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Closed expansion vessels with built in diaphragm for installation in water

Vases d'expansion fermés avec membrane incorporée pour
installation dans des systèmes à eau

Ausdehnungsgefäße mit eingebauter Membrane für den
Einbau in Wassersystemen

This European Standard was approved by CEN on 26 July 2007.

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Foreword

This document (EN 13831:2007) has been prepared by Technical Committee CEN/TC 54 “Unfired pressure vessels”, the secretariat of which is held by BSI.

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of an identical text or by endorsement, at the latest by February 2008, and conflicting national standards shall be withdrawn at the latest by February 2008.

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

For relationship with EU Directive(s), see informative Annex ZA, which is an integral part of this document.

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

Introduction

Closed expansion vessels with built in diaphragm made their commercial début in the early 1950s. They were employed in heating systems, or for fresh water supply systems.

When used in heating systems they take up the increase of the water volume due to the heating up. The gas pressure (on the other side of the diaphragm) pushes the water back into the system when due to cooling down the water volume in the heating system is decreasing. Expansions vessels with built in diaphragm are an undisputed standard in European heating engineering. When used in fresh water circuits, vessels with built in diaphragm serve to accommodate the extra volume caused by water heaters warming up,, thus saving valuable drinking water from flowing down the drain. The other main application is to store water under pressure in connection with booster systems allowing an energy efficient pump operation.

Though the development of the closed expansion vessel with built in diaphragm constituted a real revolution in the domains of heating and drinking water, industry in general took only limited note of it. Nevertheless this has not prevented the manufacturers from refining the product and the manufacturing technique over the last 40 years, often charting entirely new paths. As a consequence, the production of closed expansion vessels can differ considerably from conventional pressure vessel production. This is especially true in respect to the highly developed deep drawing technology.

1 Scope

This European Standard specifies requirements for the design, manufacture and testing of closed expansion vessels with built in diaphragm, which will hereinafter be called "vessels", and

- a) whose diaphragm serves to separate water on the one hand and air / nitrogen on the other hand in heating/cooling systems or fresh water systems;
- b) which are manufactured singly or in series;
- c) which may consist partly or entirely of (cold) deep-drawn parts;
- d) whose parts may be joined by welding, clenching or flanges;
- e) whose size is not limited;
- f) whose maximum allowable pressure is greater than 0,5 bar, yet not exceeding 30 bar;
- g) whose upper wall thickness is limited to 12 mm for austenitic steels and 15 mm for ferritic steels;
- h) whose minimum operating temperature is not below $-10\text{ }^{\circ}\text{C}$ and whose maximum operating temperature is not above $70\text{ }^{\circ}\text{C}$.

NOTE The maximum operating temperature of $70\text{ }^{\circ}\text{C}$ is determined by the characteristics of the diaphragm materials. It may be higher, if suitability of diaphragm material is proven.

Whatever the temperature in the heating system, for the vessel operation the decisive factor is the maximum operating temperature of the diaphragm. It is the system designer's responsibility to prescribe measures to protect the diaphragm from unsuitable temperatures (e.g. connection to the coldest part of the system in a heating system, to the warmest in a refrigeration circuit; thermostatic monitoring of connection to vessel or intermediate vessel).

For cases where operating temperatures above $70\text{ }^{\circ}\text{C}$ cannot be avoided the suitability of the diaphragm material is to be proven (see Clause 8).

When reference is made in this European Standard to EN 13445-1, EN 13445-2, EN 13445-3, EN 13445-4 and EN 13445-5 respectively, all relevant provisions in the concerned clauses of these standards need to apply.

2 Normative references

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

EN 287-1:2004, *Qualification test of welders — Fusion welding — Part 1: Steels*

EN 473:2000, *Non destructive testing — Qualification and certification of NDT personnel — General principles*

EN 764-1:2004, *Pressure equipment — Part 1: Terminology — Pressure, temperature, volume, nominal size*

EN 764-2:2002, *Pressure equipment: terminology — Part 2: Quantities, symbols and units*

EN 764-3:2002, *Pressure equipment — Part 3: Definition of parties involved*

EN 895:1995, *Destructive tests on welds in metallic materials — Transverse tensile test*

EN 910:1996, *Destructive test on welds in metallic materials — Bend tests*

EN 1418:1997, *Welding personnel — Approval testing of welding operators for fusion welding and resistance weld setters for fully mechanized and automatic welding of metallic materials*

EN 1435:1997, *Non-destructive examination of welds — Radiographic examination of welded joints*

EN 10204:2004, *Metallic products — Types of inspection documents*

EN 10269:1999, *Steels and nickel alloys for fasteners with specified elevated and/or low temperature properties*

EN 13445-1:2002, *Unfired pressure vessels — Part 1: General*

EN 13445-2:2002, *Unfired pressure vessels — Part 2: Materials*

EN 13445-3:2002, *Unfired pressure vessels — Part 3: Design*

EN 13445-4:2002, *Unfired pressure vessels — Part 4: Fabrication*

EN 13445-5:2002, *Unfired pressure vessels — Part 5: Inspection and testing*

EN ISO 898-1:1999, *Mechanical properties of fasteners made of carbon steel and alloy steel — Part 1: Bolts, screws and studs (ISO 898-1:1999)*

EN ISO 15609-1:2004, *Specification and qualification of welding procedures for metallic materials — Welding procedure specification — Part 1: Arc welding (ISO 15609-1:2004)*

EN ISO 15613:2004, *Specification and qualification of welding procedures for metallic materials — Qualification based on pre-production welding test (ISO 15613:2004)*

EN ISO 15614-1:2004, *Specification and qualification of welding procedures for metallic materials — Welding procedure test — Part 1: Arc and gas welding of steels and arc welding of nickel and nickel alloys (ISO 15614-1:2004)*

ISO 898-2:1998, *Mechanical properties of fasteners — Part 2: Nuts with specified proof load values — Coarse thread*

3 Terms and conditions

For the purposes of this document, the terms and definitions given in EN 764-1:2004, EN 764-2:2002, EN 764-3:2002 and the following apply.

3.1

automatic welding

welding in which all the parameters are automatically controlled, some of these parameters may be adjusted to a limited amount (manually or automatically by mechanical or electronic devices) during welding to maintain the specified welding conditions

3.2

clench joints

separate metal ring holding together two vessel parts or a rolled joint holding together two vessel parts in a permanent way. Its design is always done according to the experimental design method

3.3

expansion vessel

vessel to take up the volume variations of a liquid due to changes of temperature. The expansion vessel is called "closed", if the liquid contained is not in contact with any gaseous or liquid medium

3.4

deep drawing

forming of vessel parts from a flat state into a three dimensional state by means of a press and tools whereby no material is taken off or added

3.5

diaphragm

flexible and / or elastic wall which is fastened into the vessel inside in a gas tight way and separates the vessel into a water and a gas space

3.6

experimental test

any kind of test used to substitute for the calculation of a vessel part or the whole vessel, within the framework of the experimental design method

3.7

inspection document

document according to EN 10204:2004

3.8

family of welded joints

welded joints covered by a specific welding procedure approval document

3.9

vessel family

vessels belong to one vessel family if they have similar geometrical proportions, same design and fall within the validity of one weld procedure approval

3.10

freshwater

water coming from a supply system (mains, well etc.), untreated apart from possible hygienic measures, with natural content of oxygen

4 Symbols and units

For the purposes of this document, the symbols and units given in EN 13445-1:2002, EN 13445-2:2002, EN 13445-3:2002, EN 13445-4:2002 and EN 13445-5:2002, EN 764-1:2004 and EN 764-2:2002 and the following apply.

Other symbols used in specific clauses of this European Standard are tabulated there.

Symbol	Description	Unit
A	Area	mm^2
A	elongation after fracture	%
d, D	Diameter	mm
e	Thickness	mm
f	nominal design stress for design conditions	MPa or N/mm^2
f_{test}	nominal design stress for testing conditions	MPa or N/mm^2
l	Length	mm
p	design pressure	bar, MPa or N/mm^2 ¹⁾
PS	maximum allowable pressure	bar, MPa or N/mm^2 ¹⁾
PT	test pressure	bar, MPa or N/mm^2 ¹⁾
r, R	Radius	mm
R_e	yield strength	MPa or N/mm^2
R_{eH}	upper yield strength	MPa or N/mm^2
R_m	tensile strength	MPa or N/mm^2
$R_{m/t}$	tensile strength at temperature t °C	MPa or N/mm^2
$R_{p0,2/t}$	0,2 % proof strength at temperature t °C	MPa or N/mm^2
$R_{p1,0/t}$	1,0 % proof strength at temperature t °C	MPa or N/mm^2
TS	temperature	°C
t_c	calculation temperature	°C
t_t	test temperature	°C
V	volume, capacity	m^3 , L, (l)
z	weld joint coefficient	[]

The unit bar is needed to meet the general terminology, and thus to be used on nameplates, certificates, drawings, pressure gauges and instrumentation.

1) MPA or N/mm^2 for calculation purpose only.

5 Materials

5.1 General

The grouping according to Table 5.1-1 is based on materials corresponding to steel groups 1; 1.1; 8.1 as defined in EN 13445-2:2002.

Table 1 — Definitions of steel groups (CEN ISO/TR 15608:2005)

Group/ Subgroup	Type of steel																														
1	Steels with a minimum yield strength $R_{eH} \leq 460 \text{ N/mm}^2$ ^a and with analysis in % (ladle analysis): <table style="margin-left: 20px; border: none;"> <tr> <td>C</td><td>$\leq 0,25$</td> <td>Cu</td><td>$\leq 0,40$</td> <td>^b</td> </tr> <tr> <td>Si</td><td>$\leq 0,60$</td> <td>Ni</td><td>$\leq 0,5$</td> <td>^b</td> </tr> <tr> <td>Mn</td><td>$\leq 1,70$</td> <td>Cr</td><td>$\leq 0,3$</td> <td>(0,4 for castings) ^b</td> </tr> <tr> <td>Mo</td><td>$\leq 0,70$</td> <td>Nb</td><td>$\leq 0,05$</td> <td></td> </tr> <tr> <td>S</td><td>$\leq 0,045$</td> <td>V</td><td>$\leq 0,12$</td> <td>^b</td> </tr> <tr> <td>P</td><td>$\leq 0,045$</td> <td>Ti</td><td>$\leq 0,05$</td> <td></td> </tr> </table>	C	$\leq 0,25$	Cu	$\leq 0,40$	^b	Si	$\leq 0,60$	Ni	$\leq 0,5$	^b	Mn	$\leq 1,70$	Cr	$\leq 0,3$	(0,4 for castings) ^b	Mo	$\leq 0,70$	Nb	$\leq 0,05$		S	$\leq 0,045$	V	$\leq 0,12$	^b	P	$\leq 0,045$	Ti	$\leq 0,05$	
C	$\leq 0,25$	Cu	$\leq 0,40$	^b																											
Si	$\leq 0,60$	Ni	$\leq 0,5$	^b																											
Mn	$\leq 1,70$	Cr	$\leq 0,3$	(0,4 for castings) ^b																											
Mo	$\leq 0,70$	Nb	$\leq 0,05$																												
S	$\leq 0,045$	V	$\leq 0,12$	^b																											
P	$\leq 0,045$	Ti	$\leq 0,05$																												
1.1	Steels with a minimum yield strength $R_{eH} \leq 275 \text{ N/mm}^2$ and composition as indicated under 1																														
8.1	Austenitic stainless steels with Cr ≤ 19 % (content used in steel designation)																														
^a In accordance with the specification of the steel product standards, R_{eH} may be replaced by $R_{p0,2}$ or $R_{t0,5}$.																															
^b A higher value is accepted provided that $\text{Cr} + \text{Mo} + \text{Ni} + \text{Cu} + \text{V} \leq 0,75$ %.																															

For a complete overview of steel grades falling into the above mentioned groups reference is made to EN 13445-2:2002.

5.2 Materials proven by experience and current use

The following materials do not fulfil all the requirements of groups 1, 1.1 and 8.1, but may be used for this type of product under the condition that there is sufficient ductility of the material after forming as it will be used is proven:

- EN 10025-2²⁾ grades S 235 J2, S 235 JR;
- EN 10130³⁾ grades Dc01, Dc03, Dc04;
- EN 10111³⁾ grades DD11, DD12, DD13, DD14.

2) See 6.3.2.5.

3) Restricted use, see 6.2.

5.3 Fasteners

Fasteners (bolts, nuts, studs) shall not be made from free cutting steel. Used steels shall have an elongation after fracture, A , of at least 14 %.

Bolts and screws in accordance with EN ISO 898-1:1999 property classes 5.6 or 8.8 and nuts to ISO 898-2:1998 property classes 5 or 8 but with an elongation of at least 12 %, shall be considered suitable.

EN 10269:1999 shall be taken into account.

5.4 Non-pressurised parts

For non-pressure parts welded to pressure vessels, materials shall be used which are supplied to material specifications covering at least the requirements for the chemical composition and the tensile properties. These materials shall not limit the operating conditions of the material to which they are attached.

6 Design and calculation

6.1 Design

6.1.1 Requirements pertaining to the diaphragm

Sharp edges and corners (grooves, welding beads etc.) are not permitted in those areas of the inside surface which will come into contact with the diaphragm.

Parts projecting into the vessel in such a way, that damage of the diaphragm can occur are not permitted.

Local concavities on the inner surface are only permitted if the maximum possible linear stretching of the diaphragm being pressed into the concavity is not above 10 % of the elongation at rupture of the diaphragm material.

Openings in the vessel wall shall be designed in such a way that the diaphragm cannot be damaged through impingement.

The gaps of joggled welds shall nowhere be bigger than twice the diaphragm wall-thickness.

The dimensions of the vessel and the diaphragm shall match so as to ensure that irrespective of charge pressure the diaphragm cannot be stretched to the point where it is damaged.

6.1.2 Requirements pertaining to fresh water application

Metal parts in contact with the water during normal operation of the vessel shall be of stainless steel, corrosion resistant or adequately protected against corrosion.

6.1.3 Outside finish

The vessel and its outside parts shall be so finished as to avoid injury (e.g. from burs and sharp edges). Vessels made of carbon steel shall be protected against ambient corrosion.

6.1.4 Inspection openings

6.1.4.1 Vessels with fixed diaphragms do not require openings.

6.1.4.2 Vessels with a removable diaphragm shall have an opening of sufficient size to exchange the diaphragm. This opening serves also for inspection purposes.

6.1.4.3 Vessels with removable diaphragm, with additional compartments, shall have an inspection opening of ≥ 30 mm inside diameter in the additional compartment such that the vessel can be inspected.

6.1.5 Connections

6.1.5.1 Minimum size of water connections shall be according to Table 2.

Table 2 — Minimum size of water connections

Vessel volume [L]	Connection
$V \leq 12$	DN 12 (0,375 ")
$12 < V \leq 25$	DN 15 (0,5 ")
$25 < V \leq 600$	DN 20 (0,75 ")
$600 < V \leq 1\ 000$	DN 25 (1 ")
$1\ 000 < V \leq 2\ 000$	DN 32 (1,25 ")
$2\ 000 < V$	DN 40 (1,5 ")

NOTE For vessels used in freshwater applications larger connections could be required, depending on the flow rate.

6.1.5.2 If the water connection is covered by any kind of sieve, its total free section shall be a least equal to the free section of the connection pipe as specified in 6.1.5.1.

6.1.6 Clenched joints

In the case of clenched joints the experimental design method shall be used to determine minimum wall thickness. There is therefore no calculation method for them in the European Standard.

Within the framework of a type approval, in deviation from 6.2.2, at least 6 vessels have to be tested according to 6.2.3 or 6.2.4.

In the case of different vessel sizes within a family, a minimum of 2 vessels per vessel size have to be tested according to 6.2.3 or 6.2.4 if $PS \times V \leq 1\ 000$ bar \times L.

In the case of different vessel sizes within a family, a minimum of 1 vessel per vessel size has to be tested according to 6.2.3 or 6.2.4 if $1\ 000$ bar \times L $<$ $PS \times V <$ $6\ 000$ bar \times L.

Circumferential measurements have to be carried out in the cylindrical part of the vessel above and below the clenched joint. The maximum allowable permanent deformation shall not be higher than 1 % (see 6.2.3 and 6.2.4).

NOTE Owing to the wide variety of versions and designs of this type of joint it is impossible to indicate further dimensions or physical properties.

6.1.7 Volume tolerance of vessels

The actual volume of the vessel measured without the diaphragm, shall be a minimum of 95 % of the (nominal) volume declared by the manufacturer.

6.1.8 Fatigue

Expansion vessels as covered by this European Standard are operated in such a way that no relevant fatigue load occurs.

6.1.9 Loadings

Expansion vessels are normally installed inside buildings and external loadings are not significant. For vessels where weight or static head of the fluid have to be taken into account, reference should be made to EN 13445-3:2002. The same applies where the specification for the vessel stipulates conditions which lead to special loadings (e.g. earthquake loadings).

6.2 Experimental design method

6.2.1 General

The design for adequate strength may be determined by the use of the experimental design method for vessels with a $PS \times V < 6\,000 \text{ bar } L$.

