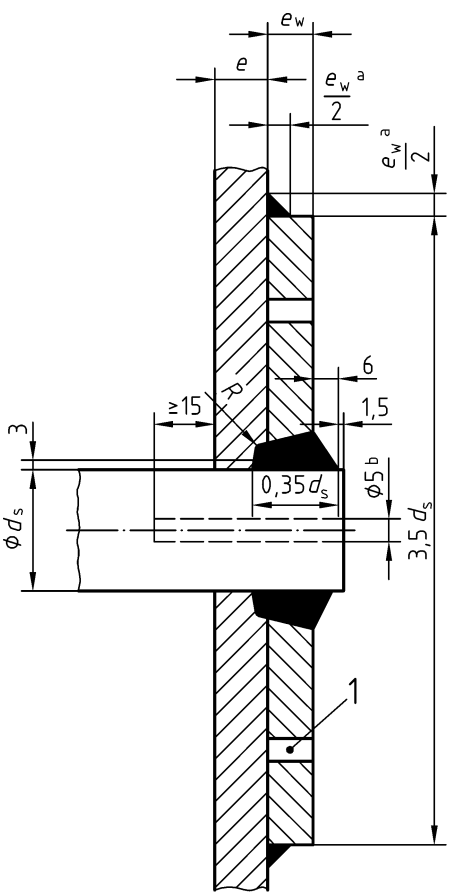
Dimensions in millimetres

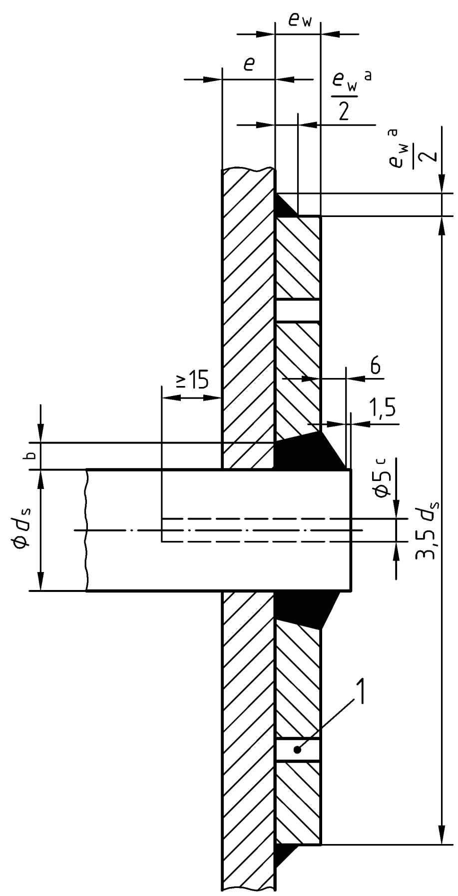


Key
1 Cooling groove, no weld

Figure 10.2-7 — Permitted weld details of stay tubes

Dimensions in millimetres





Key

1 Tell-tale hole

a with a minimum of 10 mm

b nominal

NOTE $e_w = \frac{2}{3} e$

a)

Key

1 Tell-tale hole

a with a minimum of 10 mm

b 0,15 d_s or 8 mm whichever is the greater

c nominal

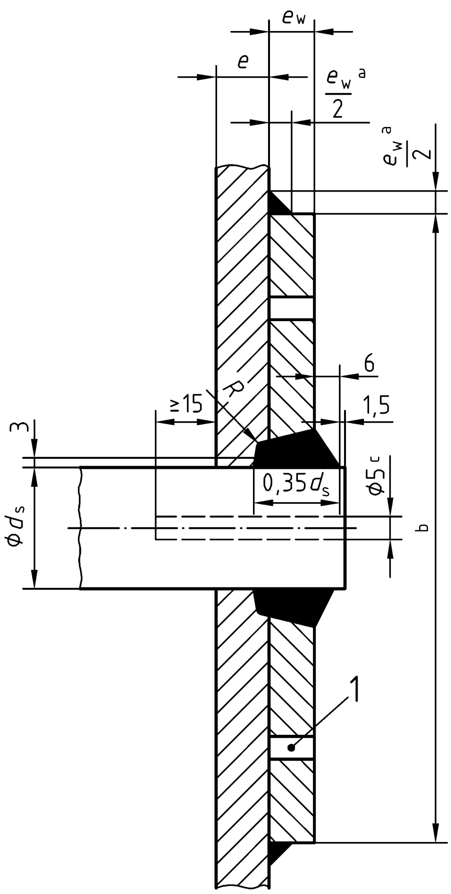
NOTE 1 $e_w = \frac{2}{3} e$

NOTE 2 Where e_w is less than 0,35 d_s, the form of construction shown in a) shall be used.

b)

Figure 10.2-8 — Permitted weld details of bar stays with washers

Dimensions in millimetres



Key

1 Tell-tale hole

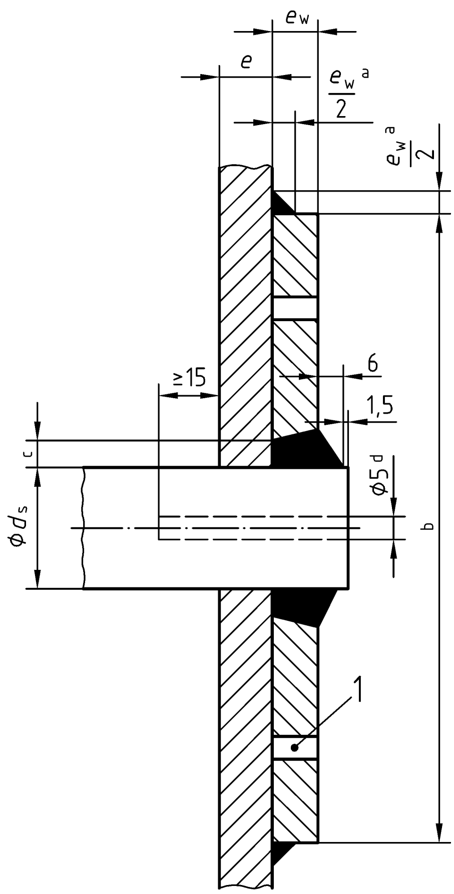
a with a minimum of 10 mm

b $^2/_3$ pitch of stays

c nominal

NOTE $e_w = \frac{2}{3} e$

c)



Key

1 Tell-tale hole

a with a minimum of 10 mm

b $^2/_3$ pitch of stays

c 0,15 d_s or 8 mm whichever is the greater

d nominal

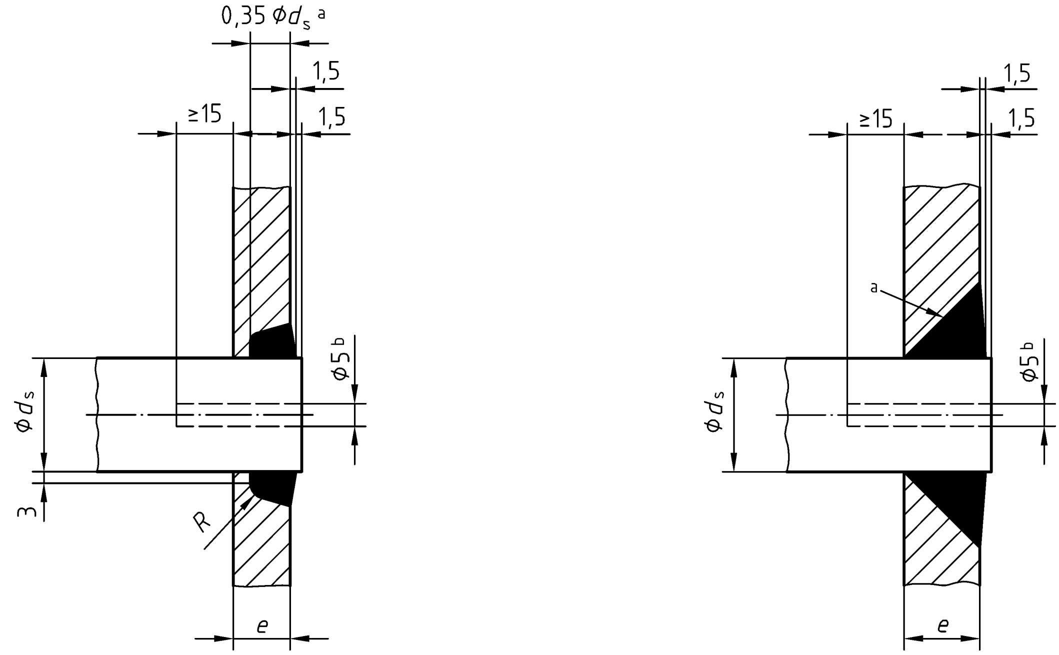
NOTE 1 $e_w = \frac{2}{3} e$

NOTE 2 Where e_w is less than 0,35 d_s, the form of construction shown in c) shall be used.

d)

Figure 10.2-8 — Permitted weld details of bar stays with washers *(concluded)*

Dimensions in millimetres



Key

a with a minimum of 8 mm

b nominal

a)

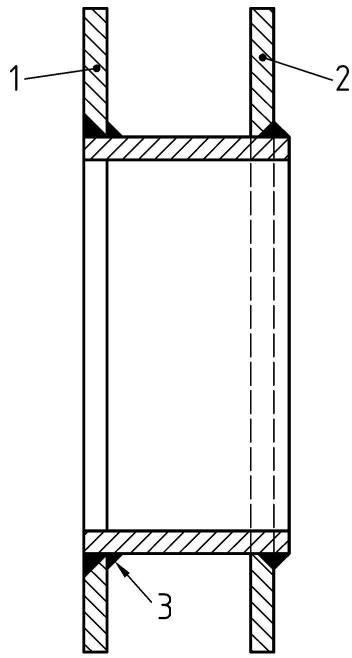
Key

a 45° weld preparation

b nominal

b)

Figure 10.2-9 — Permitted weld details of reversal chamber bar stays



Key

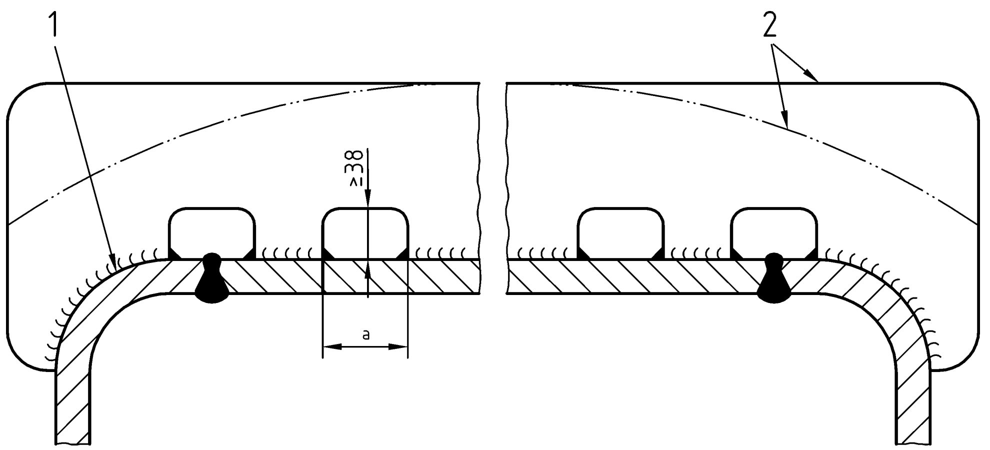
1 Combustion chamber plate

2 Back end plate

3 Seal weld

Figure 10.2-10 — Access opening for wet back boilers

Dimension in millimetres



Key

1 May be welded for fixed attachment

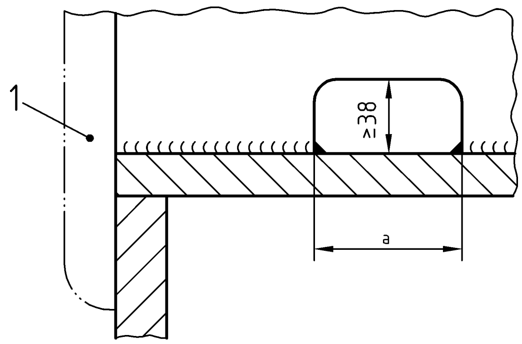
2 Alternative shape of grinder (see the note)

a Width of waterway

NOTE Girders may be shaped to either the full or the thin chain line shown.

a) Method of welding girder to a reversal chamber with flanged ends

Dimension in millimetres



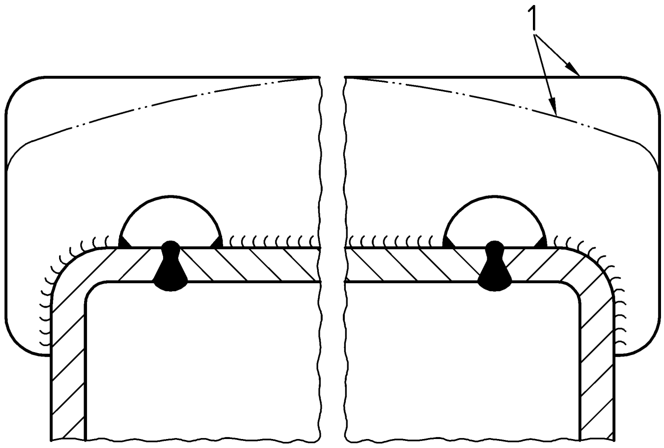
Key

1 Girder may be carried over the ends of the reversal chamber

a Width of waterway

b) Method of welding girder to a reversal chamber with flat ends

Figure 10.2-11 — Typical methods of welding girder stays to reversal chambers

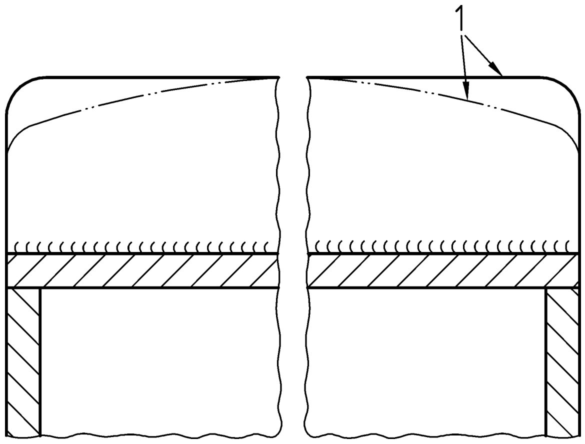


Key
1 Alternative shape of girder

NOTE Girders may be shaped to either the full or the thin chain line shown.

c) Welded-on girder to a reversal chamber having flanged tube plate and back plate

Figure 10.2-11 — Typical methods of welding girder stays to reversal chambers *(continued)*

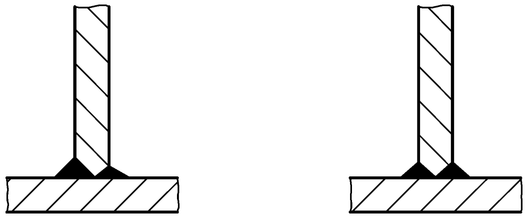


Key

1 Alternative shape of girder

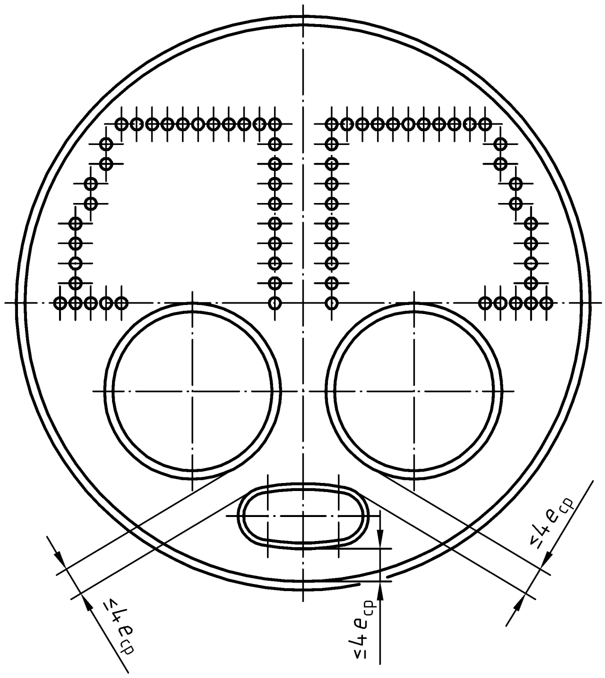
NOTE Girders may be shaped to either the full or the thin chain line shown.

d) Welded-on girder to a reversal chamber having square corners

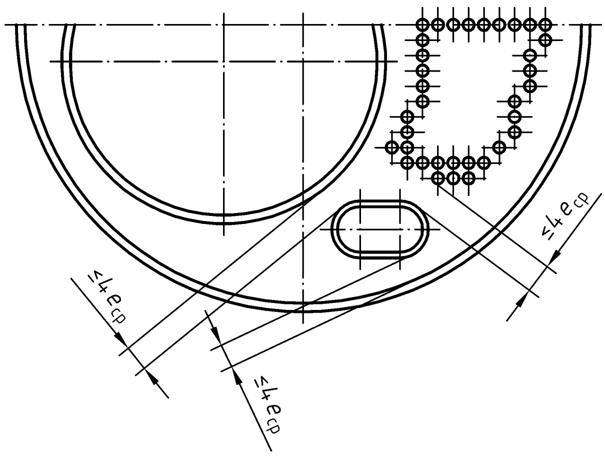


e) Alternative methods of welding girders to the reversal chamber top

Figure 10.2-11 — Typical methods of welding girder stays to reversal chambers *(concluded)*



a) Distance between manhole reinforcing ring, furnace tube and shell



b) Distance between manhole reinforcing ring, furnace tube, tube nest and shell

NOTE See Table 10.2-1

Figure 10.2-12 — Distances from manhole reinforcing ring

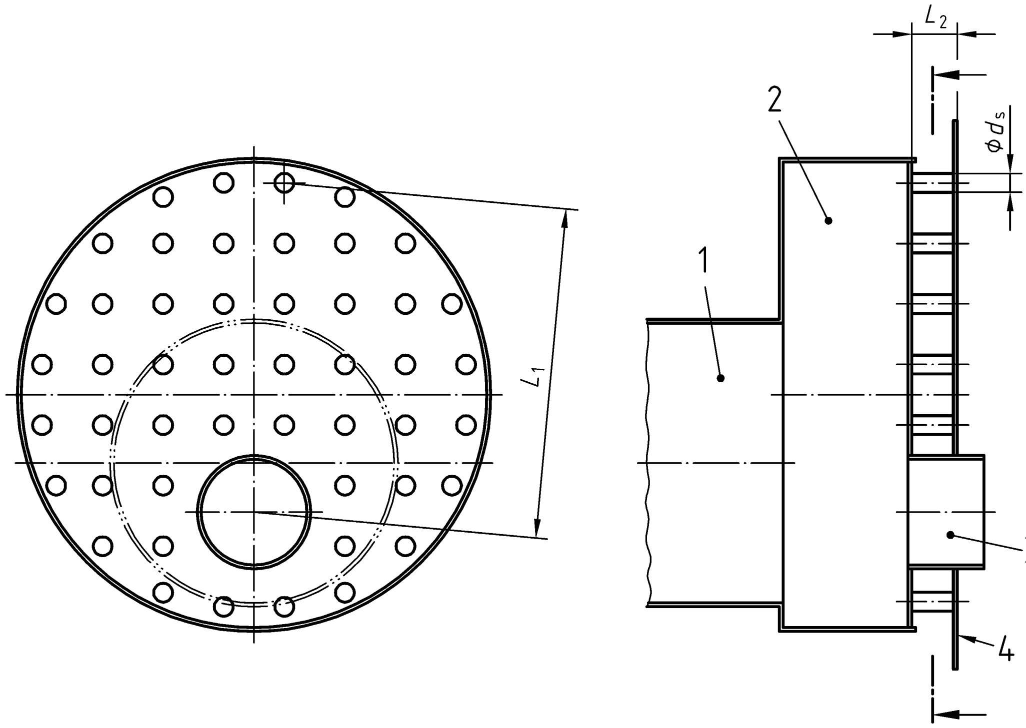
10.2.6 Stays for wet back reversal chambers

The permissible stress in the stays (solid or otherwise) calculated on the cross sectional area shall not exceed 80 N/mm^2. The diameter of any stay shall be not less than 20 mm.

