BRITISH STANDARD

BS EN 12516-2:2004

Industrial valves — Shell design strength —

Part 2: Calculation method for steel valve shells

The European Standard EN 12516-2:2004 has the status of a British Standard

ICS 23.060.01

British Standards

National foreword

This British Standard is the official English language version of EN 12516-2:2004.

The UK participation in its preparation was entrusted by Technical Committee PSE/7, Industrial valves, to Subcommittee PSE/7/6, Industrial valves: steel valves, which has the responsibility to:

— aid enquirers to understand the text;

— present to the responsible international/European committee any enquiries on the interpretation, or proposals for change, and keep the UK interests informed;

— monitor related international and European developments and promulgate them in the UK.

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

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This European Standard was approved by CEN on 16 April 2004.

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This European Standard exists in three official versions (English, French, German). A version in any other language made by translation under the responsibility of a CEN member into its own language and notified to the Central Secretariat has the same status as the official versions.

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Contents

Foreword

This document (EN 12516-2:2004) has been prepared by Technical Committee CEN/TC 69 “Industrial valves”, the secretariat of which is held by AFNOR.

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of an identical text or by endorsement, at the latest by January 2005, and conflicting national standards shall be withdrawn at the latest by January 2005.

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.

EN 12516*, Industrial valves – Shell design strength,* consists of four parts:

— *Part 1: Tabulation method for steel valve shells*

— *Part 2: Calculation method for steel valve shells*

— *Part 3: Experimental method*

— *Part 4: Calculation method for valve shells in metallic materials other than steel*

The annexes A, B and C are informative.

This document includes a Bibliography.

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

Introduction

EN 12516, *Industrial valves — Shell design strength*, is in four parts. Parts 1 and 2 specify methods for determining the thickness of steel valve shells by tabulation or calculation methods respectively. Part 3 establishes an experimental method for assessing the strength of valve shells in steel, cast iron and copper alloy by applying an elevated hydrostatic pressure at ambient temperature. Part 4 specifies methods for calculating the thickness for valve shells in metallic materials other than steel.

The calculation method, Part 2 is similar in approach to DIN 3840 where the designer calculates the wall thickness for each point on the pressure temperature curve using the allowable stress at temperature for the material he has chosen (see Bibliography, reference [1]). The allowable stress is calculated from the material properties using safety factors that are defined in Part 2. The equations in Part 2 consider the valve as a pressure vessel and ensure that there is no excessive deformation or plastic instability.

The tabulation method, Part 1 is similar in approach to ASME B16.34 in that the designer can look up the required minimum wall thickness of the valve body from a table (see Bibliography, reference [2]). The internal diameter of the straight pipe, into which the valve is to be mounted, gives the reference dimension from which the tabulated wall thicknesses of the body are calculated.

The tabulated thicknesses in Part 1 are the minimum thickness in the crotch region and are calculated using an allowable stress equal to 118 N/mm^2 and a calculation pressure, p_c, in N/mm^2. The values of the calculation pressure, p_c, and the equation used for calculating the thickness are given in Part 1.

Part 1 specifies Standard and Special pressure temperature ratings for valve bodies having the tabulated thickness. These tabulated pressure temperature ratings are applicable to a group of materials and are calculated using a selected stress, which is determined from the material properties representative of the group, using safety factors defined in Part 1.

Each tabulated pressure temperature rating is given a reference pressure designation to identify it. The B (Body) pressure designation is used to differentiate it from the PN pressure designation that is used for flanges because the rules for determining the pressure temperature ratings for B and PN designations are different.

In the case where a valve body designed to Part 1 is having PN designated flanged ends, the designer considers the requirements laid down in Part 1 to ensure that the valve body is not weaker than the flange.

Maximum allowable pressures for Special ratings are higher than those for Standard ratings as additional non-destructive examination of the body used for Special rating allows the use of lower safety factors in calculating the allowable pressure.

A merit of the calculation method is that it allows the most efficient design for a specific application using the allowable stresses for the actual material selected for the application.