EN 10130, grades Dc01, Dc03 and Dc04 and EN 10111, grades DD11, DD12, DD13 and DD14 may only be used in accordance with 5.2 when the design is verified according to the experimental design method in this subclause. Since these are intended for deep drawing the mechanical values in their respective standards do not lend themselves to the calculation method of 6.3.

The minimum wall thickness shall not be less than 0,8 mm at any point.

6.2.2 Preparations

The experimental test is performed on that vessel size within a family which will give the least favourable results under pressure (normally the one with the biggest diameter).

Vessels to be submitted to the test have to be identical to normal production with the exception that they shall not contain a complete diaphragm.

If for production reasons the complete diaphragm has to be built into the vessel, holes shall be made into the diaphragm to ensure that there is water on both sides of the diaphragm, so that leaks will then be visible irrespective of their position relative to the diaphragm.

If the vessel selected fails, two more vessels of the same size shall be submitted to the same test. The design is only acceptable if both vessels then pass the test. The water used for testing shall be at room temperature. The permanent deformation (elongation of shell) shall be measured along the shortest circumference of the vessel.

A report of the test shall be drawn up giving all necessary information so as to validate the test results including material certificates for the main parts of the vessel.

6.2.3 Vessels with $PS \times V \leq 1\,000 \text{ bar } \times L$

The vessels to be tested shall be completely filled with water, then pressurised up to $2 \times PS (-0\% + 5\%)$ and held at this pressure for 5 min. No leaks shall occur during this time. The permanent deformation shall not be higher than 1 %.

6.2.4 Vessels with $1\,000 \text{ bar } \times L < PS \times V < 6\,000 \text{ bar } \times L$

The vessels to be tested shall be completely filled with water, then pressurised up to $3 \times PS (-0\% + 5\%)$ and held at this pressure for 5 min. No leaks shall occur during this time. The permanent deformation shall not be higher than 1 %.

6.2.5 Vessel parts and components

When the experimental design method is used for vessel parts and components that are pressurized, they shall be subjected to an experimental test of $3 \times PS (-0\% + 5\%)$ and held at this pressure for 5 min. No leaks shall occur during this time.

6.3 Calculation method

6.3.1 General

This clause covers the calculation of those parts and components which are normally used for expansion vessels.

For cases not covered by this clause, reference should be made to EN 13445-3:2002.

6.3.2 Symbols

6.3.2.1 General

For symbols in general, reference should be made to Clause 4. Symbols used only in specific clauses are listed at the beginning of the respective clause.

6.3.2.2 Design pressure

The design pressure p is determined by the vessel manufacturer and cannot be lower than PS .

6.3.2.3 Calculation temperature

The temperature t_c to be used for the calculation of the vessel shall be determined by the vessel manufacturer within the limits $-10\text{ °C} / +110\text{ °C}$. It shall not be lower than the design temperature.

NOTE The upper limit of 110 °C is purely for calculation purposes. It is not relevant for the vessel operation since apart from very special rubber materials the diaphragm in the vessel is not to be operated at temperatures above 70 °C .

6.3.2.4 Nominal design stresses

Steels other than austenitic

$$f = \text{MIN} \left(\frac{R_{p0,2/t}}{1,5}; \frac{R_{m/20}}{2,4} \right) \quad (1)$$

When $R_{p0,2t}$ is not specified in the material standard $f = \frac{R_{m/20}}{2,8}$

Austenitic steels with $A \geq 30\%$

$$f = \frac{R_{p1,0/t}}{1,5} \quad (2)$$

Austenitic steels with $A \geq 35\%$

$$f = \frac{R_{p1,0/t}}{1,5} \text{ or } \text{MIN} \left(\frac{R_{p1,0/t}}{1,2}; \frac{R_{m/t}}{3} \right) \quad (3)$$

6.3.2.5 Weld joint coefficient

The weld joint coefficient shall be chosen by the manufacturer:

$z = 1$ NDT on every vessel;

$z = 0,85$ spot NDT;

$z = 0,7$ only visual examination.

6.3.2.6 Allowances

6.3.2.6.1 Allowance to compensate for plate thickness and manufacturing tolerances

For ferritic steels, the thickness tolerance to be used in the calculation is the negative tolerance in the relevant dimensional standard for the finished component.

Where the manufacturing process entails reduction in thickness the minimum required wall thickness shall be stated on the drawing.

6.3.2.6.2 Corrosion allowance

No allowance for corrosion is made in this European Standard.

NOTE Water in normal heating systems is not considered corrosive. Vessels in drinking water systems need to have metal parts protected against corrosion (see 6.1.2).

6.3.3 Cylindrical and spherical shells under internal pressure

6.3.3.1 Specific symbols

D_e outside diameter of the shell

e required shell thickness

6.3.3.2 Cylindrical shells

The required wall thickness e is given by

$$e = \frac{p \cdot D_e}{2f \cdot z + p} \quad (4)$$

6.3.3.3 Spherical shells

The required wall thickness e is given by

$$e = \frac{p \cdot D_e}{4f \cdot z + p} \quad (5)$$

6.3.4 Dished ends under internal pressure

6.3.4.1 General

These design rules apply to dished ends of the torispherical, ellipsoidal and hemispherical type, not welded (i.e. made of one piece of sheet metal).

6.3.4.2 Specific symbols

β function of e/R and r/D_i for torispherical ends given by Figure 2

e required thickness of the knuckle

D_e outside diameter of end

D_i inside diameter of end

- h_i inside height of ellipsoidal end
- $K = D_i / (2 h_i)$, shape factor for an ellipsoidal end
- R inside spherical radius of central part of torispherical end
- r inner knuckle radius

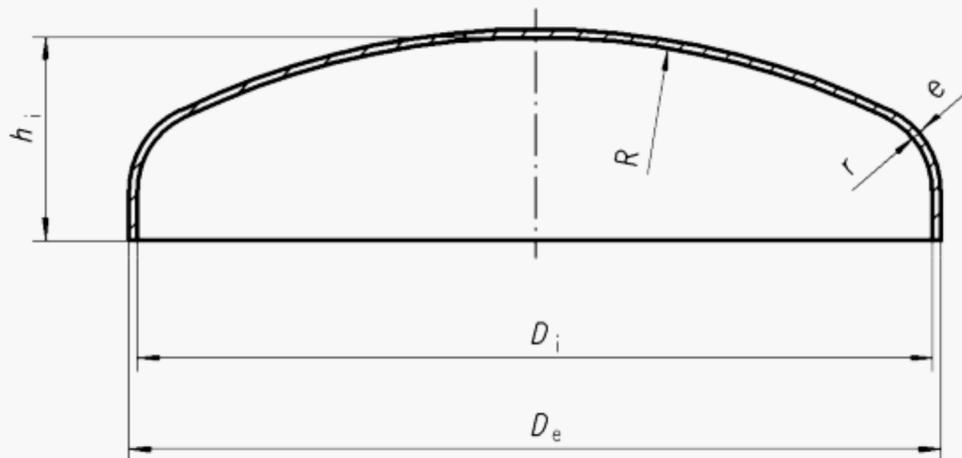


Figure 1 — Dished end

6.3.4.3 Hemispherical ends

The required thickness of a hemispherical end is given by Equation (5).

6.3.4.4 Torispherical ends

A torispherical end is made up of a spherical cap, a toroidal knuckle and a cylindrical shell, the three components having tangents where they meet.

The following rules are limited to ends for which:

- a) $r \leq 0,2 D_i$
- b) $r \geq 0,06 D_i$
- c) $r \geq 2 e$
- d) $e \leq 0,08 D_e$
- e) $e > 0,005 D_i$ (if this condition is not fulfilled, e has to be checked additionally for plastic buckling. See EN 13445-3)
- f) $R \leq D_e$

The required thickness e is the greater of the following values

$$e = \frac{\beta \cdot p \cdot (0,75 \cdot R + 0,2 \cdot D_i)}{f} \quad (6)$$

$$e = \frac{p \cdot R}{2f \cdot z - 0,5p} \quad (7)$$

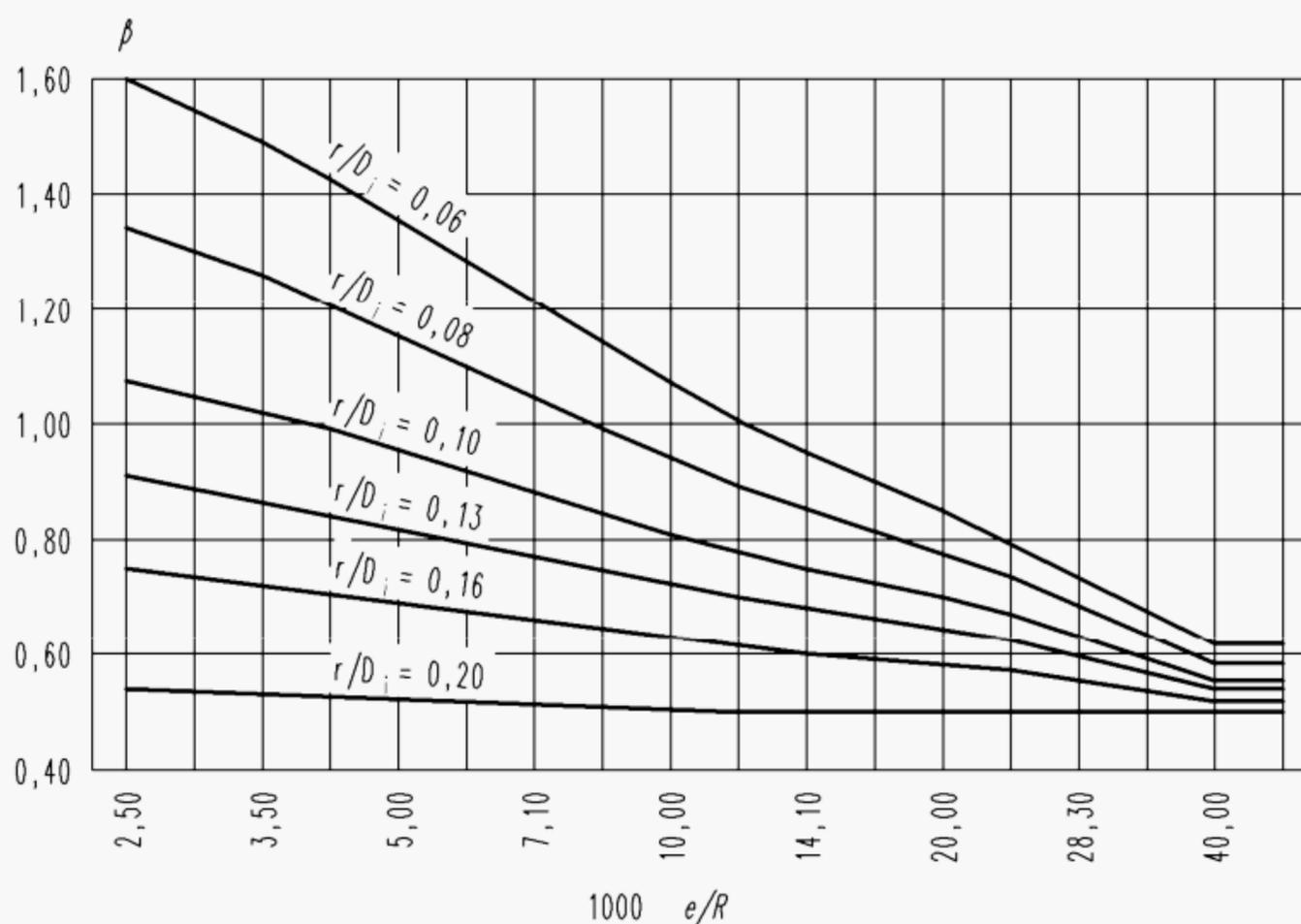


Figure 2 — Torispherical end design (factor β)

6.3.4.5 Ellipsoidal ends

An ellipsoidal end is made on a truly ellipsoidal former.

These rules apply only to ends for which $1,7 < K < 2,2$.

Ellipsoidal ends shall be designed as nominally equivalent torispherical ends with:

$$r / D_i = 0,5 / K - 0,08 \text{ and} \quad (8)$$

$$R / D_i = 0,44 \cdot K + 0,02 \quad (9)$$

6.3.4.6 Nozzles intruding into the knuckle region

For calculation of nozzles intruding into the knuckle region see EN 13445-3.

6.3.5 Openings in cylindrical shells, spherical shells and dished ends

6.3.5.1 General

The design method in this clause is applicable to cylindrical shells, spherical shells and dished ends having circular openings. It does not apply for adjacent openings, elliptical or obround openings, oblique nozzles or openings close to discontinuities (e.g. weld seams). For this, reference should be made to EN 13445-3:2002.

Cylindrical shells, spherical shells and dished ends having an opening shall be adequately reinforced. The reinforcement area of a shell having an opening cannot be calculated directly but shall be assumed in the first instance. This assumption has to be verified. If the verification's result is not sufficient, the calculation shall be repeated using a modified assumption. Nozzles, reinforcing rings and compensating plates can only be considered for reinforcement of openings if they are welded to the shell with adequate welds. They can be used in combination with one another or singly. A reinforcement shall be sufficient in all planes through the axis of the opening or nozzle.

This subclause does not apply for reinforcing by increased wall thickness of a nozzle or / and the shell.

6.3.5.2 Symbols and Units

a	distance, taken along the average wall surface on the clause where the reinforcement of an opening has to be calculated between the openings centre and the external edge of a nozzle or of a ring; if no nozzle or ring is present, a is the distance between the centre and the internal edge of the opening	mm
A_f	stress loaded cross sectional area effective as compensation	mm ²
A_{fb}	A_f of nozzle	mm ²
A_{fp}	A_f of compensating plate	mm ²
A_{fr}	A_f of reinforcing ring	mm ²
A_{fs}	A_f of shell wall (main body)	mm ²
A_p	pressure loaded area	mm ²
A_{ps}	A_p of shell (main body)	mm ²
A_{pb}	A_p of nozzle	mm ²
A_{pr}	A_p of reinforcing ring	mm ²
d	diameter (or maximum width) of opening, or inside diameter of a nozzle	mm
d_e	outside diameter	mm
d_{eb}	d_e of nozzle	mm
d_{ep}	d_e of compensating plate	mm
d_{er}	d_e of reinforcing ring	mm
d_i	inside diameter	mm
d_{ib}	d_i of nozzle (d of set-in nozzle)	mm
d_{ip}	d_i of compensating plate	mm
d_{ir}	d_i of reinforcing ring	mm
D_e	external diameter of cylindrical or spherical shell at the centre of an opening	mm
D_i	internal diameter of cylindrical or spherical shell at the centre of an opening	mm
e_b	required thickness of nozzle (or mean thickness within the length l_{bo} or l_{bo})	mm
$e_{a, b}$	analysis thickness of nozzle useful for reinforcement	mm
e_p	required thickness of compensating plate	mm

$e_{a,p}$	analysis thickness of compensating plate useful for reinforcement	mm
e_r	required thickness of reinforcing ring	mm
$e_{a,r}$	analysis thickness of reinforcing ring useful for reinforcement	mm
$e_{a,s}$	analysis thickness of shell wall or mean analysis thickness within the length l'_s	mm
e'_s	length of penetration of nozzle into shell wall for set-in nozzles with partial penetration	mm
f	nominal design stress at design temperature	MPa
f_b	f of nozzle material	MPa
f_p	f of compensating plate material	MPa
f_r	f of reinforcing ring material	MPa
f_s	f of shell (main body) material	MPa
h	inside height of a dished end, excluding cylindrical skirt	mm
l_b	length of nozzle extending outside the shell	mm
l'_b	effective length of nozzle outside the shell, useful for reinforcement	mm
l_{bi}	length of nozzle extending inside the shell	mm
l'_{bi}	effective length of nozzle inside the shell, useful for reinforcement	mm
l_{bo}	maximum length of nozzle outside the shell, useful for reinforcement	mm
l_{bio}	maximum length of nozzle inside the shell, useful for reinforcement	mm
l_s	length of shell, from edge of opening or from external diameter of nozzle, to a shell discontinuity	mm
l'_s	effective length of shell useful for opening reinforcement	mm
l_p	width of compensating plate	mm
l'_p	effective width of compensating plate useful for reinforcement	mm
l_r	width of reinforcing ring	mm
l'_r	effective width of reinforcing ring useful for reinforcement	mm
l'_{so}	maximum length of shell (main body) contributing to opening reinforcement, taken on the mean radius of curvature of the shell wall	mm
L_b	centre-to-centre distance between two openings or nozzles taken on the mean radius of curvature of the shell (main body) wall	mm
p	pressure	MPa
r_{is}	inside radius of curvature of the shell (main body) at the opening centre	mm
R	inside radius of curvature of the spherical part of a torispherical end or inside radius of a hemispherical end	mm
w	distance between edge of opening and shell discontinuity	mm
w_{min}	minimum allowed value for w	mm

6.3.5.3 Limitations

Shell reinforced openings without a nozzle and/or reinforcing plate as well as those reinforced exclusively by a reinforcing plate the ratio $d / (2 \times r_{is})$ shall not exceed 0,5.

For openings in cylindrical shells reinforced by nozzles it shall not exceed 1.

On dished ends openings, nozzles, compensating plates and reinforcing rings shall be completely located inside the central area limited by a radius equal to $0,4 D_e$.

For spherical shells and dished ends the ratio d / D_e shall not exceed the value of 0,6.