The stays shall comply with the following rule (see Figure 10.2-13):

$$\frac{d_s\ L_1}{L_2^2} \leq 2 \qquad (10.2\text{-}3)$$

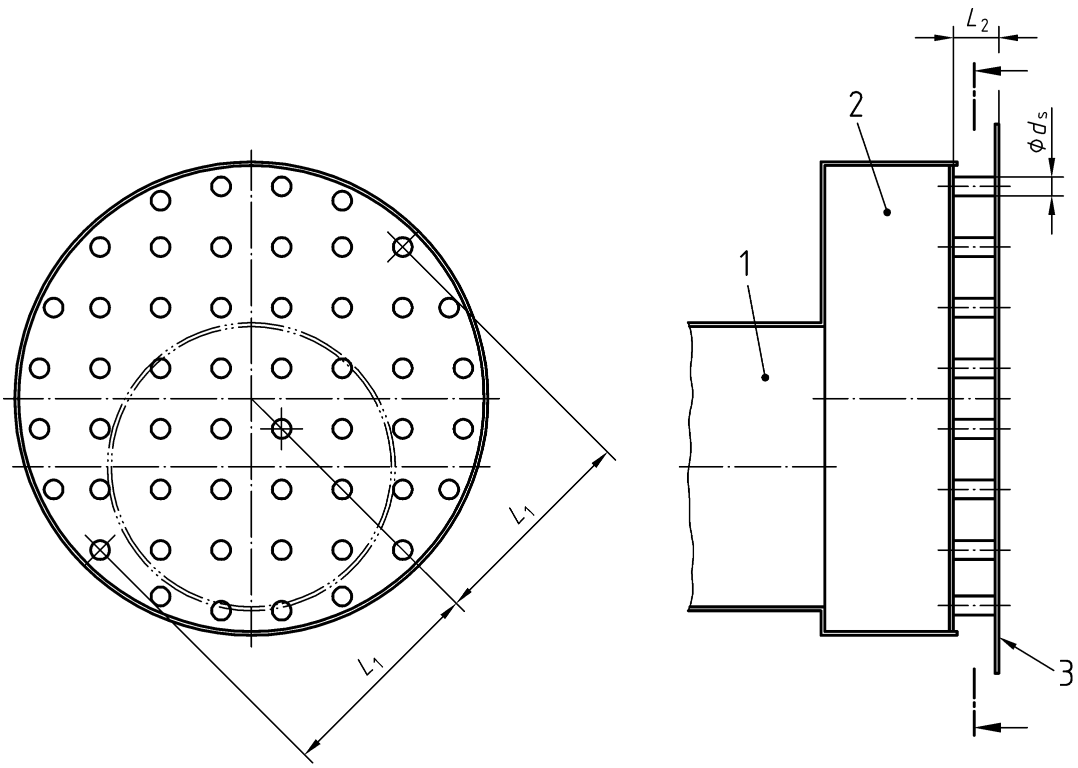
The stays may be welded as indicated in Figure 10.2-9 or with an alternative attachment provided the weld shear cross sectional area shall be at least 1,25 times the cross sectional area of the stay.



Key

1 Furnace tube
2 Reversal chamber
3 Access opening
4 Endplate

a)



Key

1 Furnace tube
2 Reversal chamber
3 Endplate

b)

Figure 10.2-13 — Location of stays in reversal chamber back plates

10.2.7 Longitudinal bar stays

The permissible stress in the stay (solid or otherwise), calculated on the cross sectional area, shall not exceed 80 N/mm^2. The diameter of any stay at any part shall be not less than 25 mm.

The stays may be welded as indicated in Figure 10.2-6 or 10.2-8 or with an alternative attachment provided the weld shear cross sectional area shall be at least 1,25 times the cross sectional area of the stay.

10.2.8 Loads on stay tubes and bar stays

Stay tubes and bar stays shall be designed to carry the whole load due to the pressure on the area to be supported, the area being calculated as follows.

a) For a stay tube within the tube nest, the net area to be supported shall be the product of the horizontal and vertical pitches of the stay tubes less the area of the tube holes embraced. Where the pitch of the stay tubes is irregular, the area shall be taken as the square of the mean pitch of the stay tubes (i.e. the square of one-quarter of the sum of the four sides of any quadrilateral bounded by four adjacent stay tubes) less the area of the tube holes embraced.

b) For a stay tube in the boundary row, or for a bar stay, the area to be supported shall be the area enclosed by a line through the midpoints of the lines joining the stay and the adjacent point of support, less the area of any tubes or stays embraced as shown in Figure 10.1-1.

c) For a bar stay where there are no stay tubes in the tube nest, the area to be supported shall extend to the tangential boundary of the tube nest.

10.2.9 Gusset stays

10.2.9.1 Principals for staying

The supporting of flat ends using too few gusset stays could lead to unacceptable local deformation of the shell. Therefore, the total load shall be divided into a larger number of gusset stays. The segmental shaped areas of unflanged plates within the free upper space (e.g. the steam space in case of steam boilers) shall be supported by at least two gusset stays.

10.2.9.2 Load on each stay

Each gusset stay supporting the flat end plate of a boiler shall be designed to carry the whole load due to the pressure on the area it supports. The area supported by any one stay shall be obtained by considering the total area to be supported and dividing this area by boundary lines drawn between the stays and the adjacent points of support (furnace tube, boundary rows of tube nests or the shell). These boundary lines shall be at all points equidistant from the adjacent points of support in the area under consideration (see Figure 10.2-5).

10.2.9.3 Calculation of gusset stays

Gusset stays shall be so proportioned that the angle V (see Figures 10.2-14) shall be not less than 60°. The minimum cross-section of the gusset shall be determined in accordance with the following equation:

$$e_g \; h = \frac{f_G \; W}{f \; \sin \; V} \qquad (10.2\text{-}4)$$

where

$e_g \leq 1{,}5 \times$ shell plate thickness

and

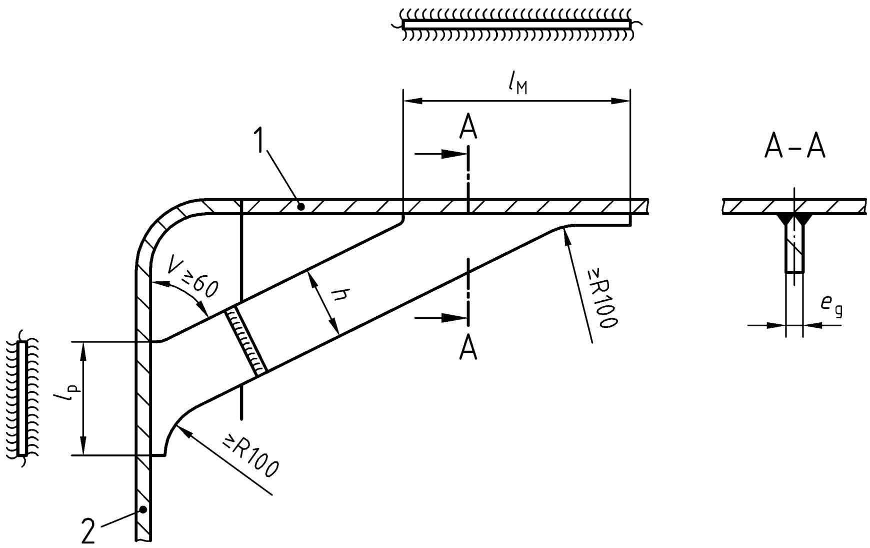
$e_g \geq 0{,}5 \times$ shell plate thickness

$e_g \geq 0{,}5 \times$ end plate thickness

However, the required minimum thickness of the gusset stay need not exceed the shell plate thickness.

The size and shape of the parts of the end plate supported by each gusset shall be such that the entire surface area of the end plate in each gusset stay zone is supported.

Dimensions in millimetres



Key
1 Shell
2 End plate

Figure 10.2-14 — Details of welded gusset stays

10.2.9.4 Weld attachments

Where gusset stays are welded to the shell and end plates, the attachment shall be by means of full penetration welds in accordance with Figure 10.2-14.

The weld profile shall be free from notches and abrupt changes of contour.

10.2.10 Additional requirements for unflanged flat end plates

10.2.10.1 General

In addition to the applicable requirements, especially in respect of acceptable weld details (see EN 12953-4), and to the general requirements for flat end plates given in 10.2.3 to 10.2.9, the requirements for welded-on or welded-in unflanged flat end plates given in 10.2.10.2 and 10.2.10.3 shall be taken into consideration.

10.2.10.2 Shell plate thickness local to the corner joint

For the determination of the shell plate thickness local to the T-butt weld, equations corresponding to equations (7.1-2) and (7.2-2) shall be used:

$$e_{s'} = e_{cs} + c_2 \qquad (10.2\text{-}5)$$

$$e_{cs} = \frac{p_c\, d_{os}}{2\, f_s\, x + p_c} \qquad (10.2\text{-}6)$$

The stress reduction factor *x* in equation (10.2-6) depends on the ratio of the end plate to shell plate thickness, and shall be:

$e_{ch}/e_{cs} \geq 1{,}4$ $x = 0{,}8$

$e_{ch}/e_{cs} \leq 1$ $x = 1$

If the shell thickness e_{cs} is calculated in accordance with equation (7.2-2) with a weld factor $v = 0{,}85$ the stress reduction factor x need not be considered.

For values of e_{ch}/e_{cs} between 1 and 1,4 the values of x shall be determined by linear interpolation.

10.2.10.3 Design parameters

Unflanged flat end plates shall comply with the parameters given in Table 10.2-2, and with the requirements given in a) to d) as follows:

a) The wall thickness of the shell shall be calculated in accordance with equation (10.2-5), including the stress reduction factor x (see equation (10.2-6));

b) The actual wall thickness of the end plate shall not exceed 30 mm;

c) When the wall thickness of the shell exceeds 30 mm, the through thickness properties of the material of the shell shall be proved by tests on the material properties;

d) Shell to end plate, furnace tube to end plate and reversal chamber end plate to wrapper plate seams shall be completely back welded, except in the case of small boilers as permitted in accordance with Table 10.2-3.

Table 10.2-2 — Design parameters for unflanged flat end plates

Shell inside diameter d_{is} mm	Length between boiler end plates [a] L_b mm	Maximum allowable pressure N/mm²	Ratio of end plate and shell thickness e_h/e_s
$d_{is} \leq 1\,500$	$L_b \leq 5\,500$	≤ 2	≤ 2
		> 2	$\leq 1{,}6$
	$5\,500 < L_b \leq 7\,000$	≤ 2	$\leq 1{,}8$
		> 2	$\leq 1{,}4$
$1\,500 < d_{is} \leq 1\,800$	$L_b \leq 5\,500$	$\leq 1{,}6$	$\leq 1{,}8$
		$> 1{,}6$	$\leq 1{,}4$
	$5\,500 < L_b \leq 7\,000$	$\leq 1{,}6$	$\leq 1{,}6$
		$> 1{,}6$	$\leq 1{,}2$
$1\,800 < d_{is} \leq 2\,500$	$L_b \leq 5\,500$	$\leq 1{,}6$	$\leq 1{,}7$
		$> 1{,}6$	$\leq 1{,}3$
	$5\,500 < L_b \leq 7\,000$	$\leq 1{,}6$	$\leq 1{,}5$
		$> 1{,}6$	$\leq 1{,}1$
$d_{is} > 2\,500$	$L_b \leq 5\,500$	$\leq 1{,}6$	$\leq 1{,}6$
		$> 1{,}6$	$\leq 1{,}2$
	$5\,500 < L_b \leq 7\,000$	$\leq 1{,}6$	$\leq 1{,}4$
		$> 1{,}6$	≤ 1

[a] The 7 000 limitation for the length does not apply to tubular waste heat boilers.

Table 10.2-3 — Conditions for omitting sections of fillet welds (back welds) from corner joints of flat end plates

Unwelded length mm	Boiler length between end plates L_b mm	Outside diameter of shell d_0 mm	Minimum breathing space between furnace tube and shell mm	End plate thickness e_{rh} mm	Ratio of end plate to furnace tube wall thickness e_{rh}/e_{rf}	General requirements for welded factions
≤ 250	≤ 3 000	≤ 1 400	6,5 % of d_0 or 65 mm, whichever is the greater	≤ 20	≥ 1,4	The weld shows full penetration. [a] Thorough inspection of the weld is possible. The weld is not heated directly.
> 250	≤ 2 500	≤ 1 000	≥ 65	≤ 15		
[b]	≤ 2 000	≤ 1 200	≥ 80	≤ 20		

[a] To be proved by special procedure tests. The procedure test piece shall reproduce the geometry of the production weld and shall be sectioned for visual and macro examination.

[b] One length equal to the furnace tube diameter for the flat end to furnace tube connection and one length equal to the shell diameter for the flat end to shell connection.

For LPB the fillet welds (back welds) may be omitted.

10.2.11 Girder stays supporting the flat section of a reversal chamber

The thickness of welded-on girders *e* in accordance with Figures 10.2-11a) to 10.2-11e) shall be calculated in accordance with the following equation, but in no case shall the thickness exceed 35 mm:

$$e = \frac{3\, p_c\, L_g^2\, P_g}{4\, d_g^2\, f} \qquad (10.2\text{-}7)$$

11 Design of isolated openings in shell boiler flat end plates

11.1 Unreinforced isolated openings

The maximum diameter of an unreinforced opening in a flat end plate shall be determined from the equation:

$$d_{max} = 8\, e_{rh} \left(1{,}5\, \frac{e_{rh}^2}{e_{ch}^2} - 1 \right) \qquad (11.1\text{-}1)$$

11.2 Branch openings

Reinforcement of branch openings shall be achieved by taking account of locally disposed material, including the attachment welds, in excess of the minimum requirements for end plate and branch thickness as shown in Figure 11.2-1. The branch thickness shall be increased where required. Compensation shall be considered adequate when the compensating area Y is equal to or greater than the area X requiring compensation.

Area X shall be obtained by multiplying 25 % of the inside radius of the branch by the thickness of the flat end plate, calculated from equation (10.2-2) for the part of the end plate under consideration.

Area Y shall be measured in a plane through the axis of the branch parallel to the surface of the flat end plate, and shall be calculated as follows:

a) For that part of the branch which projects outside the boiler, calculate the full sectional area of the branch up to a distance l_b from the actual outer surface of the flat end plate and deduct from it the sectional area that the branch would have within the same distance if its thickness were calculated in accordance with equations (7.3-1) and (7.3-2) taking $v = 1$;

b) Add to it the full sectional area of that part of the branch that projects inside the boiler (if any) up to a distance l_b from the inside surface of the flat end plate;

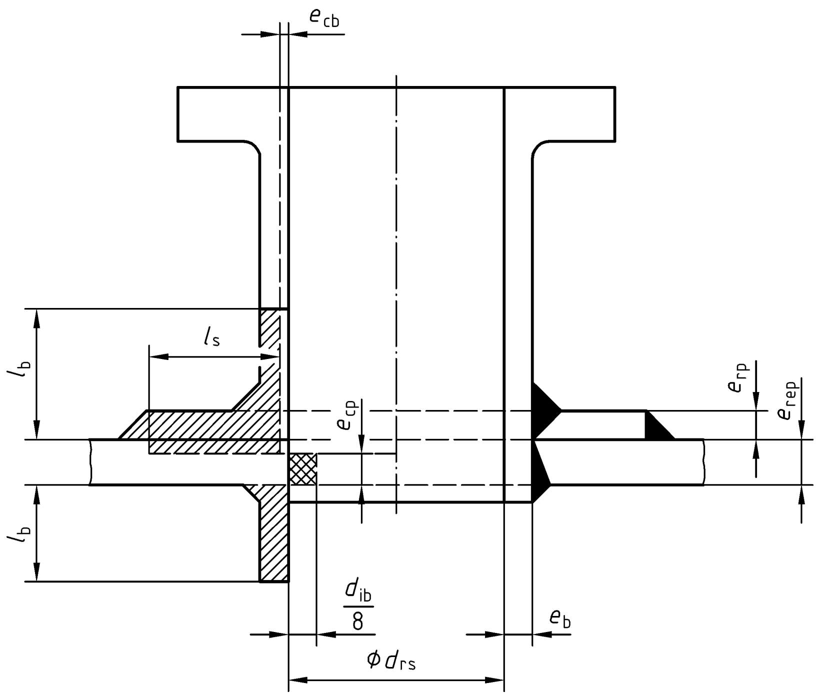
c) Add to it the sectional area of the fillet welds;

d) Add to it the area obtained by multiplying the difference between the actual flat end plate thickness and its thickness calculated from equation (10.2-2) for the part of the end plate under consideration by the length l_s ;

e) Add to it the area of the compensating plate (if any) within the limits of reinforcement shown in Figure 11.2-1.