A merit of the tabulation method, which has a fixed set of shell dimensions irrespective of the material of the shell, is that it is possible to have common patterns and forging dies. The allowable pressure temperature rating for each material varies proportionally to the selected stresses of the material group to which the material belongs.

The two methods are based on different assumptions, and as a consequence the detail of the analysis is different (see Bibliography, reference [7]). Both methods offer a safe and proven method of designing pressure-bearing components for valve shells.

1 Scope

This part of EN 12516 specifies the method for the strength calculation of the shell with respect to internal pressure of the valve.

2 Normative references

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

EN 19, *Industrial valves — Marking of metallic valves.*

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

EN 1515-1, *Flanges and their joints — Bolting — Part 1: Selection of bolting.*

EN 1591-1, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method.*

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

3 Symbols and units

The following symbols are used:

Table 1 — Symbols characteristics and units

Symbol	Characteristic	Unit
A	elongation after rupture	%
B_n	calculation coefficient for oval cross-sections	—
E	modulus of elasticity	MPa or N/mm^2
e	thickness	mm
f	nominal design stress	MPa or N/mm^2
f_d	maximum value of the nominal design stress for normal operating load cases	MPa or N/mm^2
$f_{d/t}$	nominal design stress for design conditions at temperature t °C	MPa or N/mm^2
f_{exp}	nominal design stress for exceptional conditions	MPa or N/mm^2
k_c	welding factor	—
p	pressure	MPa or N/mm^2
p_c	calculation pressure	MPa or N/mm^2
p_d	design pressure	MPa or N/mm^2
PS	maximum allowable pressure	MPa or N/mm^2
R_e	yield strength	MPa or N/mm^2
$R_{eH/t}$	upper yield strength at temperature t °C	MPa or N/mm^2

Table 1 — (*concluded*)

Symbol	Characteristic	Unit
R_m	tensile strength	MPa or N/mm^2
$R_{m/t}$	tensile strength at temperature t °C	MPa or N/mm^2
$R_{m/T/t}$	creep rupture strength for T hours at temperature t °C	MPa or N/mm^2
$R_{p0,2}$	0,2 % - proof strength	MPa or N/mm^2
$R_{p0,2/t}$	0,2 % - proof strength at temperature t °C	MPa or N/mm^2
$R_{p1,0}$	1,0 % - proof strength	MPa or N/mm^2
$R_{p1,0/t}$	1,0 % - proof strength at temperature t °C	MPa or N/mm^2
$R_{p1,0/T/t}$	1,0 % - creep proof strength for T hours at temperature t °C	MPa or N/mm^2
SF	safety factor	—
T	time	h
t	temperature	°C
t_c	calculation temperature	°C
t_d	design temperature	°C
α	linear expansion factor	K^{-1}
β	cone calculation coefficient	—
ε	strain	%
μ	Poisson's ratio	—

4 General conditions for strength calculation

Equations 1 and 2 apply to mainly static internal pressure stressing. The extent to which these equations can also be applied to pulsating internal pressure stressing is described in clause 12.

The total wall thickness is found by adding the following allowances:

$$e_0 = e_{c0} + c_1 + c_2 \tag{1}$$

$$e_1 = e_{c1} + c_1 + c_2 \tag{2}$$

where

e_{c0}, e_{c1} are the calculated wall thicknesses in accordance with the rules given in this standard at different locations on the valve shell (see Figures 1, 2, 5 and 8 to 20);

c_1 is a manufacturer tolerance allowance;

c_2 is a corrosion allowance.

The values of the corrosion allowance are:

c_2 = 1 mm for ferritic and ferritic-martensitic steels;

c_2 = 0 mm for all other steels.

When checking the wall thickness of existing pressure retaining shells these allowances shall be subtracted from the actual wall thickness.

5 Design pressure

All reasonably foreseeable conditions shall be taken into account, which occur during operation and standby.

Therefore the design pressure p_d shall be not less than the maximum allowable pressure PS.

6 Nominal design stresses for pressure parts other than bolts

6.1 General

The nominal design stresses (allowable stresses) for steels with a minimum elongation after rupture of ≥ 14 % and a minimum impact energy measured on a Charpy-V-notch impact test specimen of ≥ 27 J should be calculated in accordance with Table 2.