Openings on cylindrical shells close to discontinuities (Figure 3) shall satisfy the following condition:

$$w \geq w_{\min} = \text{MAX} (0,2 \cdot l_{\text{so}}, 3e_{\text{a,s}}) \tag{10}$$

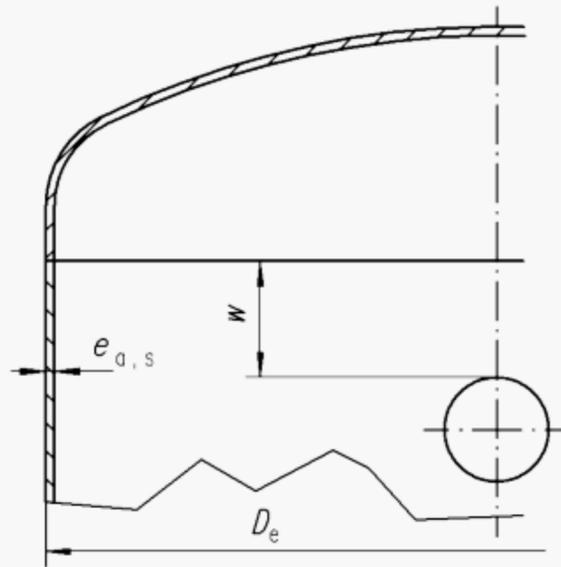


Figure 3 — Discontinuities in cylindrical shells

If an opening diameter (or maximum width) d satisfying the following condition:

$$d \leq 0,15 \sqrt{(2 \cdot r_{\text{is}} + e_{\text{a,s}}) \cdot e_{\text{a,s}}} \tag{11}$$

it is considered a “small opening” and needs no reinforcement.

6.3.5.4 Cylindrical shells, spherical shells and dished ends with isolated openings

6.3.5.4.1 Isolated openings

Adjacent openings or nozzles may be regarded as isolated openings if the minimum centre-to-centre distance L_b between the openings or nozzles taken along the average shell surface satisfies the following condition:

$$L_b \geq a_1 + a_2 + l_{\text{so1}} + l_{\text{so2}} \tag{12}$$

where

l_{so1} and l_{so2} are calculated for each opening, and where a_1 and a_2 are the straight or arched distances (taken on the mean radius) along L_b from the centre until the external diameter of each opening as shown in Figure 4:

$$l_{\text{so}} = \sqrt{(2 \cdot r_{\text{is}} + e_{\text{a,s}}) \cdot e_{\text{a,s}}} \tag{13}$$

where

— r_{is} is the inside radius of curvature of the shell at the centre of each opening, i.e.:

$$\text{for cylindrical or spherical shells } r_{\text{is}} = (D / 2) - e_s \tag{14}$$

$$\text{for hemispherical or torispherical ends } r_{\text{is}} = R \tag{15}$$

$$\text{for elliptical ends } r_{\text{is}} = 0,44 \cdot D_i^2 / (2h) + 0,02 D_i \tag{16}$$

- $e_{a,s}$ is the analysis wall thickness of the shell, or the mean analysis thickness on the length l_{so} without taking into account any compensation plate thickness.

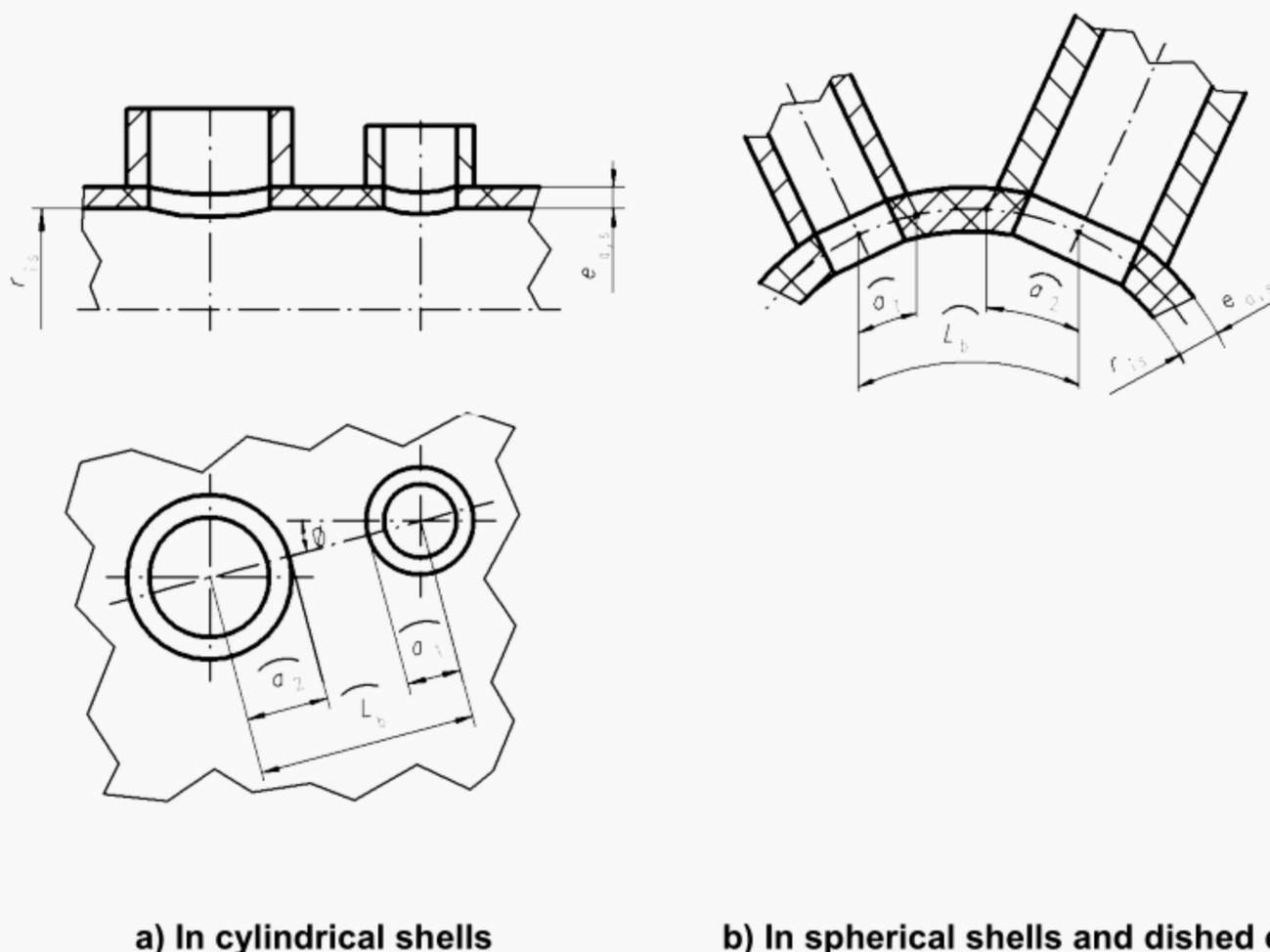


Figure 4 — Openings

6.3.5.4.2 General formula for reinforcement of openings

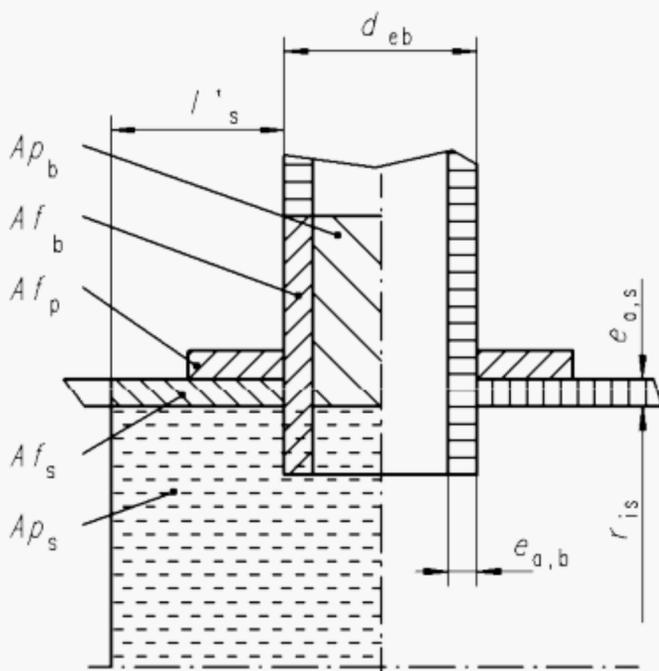
For isolated openings the following general condition shall be satisfied (Figure 5):

$$A f_s \cdot (f_s - 0,5p) + A f_p \cdot (f_{op} - 0,5p) + A f_b \cdot (f_{ob} - 0,5p) \geq p \cdot (A p_s + A p_b) \quad (17)$$

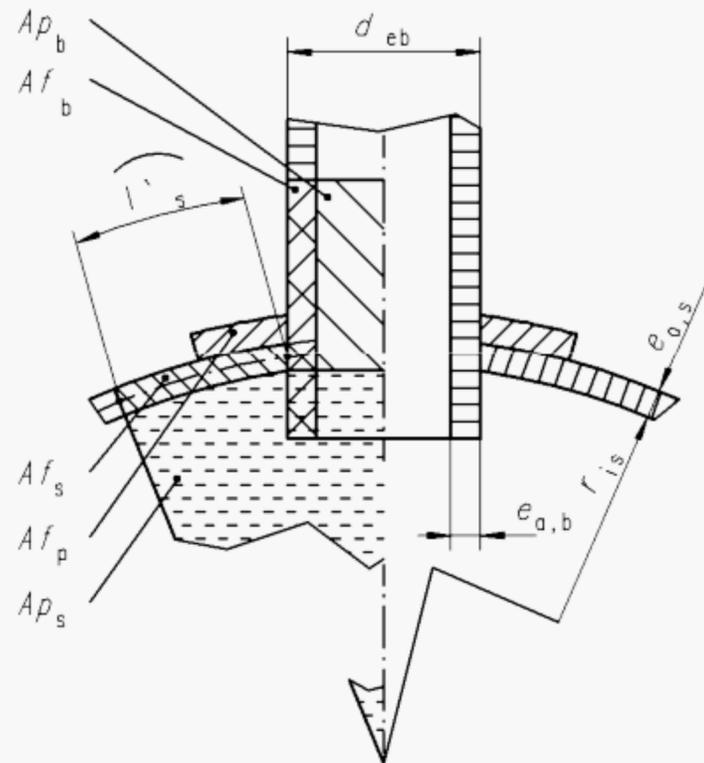
Terms related to nozzle may be substituted by terms related to reinforcing ring; moreover

$$f_{ob} = \text{MIN} (f_s; f_b) \quad (18)$$

$$f_{op} = \text{MIN} (f_s; f_p) \quad (19)$$



a) In cylindrical shells



b) In spherical shells and dished ends

Figure 5 — Reinforcement of isolated opening

For openings reinforced by nozzles, plates or rings the following formulae apply (where $l'_s = \text{MIN}(l_{so}; l_s)$):

- for cylindrical shells, calculation on the longitudinal cross section

$$Ap_s = r_{is} \cdot (l'_s + a) \tag{20}$$

$$\text{where } a = 0,5 d_{eb} \tag{21}$$

- for dished ends, spherical shells, cylindrical shells when the calculation is needed in the transverse cross section:

$$Ap_s = 0,5 r_{is}^2 \cdot (l'_s + a) / (0,5 e_{a,s} + r_{is}) \tag{22}$$

$$a = r_{ms} \arcsin(d_{eb} / (2 \cdot r_{ms})) \tag{23}$$

arcsin in radiant

where

$$r_{ms} = r_{is} + 0,5 e_{a,s} \tag{24}$$

Moreover:

- for set-in nozzles, reinforcing plates or rings:

$$Af_s = l'_s \cdot e_{a,s} \tag{25}$$

- for set-on nozzles:

$$Af_s = e_{a,s} \cdot (e_{a,b} + l'_s) \tag{26}$$

6.3.5.4.3 Reinforcement by compensating plates

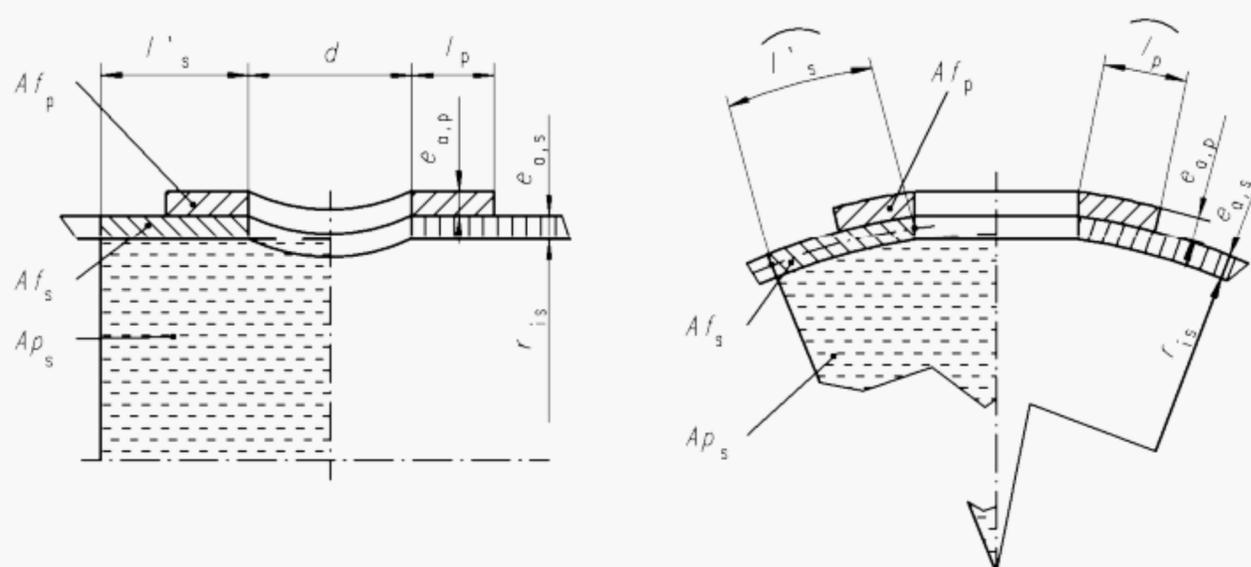
The width of a compensating plate l'_p to be considered as contributing to reinforcement is (see Equation (6.3–13) and Figure 6.3–6):

$$l'_p = \text{MIN} (l_{so}; l_p) \quad (27)$$

The value of e_p used for the calculation of Af_p shall not exceed the thickness $e_{a,s}$ of the shell

$$e_p = \text{MIN} (e_{a,p}; e_{a,s}) \quad (28)$$

Furthermore condition 6.3–17 shall be satisfied, where $Af_p = e_{a,p} \cdot l'_p$. (29)



a) In cylindrical shells

b) In spherical shells and dished ends

Figure 6 — Reinforcement by compensating plates

6.3.5.4.4 Reinforcement by reinforcing rings

The width l'_r of the ring considered as contributing to the reinforcement shall be (see Equation (13) and Figure 6.3–7):

$$l'_r = \text{MIN} (l_{so}; l_r) \quad (30)$$

The value of e_r used for the calculation Af_r shall be:

$$e_r = \text{MIN} (e_{a,r}; \text{MAX}(3e_{a,s}; 3l_r)) \quad (31)$$

Furthermore equation (6.3–17) shall be satisfied, where

$$Af_r = e_r \cdot l'_r \quad (32)$$

$$Ap_r = 0,5d_{ir} e_r \quad (33)$$

$$Af_s = e_{a,s} \cdot l_s \quad (34)$$

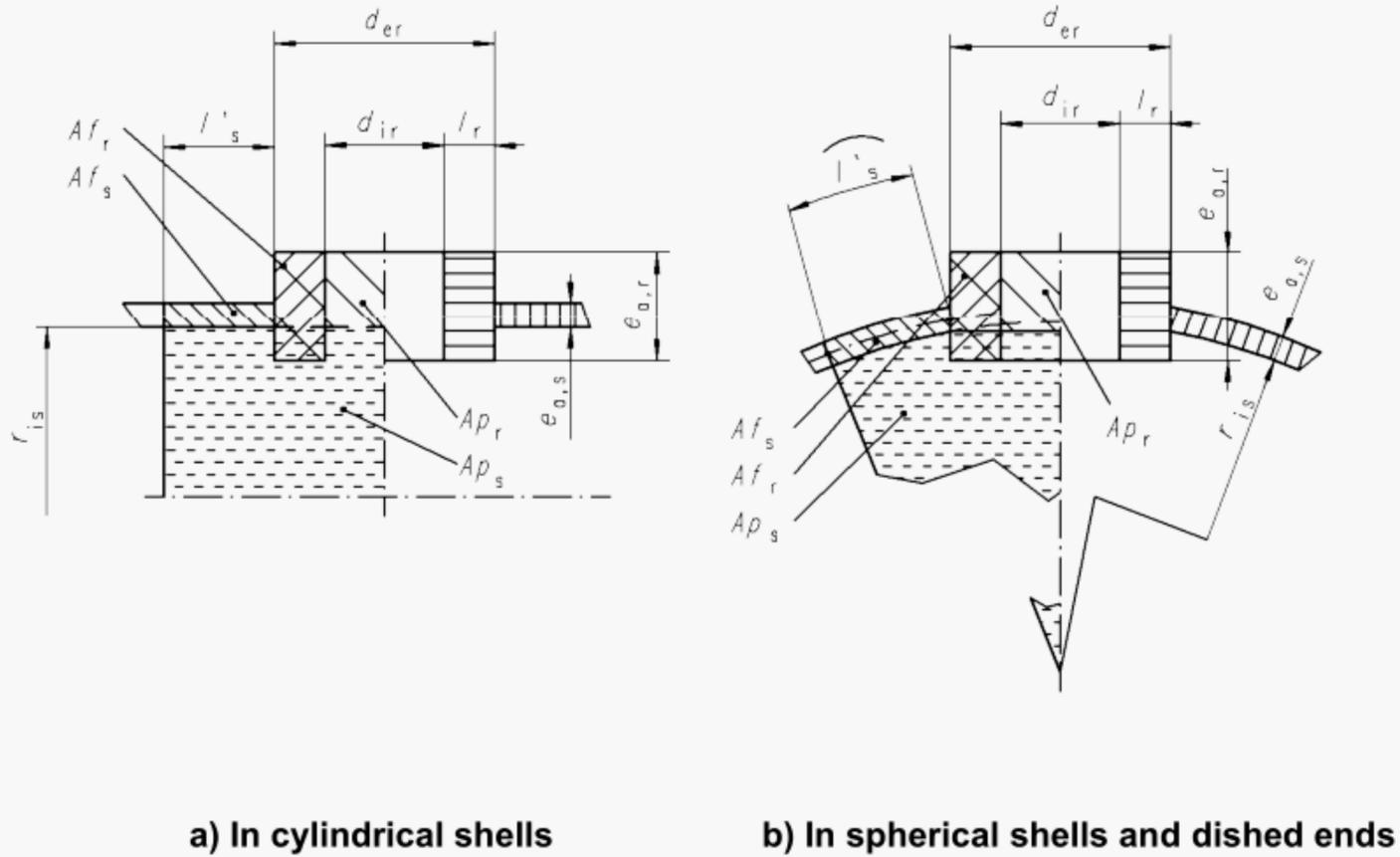


Figure 7 — Reinforcement by reinforcing rings

6.3.5.4.5 Reinforcement by nozzles (branches)

The maximum nozzle length contributing to the reinforcement shall not be greater than

l_{bo} with:

$$l_{bo} = \sqrt{(d_{eb} + e_{a,b}) \cdot e_{a,b}} \tag{35}$$

The maximum value to be used in the calculation for the inside protruding part of the nozzle in the case of set-through nozzles shall be:

$$l'_{bi} = \text{MIN} (l_{bi}; 0,5 l_{bo}) \tag{36}$$

The condition of equation 17 shall be satisfied, where (see Figure 8):

— for set-in nozzle:

$$Af_b = e_{a,b} \cdot (l_b' + l_{bi}' + e_s') \tag{37}$$

— for set-on nozzle:

$$Af_b = e_{a,b} \cdot l_b' \tag{38}$$

with: $l_b' = \text{MIN} (l_{bo}; l_b)$ (39)

e_s' = length of penetration (full or partial) of set-in nozzle into shell wall ($\leq e_{a,s}$)

$$Ap_b = 0,5 \cdot d_{ib} \cdot (l_b' + e_{a,s}) \tag{40}$$

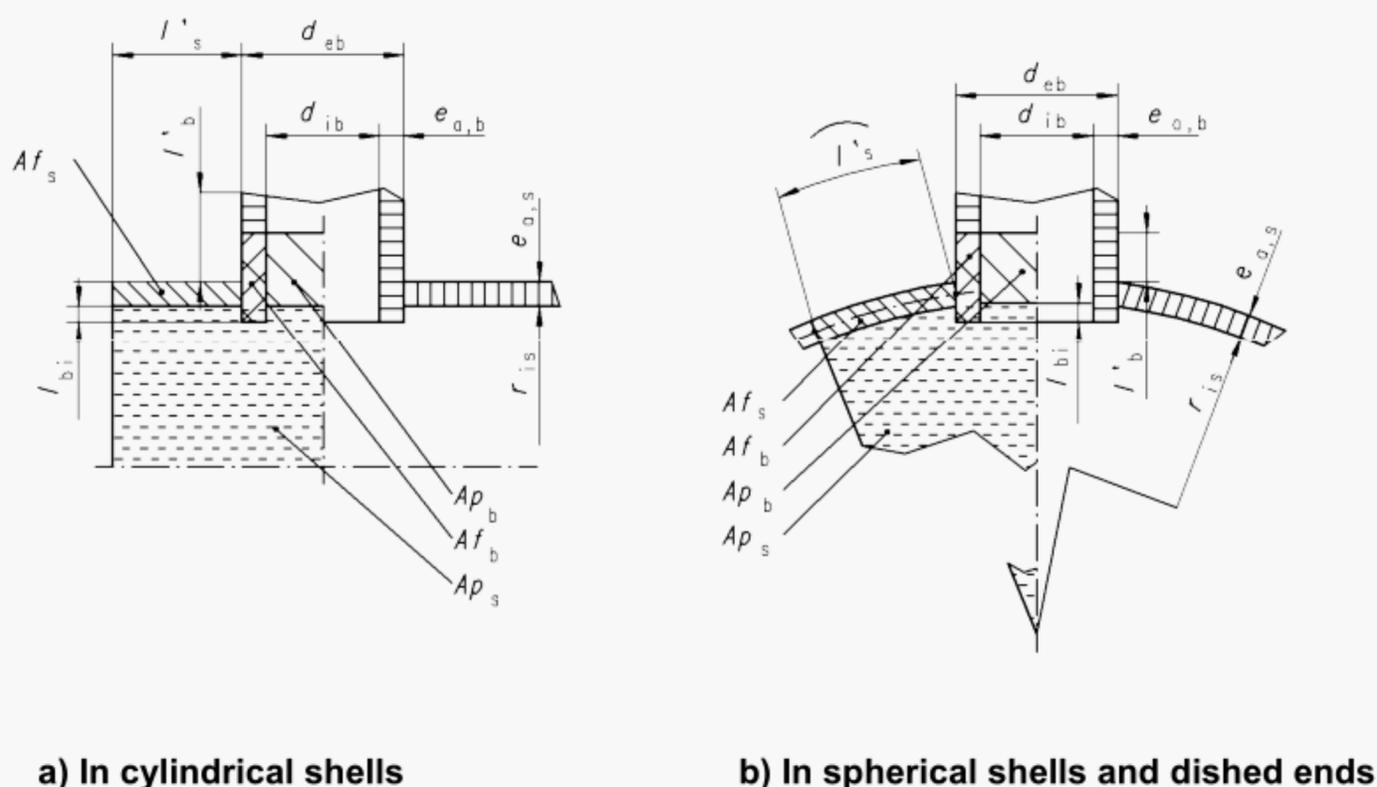


Figure 8 — Reinforcement by nozzles

6.3.6 Bolted circular flat ends under internal pressure

6.3.6.1 General

These design rules apply to unstayed circular flat ends under internal pressure connected to the vessel by bolting, and take into account reinforcement of openings.

For all other kinds of flat ends reference should be made to EN 13445-3:2002.

6.3.6.2 Symbols

A	nozzle reinforcement area	mm^2
b	effective gasket or joint width, $b = b_0$ when $b_0 < 6,3$ mm $b = 2,25 \sqrt{b_0}$ when $b_0 > 6,3$ mm, (b_0 is the basic gasket or joint seating width = $w / 2$ with the exception of the ring-joint for which $b_0 = w / 8$)	mm
d	diameter of an opening	mm
d_i	nozzle inside diameter	mm
d_e	nozzle outside diameter	mm
D_i	inside diameter of the cylindrical shell / pipe / opening to which the end is bolted	mm
C	diameter of the bolt circle	mm
G	mean diameter of gasket	mm
e	minimum required wall thickness of the end	mm

e_b	minimum required wall thickness of nozzle under internal pressure	mm
e_o	minimum required wall thickness of end without opening	mm
e_1	minimum required wall thickness for the peripheral area of the end	mm
f	nominal design stress of end material at temperature	MPa
f_{amb}	nominal design stress of end material at room temperature	MPa
W_{amb}	total tensile force of bolts for the gasket seating condition	N
k	distance between centre lines of two adjacent openings	mm
m	tightening coefficient of gasket (Table 1)	[]
$e_{a,b}$	analysis thickness of external section of a nozzle	mm
$e'_{a,b}$	analysis thickness of the internal protrusion of a nozzle	mm
f_b	nominal design stress at calculation temperature of the nozzle	MPa
l	external length of a nozzle effective for reinforcement	mm
l'	internal length on a protruding nozzle effective for reinforcement	mm
w	contact width of gasket, as limited by gasket width and flange facing	mm
Y_2	calculation coefficient of opening reinforcement	[]
y	gasket seating pressure (Table 1)	N/mm ²

6.3.6.3 Unstayed flat circular bolted ends without openings

6.3.6.3.1 General

These design rules relate to the following ends:

- gaskets entirely within the bolt circle (narrow-face);
- gaskets on both sides of the bolt circle (full-face).

The ends may or may not be of uniform thickness. However the minimum required wall thickness has to be extended to the entire surface located inside the gasket.

6.3.6.3.2 Circular ends with gasket entirely within the bolt circle

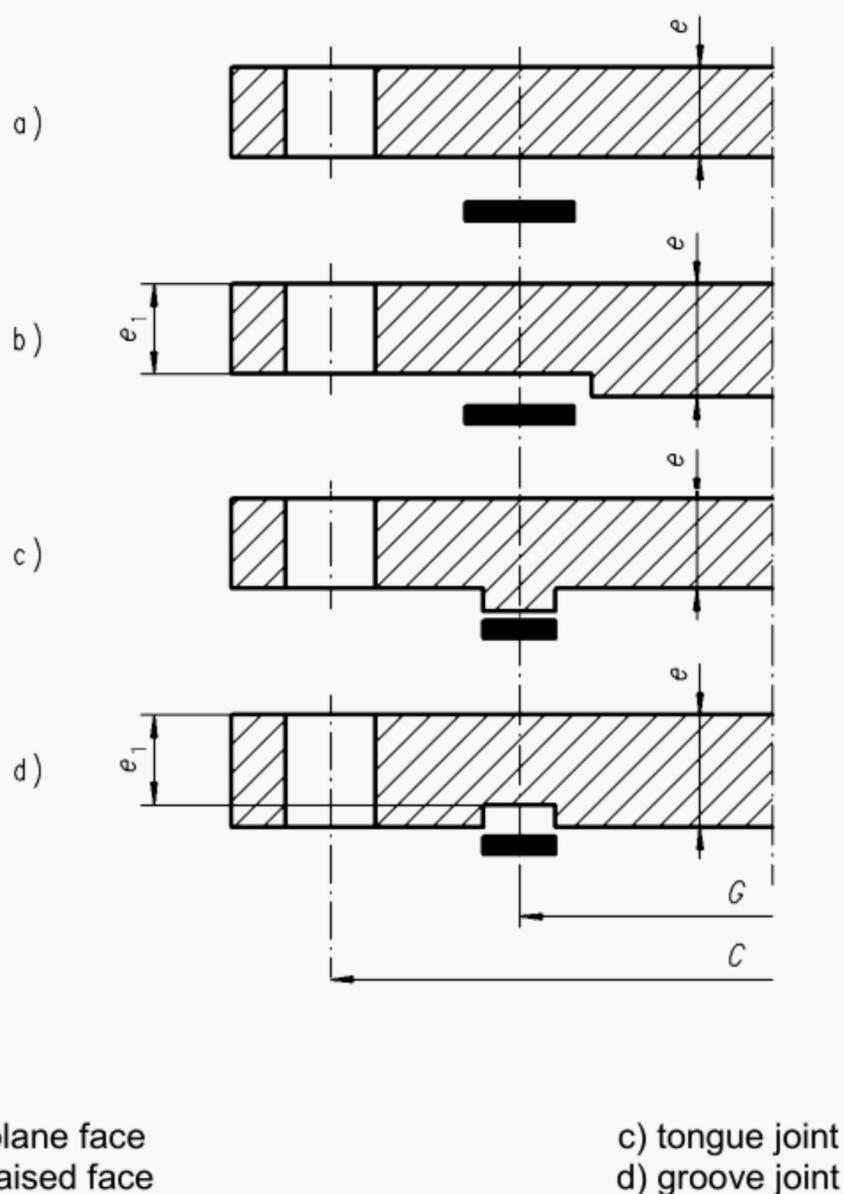


Figure 9 — Bolted circular flat ends with gasket entirely within the bolt circle

The minimum required wall thickness e for the end is the greater value calculated from Equations (41) and (42) respectively.

— For the gasket seating condition:

$$e_A = \sqrt{\frac{3}{\pi} \frac{(C - G) \cdot W_{amb}}{G \cdot f_{amb}}} \quad (41)$$

— For each pressure condition:

$$e_p = \sqrt{\left[0,31 \cdot G^2 + 3 \left(\frac{G}{4} + 2 \cdot b \cdot m \right) \cdot (C - G) \right] \cdot \frac{P}{f}} \quad (42)$$

W_{amb} is given by the following equation:

$$W_{amb} = \pi \cdot b \cdot G \cdot y \quad (43)$$

The minimum required wall thickness e_1 for the peripheral area end is given by Equation (41) or the following for each pressure condition whichever is greater.

$$e_{pl} = \sqrt{3 \left(\frac{G}{4} + 2 \cdot b \cdot m \right) \cdot (C - G) \cdot \frac{p}{f}} \quad (44)$$

6.3.6.3.3 Circular ends with gasket on both sides of the bolt circle (full faced gasket flange)

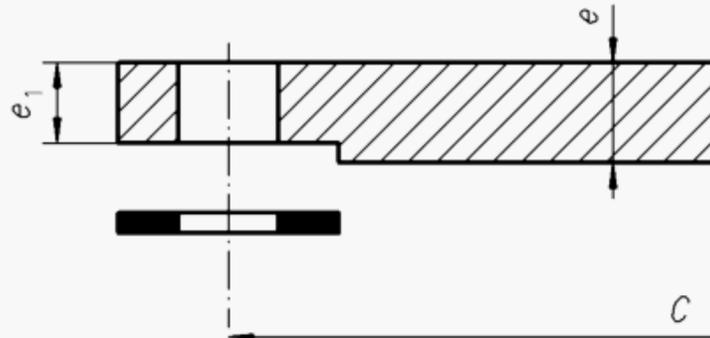


Figure 10 — Bolted circular flat end with a gasket on both sides of the bolts

The minimum required wall thickness for the end is given by the following equation:

$$e = 0,41 \cdot C \cdot \sqrt{\frac{p}{f}} \quad (45)$$

The minimum required wall thickness for the peripheral area of the end is given by:

$$e_1 = 0,8 e \quad (46)$$

The reduced thickness of the flanged extension shall be limited to a crown area whose internal circumference is not smaller than 0,7 C.

Table 3 — Gasket factors (m) and minimum design seating pressures (y)

Gasket material	Gasket factor m	Minimum design seating stress y N/mm ²
Rubber without fabric or high percentage of asbestos ^a fibre: below 75° ^b BS and IRH 75° ^b BS and IRH or higher	0,50 1,00	0,0 1,4
Asbestos ^a with a suitable binder for the operating conditions	3,2 mm thick 1,6 mm thick 0,8 mm thick	2,00 2,75 3,50
Rubber with cotton fabric insertion		1,25 2,8
Rubber with asbestos ^a fabric insertion, with or without wire reinforcement	3-play 2-play 1-play	2,25 2,5 2,75
Vegetable fibre		1,75 7,6
Rubber O-rings: below 75° ^b BS 75° ^b BS and 85° ^b BS and IRH		0 to 0,25 0,7 1,4
Rubber square section rings: below 75° ^b BS and IRH 75° ^b BS and 85° ^b BS and IRH		0 to 0,25 1,0 2,8
For other gasket material reference should be made to EN 13445-3.		
^a In many countries materials containing asbestos are prohibited. New non-asbestos bonded fibre sheet gaskets are not necessarily direct substitutes for asbestos based materials. Use within the manufacturer's recommendations.		
^b 75° corresponds to 85 Shore A, 85° to 95° Shore A.		

6.3.6.4 Reinforcement of openings in unstayed flat circular ends

These design rules apply to reinforcement of single or multiple openings in unstayed circular flat ends, provided their diameter is smaller than 50 % of the gasket mean diameter G/C .

Blind screwed holes of stud-bolts for connection to standard pipe flanges do not need reinforcement, provided they are located around an opening having a diameter not greater than the maximum bore diameter of the standard flange which should be bolted to that opening, and provided the thickness at the bottom of the bore is at least 50 % of the stud-bolt diameter.