Where material having a lower allowable stress than that of the flat end plate is taken as compensation, its effective area shall be reduced in the ratio of the allowable stresses at the calculation temperature. No credit shall be taken for the additional strength of material having a higher allowable stress than that of the flat end plate.

Welds attaching branches and compensating plates shall be capable of transmitting the full strength of the reinforcing area and all other loadings to which they may be subjected.



Key

Area X

Area Y

e_{cp} is the thickness calculated in accordance with equation (10.2-1) for the part under consideration.

e_{cb} is the thickness calculated in accordance with equation (7.3-2a) taking $v = 1$.

l_b is the smaller of the two values $2,5\ e_{rep}$ and $(2,5\ e_b + e_{rp})$.

l_s is the greater of the two values $(e_{rep} + 75)$ and $(d_{ib}/4)$.

Area Y shall not be less than area X.

NOTE The compensating plate is required only in cases where area Y would otherwise be less than area X.

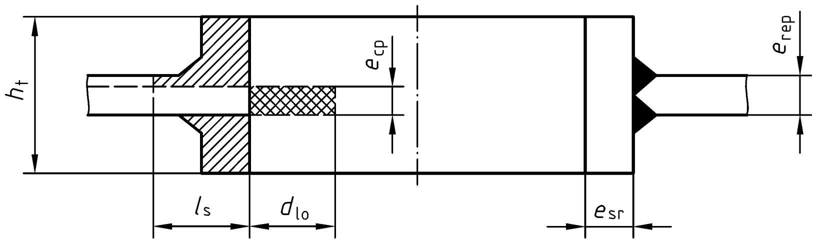
Figure 11.2-1 — Compensation for branch in flat end plate

11.3 Manholes, headholes and handholes

When elliptical manholes, headholes or handholes are located in flat end plates, the openings shall be compensated (see Figure 11.3-1). The method given in 11.2 for calculating the required area of reinforcement shall be used where applicable except that the thickness of the stiffening ring e_{sr} shall be not less than 19 mm for manholes, 15 mm for headholes and 10 mm for handholes.

Area X shall be obtained by multiplying half the mean of the major and minor semi-axes of the opening by the thickness of the flat end plate, calculated from equation (10.2-2), for the part of the end plate under consideration.

The full thickness of the stiffening ring may be used when calculating area Y.



Key

Area X

Area Y

d_{i0} is half the mean of the major and minor semi-axes of the opening.

e_{cp} is the thickness calculated in accordance with equation (10.2-1) for the part under consideration.

e_{sr} is the thickness of the stiffening ring

h_t is the weighted average heat transfer coefficient

l_s is the greater of the two values (e_{rep} + 75) and d_{i0}.

Area Y shall not be less than area X.

Figure 11.3-1 — Compensation for elliptical manholes or inspection openings in flat end plates

12 Unpierced tubes and tube plates

12.1 Thickness of straight tubes subject to external pressure

The thickness of straight tubes ≤ 170 mm nominal outside diameter, subject to external pressure, shall be given by equation (12.1-1) or Table 12.1-1, whichever is the larger.

$$e = e_{ct} + c_1 + c_2 \qquad (12.1\text{-}1)$$

with

$$e_{ct} = \frac{p\, d_o}{1{,}6\ f} \qquad (12.1\text{-}2)$$

Table 12.1-1 — Lowest nominal thickness of tubes

Dimensions in millimetres

Nominal outside diameter	Lowest nominal thickness
$d_0 \leq$ 26,9	1,90
26,9 < $d_0 \leq$ 54,0	2,20
54,0 < $d_0 \leq$ 76,1	2,50
76,1 < $d_0 \leq$ 88,9	2,80
88,9 < $d_0 \leq$ 114,3	3,15
114,3 < $d_0 \leq$ 139,7	3,50
139,7 < $d_0 \leq$ 168,3	3,99

12.2 Thickness of straight tubes subject to internal pressure

The thickness of straight tube subject to internal pressure shall be given by equation (12.2-1) or Table 12.1-1, whichever is the larger:

$$e_{t'} = e_{ct} + c_1 + c_2 \qquad (12.2\text{-}1)$$

where

c_2 = 0,75 mm

$$e_{ct} = \frac{p_c\, d_o}{2\, f + p_c} \qquad (12.2\text{-}2)$$

For nozzles, see 8.3.

12.3 Wall thickness and ovality of elbows and tube bends

The wall thickness of elbows (see Figure 12.3-1), tube bends ≤ 170 mm nominal outside diameter shall be not less than that given by the following equations:

wall thickness at the intrados

$$e_{ti} = e_{ct}\, C_i + c_1 + c_2 \qquad (12.3\text{-}1)$$

wall thickness at the extrados

$$e_{to} = e_{ct}\, C_o + c_1 + c_2 \qquad (12.3\text{-}2)$$

where e_{ct} is the thickness calculated for a straight tube in accordance with 12.1 or 12.2 as appropriate, and C_i and C_o are factors to be taken from Figure 12.3-2 and equations (12.3-3) and (12.3-4).

$$C_i = \frac{\dfrac{2\,R}{d_o} - 0{,}5}{\dfrac{2\,R}{d_o} - 1} \qquad (12.3\text{-}3)$$

$$C_o = \frac{\dfrac{2\,R}{d_o} + 0{,}5}{\dfrac{2\,R}{d_o} + 1} \qquad (12.3\text{-}4)$$

where

R is the centre line of the bore of the bend.

The factors C_i and C_o shall be applicable for elbows and tube bends where the ratio R/d_o is greater than or equal to 1 and less than or equal to 4,5. Bends with R/d_o greater than 4,5 shall be treated as straight tubes.

Minimum wall thicknesses shall be taken into account in accordance with 12.1 or 12.2 as appropriate.

The value of the ratio d_m/e shall not exceed 40.

The departure from circularity u of tube bends given by:

$$u = \frac{2\left(\hat{d}_o - \breve{d}_o\right)}{\hat{d}_o + \breve{d}_o} \times 100 \qquad (12.3\text{-}5)$$

shall not exceed the limits given in EN 12953-4.

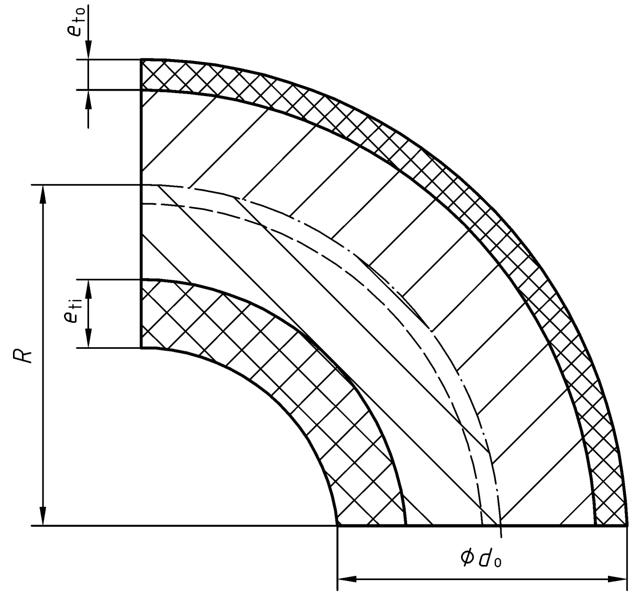


Figure 12.3-1 — Notation used for tube bends

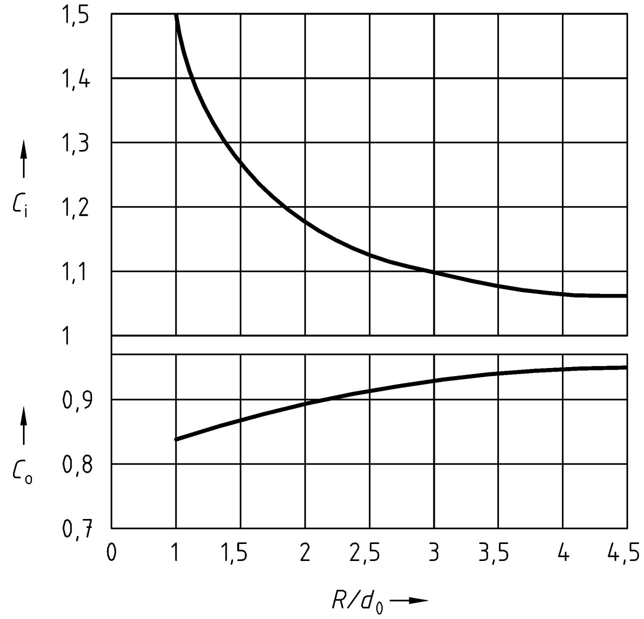


Figure 12.3-2 — Design factors C_i and C_o

12.4 Stay tubes

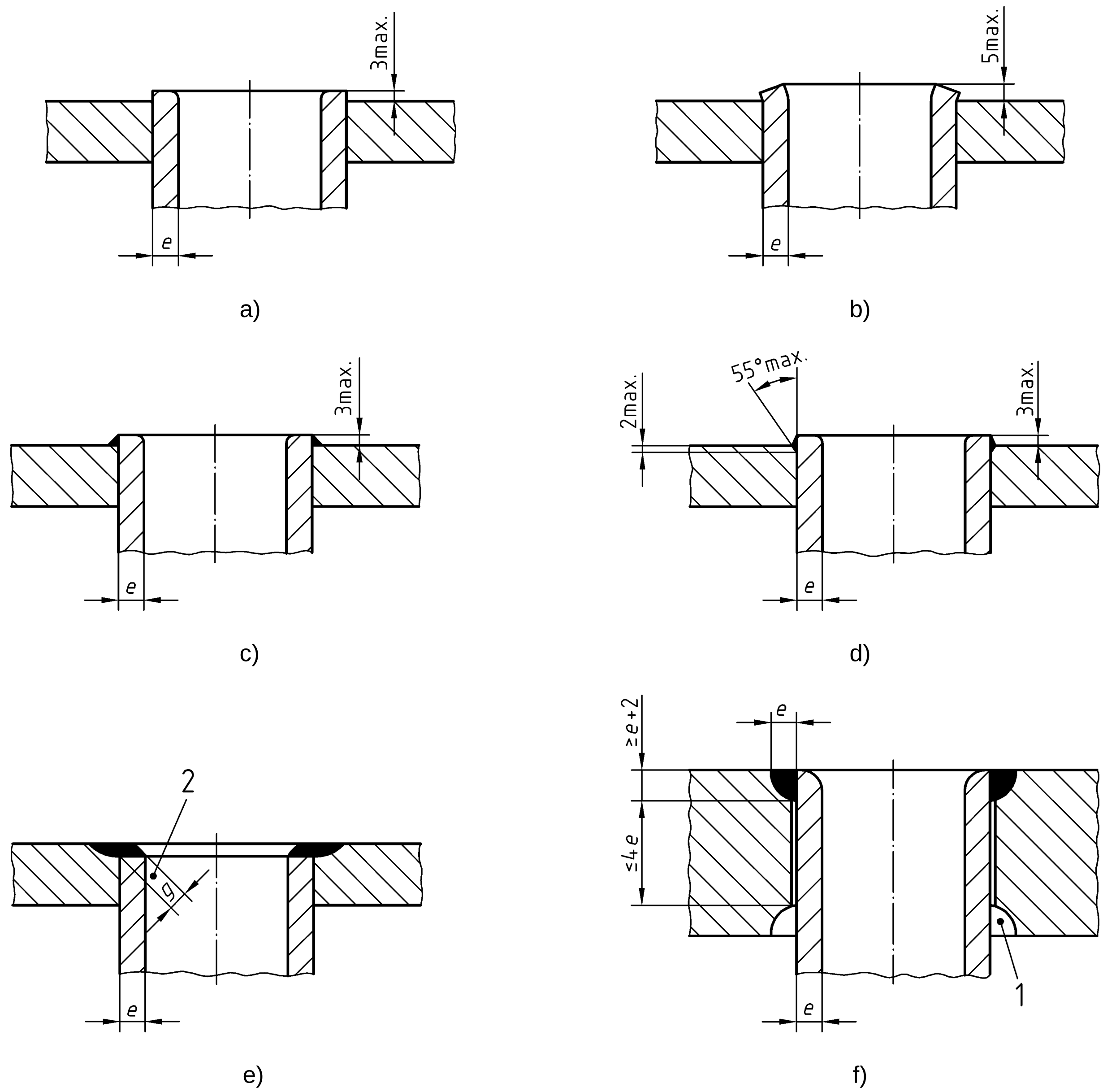
Stay tubes are tubes welded in accordance with Figure 10.2-7 having a weld depth equal to the tube thickness plus 3 mm, or the weld shear cross section shall be equal to or greater than 1,25 multiplied by the required cross sectional tube area, with a minimum weld depth of 3 mm. These stay tubes shall not be required within tube nests except when the tube nests comprise tubes which are expanded only (see Figure 12.4-1a)). In addition, these stay tubes shall not be required for small boilers with inside diameter ≤ 1 400 mm operating ≤ 12 bar maximum allowable working pressure and 0,75 MW heat input.

When the tube nests shall be comprised of plain tubes which are expanded and belled (see Figure 12.4-1 b)), or expanded and welded in accordance with Figures 12.4-1 c) to e), welded stay tubes in accordance with Figure 10.2-7 are required in the boundary rows to carry the flat plate loadings.

Where tube nests comprise plain tubes welded in accordance with Figure 12.4-1 f), or having a weld depth equal to the tube thickness plus 2 mm, stay tubes shall not be required.

Each stay tube shall be designed to carry its due proportion of the load on the plates which it supports. The thickness of stay tubes welded into tube plates shall be such that the axial stress on the thinnest part of the tube does not exceed 80 N/mm^2.

All dimensions in millimetres



Key

1 Cooling groove, no weld

2 $g \geq e$

Figure 12.4-1 — Permitted methods of attaching plain tube

12.5 Smoke tubes

If tube ends are exposed to flame or to a flue gas temperature of 600 °C and above at the inlet, or 700 °C and above at the outlet, the ends shall be flush to the welding or, if the tubes are expanded they shall be in accordance with Figure 12.4-1a) or 12.4-1b).

If the tube ends are exposed to a flue gas temperature below these limits, but higher than 500 °C, the tubes may stand out up to 18 mm from the tube plate.

At flue gas temperatures below 500 °C the tubes may stand out up to 25 mm from the tube plate.

12.6 Pitch of tubes

The spacing of tube holes shall be such that the minimum width, in millimetres, of any ligament between the tube holes shall be not less than:

a) for expanded tubes,

   0,125 $d + 12{,}5$ mm

b) for welded tubes,

   1) for gas entry temperatures greater than 800 °C,

      0,125 $d + 9$ mm, but need not exceed 15 mm;

   2) for gas entry temperatures less than or equal to 800 °C,

      0,125 $d + 7$ mm, but need not exceed 15 mm.

12.7 Thickness of the tube plates within tube nests

The thickness of tube plates shall be calculated from equations (10.2-1) and (10.2-2), but shall be not less than the following:

a) 12 mm where the tubes are expanded into the tube plate when the diameter of the tube hole does not exceed 50 mm, or 14 mm when the diameter of the tube hole is greater than 50 mm, or

b) 6 mm where the tubes are attached to the tube plate by welding only.

13 Furnaces tubes, furnace tube components and reversal chambers of cylindrical form subject to external pressure

13.1 Furnaces tubes

13.1.1 Plain furnaces tubes

The calculation pressure of plain furnaces tubes shall be the lower of the calculation pressures obtained using the following equations:

$$p_c = \frac{R_{p\,0,2\,tc}}{S_1} \frac{2\,e_{cf}}{d_m} \frac{1 + 0{,}1\,d_m / L}{1 + \left(0{,}03\,d_m / e_{cf}\right)\left[u / \left(1 + 5\,d_m / L\right)\right]} \quad (13.1\text{-}1)$$

$$p_c = \frac{2{,}6\,E}{S_2\,L} \left(\frac{e_{cf}}{d_m}\right)^2 \sqrt{d_m\,e_{cf}} \quad (13.1\text{-}2)$$

The preceding equations may be expressed in terms of thickness as shown in equations (13.1-3) and (13.1-5). The greater of the thicknesses obtained shall be used, but the thickness of plain furnaces tubes with diameters less than or equal to 400 mm shall be not less than 6 mm, and for diameters greater than 400 mm, shall be not less than 7 mm. Bowling hoop furnaces tubes shall have a minimum wall thickness of not less than 10 mm. In no case shall the thickness exceed 22 mm.

$$e_{cf} = \frac{B}{2}\left[1 + \sqrt{1 + \frac{0{,}12\ d_m\ u}{\left(1 + 5\ d_m / L\right) B}}\right] \qquad (13.1\text{-}3)$$

where

$$B = \frac{p_c\ d_m\ S_1}{2\ R_{p\ 0{,}2\ tc}\ \left(1 + 0{,}1\ d_m / L\right)} \qquad (13.1\text{-}4)$$

$$e_{cf} = d_m^{0{,}6}\ \left[\left(L\ S_2\ p_c\right)/\left(2{,}6\ E\right)\right]^{0{,}4} \qquad (13.1\text{-}5)$$

$$e_{fa} = e_{cf} + c_1 + c_2 \qquad (13.1\text{-}6)$$

where

c_2 = 0,75 mm (corrosion allowance)

S_1 and S_2 are safety factors (see 13.1.3).

Equations (13.1-1) to (13.1-6) shall apply to furnaces tubes having diameters ≤ 1 800 mm.