Table 2 — Nominal design stresses (allowable stresses)

Material	Design conditions	Creep conditions
Steel as defined in 6.2	f = min ($R_{p0,2/t}$ / 1,5 ; $R_{m/20}$ / 2,4)	$f = R_{m/100\,000/t}$ / 1,5
Austenitic steel and cast steel as defined in 6.2	f = min ($R_{p1,0/t}$ / 1,5 ; $R_{m/20}$ / 2,4)	$f = R_{m/100\,000/t}$ / 1,5
Austenitic steel as defined in 6.3 with rupture elongation ≥ 30 %	$f = R_{p1,0/t}$ / 1,5	$f = R_{m/100\,000/t}$ / 1,5
Austenitic steel as defined in 6.4 with rupture elongation ≥ 35 %	f = max [$R_{p1,0/t}$ / 1,5; min ($R_{p1,0/t}$ / 1,2; $R_{m/t}$ / 3,0)]	$f = R_{m/100\,000/t}$ / 1,5
Cast steel as defined in 6.5	f = min ($R_{p0,2/t}$ / 1,9; $R_{m/20}$ / 3,0)	$f = R_{m/100\,000/t}$ / 1,9
Weld-on ends on cast steel as defined in 6.5	f = min ($R_{p0,2/t}$ / 1,5; $R_{m/20}$ / 2,4) [a]	$f = R_{m/100\,000/t}$ / 1,5

[a] The transition zone situated immediately outside the effective length l_0 or l_1 may be calculated with this higher nominal design strength if the length of the transition zone ≥ 3 × e_v , however = 50 mm min. and the angle of the transition ≤ 30°.

However, materials with lower elongation values and/ore lower values for a Charpy-V-notch impact test may also be applied, provided that appropriate measures are taken to compensate for these lower values and the specific requirements are verifiable.

6.2 Steels and cast steels other than defined in 6.3, 6.4 or 6.5

The maximum value of the nominal design stress for normal operating load cases f_d shall not exceed the smaller of the following two values:

— the yield strength $R_{eH/t}$ or 0,2 % proof strength $R_{p0,2/t}$ at calculation temperature, as given in the material standard, divided by the safety factor SF = 1,5. For austenitic steels and cast steels with a rupture elongation less than 30 % and with a relationship at 20 °C between proof and tensile strength less than or equal 0,5 the 1,0 % proof strength $R_{p1,0/t}$ can by used, divided by the safety factor SF = 1,5;

— the minimum tensile strength R_m at 20 °C as given in the material standard, divided by the safety factor SF = 2,4.

6.3 Austenitic steel and austenitic cast steel with a minimum rupture elongation not less than 30 %

The maximum value of the nominal design stress for normal operating load cases f_d shall not exceed the 1,0 % proof strength $R_{p1,0/t}$ at calculation temperature, as given in the material standard, divided by the safety factor SF = 1,5.

NOTE The nominal design stresses of this clause are in accordance with the Pressure Equipment Directive 97/23/EC Annex 1, Clause 7. The term "nominal design stress" means the "permissible general membrane stress" in the context of this Directive.

6.4 Austenitic steel and austenitic cast steel with a minimum rupture elongation not less than 35 %

The maximum value of the nominal design stress for normal operating load cases f_d shall not exceed the greater of the following two values:

- the 1,0 % proof strength $R_{p1,0/t}$ at calculation temperature, as given in the material standard, divided by the safety factor SF = 1,5;
- the smaller of the two values:
  - the 1,0 % proof strength $R_{p1,0/t}$ at calculation temperature, as given in the material standard, divided by the safety factor SF = 1,2;
  - the minimum tensile strength $R_{m/t}$ at calculation temperature divided by the safety factor SF = 3,0.

6.5 Non-alloy and low-alloy cast steel

The maximum value of the nominal design stress for normal operating load cases f_d shall not exceed the smaller of the following two values:

- the yield strength $R_{eH/t}$ or 0,2 % proof strength $R_{p0,2/t}$ at calculation temperature, as given in the material standard, divided by the safety factor SF = 1,9;
- the minimum tensile strength R_m at 20 °C as given in the material standard, divided by the safety factor SF = 3,0.