For bolted ends the wall thickness shall be:

$$e = Y_2 e_0 \quad (47)$$

where

$$Y_2 = \sqrt{\frac{k}{k-d}} \quad (48)$$

where k for a single opening is equal to D_i

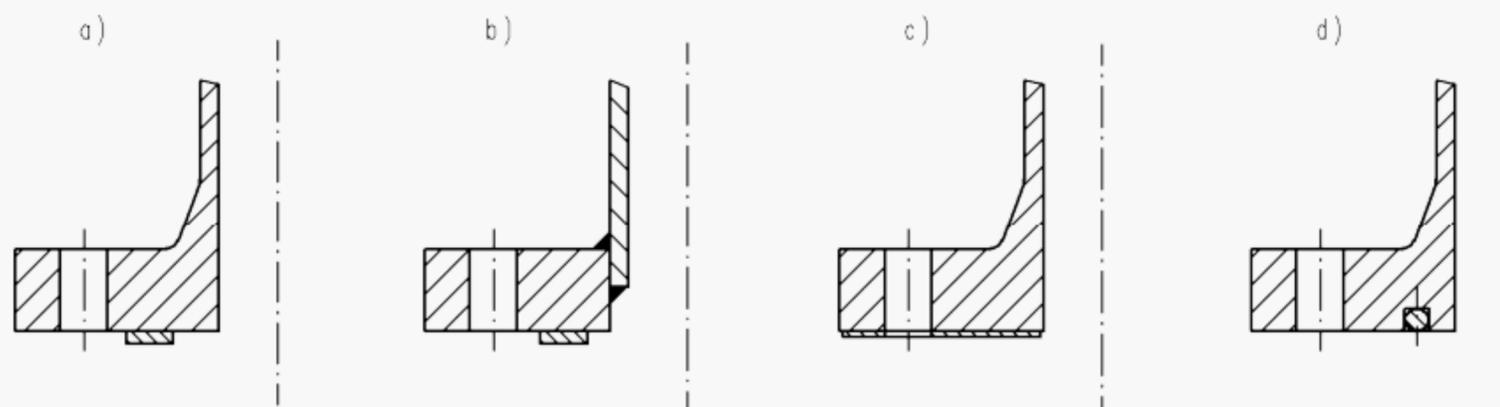
when the opening has a nozzle, d is given by:

in case of set on nozzles:

6.3.7 Flanges and boltings

6.3.7.1 General

This clause gives rules for the design of circular bolted flange connections as shown in Figure 12, subject to internal pressure. For all other types of flanges reference should be made to EN 13445-3:2002.



- a) Narrow-face flange design, smooth bore
- b) Narrow-face flange design, stepped bore
- c) Full-face flange design (soft gasket)
- d) Full-face metal / metal contact and an O-ring seal

Figure 12 — Flange working forms

Both flanges of a mating pair shall be designed to the same standard or set of rules. This applies too when one of the pairs is bolted flat end or cover.

A flange is attached to and supported by a nozzle neck, pipe or vessel wall, which will be referred to as a shell. Any fillet radius between flange and hub or shell shall be not less than $0,25 g_0$ and not less than 5 mm.

Hub flanges shall not be made by machining the hub directly from plate material without special consideration.

6.3.7.2 Definitions

Bolting-up condition: This condition applies when the gasket or joint contact surface is seated during assembly of the joint at ambient temperature and with the only loading coming from the bolts.

Operating condition: The condition when the end force due to the design pressure acts on the flange.

Narrow-faced flanges: These are flanges where all the face contact area lies inside the circle enclosed by the bolts.

Full-faced flanges: These are flanges where the face contact area, either direct or via a gasket or spacer, extends outside the circle enclosing the bolts.

6.3.7.3 Symbols and units

A	outside diameter of the flange or, where slotted holes extend to outside of flange, the diameter to bottom of slots
A_B	total cross-sectional area of bolts at the section of least diameter under load
$A_{B,min}$	total required cross-sectional area of bolts
B	inside diameter of flange

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b_0	basic gasket or joint seating width = $w/2$ with the exception of the ring-joint for which $b_0 = w/8$
b	effective gasket or joint seating width, $b = b_0$ when $b_0 < 6,3$ mm, $b = 2,52 \sqrt{b_0}$ when $b_0 > 6,3$ mm
C	bolt circle diameter
C_F	bolt pitch correction factor = $\sqrt{\frac{\text{bolt - spacing}}{2 \cdot d_b + \frac{6 \cdot e}{m + 0,5}}}$
	where "bolt spacing" is the distance between bolt centre lines (if calculated value < 1 , $C_F = 1$)
D	inside diameter of shell
d_b	bolt outside diameter
G	assumed diameter of gasket load reaction. When $b_0 < 6,3$ mm, $G =$ mean diameter of gasket contact face; when $b_0 > 6,3$ mm, $G =$ outside diameter of gasket contact face less $2 \cdot b$
g_0	thickness of hub at small end
g_1	thickness of hub at back of flange
H	total hydrostatic end force, $0,785 G^2 \cdot p$
H_D	hydrostatic end force applied via shell to flange, $= 0,785 B^2 \cdot p$
H_T	hydrostatic end force due to pressure on flange face $= H - H_D$
H_G	compression load on gasket to ensure tight joint, $= 6,28 b \cdot G \cdot m \cdot p$
h	hub length
l_0	$= \sqrt{B \cdot g_0}$
h_D	radial distance from bolt circle to circle on which H_D acts $= (C - B - g_1)/2$ except for stepped bore flanges for which $h_D = (C - B)/2$
h_G	radial distance from gasket load reaction to bolt circle, $(C - G)/2$
h_T	radial distance from bolt circle to circle on which H_T acts, $= (2C - B - G)/4$
k	stress factor
K	$= A/B$
M	$= M_{amb} \cdot C_F/B$ (bolting-up condition) or $= M_{op} \cdot C_F/B$ (operating condition)
M_{amb}	total moment acting upon flange for bolting-up condition
M_{op}	total moment action upon flange for operating condition
m	gasket factor (Table 6.3–1)
w	contact width of gasket, as limited by gasket width and flange facing
p_c	calculation pressure
$f_{B, amb}$	bolt nominal design stress at atmospheric temperature (see 6.3.7.5)
f_B	bolt nominal design stress at design temperature (see 6.3.7.5)
f_{amb}	design stress of flange material at ambient temperature
f	design stress of flange material at design temperature
$f_{H, amb}$	lower of design stresses of hub and shell materials at ambient temperature

f_H	lower of design stresses of hub and shell materials at design temperature
σ_θ	calculated tangential stress in flange
e	minimum flange thickness, measured at the thinnest section
β_y	$= \frac{1}{k-1} \left(0,66845 + 5,71690 \frac{K^2 \log K}{K^2 - 1} \right)$
W_{op}	minimum required bolt load for operating condition
W_{amb}	minimum required bolt load for operating bolting-up condition
W	flange design bolt load = $0,5 (A_{B, \min} + A_B) f_{B, amb}$
y	gasket or joint seating pressure (Table 6.3–1)

6.3.7.4 Use of standard flanges

PN designated flanges according to EN 1092-1 or Class designated flanges according to EN 1759-1 may be used as pressure vessel components without any calculation, provided the following conditions are fulfilled:

- under normal operating conditions, the calculation pressure does not exceed the rating pressure for the calculation temperature, according to the values given in EN 1092-1 for PN designated flanges or in EN 1759-1 for Class designated flanges, for the flange and the material under consideration;
- under testing conditions or exceptional operating conditions, the calculation pressure does not exceed 1,5 times the rating pressure given in the same tables, at appropriate temperature;
- gasket is one of the non metallic flat types with or without jacket;
- bolts are at least of the low strength category ($R_{p, bolt} / R_{p, flange} \geq 1$).

6.3.7.5 Requirements for bolting

There shall be at least four bolts.

If steel bolts or studs smaller than 12 mm are to be used, the bolting material shall have a design stress at 50 °C of more than 150 N/mm².

Recommended bolt stresses for determining the minimum bolt area are:

- for carbon and other non-austenitic steel, the lesser of $R_{p0,2} / 3$ measured at design temperature and $R_m / 4$ measured at room temperature;
- for austenitic stainless steel, $R_m / 4$ measured at design temperature;
- these allowable stresses may be multiplied by 1,5 at test conditions.

The bearing surface for the nuts shall be parallel to the flange face to within 1°. Any back facing or spot facing to accomplish this shall not reduce the flange thickness or hub thickness below design values. The radius between the back of the flange and the hub or shell shall be maintained.

The surface finish of the gasket contact face should be in accordance with the gasket manufacturer's recommendations if any or be based on experience.

6.3.7.6 Narrow-faced gasketed flanges

6.3.7.6.1 General

For calculation the "loose method" is used in which the flange is assumed to get no support in bending from the shell and correspondingly imposes no bending stresses on it. The loose method can only be applied, if all of the following requirements are met:

- a) $g_0 \leq 16 \text{ mm}$
- b) $p \leq 2 \text{ N/mm}^2$
- c) $B / g_0 \leq 300$.

If this is not the case the "integral method" shall apply (see EN 13445–3:2002).

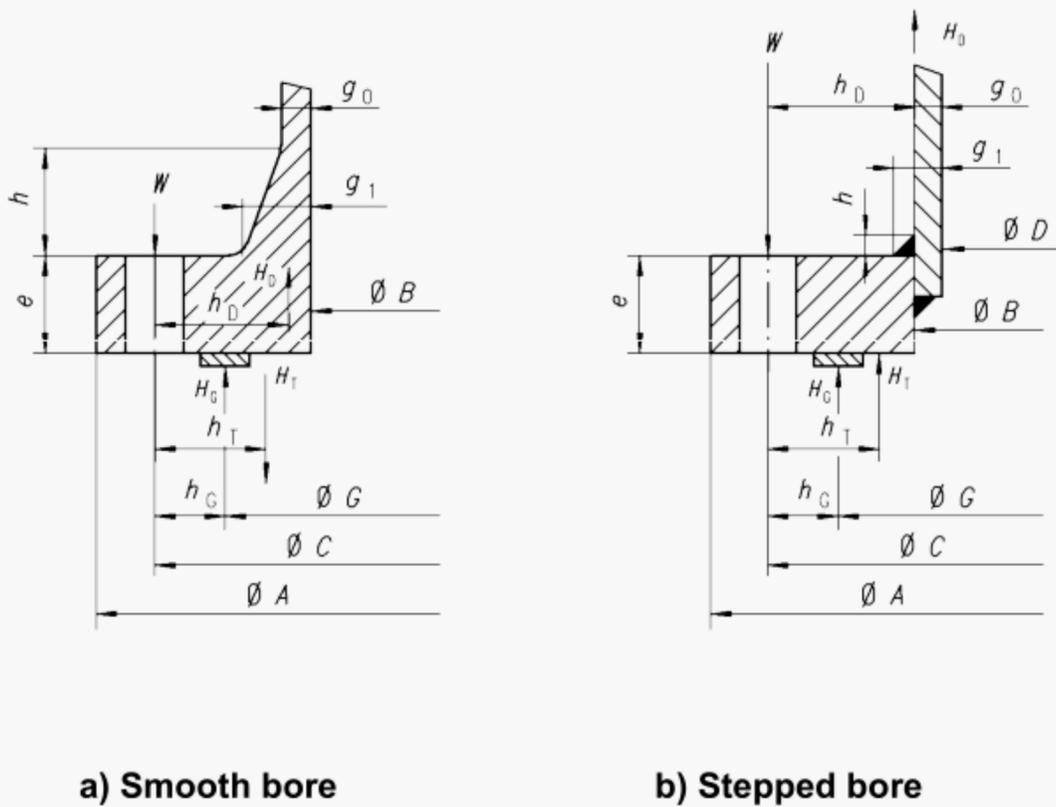


Figure 13 — Narrow-faced gasketed flanges

6.3.7.6.2 Bolt loads and areas

Bolt loads and areas shall be calculated for both the bolting-up and operating conditions.

- a) Bolting-up condition. The minimum bolt load is given by

$$W_{amb} = 3,14 b \cdot G \cdot y \tag{53}$$

- b) Operating condition. The minimum bolt load is given by

$$W_{op} = H + H_G \tag{54}$$

The required minimum bolt area $A_{B,min}$ is given by

$$A_{B,min} = \text{MAX} \left(\frac{W_{op}}{f_B}, \frac{W_{amb}}{f_{B,amb}} \right) \tag{55}$$

6.3.7.6.3 Flange moments

Flange moments shall be calculated for both the bolting-up and operating conditions.

a) Bolting-up condition. The total flange moment shall be:

$$M_{\text{amb}} = W \cdot h_G \quad (56)$$

b) Operating condition. The total flange moment shall be:

$$M_{\text{op}} = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad (57)$$

6.3.7.6.4 Flange stresses and stress limits

Flange stresses shall be determined for both bolting-up and operating conditions from the moment, M , as follows, where:

$$M = M_{\text{amb}} \frac{C_F}{B} \quad (58)$$

$$M = M_{\text{op}} \frac{C_F}{B} \quad (59)$$

for the bolting up and operating conditions respectively.

Tangential flange stress

$$\sigma_{\theta} = \frac{\beta_y \cdot M}{e^2} \quad (60)$$

Design stresses are the nominal design stresses f as defined in 6.3.2.4, except that the $R_m / 3$ alternative for austenitic stainless steel is not allowed.

The flange stresses as calculated above shall not exceed the following values:

operating condition: $k \sigma_{\theta} \leq f$;

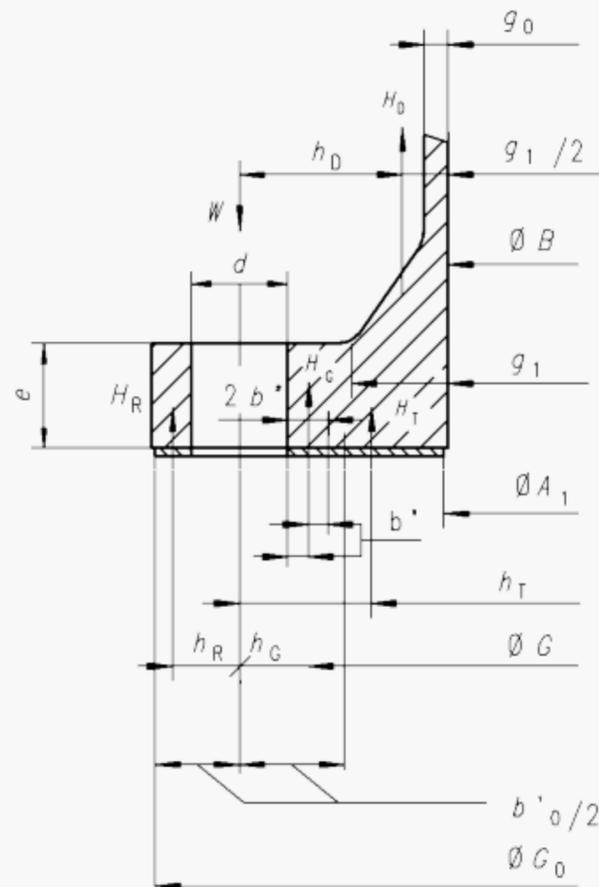
bolting up condition: $k \cdot \sigma_{\theta} \leq f_{\text{amb}}$;

If $B \leq 1\,000$ mm, $k = 1,0$.

6.3.7.7 Full-faced flanges with soft ring type gasket

6.3.7.7.1 General

This design method is applicable for full-faced flanges with non-metallic gasket not less than 1,5 mm thick and extending beyond the circle, enclosing the bolt holes.



Key:

See 6.3.7.7.2

Figure 14 — Full-faced flanges with soft ring type gasket

6.3.7.7.2 Additional symbols

- A_1 inside diameter of gasket contact face;
- b_0' basic gasket seating width effective under initial tightening up = lesser of $(G_0 - C)$ and $(C - A_1)$;
- b' effective gasket seating width = $4 \cdot \sqrt{b_0'}$ (This expression is valid only with dimensions expressed in millimetres);
- $2b''$ effective gasket pressure width, taken as 5 mm;
- G diameter at location of gasket load reaction, = $C - (d_h + 2b'')$;
- G_0 outside diameter of gasket or outside diameter of flange, whichever is less;
- H total hydrostatic end force, = $0,785 (C - d_h)^2 \cdot p$;
- H_R balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle;
- h_T radial distance from bolt circle to circle on which H_G acts, = $(d_h + 2b'')/2$;
- h_R radial distance from bolt circle to circle on which H_R acts, = $(G_0 - C + d_h)/4$;
- h_S radial distance from bolt circle to circle on which H_T acts, = $(C + d_h + 2b'' - B)/4$;

M_R balancing radial moment in flange along line of bolt holes;

n number of bolts.

6.3.7.7.3 Bolt loads and areas

Bolt loads shall be calculated, taking:

$$W_{op} = H + H_G + H_R \quad (61)$$

where

$$H_R = \frac{M_R}{h_R} \quad (62)$$

$$M_R = H_D \cdot h_D + H_T \cdot h_T + H_G \cdot h_G \quad (63)$$

$$W_{amb} = 3,14 C \cdot b' \cdot y \quad (64)$$

6.3.7.7.4 Flange design

The flange thickness shall be not less than the value of e from the following equation:

$$e = \sqrt{\frac{6 \cdot M_R}{f \cdot (3,14 \cdot C - n \cdot d_h)}} \quad (65)$$

The bolt spacing shall not exceed:

$$2 \cdot d_b + \frac{6 \cdot e}{m + 0,5} \quad (66)$$

6.3.7.8 Full face flanges with metal to metal contact

6.3.7.8.1 General

The rules shall be applied when there is metal to metal contact both inside and outside the bolt circle before the bolts are tightened with more than a small amount of preload and the seal is provided by an O-ring or equivalent. Manufacturing procedures and tolerances shall ensure that the flange is not dished so as to give initial contact outside bolt circle.