NOTE Equations (13.1-1) and (13.1-3) are based on considerations of plastic deformation. Equations (13.1-2) and (13.1-5) are based on considerations of elastic instability.

13.1.2 Corrugated furnaces tubes

The calculation pressure of corrugated furnaces tubes shall be determined using the following equation, but the nominal thickness shall be not less than 10 mm nor greater than 22 mm:

$$p_c = \frac{R_{p\ 0{,}2\ tc}}{S_1}\ \frac{2\ X_2}{P_{cor}\ d_m}\ \frac{1 + 0{,}1\ d_m / L}{1 + \left[\frac{X_2\ w\ d_m}{800\ I_1}\ \frac{u}{1 + \left(5\ d_m / L\right)\left(e_{cf} / w\right)^3}\right]} \qquad (13.1\text{-}7)$$

where

d_m is the mean diameter.

NOTE 1 For corrugated furnace tubes, the mean diameter is equal to the inside diameter plus the full depth of one corrugation, i.e. inside diameter plus w (see Figure 13.1-1).

NOTE 2 Values of X_2 and I_1 for Fox corrugations are given in Tables 13.1-1 and 13.1-2.

13.1.3 Safety factors

The value of the safety factor shall be:

S_1 = 2,5 for furnaces tubes and wrapper plates exposed to flame with $p_c > 0{,}6$ N/mm^2 or;
$p_c \leq 0{,}6$ N/mm^2 and $d_m/L < 0{,}25$

S_1 = 2,0 for furnaces tubes and wrapper plates exposed to flame with $p_c < 0{,}6$ N/mm^2 and $d_m/L \geq 0{,}25$;

S_1 = 2,0 for furnaces tubes and wrapper plates not exposed to flame;

S_2 = 3,0 for all conditions.

13.1.4 Furnace tube components

The thickness of the furnace tube components e.g. ash drop tubes and fuel inlet connections, shall be calculated in accordance with 13.1.1 with a minimum thickness of 10 mm and a maximum thickness of 22 mm.

Compensation for openings in furnaces tubes shall be provided in accordance with 8.3, except that the use of reinforcing pads shall not be permitted.

The thickness of access tubes shall be calculated in accordance with 13.1.1 with a minimum thickness of 10 mm.

13.1.5 Reversal chambers

The thickness of wrapper plates of cylindrical reversal chambers shall be calculated in accordance with the equations given in 13.1.1. Where non-circular geometry is employed using plates of different radii, the thickness shall be calculated using the maximum radius.

Where the use of reversed curvature sections is involved, a check shall be made that the sections can sustain the maximum allowable pressure without the design stress being exceeded and, if necessary, the thickness appropriately increased. A suggested method is given in Annex B.

The thickness shall be not less than 10 mm and not greater than 35 mm.

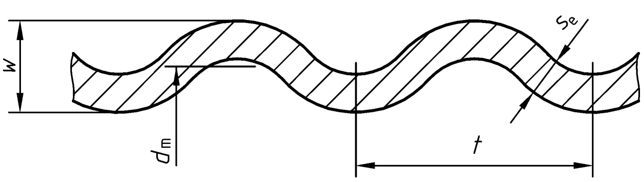


Figure 13.1-1 — Cross sectional area for Fox type corrugated tube

Table 13.1-1 — Second moments of area for Fox type corrugated tube (150 corrugation and 50 overall depth)

Wall thickness without corrosion allowances	Second moment of area	Cross sectional area
$e_{rf} - c$	I_1	X_2
mm	10^4 mm^4	10^2 mm^2
9,25	35,6	16,4
10,25	37,7	18,0
11,25	39,6	19,6
12,25	41,2	21,2
13,25	42,7	22,8
14,25	44,1	24,4
15,25	45,3	25,9
16,25	46,4	27,4
17,25	47,4	28,9
18,25	48,3	30,4
19,25	49,2	31,9
20,25	50,1	33,3
21,25	51,0	34,8

Table 13.1-2 — Second moments of area for Fox type corrugated tube (200 corrugation and 75 overall depth)

Wall thickness without corrosion allowances	Second moment of area	Cross sectional area
$e_{rf} - c$	I_1	X_2
mm	10^4 mm^4	10^2 mm^2
9,25	129,4	23,3
10,25	138,9	25,7
11,25	147,7	28,0
12,25	155,9	30,4
13,25	163,5	32,6
14,25	170,5	34,9
15,25	177,0	37,1
16,25	183,0	39,4
17,25	188,5	41,5
18,25	193,6	43,7
19,25	198,4	45,8
20,25	202,8	48,0
21,25	206,9	50,0

13.2 Calculation length of composite furnaces tubes

When the length of the plain portion of a corrugated tube exceeds 250 mm, the total length of both sections shall be used for calculating the thickness of the corrugated section, and 1,5 times the length of the plain section shall be used for calculating the thickness of the plain section.

13.3 Tolerances of furnaces tubes

For corrugated furnaces tubes, the calculated wall thickness shall be the minimum thickness of the finished furnaces tubes.

The departure from circularity *u* of furnaces tubes and reversal chambers shall be calculated as follows:

$$u = \frac{2\left(\hat{d} - \breve{d}\right)}{\hat{d} + \breve{d}} \times 100 \qquad (13.3\text{-}1)$$

where

$\hat{d}$ is the maximum mean diameter of the furnace tube;

$\breve{d}$ is the minimum mean diameter of the furnace tube.

For tolerances of furnaces tubes, see EN 12953-4.

The value of *u* in equations 13.1-1, 13.1-3 and 13.1-7 shall be taken as 1 % for corrugated furnaces tubes and 1,5 % for plain furnaces tubes.

13.4 Stiffeners

13.4.1 General

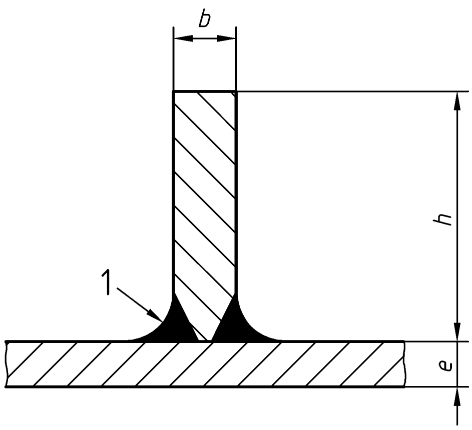
Stiffeners welded to furnaces tubes with the maximum dimensions in accordance with Figures 13.4-1 and 13.4-2 shall be regarded as fulfilling the requirements of this section and need not be calculated.

Stiffeners not conforming to Figures 13.4-1 and 13.4-2 shall have a second moment of area not less than that given by the following equation:

$$I_2 = \frac{p_c \, d_m^3 \, L}{1{,}33 \times 10^6} \qquad (13.4\text{-}1)$$

The second moment of area of the stiffener about its neutral axis I_2 shall be related to the stiffener section, including a length of furnace tube equal to $0{,}55\sqrt{d_m \, e_{cf}}$ on either side of the stiffener.

Dimensions in millimetres

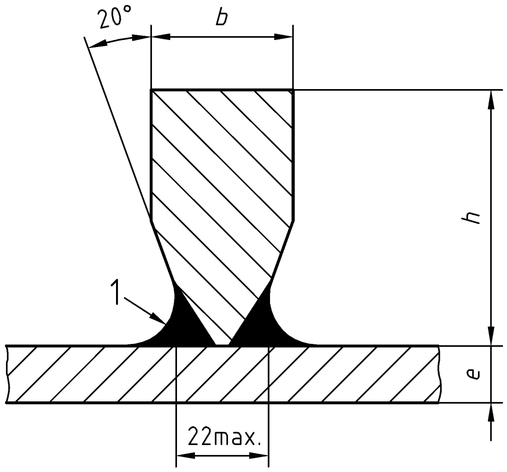


$2\,e \le b \le 22$
$5\,e \le h \le 6\,b$

Key
1 Continuous full penetration weld

Figure 13.4-1 — Furnace tube stiffeners up to and including 22 mm thick for plain and corrugated sections

Dimensions in millimetres



Key
1 Continuous full penetration weld

Figure 13.4-2 — Furnace tube stiffeners thicker than 22 mm for plain and corrugated sections

13.4.2 Stiffener sections made from bar or plate

Stiffener sections made from bar or plate shall be joined by full penetration welds.

The thickness of the stiffening ring shall be kept to the minimum required by 13.4.1. If it exceeds 22 mm, or twice the furnace tube thickness, it shall be tapered as shown in Figure 13.4-2.

Stiffening rings need not be manufactured from the same material as the furnace tube but shall have the same modulus of elasticity and coefficient of linear expansion as the furnace tube material. The stiffening ring materials shall be selected from those materials specified in EN 12953-2.

Full penetration welds shall be used to attach stiffeners to furnaces tubes.

13.4.3 Stiffeners located within the zone of peak heat flux

When stiffeners are welded on to furnaces tubes which are more than 11 mm thick they shall not be located in the zone of peak heat flux. The zone of peak heat flux shall be considered to extend for a length equal to two times the minimum required furnace tube diameters according to 5.4, from the tip of the burner, or to the end of the grate, whichever is applicable.

NOTE The requirements of 13.4.3 need not apply to boilers with a heat input less than 2 MW.

13.4.4 Bowling hoops

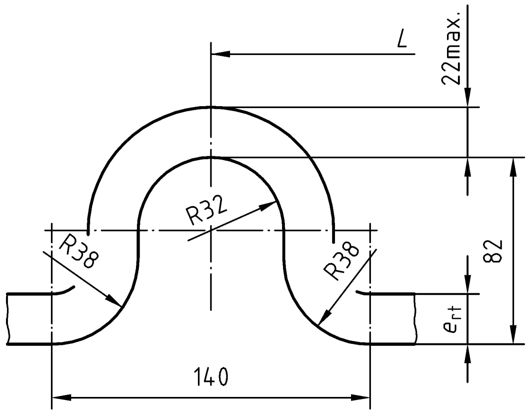
Bowling hoops shall be considered effective points of support. The minimum pitch of bowling hoop centres shall, for calculation purposes, be taken as one third of the inside diameter of the furnace tube, but not less than 500 mm. When bowling hoops are used, the furnace tube thickness shall be calculated in accordance with 13.1.1.

In the calculation for furnaces tubes supported by bowling hoops, L shall be taken as 1,5 times the actual length between bowling hoop centres.

The second moment of area of the bowling hoop shall be not less than that required by 13.4.1. The dimensions and the second moment of area of bowling hoops are given in the tables to Figures 13.4-3a) to -3c).

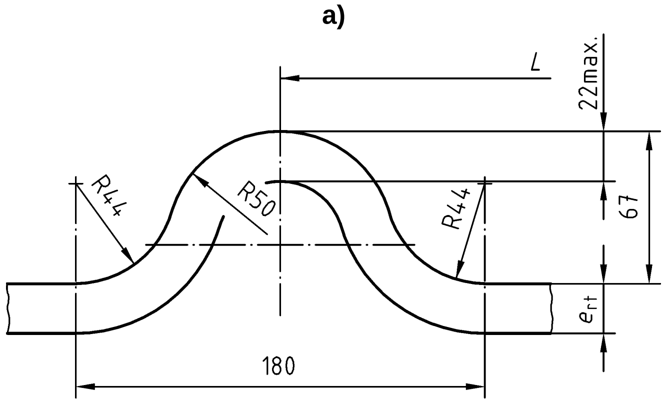
The nominal wall thickness of bowling hoops shall not be less than the nominal wall thickness of the plain furnaces tubes to which they are attached.

Dimensions in millimetres



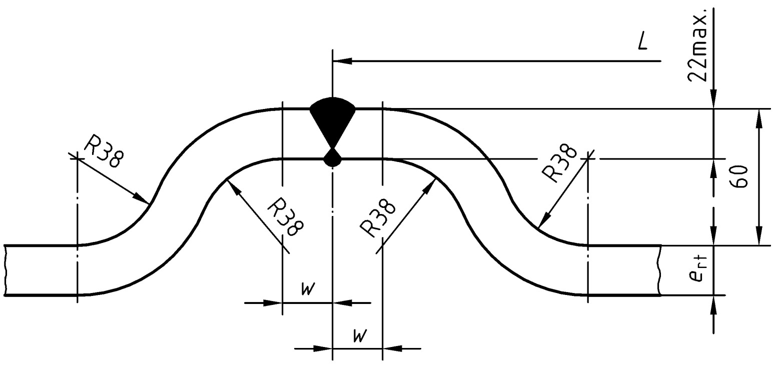
Wall thickness without corrosion allowances	Second moment of area
$e_{rt} - c$ mm	I_2 10^6 mm^4
9,25	1,90
10,25	2,11
11,25	2,32
12,25	2,53
13,25	2,74
14,25	2,96
15,25	3,18
16,25	3,40
17,25	3,62
18,25	3,85
19,25	4,08
20,25	4,31
21,25	4,55

a)



9,25	1,30
10,25	1,44
11,25	1,59
12,25	1,74
13,25	1,90
14,25	2,04
15,25	2,20
16,25	2,36
17,25	2,52
18,25	2,68
19,25	2,84
20,25	3,01
21,25	3,18

b)



9,25	1,14
10,25	1,28
11,25	1,41
12,25	1,55
13,25	1,70
14,25	1,86
15,25	2,04
16,25	2,22
17,25	2,41
18,25	2,60
19,25	2,80
20,25	3,01
21,25	3,22

NOTE $W = e_{rt} - c$, but not less than 13.

c)

Figure 13.4-3 — Bowling hoops

13.4.5 Stiffeners on corrugated furnaces tubes

When corrugated furnaces tubes are equipped with several stiffeners, e.g. one on each corrugation or on each second corrugation, the cross-sectional area and the second moment of area of the stiffeners shall be included when using equation (13.1-7). A height of not more than six times the furnace tube wall thickness shall be used for the calculation.

14 Access and inspection openings

14.1 General requirements

14.1.1 All boilers shall be provided with openings adequate in size and number to allow access for fabrication, cleaning and internal inspection (see 14.5.1). The dimensions of the openings shall be in accordance with 14.2 to 14.4.

14.1.2 Boilers with a shell diameter d_o of ≥ 1 400 mm shall be designed to permit entry of a person and shall be provided with a manhole for this purpose.

Boilers with a shell diameter d_o < 1 400 mm, which are capable of being entered by a person, shall be provided with a manhole.

Boilers with a shell diameter between 800 mm and 1 400 mm shall be provided with a headhole as a minimum requirement.

14.1.3 The number, size and location of access and inspection openings shall vary according to the boiler design. The following shall be intended to ensure that a good representative visual examination of the welded seams is possible.

14.1.4 Detachable ends or covers may replace all the other examination holes if, by their dimensions and position, a general view of the interior is provided which shall be at least equivalent to that obtained by the examination holes which otherwise would be required.

14.1.5 For LPB inspection openings can be provided by disassembling pipes from nozzles, if other possibilities of close inspection are not available.

14.2 Types and minimum dimensions of access and inspection openings

Openings shall be elliptical, circular or obround (see Figures 11.3-1 and 14.2-1a) to 14.2-1h)).

a) Handholes

A handhole for cleaning shall be not less than 80 mm × 100 mm or shall have an inside diameter of 100 mm.

A handhole for inspection shall be not less than 100 mm × 150 mm or shall have an inside diameter of 120 mm. The height of the neck or ring shall not exceed 65 mm, or 100 mm if the neck or ring is conical.

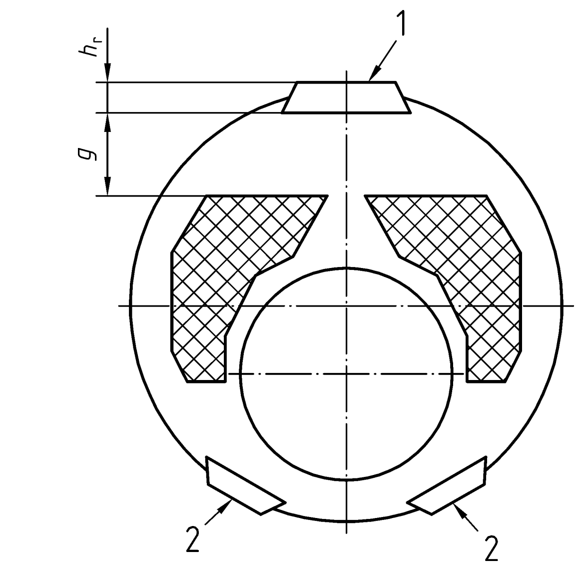
b) Headholes

Headholes shall be not less than 220 mm × 320 mm or shall have an inside diameter of 320 mm. The height of the neck or ring shall not exceed 100 mm, or 120 mm if the neck or ring is conical.

c) Manholes

Manholes shall be not less than 320 mm × 420 mm or shall have an inside diameter of 420 mm. The height of the neck or ring shall not exceed 300 mm. If, in special cases manholes of the size 300 mm × 400 mm are used, the requirements in the table in Figure 14.2-1 shall be fulfilled.

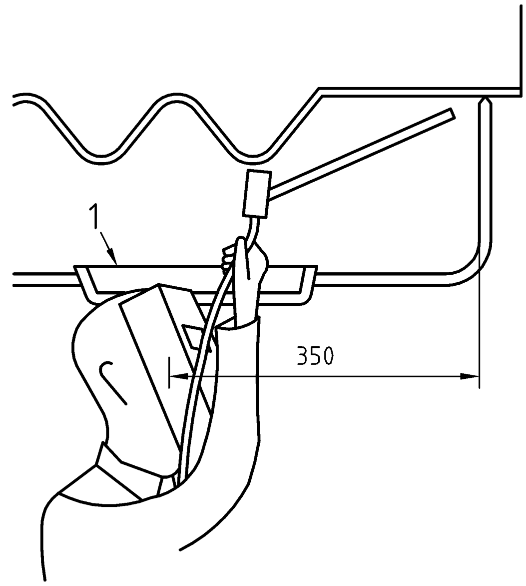
Dimensions in millimetres



Key
1 Manhole
2 Headhole

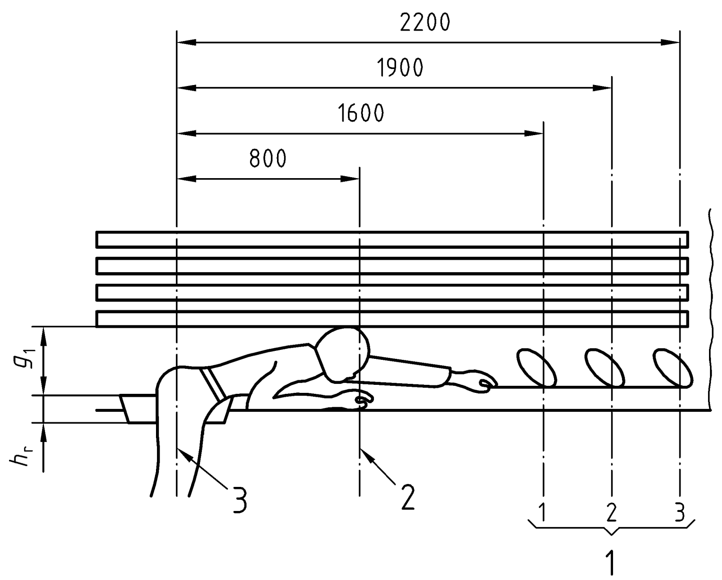
NOTE For the size of manhole and dimension g, see table in Figure 14.2-1 g).

a)



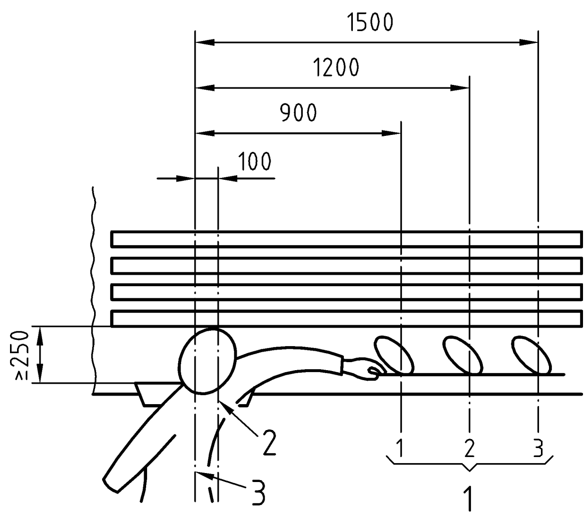
Key
1 Headhole 220 mm × 320 mm

b)



Key
1 Inspection category
2 Plane of eye
3 Centre of manhole

c)



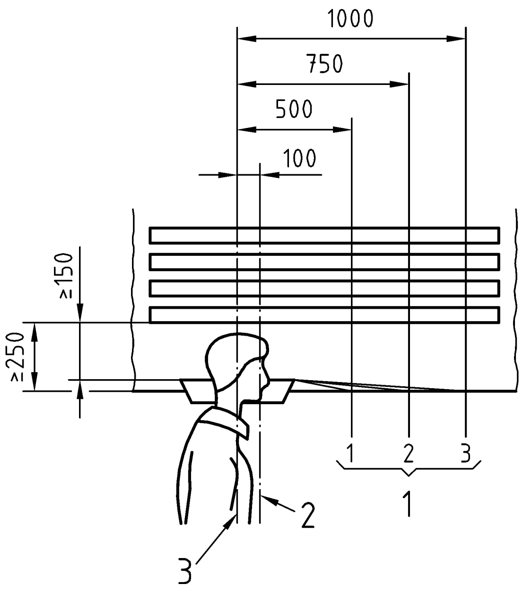
Key
1 Inspection category
2 Plane of eye
3 Centre of headhole

d)

NOTE For the size of manhole and dimension g_1, see table in Figure 14.2-1 g).

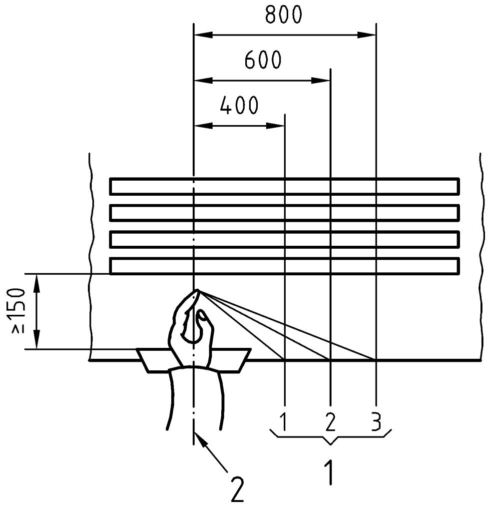
Figure 14.2-1 — Openings for access and inspection

Dimensions in millimetres



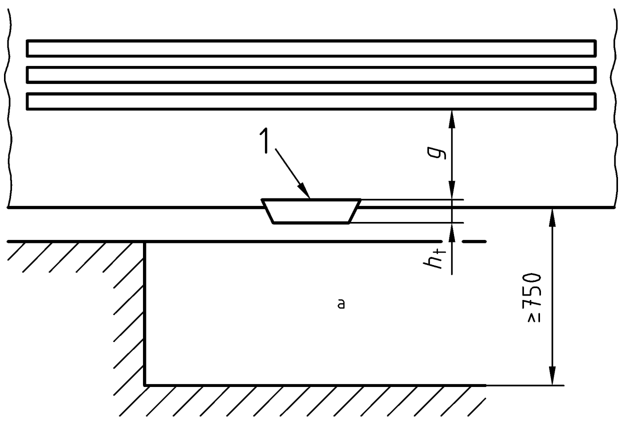
Key
2 Plane of eye

e)



Key
1 Inspection category
2 Centre of handhole

f)



Key
1 Manhole

[a] Depth of pit ≥ 500

Size of manhole	h_t	g [a]	g_1 [b]
320 × 420 300 × 400	≤ 300 ≤ 150	≥ 400 ≥ 450	≥ 320 ≥ 370

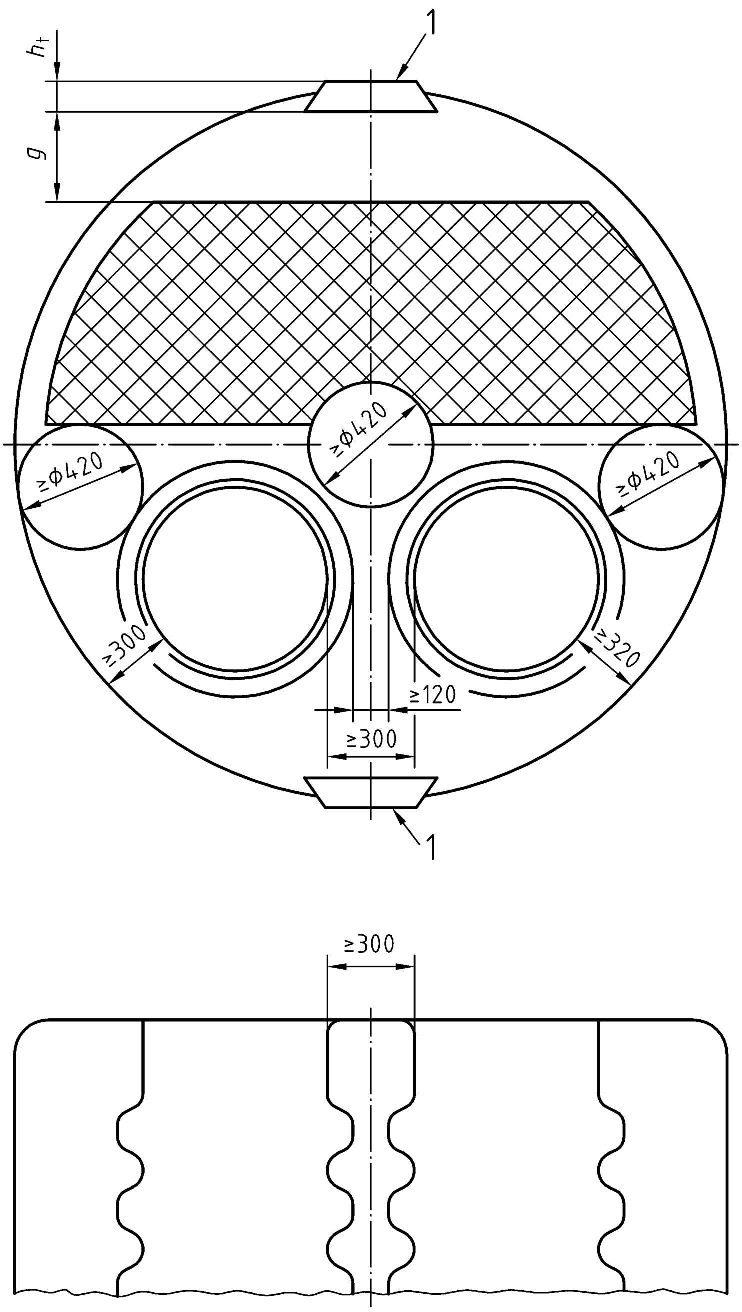
[a] For dimension g, see also Figure 14.2-1a) and Figure 14.2-1f).

[b] For dimension g_1, see Figure 14.2-1c).

g)

Figure 14.2-1 — Openings for access and inspection *(continued)*

Dimensions in millimetres



Key
1 Manhole

NOTE For the size of manhole and dimension g, see table in Figure 14.2-1 g)

h)

Figure 14.2-1 — Openings for access and inspection *(concluded)*

14.3 Minimum gasket bearing width and clearance for access and inspection doors

Access and inspection doors of the type in which the internal pressure forces the door against a flat gasket shall have a minimum gasket bearing width of 15 mm for manholes and headholes. For handholes, the gasket bearing width may be reduced to 10 mm. The total clearance between the door frame and the spigot or recess of such doors shall not exceed 3 mm, i.e. 1,5 mm all round, and the spigot depth shall be sufficient to trap the gasket.

14.4 Access and inspection openings in flat plates

Where access and inspection openings are located in flat plates, the openings shall be suitably reinforced (see Figure 11.3-1).

14.5 Inspection requirements

14.5.1 The effectiveness of visual inspections depends in particular on the distance between eye and object and the angle under which the surface of the object is seen. Effective visual inspection is best achieved by entering the boiler. This subclause is intended to ensure that a good visual examination of representative parts of the various welded seams is possible. It is recognized that it is not practicable to provide access for close visual inspection of every weld after the boiler has been constructed, particularly in the case of small boilers. However, it is also recognized that some parts of boilers are more prone than others to the occurrence of cracking or corrosion. The following inspection categories are arranged in order of relative importance and, for each category, appropriate facilities for the inspection of representative parts shall be provided. Means of achieving various degrees of visual inspection are illustrated in Figure 14.2-1.

a) Inspection category 1

- T-butt welds (e.g. end plate to shell welds and furnace tube to end plate welds) except where hidden by tube nests;
- Attachment welds of stays and stiffeners.

NOTE In the case of category 1 inspection, the representative parts of the welds are those where high bending moments are likely to occur, and would typically include half of the end plate to shell welds and at least 50 % of the end plate to furnace tube welds.

b) Inspection category 2

- Attachment welds not covered under a);
- Large components subjected to high heat flux (e.g. furnaces tubes);
- Flanged corners where the boiler is not stayed end to end;
- Parts where deposits could accumulate;
- Parts located in the vicinity of the water inlet;
- Parts where there are changes in the water level during operation;
- Staybars and staybolts between reversal chambers and endplates which are used for anchoring.

c) Inspection category 3

- All parts not covered under a) and b) which are exposed to low flux (flue gas temperature less than or equal to 400 °C) or which are not exposed to flame.

The positions of inspection openings are based on two factors, the size of the opening and the length of the visual path appropriate to the inspection category. These positions are shown in Figures 14.2-1c) to 14.2-1f). The distance may be increased in the case of nests of stay bars or stay tubes if appropriate inspection devices can be applied.

14.5.2 In the case of circumferential T-butt welds, not more than one single length equal to half the shell outside diameter or a number of lengths totalling one shell outside diameter shall be hidden by tube nests. Where necessary, the tube nest shall have suitable inspection lanes and the shell shall have headholes or handholes in sufficient number.

This shall not apply to waste heat boilers with smoke tubes only, i.e. without furnace tube.

NOTE In determining the hidden area between tubes and shell, it is assumed that the eye can be brought to within 80 mm of the internal surface of the boiler shell.

14.5.3 In all cases it shall be possible to inspect the bottom of the shell and the longitudinal welds of the shell.

14.6 Requirements for entry into boilers with a shell outside diameter greater than 1 400 mm

14.6.1 The space available for entry along the length of the boiler shall include at least one cross-section which is comparable with one measuring 600 mm diameter. This requirement shall be considered satisfied if the space includes an inscribed circle of at least 420 mm diameter and adjacent wedge-shaped spaces which guarantee sufficient freedom of movement. When entering along the bottom of the boiler (or in similar situations of movement, e.g. above tube assemblies), as well as when climbing in through a bottom manhole, as shown in Figure 14.2-1h), or a top manhole, as shown in Figure 14.2-1a), a clear height of 400 mm between the boiler shell (manhole surround) and the tube assembly shall be sufficient, if one width of the entry space (if possible wedge shaped) is at least 600 mm. For smaller spaces, only the upper part of the body need enter as shown in Figure 14.2-1c).

14.6.2 If it is necessary to pass from one inspection space into another, for instance a lateral space, it shall be sufficient to have a hole with a height of at least 300 mm at its narrowest point (see Figure 14.2-1h)). Figure 14.2-1h) merely explains the idea of such a "hole". The possible types need not all be available at the same time.

14.7 Accessibility and arrangement of entry and inspection openings

All entry and inspection openings shall be accessible or shall be easily made accessible. When installing pumps, valves, preheaters, frame constructions, foundations, etc., accessibility shall be taken into account. In each case, the arrangement of the entry and inspection opening, along or at right angles to the boiler axis, shall be used to make the inspection conditions as favourable as possible.

Annex A
(informative)

Calculation of tube plate temperatures

A.1 General

This annex provides a method for the calculation of the hot-face metal temperature and the average (design) temperature of tube plates within the tube nest.

The calculation takes into account the effects under steady state conditions of heat transfer

a) from the hot gas to the tube plate face and tube inside surfaces by convection, including the tube entrance effect, and radiation, including radiation interchange in the reversal chamber;

b) by thermal conduction through the tube plate and tube walls from the tube plate face and tube inside surfaces to the water side surfaces, assuming adequate thermal contact between tube and plate; and

c) by nucleate boiling from the water side surfaces.

The method and design curves have been developed from published heat transfer data and contain some simplifying approximations which tend to be self-compensating. Calculated and measured temperatures have shown good agreement where complete data are available.

A.2 Symbols

In Table A.2 are specific symbols for this annex which could deviate from Table 4.1 of prEN 12953-1:2001 and the main part.

Table A.2 — Symbols

Symbol	Description	Unit
a	Heat input area to the tube plate element from the tube plate face (see Figure A.8).	mm^2
A	Heat input area to the tube plate element from the tube inside surfaces (see Figure A.7).	mm^2
A_C	Total effective water-cooled surface area in the reversal chamber.	mm^2
A_R	Total refractory surface area in the reversal chamber.	mm^2
C	Correction factor for tube to tube plate contact thermal resistance.	
d	Inside diameter of the convection tube.	mm
D	Reversal chamber inside diameter (for cylindrical chambers).	mm
e	Tube plate thickness.	mm
F	Overall exchange factor for radiation interchange in the reversal chamber (see Figure A.2).	
G	Tube specific gas flow rate.	kg/(m^2 · s)
h_{CE}	Tube entrance convection coefficient (see Figure A.6)	W/(m^2 · K)
h_{CO}	Corrected basis convection coefficient (see Figure A.5)	W/(m^2 · K)
h'_{CO}	Hypothetical basis convection coefficient (see Figure A.4)	W/(m^2 · K)
h_m	Tube plate thermal conductance.	W/(m^2 · K)
h_R	Radiation coefficient for the tube plate face.	W/(m^2 · K)

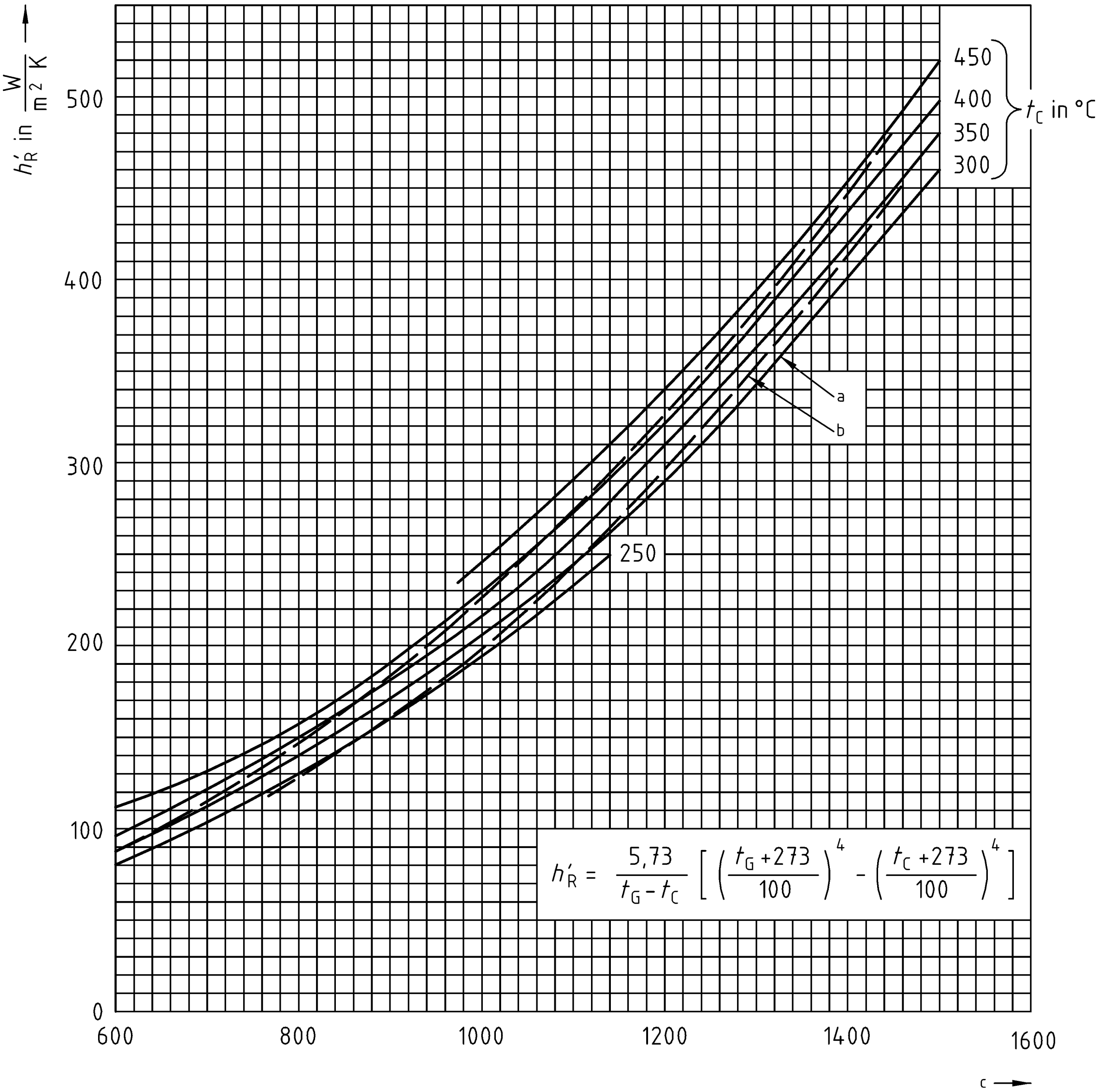
Table A.2 *(continued)*

Symbol	Description	Unit
h'_R	Radiation coefficient for black exchange (see Figure A.1).	W/(m^2 · K)
h_t	Weighted average heat transfer coefficient.	W/(m^2 · K)
L	Reversal chamber inside length (for cylindrical chambers).	mm
L_B	Reversal chamber radiation beam length.	mm
N	Water side constant, = 4 000.	W/(m^2 · K)
p	Average pitch between the tube centres.	mm
t	Tube plate average (design) temperature	°C
t_C	Initial estimate of t_M	°C
t_G	True gas temperature at the tube entrance.	°C
t_M	Tube plate hot-face metal temperature.	°C
t_S	Boiler water temperature.	°C
β	Tube plate average temperature factor (see Figure A.12).	
η	Heat transfer factor for the tube plate element (see Figure A.10).	
λ	Tube plate thermal conductivity: for steel grades 460 and 490, = 40 000; for steel grades 400 and 430, = 45 000.	W · mm/ (m^2 · K)
Φ	Tube plate hot-face temperature factor (see Figure A.11).	