It is assumed that a self-sealing gasket is used approximately in line with the wall of the attached pipe or vessel and that the gasket seating load and any axial load from the seal may be neglected.

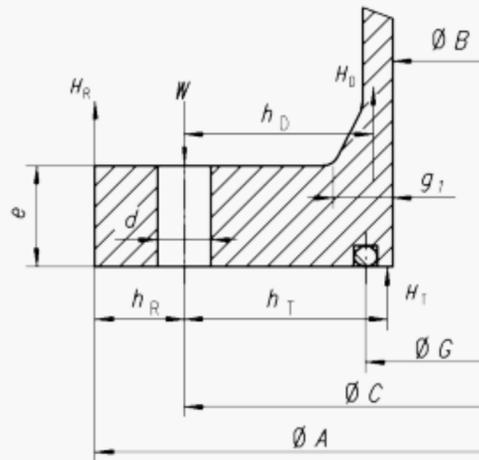


Figure 15 — Full face flange with metal to metal contact

6.3.7.8.2 Additional symbols

- G mean diameter of gasket
- H_R balancing reaction force outside bolt circle in opposition to moments due to loads inside bolt circle;
- h_R radial distance from bolt circle to circle on which H_R acts, $h_R = (A - C) / 2$;
- M_R balancing radial moment in flange along line of bolt holes;
- n number of bolts.

6.3.7.8.3 Design

The following rules apply where the flange is to be bolted to an identical flange or to a flat cover.

Bolt loads shall be calculated in accordance with 6.3.7.6.2 taking:

$$W_{op} = H + H_R \tag{67}$$

where

$$H_R = M_R / h_R \tag{68}$$

$$M_R = H_D h_D + H_T h_T \tag{69}$$

$$W_{amb} = 0$$

The flange thickness shall be not less than:

$$e = \sqrt{\frac{6 \cdot M_R}{f \cdot (3,14 \cdot C - n \cdot d_h)}} \tag{70}$$

7 Manufacturing and welding

7.1 Introduction

In case of matters not covered by this clause reference should be made to EN 13445-3:2002, EN 13445-4:2002, EN 13445-5:2002, where applicable.

7.2 General

Expansion vessels shall be manufactured according to approved drawings and specifications observing the requirements of this European Standard and sound engineering practice.

Workshops and equipment shall be adequate for the construction of the vessels and shall also allow their testing and inspection. The manufacturer has the responsibility for the competence, training and supervision of his staff.

In order to realise the specified quality of the finished expansion vessel or part of it the manufacturer shall make sure that the manufacturing procedure chosen is adequate for the purpose envisaged.

It is the responsibility of the manufacturer to operate a recognized formal system for the control of manufacturing operations which includes special process such as forming, welding, clenching etc., and which assures the appropriate traceability of materials.

7.3 Manufacturing tolerances

7.3.1 General

The geometry of welded butt and fillet joints shall comply with EN 13445-3:2002 except as modified below for shell longitudinal and circumferential seams. If the drawings of expansion vessels require more stringent manufacturing tolerances they supersede the requirements given in this clause.

7.3.2 Middle line and surface alignment

For longitudinal joints in cylinders and spherical components (whether of equal or different thicknesses) the middle lines of adjacent plates shall have a maximum misalignment d_1 (e_1 being the thinner plate thickness) according to Table 4.

Table 4 — Middle line and surface alignment

e_1	d_1
$e_1 \leq 2$	0,5
$2 < e_1 \leq 4$	$e_1 / 4$
$4 < e_1 \leq 10$	1
$10 < e_1 \leq 30$	$e_1 / 10$

For circumferential joints the middle lines of adjacent plates (whether of equal or different thicknesses) shall have a maximum misalignment $d_1 = e_1 / 10 + 1$ mm. e_1 being the thinner plate thickness.

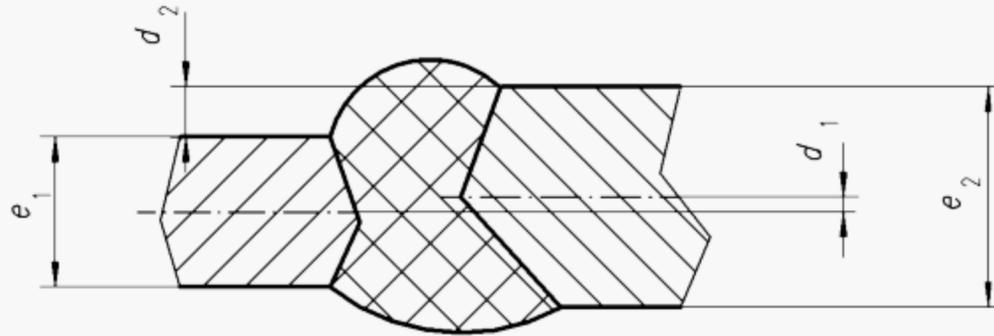


Figure 16 — Middle line and surface misalignment

For longitudinal joints in cylinders the surfaces of adjacent plates shall have a maximum misalignment of $d_2 = e_1 / 4$ but not more than 3 mm. e_1 being the thinner plate thickness.

For longitudinal joints in spherical components as well as for circumferential joints the surfaces of adjacent plates shall have a maximum misalignment of $d_2 = e_1 / 4$, e_1 being the thinner plate thickness.

If the misalignment tolerances are exceeded surfaces shall be tapered with a slope on 1 in 4 over a width that includes the width of the weld, the lower surface being built up with added weld metal if necessary to provide the required taper. Trimming of plate surfaces is not permitted where this reduces the thickness below the required minimum.

7.3.3 Tolerances for vessels

7.3.3.1 For cylindrical and spherical vessels the mean external diameter derived from the circumference shall not deviate by more than 1,5 % from the specified external diameter.

7.3.3.2 Out-of-roundness defined by the following equation:

$$O = \frac{D_{\max} - D_{\min}}{\left(\frac{D_{\max} + D_{\min}}{2}\right)} \cdot 100 \text{ [%]} \tag{71}$$

shall not exceed:

- 1,5 % for $e / D < 0,01$;
- 1,0 % for $e / D \geq 0,01$.

7.3.3.3 The deviation from the straight line shall not be more than 0,5 % of the total cylindrical length of the vessel.

7.3.3.4 Regularities in profiles occurring at longitudinal welding seams and associated with "flats" adjacent to the weld or "peakings" shall not exceed 5 mm.

7.3.3.5 Dished ends shall be aligned with the tolerances as laid down below except that the crown radius shall not be greater than specified in the design and the knuckle radius shall not be less than specified in the design.

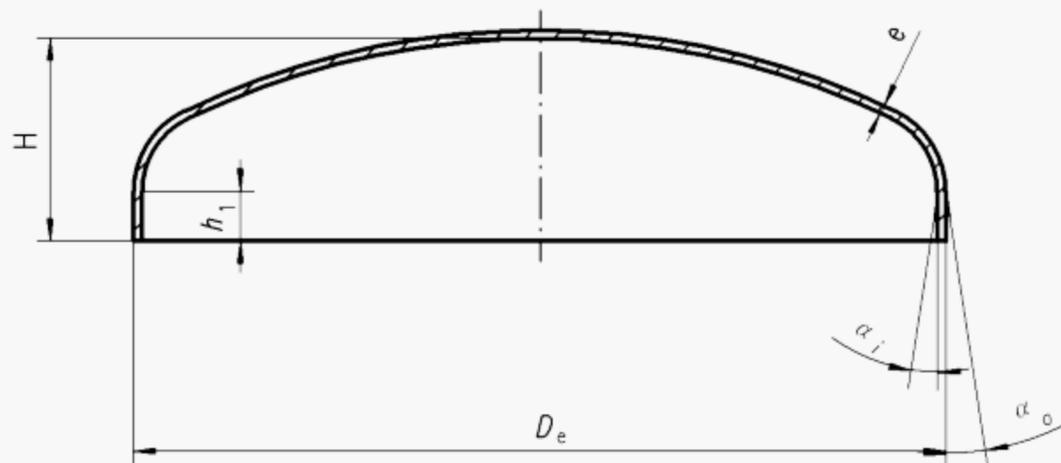


Figure 17 — Tolerances for dished ends

Table 5 — Tolerances for dished ends

Dimension	Range of application	Limit deviations
Circumference C ($C = \pi \times D$)	$D_e \leq 300$ mm	± 4 mm
	Ferritic materials: 300 mm < $D_e \leq 1\ 000$ mm: 1 000 mm < D_e :	$\pm 0,4$ % $\pm 0,3$ %
	Austenitic materials: 300 mm < D_e	+ 0,5 %, - 0,7 %
Out-of-roundness O $\left(O = \frac{2 \cdot (D_{\max} - D_{\min})}{D_{\max} + D_{\min}} \times 100 \right)$	all	≤ 1 %
Inner height H	+ 0,015 D_e , but not less than + 10 mm	
Wall thickness e (nominal)	$e \leq 10$ mm: 10 mm < e :	- 0,3 mm - 0,5 mm
Deviation from cylindrical shape α	all	$\alpha_i \leq 2^\circ$ $\alpha_o \leq 5^\circ$
Straight flange h_1	all	$3 \times e$

7.4 Weld details

7.4.1 Recommended weld details

Recommended weld details are given in EN 1708-1.

The manufacturer in selecting an appropriate weld detail should give consideration to:

- method of manufacture;
- service conditions;
- ability to carry out the necessary NDE.

Other weld details may be used provided their suitability is proven by procedure approval to EN ISO 15614-1:2004 and EN ISO 15613:2004.

7.4.2 Vessels made in more courses

Where any part of a vessel is made in two or more courses the longitudinal weld seams of adjacent courses shall be staggered by $4e$ with 10 mm minimum, measuring the distance between the edges of the longitudinal joints.

7.4.3 Joggle joints

Joggle joints may be used on expansion vessels under the following conditions:

- operating temperature is between $-10\text{ }^{\circ}\text{C}$ and $+70\text{ }^{\circ}\text{C}$ and
- for circumferential joints only, with wall thicknesses not exceeding 8 mm and
- maximum diameter not exceeding 2 000 mm.

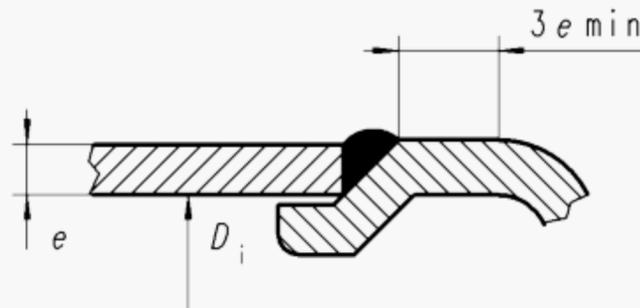


Figure 18— Details of joggle joint

7.5 Welding

7.5.1 General

Welding of joints for parts of a vessel shall only be undertaken if the following conditions are satisfied:

- a) welding procedure specification is compiled by the manufacturer;
- b) welding procedures selected by the manufacturer are qualified for the field of application;
- c) welders and welding operators are qualified for the work allocated to them and their approval is valid.

7.5.2 Welding procedure specification (WPS)

The manufacturer shall compile welding procedure specifications, in accordance with EN ISO 15609-1:2004 for each joint or family of joints.

7.5.3 Qualification of WPS

Welding procedure qualification tests in accordance with EN ISO 15614-1:2004 or EN ISO 15613:2004 shall be carried out to cover the welding procedure specifications and are to be recorded in a welding procedure approval record (WPAR).

7.5.4 Qualification of welders and welding operators

Welders and welding operators shall be approved to EN 287-1:2004 or EN 1418:1997 respectively. An up-to-date list of welders and welding operators together with their records of approval test shall be maintained by the manufacturer.

7.5.5 Preparation of edges to be welded

The edges to be welded shall be kept in position, either by mechanical means, and/or by tack welds. The tack welds shall be removed or fused again in the weld bead in such a way that no metallurgical or homogeneity defects are generated.

When welding without a sealing run, the manufacturer shall ensure the required penetration at the weld root.

The surface to be welded shall be thoroughly cleaned of foreign substance.

7.5.6 Execution of welded joints

After each weld run, the slag shall be removed and the weld cleaned and the surface defects removed.

Unless the welding process used provides effective and sound penetration, the second side of a welded joint shall be removed back to sound metal.

7.5.7 Attachments, supports and stiffeners

Attachments, whether temporary or not, supports and stiffeners shall be welded to a part subject to pressure by qualified welders using a qualified procedure.

Temporary attachments shall be removed using a technique which does not affect the properties of the metal of the pressure part to which they are welded.

7.6 NDE personnel

NDE personnel shall be qualified and certified according to EN 473:2000.

NOTE For pressure equipment in categories III to IV the personnel need to be approved by a recognized third-party.

7.7 Manufacture and testing of permanent joints

7.7.1 Welded joints

7.7.1.1 General

In order to control the compliance of the welded joints with the specification production test plates shall be welded and tested in accordance with the requirements detailed below, for a joint or family of joints of the main shell longitudinal and circumferential welds.

7.7.1.2 Reference criteria

The number of the production test plates and the extent of their testing depend on the length of the welded joints, the material thickness and the weld joint coefficient z for each qualified welding procedure as follows:

7.7.1.2.1 For vessels with $z = 0,7$ no test plates are required.

7.7.1.2.2 For vessels with $z > 0,7$ the following shall apply:

- a) one test plate per vessel in the case of $z = 1,0$;
- b) one test plate per 500 m of longitudinal joints in the case of $z = 0,85$;
- c) where the circumferential seams are welded to a procedure different to the longitudinal seams, one test plate per year.

After 10 test plates having successfully passed the tests testing may be reduced to the following:

- d) one test plate per 100 m of longitudinal joints in the case of $z = 1,0$;
- e) one test plate per 2 500 m of longitudinal joints in the case of $z = 0,85$.

7.7.1.2.3 For vessels whose design is verified by experimental design methods, regular tests performed on vessels taken from the production cover the performance of tests of welded joints.

7.7.1.2.4 When a vessel includes one or more longitudinal seams the test plates should wherever practicable be attached to the shell plate on one end of one seam so that the edges to be welded in the test plate are a continuation and duplication of the corresponding edges of the longitudinal seams. The weld metal should be deposited in the test plates continuously with the welding of the corresponding longitudinal seam so that the welding process, procedure and technique are the same. When it is necessary to weld the test plates separately, the procedure used should duplicate that used in the construction of the vessel.

When the test plates are required for circumferential welds they may be welded separately from the vessel providing the technique used in their preparation duplicates as far as possible the procedure used in the welding of the appropriate seams of the vessel.

7.7.1.3 Testing on test plates

The type and number of specimens to be taken from the test plate shall be to the requirements given below.

Table 6 — Test specimens

Designation	Symbol
Face bend test to EN 910:1996	FB
Root bend test to EN 910:1996	RB
Transverse tensile test to EN 895:1995	TT
Macro examination	Ma

Table 7 — Testing of production test plates

Group	Test specimens
Carbon steels (1.;1.1)	1 FB, 1 RB, 1 Ma
Austenitic steels (8.1)	1 FB, 1 RB, 1 TT, 1 Ma

7.7.1.4 Performances of tests

7.7.1.4.1 General

The individual test pieces shall be manufactured, tested and meet the acceptance criteria defined below:

7.7.1.4.2 Transverse tensile test (TT)

Unless otherwise specified the tensile strength shall meet the minimum value for the parent metal.

Where the weld metal and/or welded joint properties are specified below those of the base metal by design, special consideration is needed.

7.7.1.4.3 Bend test (FB / RB)

The testing and the test requirements shall comply with 7.4.2 of EN ISO 15614-1:2004.

7.7.1.4.4 Macro examination (Ma)

The macro etch shall show sound built-up of beads and sound penetration.

7.7.1.4.5 Test record

A test record shall be prepared indicating compliance of the results with the specified requirements as per example in EN ISO 15613:2004.

7.7.1.4.6 Retests

Production factors may result in a scatter of mechanical test results which may fall below the agreed specification level. Where individual tests do not comply with the above requirements the reasons for this failure shall be investigated and if no unacceptable imperfections are found the following retests should be made:

- a) tensile test: two retests;
- b) bend test: two retests.

Should any of the retests fail to comply with the requirements then the joints represented by the test plate shall be deemed not in compliance with this European Standard.

7.7.1.5 NDE of welded joints during manufacture**7.7.1.5.1 General**

As an alternative to the welding and testing of production test plates NDE in the form of radiographic or ultrasonic examination may be used to check the required quality of the vessel welds.

For the following vessels no NDE is required:

- a) vessels with a weld joint coefficient $z = 0,7$;
- b) vessels whose design is verified by experimental design methods. See also 7.7.1.2.3.

7.7.1.5.2 Radiography

The examination shall apply according to class B of EN 1435:1997 and with the following modalities for each qualified welding procedure:

- a) one film of 200 mm per vessel including one T-junction in the case of $z = 1,0$;
- b) one film of 200 mm including one T-junction per 500 m of longitudinal joints in the case of $z = 0,85$;
- c) where the circumferential seams are welded to a procedure different to the longitudinal seams, one film of 200 mm including one T-junction per year.