A.3 Calculation method

A.3.1 Radiation coefficients

Determine the radiation coefficient h'_R for black exchange, i.e. emissivity = 1, F = 1, from Figure A.1. The gas temperature t_G at tube entry shall be the true value as would be measured by a multishield high-velocity suction pyrometer. (An ordinary thermocouple will always read low; the error may be up to 300 °C.) Assume an initial value t_C for the tube plate hot-face metal temperature. Typical values shown on Figure A.1 shall usually avoid the neccessity for reiteration.



Key

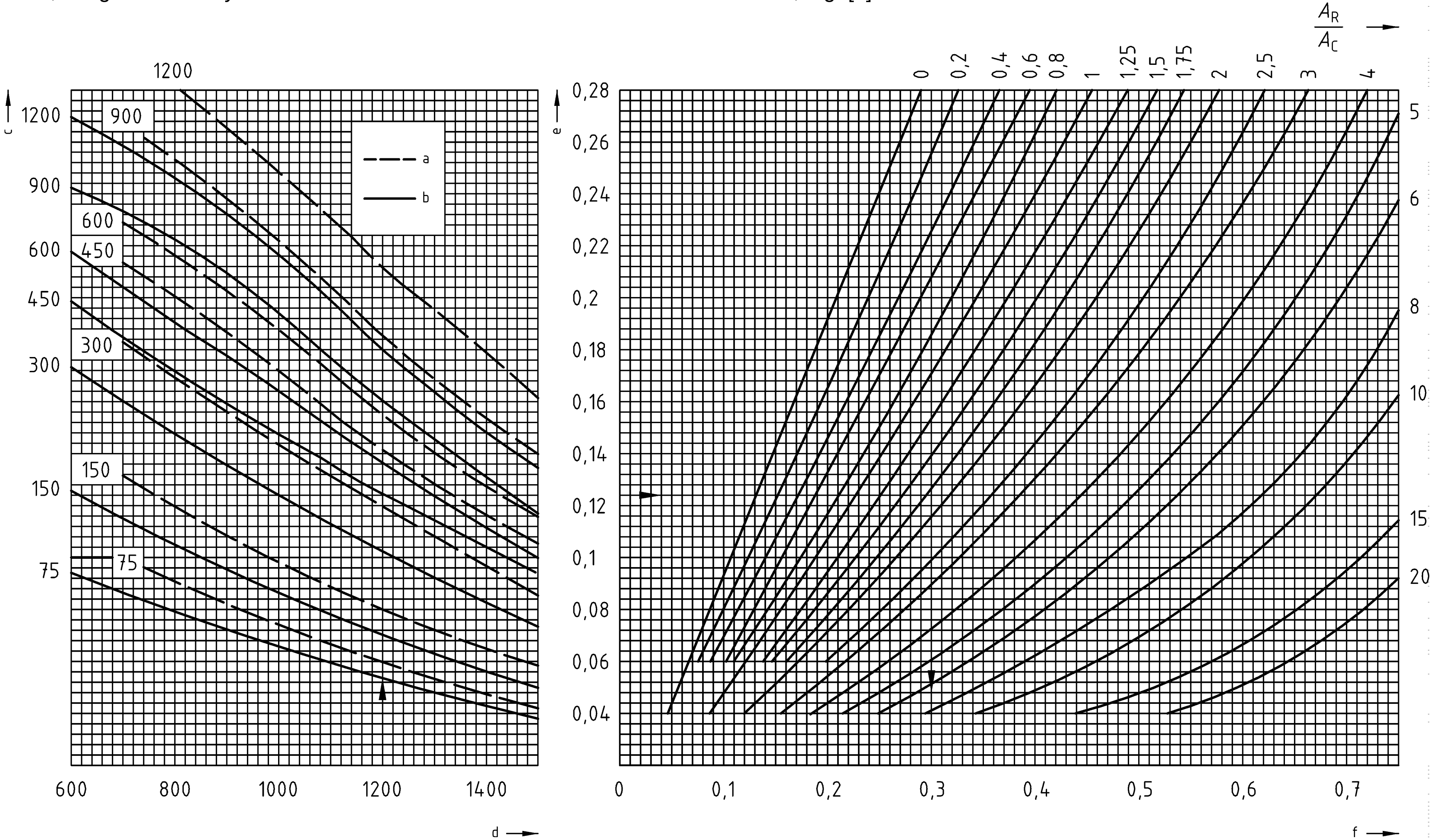
a Typical dry back curve

b Typical wet back curve

c True gas temperature at the tube entry t_G (in °C)

Figure A.1 — Radiation coefficient h'_R for black exchange (F = 1)

The emissivity of the gas shall depend on the gas analysis, temperature, partial pressures and the beam length in the reversal chamber. The curves in Figure A.2 are based on the excess air normally used in directly fired boilers. For products of coal combustion, it is recommended that the natural gas curve be used to allow for particle radiation. For other gas mixtures, the gas emissivity should be determined from a text on radiant heat transfer, e.g. [1].



Key

a Natural gas and coal

b Oil fuels

c Beam length (in mm)

d True gas temperature at the tube entry t_G (in °C)

e Gas emissivity

f Overall exchange factor F

Figure A.2 — Determination of overall exchange factor F

The radiation beam length for a cylindrical reversal chamber shall be given by the following formula:

$$L_B = \frac{0{,}83\ L}{L / D + 0{,}5} \tag{A.3-1}$$

For chambers which are not cylindrical, the radiation beam length shall be given by the following formula:

$$L_B = 3{,}3\ \frac{V_c}{A_{CS}} \tag{A.3-2}$$

where

V_C is the chamber volume;

A_{CS} is the chamber surface area.

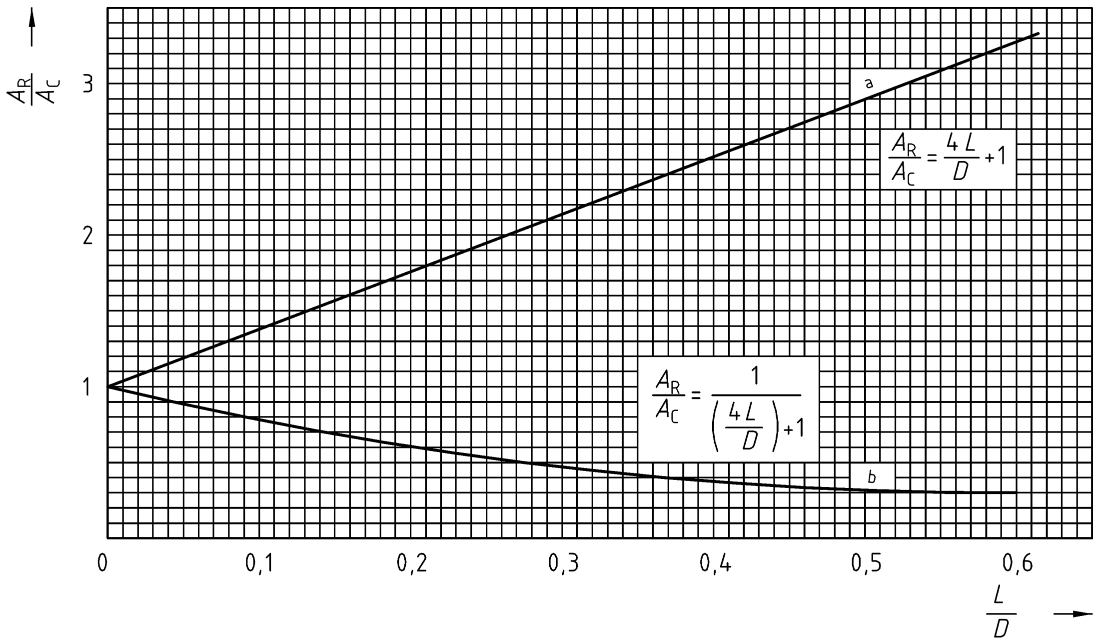
In calculating the chamber surface area no reduction shall be made for tube holes or the furnace tube opening.

For chambers containing refractory linings A_R/A_C is the ratio of the total effective (reflecting) refractory surface area to the effective cooled (absorbing) surface area in the chamber.

A_C includes the total area enclosed within the tube plate perimeter, i.e. no reduction for tube holes or the furnace tube opening.

A_R/A_C for cylindrical chambers may be obtained from Figure A.3.

For fully water-cooled chambers A_R/A_C = 0.



Key
a Dry back
b Semi-wet back

NOTE For a non-cylindrical chamber, include the total superficial area of the tube plate in A_C (no reduction for tube holes or furnace tube openings).

Figure A.3 — A_R/A_C for a cylindrical chamber with diameter D and length L

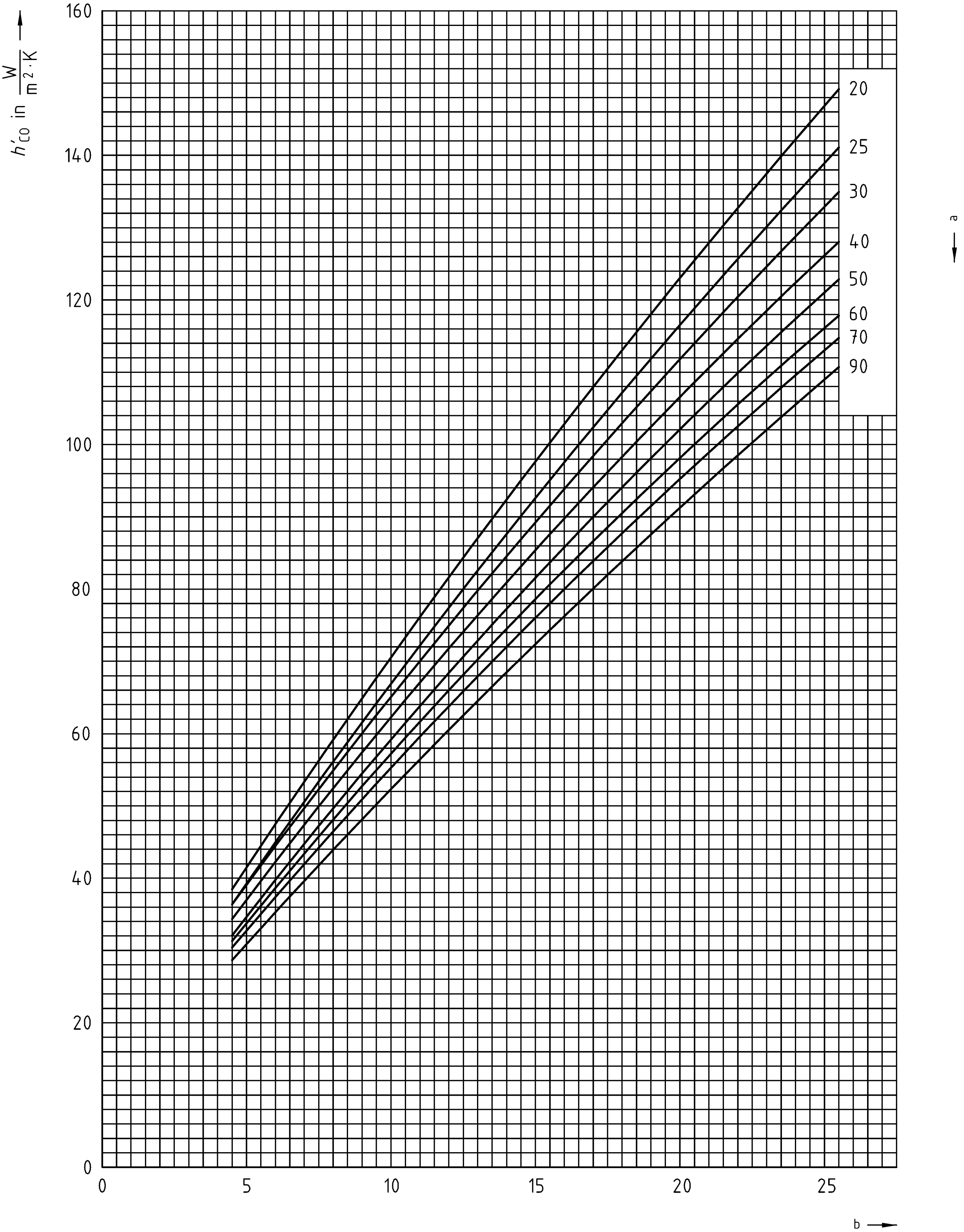
Determine the overall exchange factor F from Figure A.2, then the radiation coefficient for the tube plate face is given by the following formula:

$$h_R = F \, h'_R \tag{A.3-3}$$

Radiation to the tube inside surfaces shall be taken into account by use of the coefficient 0,5 h_R in the equation for the weighted average heat transfer coefficient h_t (see A.3.3).

A.3.2 Convection coefficients

The hypothetical basis convection coefficient h'_{CO} is dependent on the specific gas flow rate G in the convection tubes and on the tube inside diameter d. For the products of combustion of oil fuels, natural gas and coal, determine h'_{CO} from Figure A.4.



Key

a Tube inside diameter d (in mm)

b Tube flow rate G [in kg/(m^2 · s)]

NOTE $$h'_{CO} = 20{,}2 \frac{G^{0,8}}{d^{0,2}} \qquad \text{(A.3-4)}$$

Figure A.4 — Basis convection coefficient h'_{CO}

Determine the correction factor h_{CO}/h'_{CO} for the tube entry gas temperature from Figure A.5. Then the corrected basis convection coefficient for fully developed tube flow at temperature t_G is given by the following formula:

$$h_{CO} = h'_{CO} \left(\frac{h_{CO}}{h'_{CO}} \right) \quad \text{(A.3-5)}$$

For other gases where the values of specific heat, thermal conductivity or viscosity are different from those for the products of combustion of oil or natural gas, the value of h_{CO} shall be deduced from the equation for fully developed flow inside tubes as follows:

$$Nu = 0{,}023\ Re^{0{,}8}\ Pr^{\ 0{,}33} \quad \text{(A.3-6)}$$

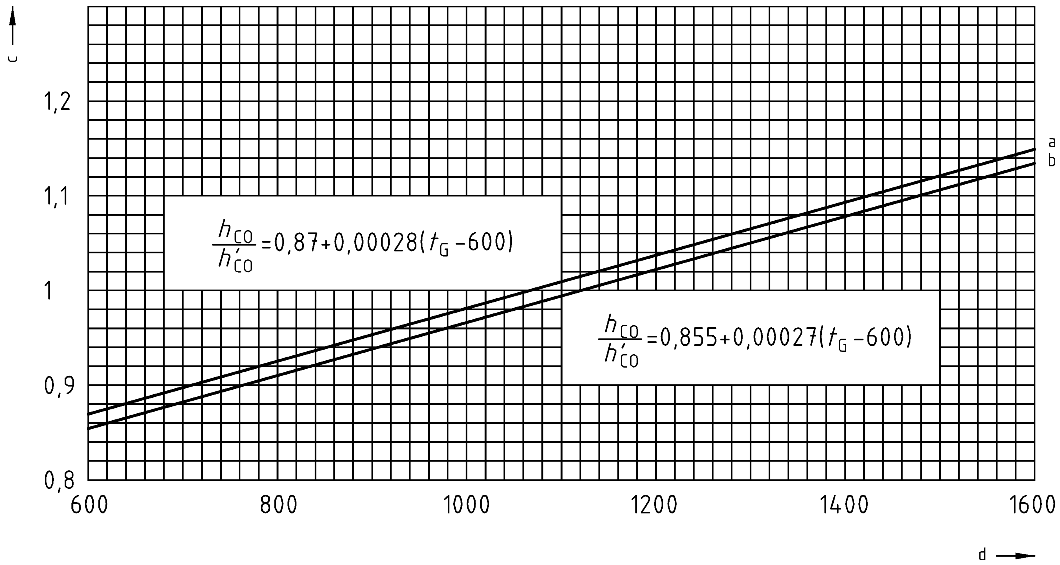
where

Nu is the Nusselt number, based on the tube inside diameter d,

Re is the Reynolds number, based on the tube inside diameter d,

Pr is the Prandtl number.

(see, for example, [1]).



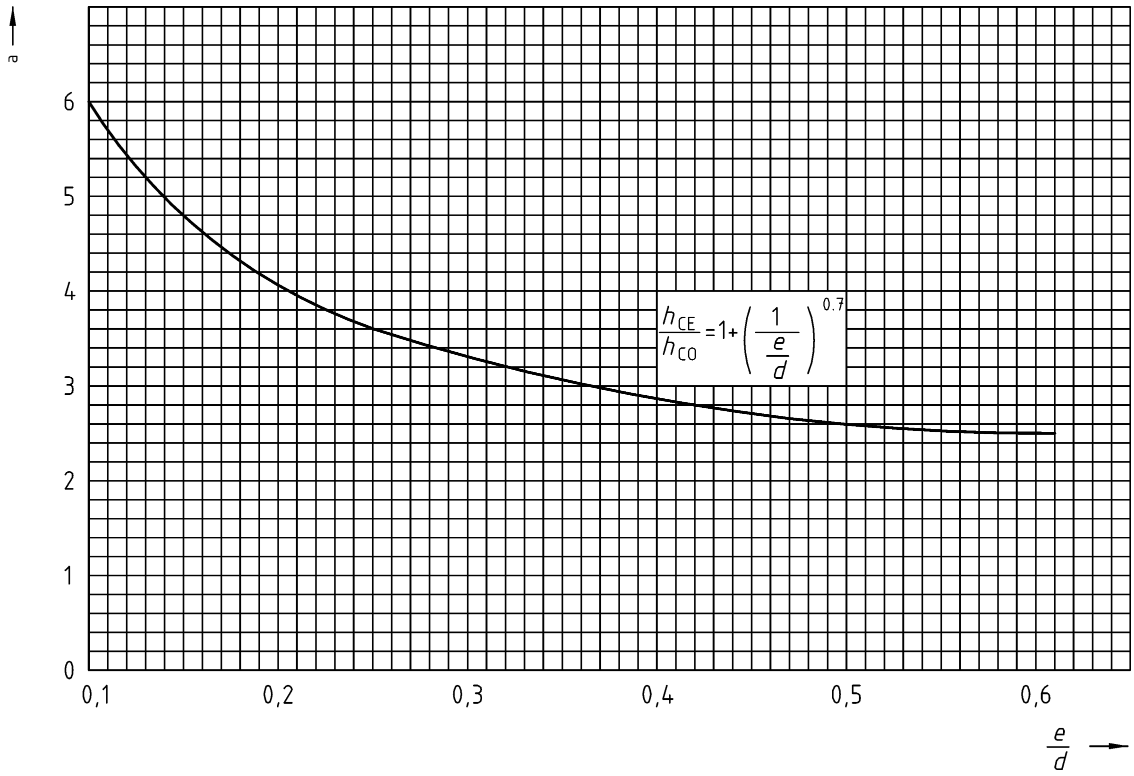
Key
a Natural gas
b Oil fuels and coal
c Factor $\left(\frac{h_{CO}}{h'_{CO}} \right)$
d True gas temperature at tube entry t_G (°C)

Figure A.5 — Determination of correction factor

Determine the correction factor h_{CE}/h_{CO} for the tube entrance region from Figure A.6. Then the average convection coefficient, h_{CE}, for the tube inside surface over the effective length for heat input to the tube plate, is given by the following formula:

$$h_{CE} = h_{CO} \left(\frac{h_{CE}}{h_{CO}} \right) \quad \text{(A.3-7)}$$

Convective heat transfer to the tube plate face shall be taken into account by the use of the coefficient h_{CO} in the equation for the weighted average heat transfer coefficient h_t (see A.3.3).



Key

a Factor $\left(\frac{h_{CE}}{h_{CO}}\right)$

Figure A.6 — Determination of correction factor

A.3.3 Weighted average gas-side heat transfer coefficient

For the tube plate element, bounded by tube inside surfaces and planes containing tube centre-lines, the heat input areas A (tube inside surfaces) and a (tube plate face) are determined from Figures A.7 and A.8.

The weighted average heat transfer coefficient is then calculated as follows:

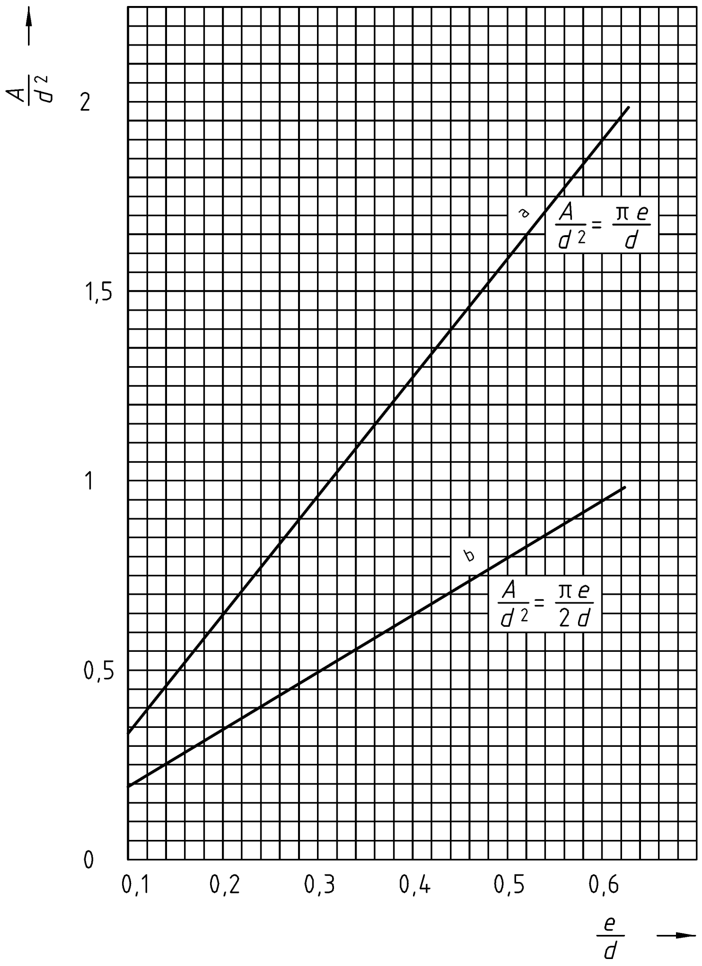
$$h_t = \frac{\frac{CA}{d_2}\left(h_{CE} + 0{,}5\, h_R\right) + \frac{a}{d^2}\left(h_{CO} + h_R\right)}{\left(\frac{A}{d^2} + \frac{a}{d^2}\right)} \quad \text{(A.3-8)}$$

where

C = 0,9 for tubes expanded only;

C = 0,95 for tubes expanded and welded,

C = 1 for tubes full penetration welded.

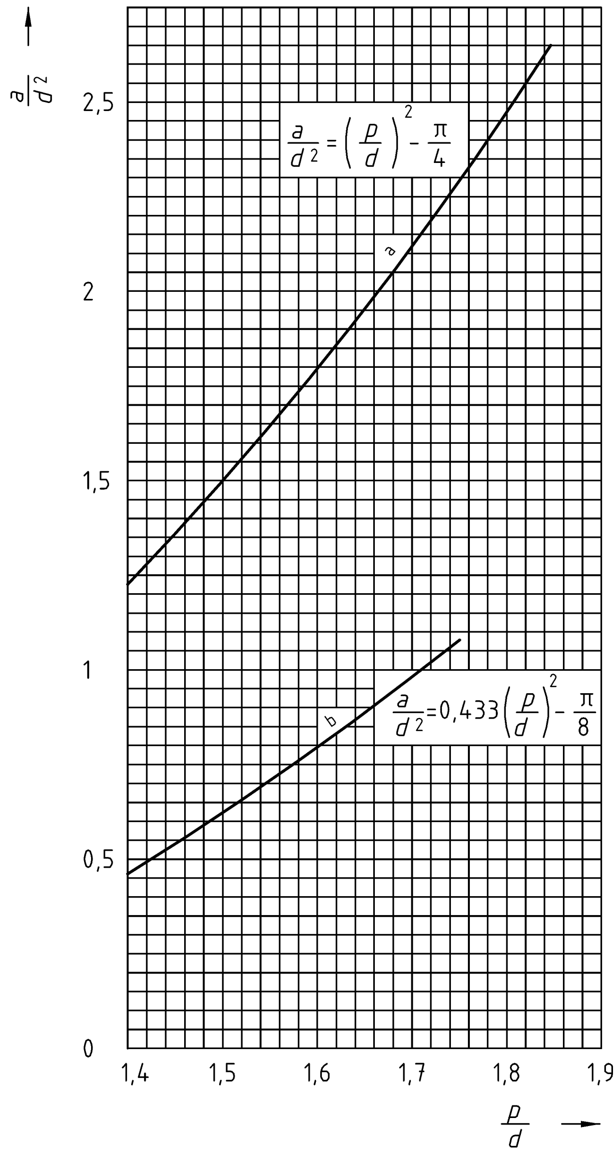


Key

a Square pitch

b Triangular pitch

Figure A.7 — Non-dimensional tube area



Key

a Square pitch

b Triangular pitch

Figure A.8 — Non-dimensional plate area

A.3.4 Tube plate thermal conductance

The tube plate thermal conductance is given by the following formula:

$$h_{\mathrm{m}} = \frac{\lambda}{e} \quad \text{(A.3-9)}$$

A.3.5 Water side heat transfer

Heat transfer conditions at the water side surfaces are taken into account in the equations for the tube plate metal temperatures by use of the constant N.

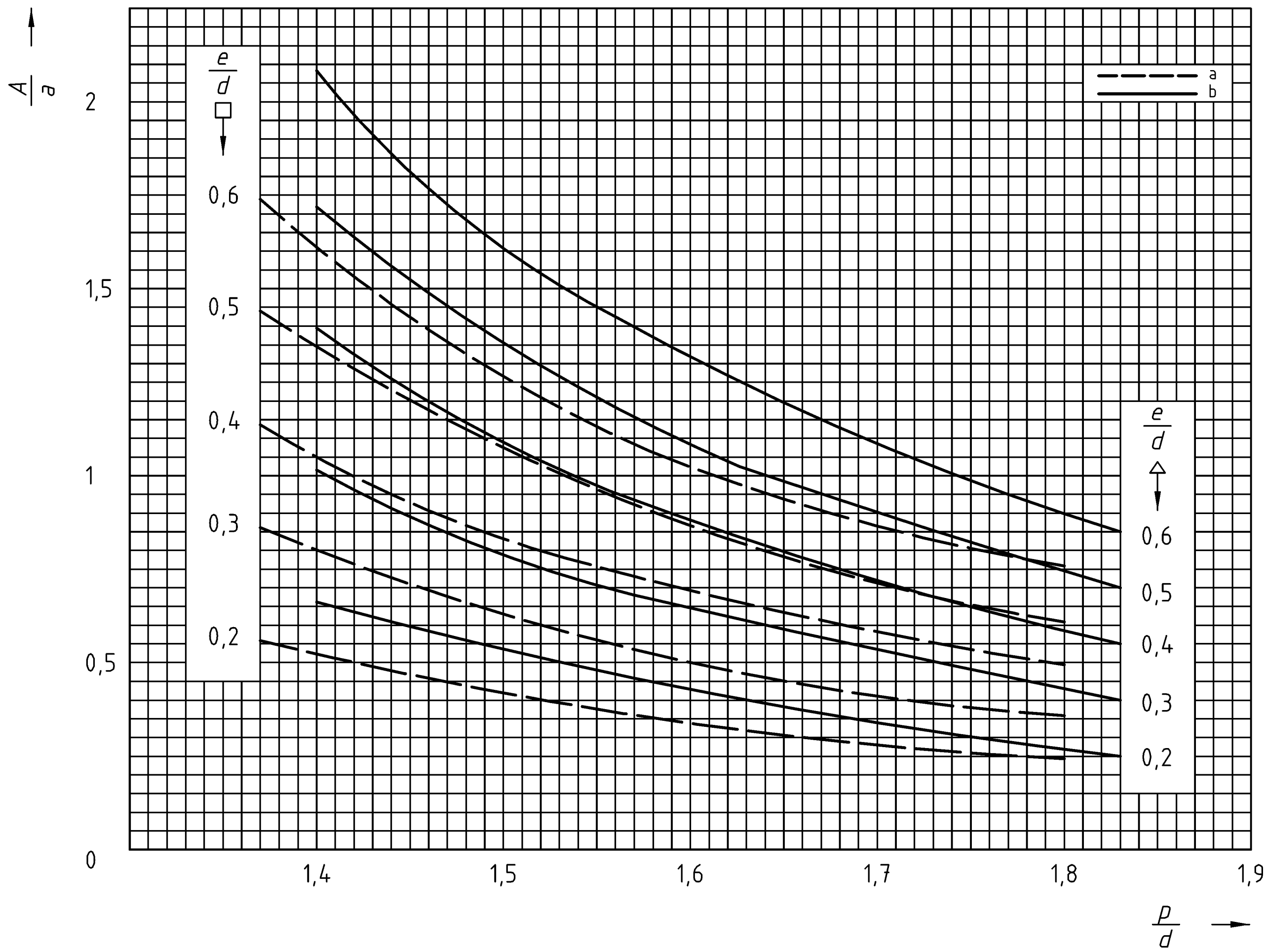
A.3.6 Tube plate temperatures

The following equations for the tube plate hot-face and average metal temperatures are based on equations developed by Gardner [2]:

$$t_M = t_S + 15 + (t_G - t_S)\left(1 - \frac{\Phi}{1 + (\eta\, h_t / N)}\right) \quad \text{(A.3-10)}$$

$$t = t_S + 15 + (t_G - t_S)\left(1 - \frac{\beta}{1 + (\eta\, h_t / N)}\right) \quad \text{(A.3-11)}$$

The factors η, Φ and β are dependent on A/a (from Figure A.9) and on h_t/h_m and are obtained from Figures A.10 to A.12.