After 10 films plates having fulfilled the acceptance criteria testing may be reduced to the following:

- d) one film per 100 m of longitudinal joints in the case of $z = 1,0$;
- e) one film per 2 500 m of longitudinal joints in the case of $z = 0,85$.

Acceptance criteria for radiography are to be taken from EN 13445-5.

7.7.2 Clenched joints

For vessels whose design is verified by the experimental design method, regular tests performed on vessels taken from the production cover the performance of tests of clenched joints.

At least tests according to 6.2.3 or 6.2.4 have to be carried out per change of construction parameters.

7.8 Forming of parts subject to pressure

7.8.1 Ratio of deformation

7.8.1.1 Dished circular products

The following equation shall be used for the calculation of deformation (F) for all finished circumferential products (e.g. dished ends, spherical caps).

$$F = 100 \ln \cdot \frac{D_b}{D_e - 2e} \text{ [%]} \tag{72}$$

where

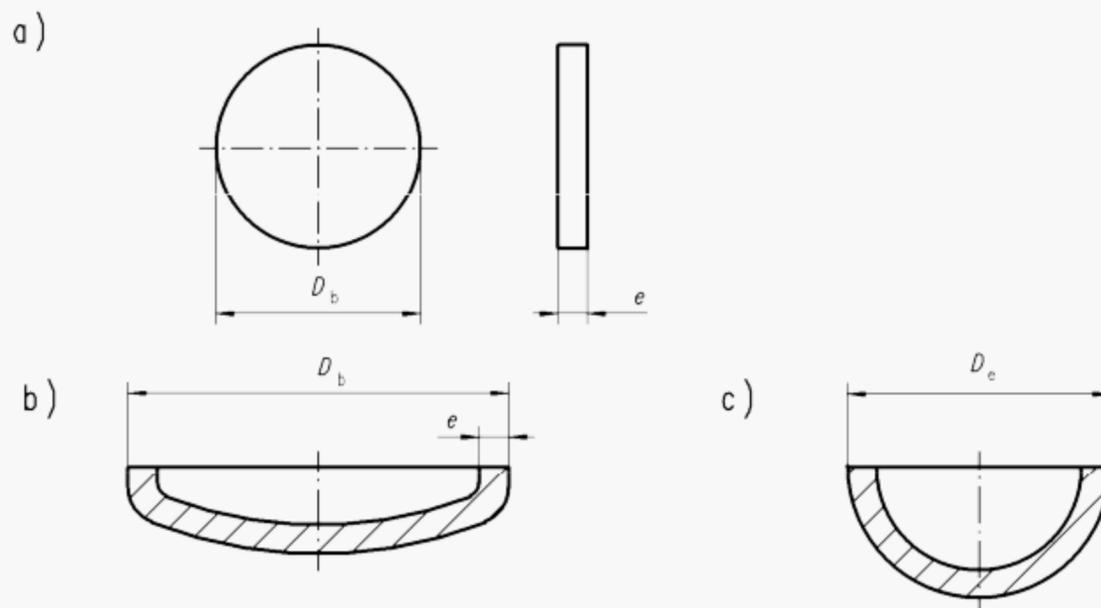
e is the thickness of the initial product;

D_b is the diameter of the blank or diameter of intermediate product.

In case of different forming steps without intermediate heat treatment, the initial diameter is to be applied. In case of intermediate heat treatment, the diameter after the last heat treatment is relevant;

D_e is the external diameter of the finished product;

ln is the naperian logarithm.



a) Initial product b) Intermediate product c) Final product

Figure 19 — Ratio of deformation of dished circular products

7.8.1.2 Cylinders made by rolling

The following equation shall be used for the calculation of deformation (F) for rolled products:

$$F = \frac{50 \cdot e}{R_m} \cdot \left(1 - \frac{R_m}{R_{mo}} \right) \quad [\%] \quad (73)$$

where

e is the thickness of the initial product;

R_{mo} is the mean radius of the initial product.

In case of different forming steps without intermediate heat treatment, the ratio of deformation is the total amount of the ratio of deformation of the individual forming steps. In case of intermediate heat treatment, the deformation is that deformation achieved after the last previous heat treatment;

R_m is the mean radius of the finished product.

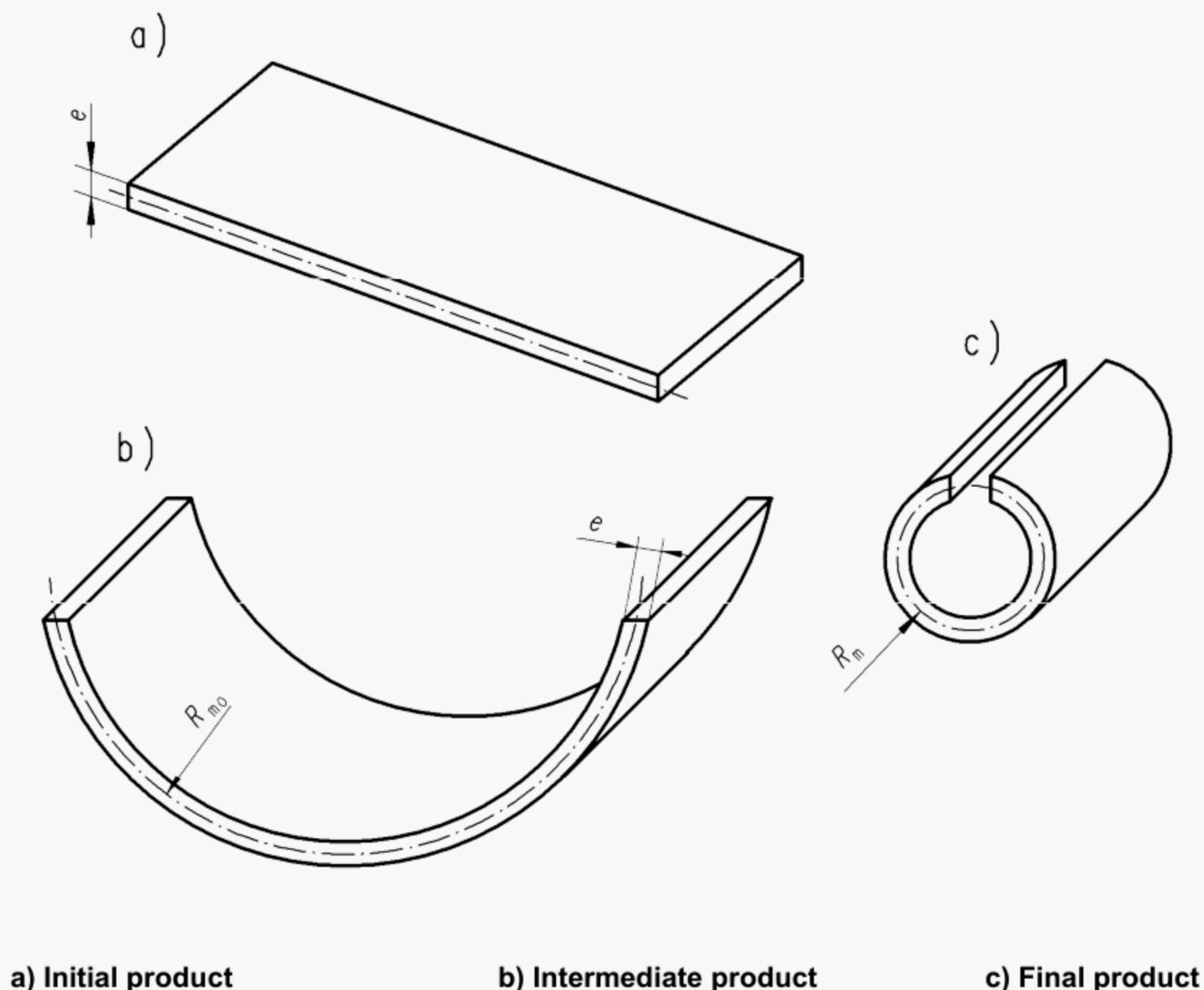


Figure 20 — Ratio of deformation of rolled cylinders

7.8.1.3 Tube bends

The following equation shall be used for the calculation of deformation (F) of tube bends:

$$F = 100 \cdot \frac{D_e}{2R} \quad [\%] \tag{74}$$

where

R is the radius of curvature for the tube;

D_e is the external diameter of the tube.

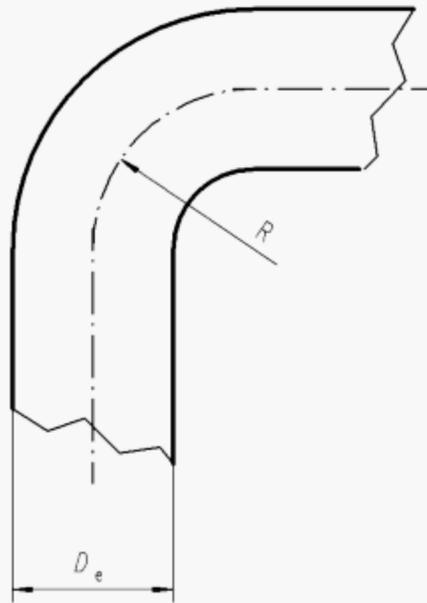


Figure 21 — Ratio of deformation for tubes

7.8.1.4 Other part types

For all other types of parts reference should be made to EN 13445-4:2002.

7.8.2 Forming conditions

7.8.2.1 Cold forming

Cold forming of ferritic steels as given in Clause 5 is defined as forming at temperatures below the maximum permissible temperature for stress relieving minus (20 to 30) °C in accordance with the applicable material standards.

Cold forming of austenitic steels is defined as forming at temperature below 300 °C.

7.8.2.2 Hot forming

Hot forming of carbon steels shall be carried out at temperatures above the maximum permissible temperature for stress relieving, usually in the temperature range of normalising, in accordance with the material specifications.

For austenitic steels of group 8.1 hot forming shall be carried out at a temperature of 300 °C or above.

7.8.3 Heat treatment

7.8.3.1 Heat treatment after cold forming of flat parts

If $F > 5 \%$, post forming heat treatment is required with the following exceptions:

- torispherical dished ends {(flange radius $\geq 0,1 D_e$ and crown radius $\geq D_e$) with $e \leq 8$ mm and $h \leq 40$ mm};
- austenitic steels of group 8.1;
- vessel and vessel parts whose design is proven by experimental design methods.

7.8.3.2 Heat treatment after cold forming for tubular products

Post forming heat treatment is required:

- if curvation radius $R \leq 1,3 D_e$ or
- if $1,3 \cdot D_e < R < 2,5 D_e$ and if external tube diameter $D_e > 142$ mm.

7.8.3.3 Heat treatment after hot forming

Heat treatment after hot forming is not required for materials falling into groups 1 1.1 and 8.1.

7.8.4 Visual examination and control of dimensions

Formed pressure parts shall be subject to visual examination and dimensional check in the delivery condition. The results of the visual examination and the check of dimensions shall be certified in the certificate.

7.8.5 Test certificate

Formed parts for pressure vessels shall be accompanied by an inspection document according to vessel category.

Where applicable the following information shall be recorded:

- a) type of heat treatment;
- b) correct state of heat treatment;
- c) correct transfer of marking, if required.

7.9 Repairs

7.9.1 Surface defects

Vessel surfaces have to undergo a visual check after production.

The surfaces shall be smooth. Irregularities are accepted if they do not influence the appropriate use of the vessels.

7.9.2 Repairs, elimination of defects

All unacceptable imperfections are to be removed by mechanical or thermal means, whereby the minimum thickness of the surface has to be maintained.

Repair weldings may only be carried out by qualified welders within the framework of approved WPS.

7.10 Finishing operations

Finishing operations are all operations carried out after the vessel has been pressure tested and before shipment/transport.

Any repairs, thermal or mechanical operations shall be followed by the pressure test (see 9.5).

8 Diaphragm

8.1 General

As an integral part of the expansion vessel the diaphragm separates the vessel into a water space and a gas space. A distinction is drawn between (closed) bag-type diaphragms and (open) cup-type diaphragms. The latter are stressed primarily by deflexion, the former by extension.

NOTE The diaphragm separates water and gas both of which are under the same pressure. Therefore the diaphragm is not a pressure bearing part.

8.2 Materials

Materials used for diaphragm shall be suitable for the intended purpose and comply with the tests specified in this clause. These tests are valid for all applications where the water in the vessel normally does not exceed 70 °C and is not lower than 5 °C.

For applications where the vessel water temperature is normally outside the range of 5 °C to 70 °C the test methods in this clause shall be adapted in such a way that the test temperature corresponds to the temperature, requirements to which the diaphragm will be subject during operation.

Diaphragms for water with antifreeze additives shall be tested with the corresponding water solution. This is not necessary if a declaration of the diaphragm manufacturer confirms the suitability of the material.

8.3 Hygienic demands

Diaphragms for vessels intended for drinking water systems are not to affect the quality of the water.

NOTE For the time being materials in contact with drinking water have to comply with the respective national regulations.

8.4 Test concept

The test program in this standard falls into 2 categories:

- tests on the diaphragm material according to 8.5.1;
- tests on the diaphragm, built into the vessel, according to 8.5.2 and 8.5.3.

Out of a range of vessels similar in design and geometrical proportions two vessels sizes are selected whose diaphragm would be stressed the most when compared with other vessels in the range.

Standards to be used for the tests are contained in Annex A and can be selected by making reference to the test number.

8.5 Testing

8.5.1 General

For each vessel size/type selected for testing two whole vessels taken from production and two corresponding diaphragms are to be provided.

8.5.2 Tests on diaphragms

One unused diaphragm per vessel size/type selected is to be used for tests according to Table 8.5–1.

Table 8 — Tests on diaphragms

Test number	Property to be tested	Requirements for cup-type diaphragms	Requirements for bag-type diaphragms
8.5.2.1	Thickness	Not less than 1 mm at any point	Not less than 1 mm at any point
8.5.2.2	Visual surface inspection	No cracks, foreign inclusions or blisters	No cracks, foreign inclusions or blisters
8.5.2.3	Tensile strength at break	> 10 N/mm ²	> 10 N/mm ²
8.5.2.4	Elongation at break	> 450 %	> 450 %
8.5.2.5	Shore A hardness	50 to 65 Shore A	45 to 65 Shore A
8.5.2.6	Compression set in air	≤ 40 %	≤ 60 %
8.5.2.7	Specific mass (tolerance)	± 0,03 g/cm ³	± 0,03 g/cm ³

Minimum requirements given in 8.5.2.1 to 8.5.2.7 are valid for proven materials as listed in Annex B.

For new materials or derivatives of proven materials minimum requirements are to be defined by the diaphragm manufacturer on the basis of the results of appropriate tests.

The second unused diaphragm shall undergo an ageing test.

The diaphragm is immersed in water with a temperature of 70 °C ± 5 °C for 28 days. After 28 days the tests 8.5.2.8, 8.5.2.9 and 8.5.2.10 are carried out. The test results shall be within the limits given in Table 9 when compared with the results obtained on the first diaphragm.

Table 9 — Limits for results of ageing test

Test number	Property to be tested	Requirements
8.5.2.8	Tensile strength at break	≤ 25 %
8.5.2.9	Elongation at break	≤ 25 % (rel.)
8.5.2.10	Shore A hardness	≤ 5 shore A

Diaphragms for fresh water application do not need to undergo an ageing test if the vessel plate explicitly limits the max. operating temperature to 40 °C.

8.5.3 Cyclic stressing on vessels

8.5.3.1 General

One vessel per each selected size is subjected to continuous cyclic stressing. Pressurised water is pumped in the vessel until it is filled to 50 % of the content of the chamber where the membrane expands and then released again.

8.5.3.2 For heating expansion vessels the test is performed at a water temperature of 70 °C ± 5 °C. The number of cycles depends on the vessel volume and can be read from Figure 22.

Linear interpolation of values is permitted.