Key
a Square pitch
b Triangular pitch

Figure A.9 — Tube/plate area ratio

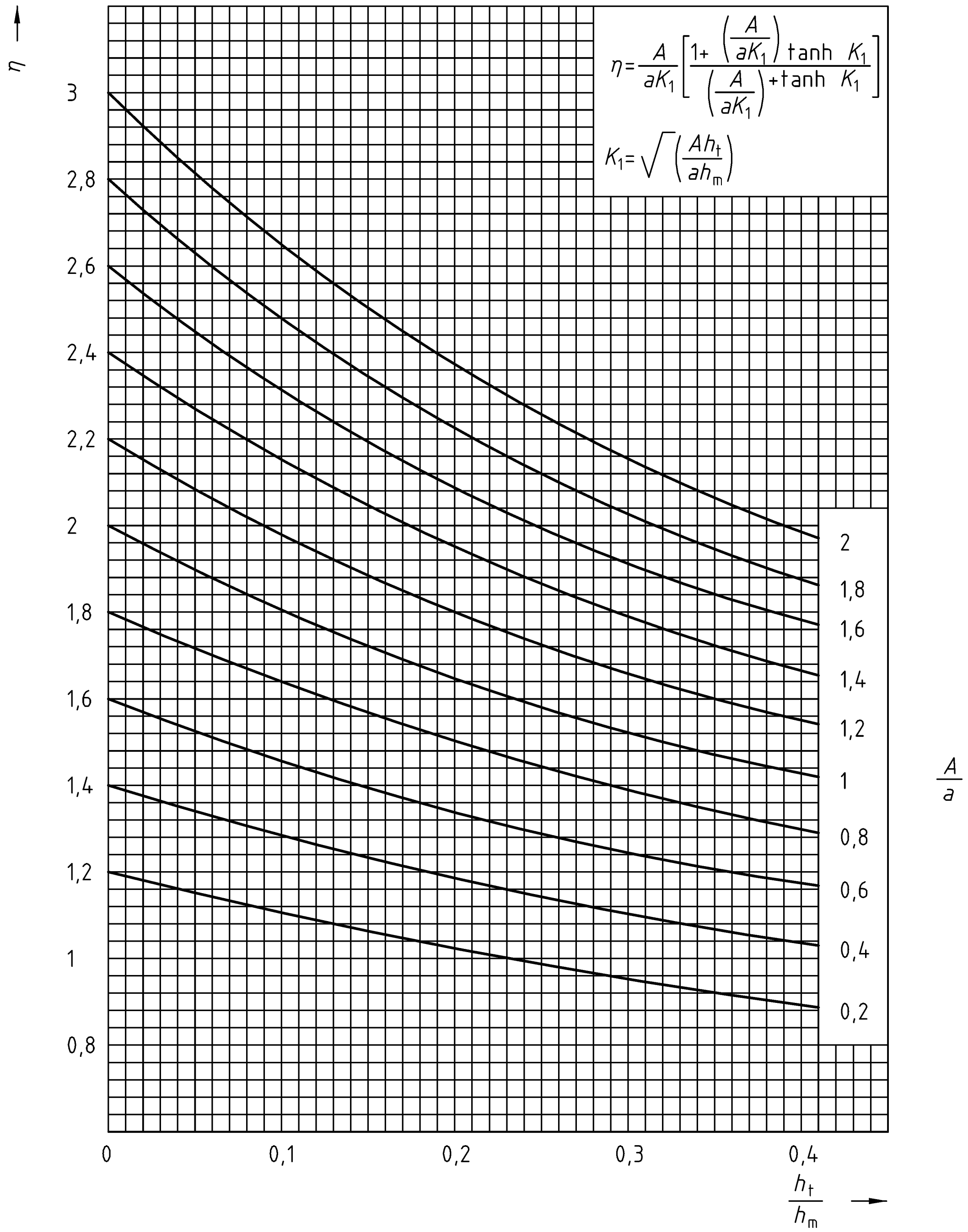


Figure A.10 — Factor η

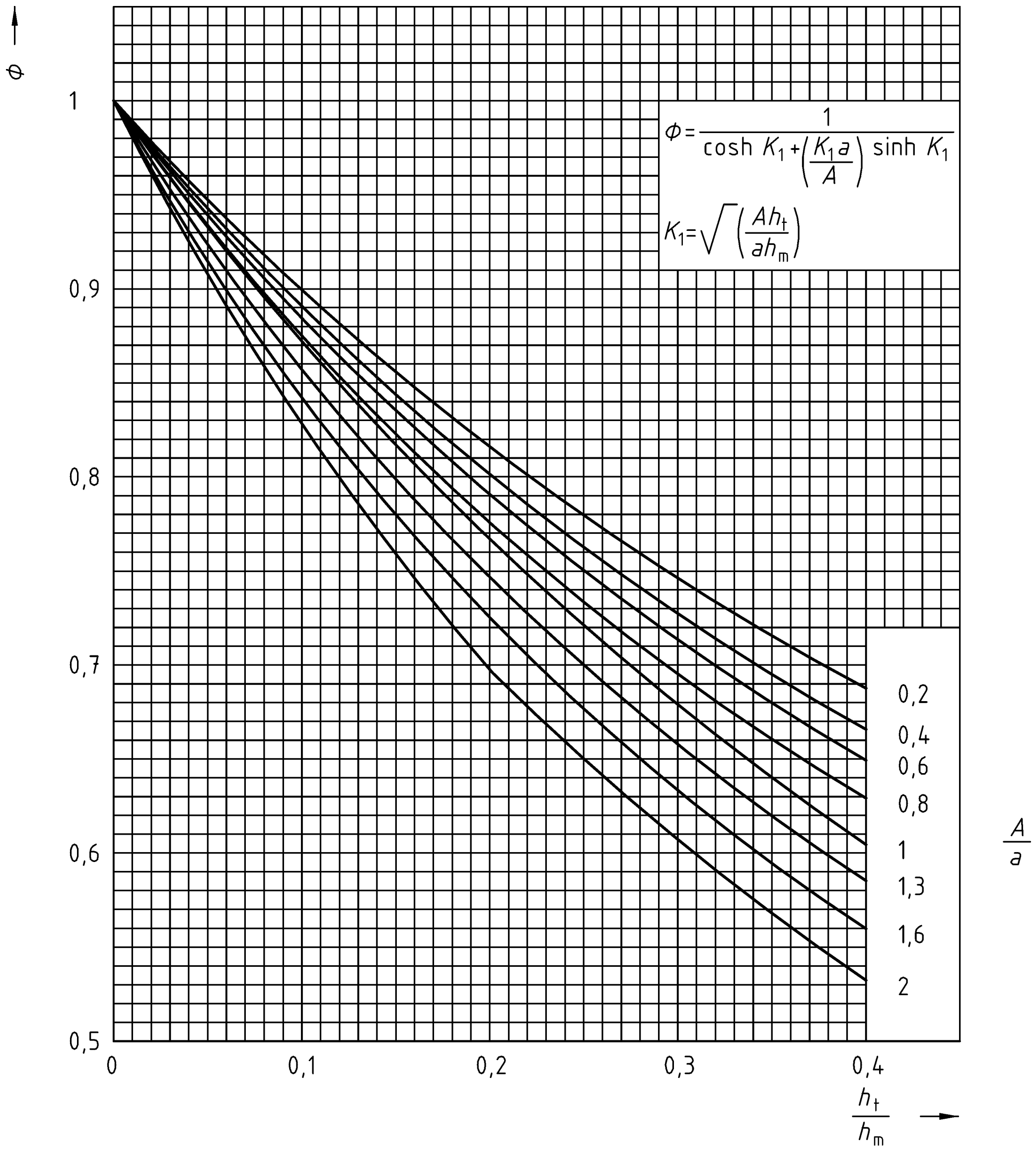


Figure A.11 — Factor Φ

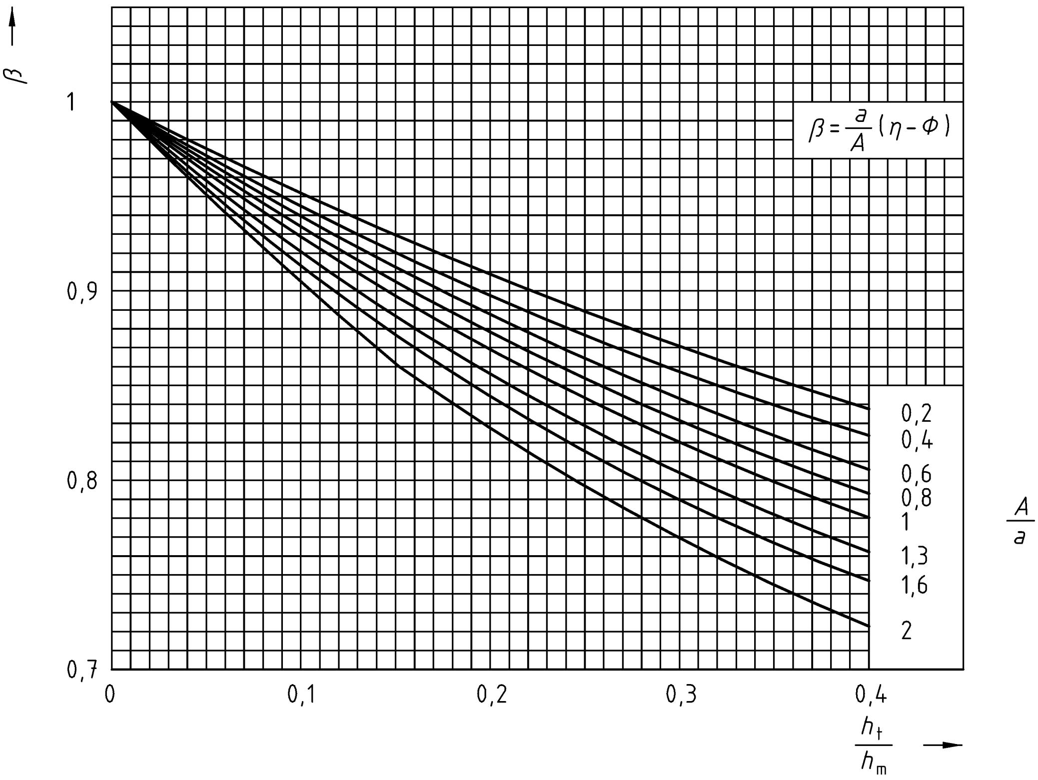


Figure A.12 — Factor β

A.4 Example of a calculation carried out using the method given in A.3

A.4.1 Design data assumed

Fuel: natural gas

Boiler: multitubular waste heat with refractory-lined hot-gas chamber

Specified inlet gas temperature: 900 °C

Boiler design pressure: 1,1 N/mm^2

Saturation temperature: t_S = 188 °C

Boiler tubes:

inside diameter d = 56,3 mm

pitch, triangular p = 88 mm

gas flow rate G = 11 kg/(m^2 · s)

Tubeplate:

thickness e = 22 mm

steel group 1

Tube end attachment: expanded and welded

Inlet gas chamber:

cylindrical, refractory lined on wrapper and back plates

inside diameter D = 1 800 mm

inside length L = 1 000 mm

A.4.2 Calculation of radiation coefficient

The calculation of the radiation coefficient h_R shall be carried out as described in A.3.1.

From Figure A.1, using an assumed value of t_C = 350 °C indicated by the for external chamber typical dry back curve, h'_R = 185 W/(m^2 · K).

Radiation beam length

$$L_B = \frac{0{,}83 \times 1\,000}{1\,000 / 1\,800 + 0{,}5} = 786 \text{ mm}$$

From Figure A.3, $\frac{A_R}{A_C} = 3{,}15$, where $\frac{L}{D} = 0{,}555$.

From Figure A.2, F = 0,58.

Therefore,

$$h_R = 0{,}58 \times 185 = 107{,}3 \text{ W/(m}^2 \cdot \text{K)}$$

A.4.3 Calculation of convection coefficients

The calculation of convection coefficients h_{CO} and h_{CE} shall be carried out as described in A.3.2.

From Figure A.4, h'_{CO} = 61 W/(m^2 · K).

From Figure A.5, $\frac{h_{CO}}{h'_{CO}} = 0{,}952$.

Therefore,

$$h_{CO} = 0{,}952 \times 61 = 58{,}1 \text{ W/(m}^2 \cdot \text{K)}$$

From Figure A.6 ist $\frac{h_{CE}}{h_{CO}} = 2{,}9$, where $\frac{e}{d} = \frac{22}{56{,}3} = 0{,}391$.

Therefore,

$$h_{CE} = 58{,}1 \times 2{,}9 = 168{,}5 \text{ W/(m}^2 \cdot \text{K)}$$

A.4.4 Calculation of weighted average gas-side heat transfer coefficient

The calculation of the weighted average gas-side heat transfer coefficient h_t shall be carried out as described in A.3.3.

From Figure A.7, $\frac{A}{d^2} = 0{,}6$, where $\frac{e}{d} = 0{,}391$ triangular pitch.

From Figure A.8, $\frac{a}{d^2} = 0{,}67$, where $\frac{p}{d} = \frac{88}{56{,}3} = 1{,}563$.

For tubes expanded and welded C = 0,95.

Therefore,

$$h_t = \frac{0{,}95 \times 0{,}6\,(168{,}5 + 0{,}5 \times 107{,}3) + 0{,}67\,(58{,}1 + 107{,}3)}{0{,}6 + 0{,}67} = 187 \text{ W}/(\text{m}^2 \cdot \text{K})$$

A.4.5 Calculation of tube plate thermal conductance

The calculation of the tube plate thermal conductance h_m shall be carried out as described in A.3.4.

For steel group 1 λ = 45 000 W · mm/(m^2 · K) (see A.2).

Therefore,

$$h_m = \frac{45\ 000}{22} = 2\ 045 \text{ W}/(\text{m}^2 \cdot \text{K})$$

A.4.6 Calculation of tube plate temperatures

The calculation of tube plate temperatures t and t_M shall be carried out as described in A.3.6.

$$\frac{h_t}{h_m} = \frac{187}{2\ 045} = 0{,}09144$$

From Figure A.9, $\frac{A}{a} = 0{,}9$.

From Figures A.10, A.11 and A.12,

η = 1,72

Φ = 0,885

β = 0,935

Therefore, the tube plate hot-face metal temperature shall be given by

$$t_M = 188 + 15 + (900 - 188)\left[1 - \frac{0{,}885}{1 + \frac{1{,}72 \times 187}{4\ 000}}\right] = 332\ °C$$

This is below the limit given in 6.1 and shall be therefore satisfactory.

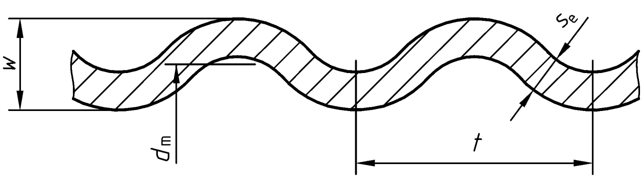
The tube plate average (design) metal temperature shall be given by

$$t = 188 + 15 + (900 - 188)\left[1 - \frac{0{,}935}{1 + \frac{1{,}72 \times 187}{4\ 000}}\right] = 299\ °C$$

Annex B
(informative)

Calculation form for "Walker"-type reverse curve sections or corrugations

Design pressure = N/mm^2	[a] Minimum in accordance with 13.1.1 but not less than t
Design temperature = °C	
Design stress, f = N/mm^2	
R_1 = mm	
R_2 = mm	
R_3 = mm	
b = mm	
t = mm (corroded)	
L, dist. between centres of support ($\leq 4\sqrt{2\,R\,t}$ where R is the greater of R1 or R3) = mm	



$r_1 = R_1 - t/2 =$ mm	$r_2 = R_2 + t/2 =$ mm	$r_3 = R_3 - t/2 =$ mm

$$\Theta = \sin^{-1}\left(\frac{r_1 - r_3}{b}\right) = \quad °$$

$$\alpha = 90 - \cos^{-1}\left[\frac{(r_1 + r_2)^2 + b^2 - (r_2 + r_3)^2}{2\,(r_1 + r_2)\,b}\right] - \Theta = \quad °$$

$$\beta = 90 - \cos^{-1}\left[\frac{(r_2 + r_3)^2 + b^2 - (r_1 + r_2)^2}{2\,(r_2 + r_3)\,b}\right] + \Theta = \quad °$$

$$d = r_2 - (r_1 + r_2)\cos\alpha + r_1 + t = \quad \text{mm}$$

considering areas of each section —

$a_1 = r_1\,\alpha\,\dfrac{t\,\pi}{180} =$ mm^2	$a_3 = r_2\,\beta\,\dfrac{t\,\pi}{180} =$ mm^2
$a_2 = r_2\,\alpha\,\dfrac{t\,\pi}{180} =$ mm^2	$a_4 = r_3\,\beta\,\dfrac{t\,\pi}{180} =$ mm^2
$A = \Sigma_a =$ mm^2	

for positions of centroids —

$$Y_1 = r_1 + t/2 - \frac{\sin\alpha}{\alpha} \times \frac{2}{3}\left[\frac{(r_1 + t/2)^3 - (r_1 - t/2)^3}{(r_1 + t/2)^2 - (r_1 - t/2)^2}\right]\frac{180}{\pi} = \quad \text{mm}$$

$$Y_2 = d - r_2 - t/2 + \frac{\sin\alpha}{\alpha} \times \frac{2}{3}\left[\frac{(r_2 + t/2)^3 - (r_2 - t/2)^3}{(r_2 + t/2)^2 - (r_2 - t/2)^2}\right]\frac{180}{\pi} = \quad \text{mm}$$

$$Y_3 = d - r_2 - t/2 + \frac{\sin\beta}{\beta} \times \frac{2}{3}\left[\frac{(r_2 + t/2)^3 - (r_2 - t/2)^3}{(r_2 + t/2)^2 - (r_2 - t/2)^2}\right]\frac{180}{\pi} = \quad \text{mm}$$

$$Y_4 = r_3 + t/2 - \frac{\sin\beta}{\beta} \times \frac{2}{3}\left[\frac{(r_3 + t/2)^3 - (r_3 - t/2)^3}{(r_3 + t/2)^2 - (r_3 - t/2)^2}\right]\frac{180}{\pi} = \quad \text{mm}$$

moments about 0-0 —

$Y_0 = \dfrac{a_1\,Y_1 + a_2\,Y_2 + a_3\,Y_3 + a_4\,Y_4}{\Sigma_a} =$ mm	$Y = d - Y_0 =$ mm

for second moments of area about neutral axis N-N —		
$l_1 = \left[\frac{\alpha\ \pi}{90} + \sin 2\,\alpha\right] \times \left[\frac{(r_1 + t/2)^4 - (r_1 - t/2)^4}{16}\right] - a_1\,(r_1 + t/2 - Y_1)^2 + a_1\,(Y_0 - Y_1)^2 =$		mm^4
$l_2 = \left[\frac{\alpha\ \pi}{90} + \sin 2\,\alpha\right] \times \left[\frac{(r_2 + t/2)^4 - (r_2 - t/2)^4}{16}\right] - a_2\,(r_2 + t/2 - d + Y_2)^2 + a_2\,(Y_2 - Y_0)^2 =$		mm^4
$l_3 = \left[\frac{\beta\ \pi}{90} + \sin 2\,\beta\right] \times \left[\frac{(r_2 + t/2)^4 - (r_2 - t/2)^4}{16}\right] - a_3\,(r_2 + t/2 - d + Y_3)^2 + a_3\,(Y_3 - Y_0)^2 =$		mm^4
$l_4 = \left[\frac{\beta\ \pi}{90} + \sin 2\,\beta\right] \times \left[\frac{(r_3 + t/2)^4 - (r_3 - t/2)^4}{16}\right] - a_4\,(r_3 + t/2 - Y_4)^2 + a_4\,(Y_0 - Y_4)^2 =$		mm^4
$l_n = \Sigma\ l =$		mm^4
$P_{max} = \frac{8\ f\ I_n}{Y\ L^2\ b \cos \Theta} =$	N/mm^2	If $P_{max} > P_{design}$ then the section under consideration is acceptable.

Annex ZA
(informative)

Clauses of this European Standard addressing essential requirements or other provisions of the Pressure Equipment Directive

This European Standard has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association and supports essential safety requirements of the Pressure Equipment Directive 97/23/EC with regard to **design and calculation for pressure parts**.

WARNING Other requirements and other EU Directives **may** be applicable to the product(s) falling within the scope of this standard.

The following clauses of this standard given in Table ZA.1 are likely to support essential safety requirements of the Pressure Equipment Directive 97/23/EC:

Table ZA.1 — Comparison between EN 12953-3 and the Pressure Equipment Directive 97/23/EC with respect to design and calculation for pressure parts for shell boilers

EN 12953-3 harmonized clauses	Content	Pressure Equipment Directive 97/23/EC Annex I
5.1 to 5.7	Design — general	2.1
5.6	Hydrostatic test pressure	7.4
5.7	Corrosion or other chemical attack	2.6
5.7	Wear	2.7
6 to 13	Design for adequate strength	2.2.2 and 2.2.3
6.1	Operating temperatures	2.2.1
6.2	Allowable stresses	7.1.2
14	Means of examination	2.4

Compliance with the clauses of this standard provides on means of conforming with the specific essential requirements of the Directive concerned and associated EFTA regulations.

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EN 1092-2*, Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories, PN designated - Part 2: Cast iron flanges.*

prEN 1092-3, *Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories - Part 3: Copper alloy and composite flanges, PN designated*.

prEN 1092-4, *Flanges and their joints - Circular flanges for pipes, valves fittings and accessories, PN designated - Part 4: Aluminium alloy flanges*.

EN 13445-1, *Unfired pressure vessels - Part 1: General.*

EN 13445-2*, Unfired pressure vessels - Part 2: Materials.*

EN 13445-4, *Unfired pressure vessels - Part 4: Manufacture*.

EN 13445-5, *Unfired pressure vessels - Part 5: Inspection and testing*.

EN 13445-6, *Unfired pressure vessels - Part 6: Additional requirements for design and fabrication of pressure vessels and vessel parts constructed of spheroidal graphite cast iron.*

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