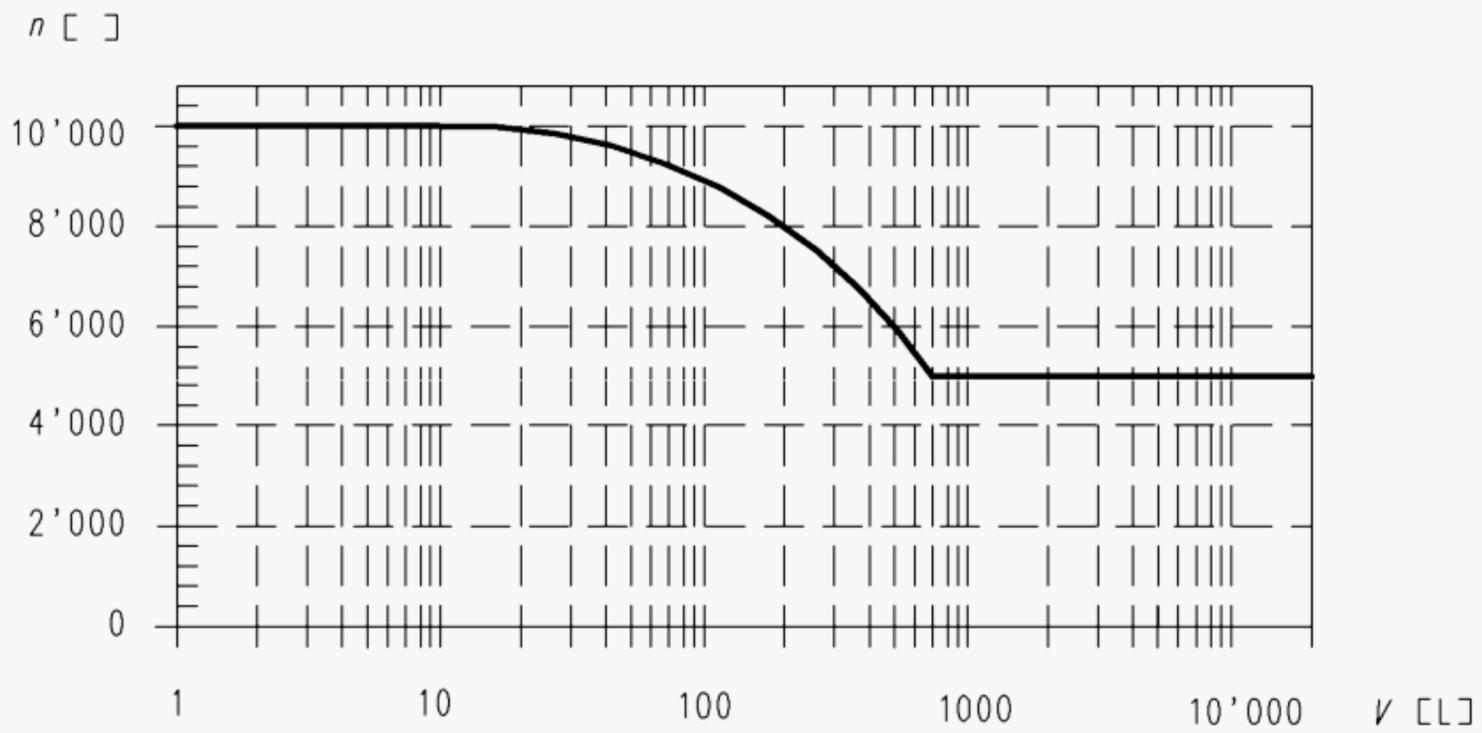


Figure 22 — Cyclic stressing for heating expansion vessels

8.5.3.3 For drinking water expansion vessels the test is performed at ambient water temperature. The number of cycles depends on the vessel volume and can be read from Figure 8.5–2. Linear interpolation of values is permitted.

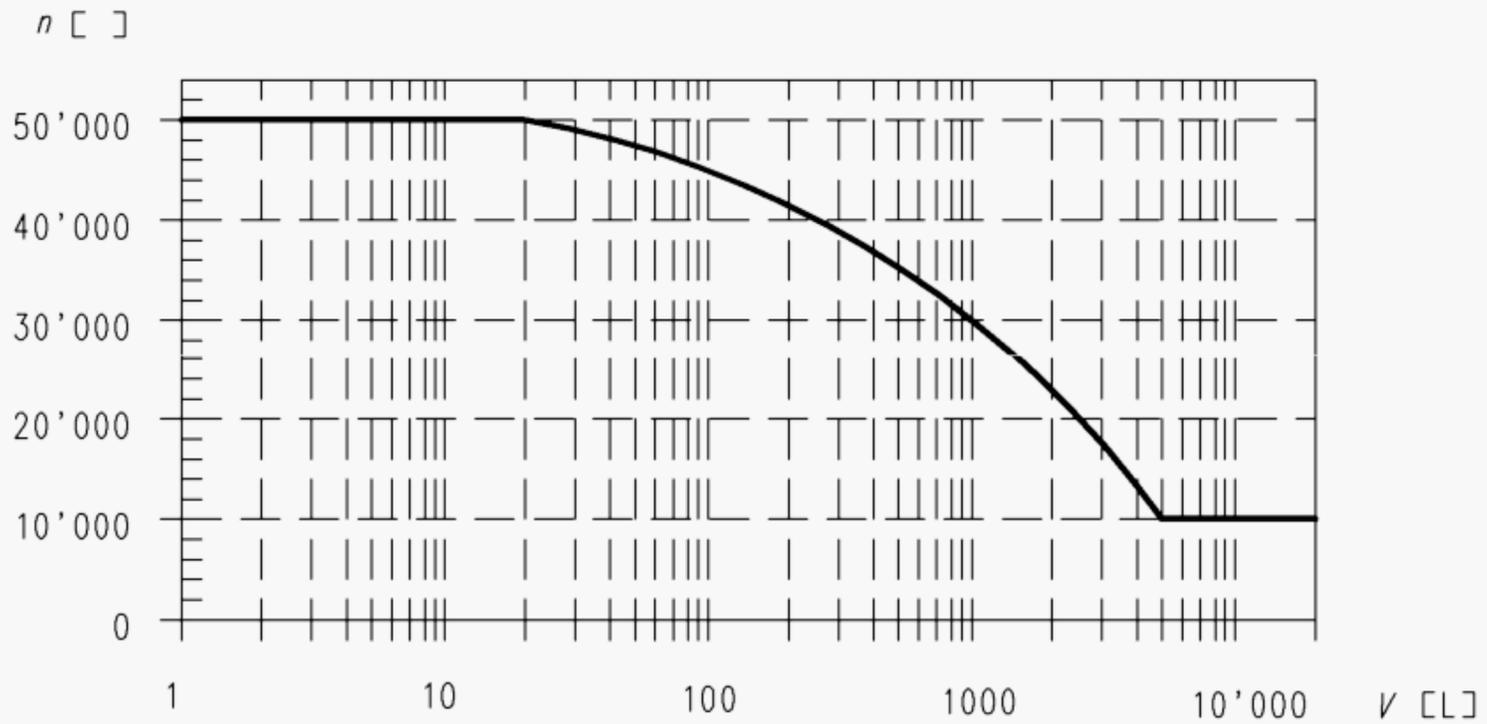
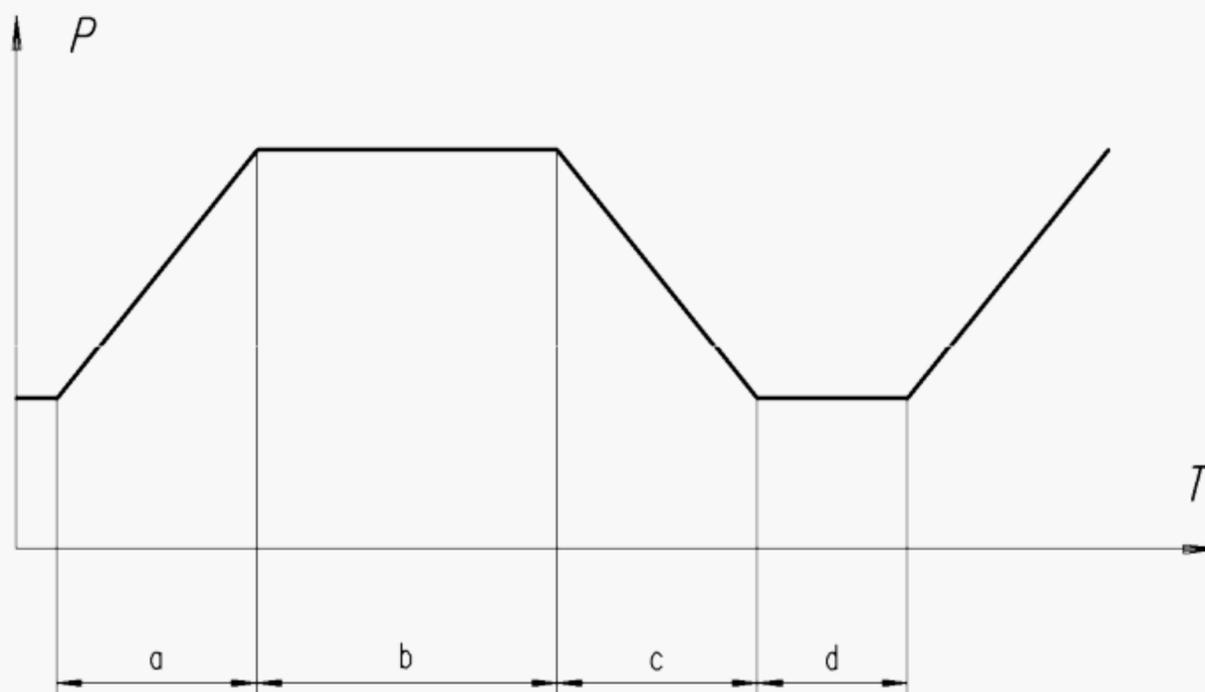


Figure 23 — Cyclic stressing for drinking water expansion vessels

8.5.3.4 In respect to the characteristics of the test cycles the minimum time values in Figure 24 have to be observed.



Key

a, c minimum 20 s
b, d minimum 10 s

Figure 24 — Time characteristics of test cycles

8.5.3.5 Acceptance criteria

After the cyclic test, once the vessel has cooled down, the gas side of the vessel is filled with air to 1,5 bar. The pressure drop within the following hour shall not exceed 0,15 bar.

Changes in ambient temperature shall be taken into account.

8.5.4 Diaphragm permeability test

This test is performed on a second vessel of each size selected. It is carried out by measuring the pressure drop in both of them after 28 days at a temperature of 20 °C. The initial pressure is the admissible operating pressure. The pressure drop shall not exceed 2 % for vessels of ≤ 100 l during this time. With vessels > 100 l it shall not exceed 1 % of the initial pressure. Air is to be used as the testing gas. Changes in ambient temperature shall be taken into account.

8.5.5 Repetition of tests

Two repeat tests have to be made for each failed test. Both shall be successful.

8.5.6 Test report

A report on the results of all the tests is to be drawn up.

8.6 Tests by the diaphragm manufacturer

Per batch (series) of diaphragms of the same size and material the manufacturer conducts the tests according to 8.5.2.1 to 8.5.2.5 and 8.5.2.7. The results shall be recorded in a certificate according to EN 10204 and kept at the disposal of the vessel manufacturer. Once a year the diaphragm manufacturer conducts tests according to 8.5.2.6 and 8.5.2.8 to 8.5.2.10 for each type of material used.

8.7 Marking of diaphragms

Each diaphragm has to be marked in such a way that the following information (possibly encoded) is contained:

- month/year of manufacturing;
- mark of manufacturer (if diaphragm manufacturer is not the vessel manufacturer);
- size/type or code number.

9 Testing and inspection

9.1 General

Type, amount and frequency of approvals, tests and inspections described in this clause as well as the responsibilities are to be determined in accordance with EN 13445-5:2002.

9.2 Technical documentation

The manufacturer establishes a technical documentation. It is to demonstrate compliance with this European Standard and consists of:

- design calculations or data of experimental design method;
- drawings;
- technical / manufacturing schedule, covering:
 - technical data / operation of vessel;
 - manufacture;
 - welding;
 - materials;
- amount and frequency of tests / inspections during manufacturing.

9.3 Inspections during manufacturing

Purpose of these inspections is to make sure the vessels manufactured meet the requirements of this European Standard and are based on the technical documentation. They cover:

- use of correct materials/ correct transfer of stamping where applicable;
- use of agreed on welding procedures/ observation of the corresponding specifications;
- use of qualified welders/ operators;
- visual examination of welds;
- correct thermal treatment where applicable;
- internal / external examination and dimensional checks;
- establishing and testing of test plates or NDE.

9.4 Pressure test

Pressure tests are normally carried out with water. Where air is used instead of water authorisation of national safety bodies may have to be obtained in order to cover the workers safety.

The pressure test is performed at ambient temperature. The vessel is pressurised to 1,43 PS and inspected for leaks. No leaks shall occur. If the vessel leaks and is then repaired the pressure test shall be repeated.

The results of the test shall be recorded by the manufacturer.

9.5 Marking

Each vessel shall be permanently and legibly marked with the following information. Direct stamping of the vessel shell is only acceptable where the vessel walls are above 6 mm.

- Nominal volume of the vessel V
- manufacturer's name / symbol and address;
- vessel identification;
- period / year of manufacture;
- maximum allowable pressure PS ;
- maximum / minimum allowable operating temperature TS ;
- initial precharge pressure;
- space for a modified precharge pressure, if applicable;
- test pressure PT .

9.6 Documentation

9.6.1 Depending on type of vessel and/or the method of production the documentation may cover one vessel only or a group/ series of vessels. It shall contain the following elements:

- vessel type and/or production batch;
- technical documentation;
- records of all inspections and tests;
- reference to relevant files containing material certificates, welding procedure approvals / specifications, welder / operator qualification, welding consumables;
- report on pressure test;
- certificate of compliance.

9.6.2 Besides the documentation covering the vessel(s) the following records are to be filed:

- manufacturer's quality system;
- material certificates;
- list of qualified welders / operators;
- welding procedure approvals;
- welding procedure specifications.

9.6.3 The vessel manufacturer is responsible for filing the records for a minimum period of 10 years.

9.6.4 With each vessel Instructions for installation and operation are to be provided.

Annex A (informative)

Standards for testing diaphragms

Testing of diaphragms for expansion vessels has been done for many years based on ISO standards and national German DIN standards as listed below; see Bibliography.

Standard number	Property to be tested on diaphragms in vessels according to EN 13831	Relevant test number
ISO 2781	Density	8.5.2.7
DIN 53504	Tensile strength at break, tensile strength at yield, elongation at break and stress values in tensile test	8.5.2.3 8.5.2.4 8.5.2.8 8.5.2.9
DIN 53505	Shore A and Shore D hardness	8.5.2.5 8.5.2.10
ISO 815	Compression set at various temperatures	8.5.2.6
ISO 1817	Behaviour of rubber in liquids	
ISO 23529	Linear dimensions of test pieces and products	8.5.2.1

Annex B (informative)

Proven diaphragm materials

The following materials have proven suitable for diaphragms on the basis of Clause 8 in water with glycol based additives of up to 49 %:

- IIR (Butyl rubber);
- NBR (Nitrile rubber);
- NR (Natural rubber);
- SBR (Styrens — butadiene rubber);
- EPDM (Ethylene — Propylens — dien rubber).

Annex ZA (informative)

Relationship between this European Standard and the Essential requirements of EU Directive 97/23/EC

This European Standard has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association to provide a means of conforming to Essential Requirements of the New Approach Directive 97/23/EC.

Once this standard is cited in the Official Journal of the European Communities under that Directive and has been implemented as a national standard in at least one Member State, compliance with the clauses of this standard given in Table ZA.1 confers, within the limits of the scope of this standard, a presumption of conformity with the corresponding Essential Requirements of that Directive and associated EFTA regulations.

Table ZA-1 — Correspondence between this European Standard and Directive 97/23/EC

Clause(s)/sub-clause(s) of this EN	Essential Requirements (ERs) of Directive 97/23/EC	Qualifying remarks/Notes
5.1	Annex I, 4.1 (d)	Material suitable for intended processing procedure
6.1.4	Annex I, 2.4	Means of examination
6.2	Annex I, 2.2.4	Experimental method
6.3	Annex I, 2.2.3	Calculation method
6.3.2.4	Annex I, 7.1.2	Permissible membrane stress
6.3.2.5	Annex I, 7.2	Joint coefficients
7 (except 7.2)	Annex I, 3.1	Manufacturing procedures
7.4 / 7.5 / 7.7 (except 7.7.2)	Annex I, 3.1.2	Joining
7.6	Annex I, 3.1.3	Non-destructive testing
7.8.3	Annex I, 3.1.4	Heat treatment
9.3/9.4	Annex I, 3.2	Final assessment
9.5	Annex I, 3.3	Labelling and marking
9.4	Annex I, 7.4	Hydrostatic test pressure

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

Bibliography

- [1] EN 1708-1:1999, *Welding — Basic weld joint details in steel — Part 1: Pressurized components*
- [2] EN 10028-7:2000, *Flat products made of steel for pressure purposes — Part 7: Stainless steels*
- [3] CEN ISO/TR 15608:2005, *Welding — Guidelines for a metallic materials grouping system (ISO/TR 15608:2005)*
- [4] ISO 815:1991⁴⁾, *Rubber, vulcanized or thermoplastic — Determination of compression set at ambient, elevated or low temperatures*
- [5] ISO 1817:2005, *Rubber, vulcanized — Determination of the effect of liquids*
- [6] ISO 2781:1988, *Rubber, vulcanized — Determination of density*
- [7] DIN 53504:1994, *Testing of rubber — Determination of tensile strength at break, tensile stress at yield, elongation at break and stress values in a tensile test*
- [8] DIN 53505:2000, *Testing of rubber — Shore A and Shore D hardness test*
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- [10] EN 10130, *Cold rolled low carbon steel flat products for cold forming — Technical delivery conditions*
- [11] EN 10111, *Continuously hot-rolled low carbon steel sheet and strip for cold forming — Technical delivery conditions*
- [12] EN 10025-2, *Hot rolled products of structural steels — Part 2: Technical delivery conditions for non-alloy structural steels*
- [13] EN 1092-1, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 1: Steel flanges*
- [14] EN 1759-1, *Flanges and their joint — Circular flanges for pipes, valves, fittings and accessories, Class designated — Part 1: Steel flanges, NPS ½ to 24*

4) Under revision.