6.6 Creep conditions

The maximum value of the nominal design stress for normal operating load cases shall not exceed the average creep rupture strength at calculation temperature $R_{m/T/t}$ divided by the safety factor SF = 1,5 for the T = 100 000 hours value.

The nominal design stress calculated in 6.2 to 6.5 has to be compared with the nominal design stress calculated in this clause and the lower value shall be used.

For cast steel defined in 6.5 the safety factor SF = 1,9 for the T = 100 000 hours value.

For limited operating times and in certain justified cases, creep rupture strength values for shorter times may be used for calculations but not less than T = 10 000 hours.

7 Calculation methods for the wall thickness of valve bodies

7.1 General

Valve bodies are considered to be hollow bodies penetrating each other with different angles i.e. basic bodies with branches.

Basic bodies and branches can be tubes, balls or conical hollow parts with cylindrical, spherical, elliptical or rectangular cross-sections.

In special cases the body consists only of a basic body.

The basic body-part is the part of the body with the larger diameter or cross-section, with the symbol d_0. For the branches, the symbols are for example, d_1, d_2.

It follows that:

$d_0 \geq d_1$; $b_2 \geq d_1$, see Figure 8

7.2 Valve bodies

7.2.1 General

The wall thickness of a valve body composed of different geometric hollow components cannot be calculated directly. The calculation needs two steps:

— the calculation of the wall thickness of the basic body and the branches outside of the intersection — or crotch area, see 7.2.2;

— the calculation of the wall thickness in the crotch area, see 7.2.3.

A check of the wall thickness of the crotch area is necessary by considering the equilibrium of forces, see 7.2.3.

7.2.2 Wall thickness of bodies and branches outside crotch area

7.2.2.1 General

Outside the intersection or crotch area, means that the calculated hollow body is without openings or cutaways in this zone (e.g. a smooth tube).

The welding factor k_c in the following equations is a calculation factor dependent on the level of destructive and non-destructive testing to which the weld or series of welds is subject.

The values of the welding factor k_c shall be:

1,0 for equipment subject to destructive and non-destructive tests, which confirm that the whole series of joints show no significant defects;

0,85 for equipment of which 10% of the welds are subject to random non-destructive testing and all welds are subject to 100% visual inspection;

0,7 for equipment not subject to non-destructive testing other than 100% visual inspection of all the welds;

1,0 for no welds.

All the calculated wall thicknesses are wall thicknesses excluding allowances.

d_i = inner diameter or radius;

d_o = outer diameter or radius.

7.2.2.2 Cylindrical bodies or branches

$d_o / d_i \leq 1{,}7$

$$e_c = \frac{d_i \times p}{(2 \times f - p) \times k_c} \tag{3}$$

or

$$e_c = \frac{d_o \times p}{(2 \times f - p) \times k_c + 2 \times p} \quad (4)$$

7.2.2.3 Both equations are equivalent when $d_i = d_o - 2 \times e_c$

7.2.2.4 Spherical bodies or branches

$d_o / d_i \leq 1{,}2$

$$e_c = \frac{r_i \times p}{(2 \times f - p) \times k_c} \quad (5)$$

or

$$e_c = \frac{r_o \times p}{(2 \times f - p) \times k_c + p} \quad (6)$$

$1{,}2 < d_o / d_i \leq 1{,}5$

$$e_c = r_i \times \left[\sqrt{1 + \frac{2 \times p}{(2 \times f - p) \times k_c}} - 1 \right] \quad (7)$$

or

$$e_c = r_o \times \frac{\sqrt{1 + \frac{2 \times p}{(2 \times f - p) \times k_c}} - 1}{\sqrt{1 + \frac{2 \times p}{(2 \times f - p) \times k_c}}} \quad (8)$$

Both equations are equivalent when $r_i = r_o - e_c$

7.2.2.5 Conical bodies or branches

$e_c / d_o > 0{,}005$

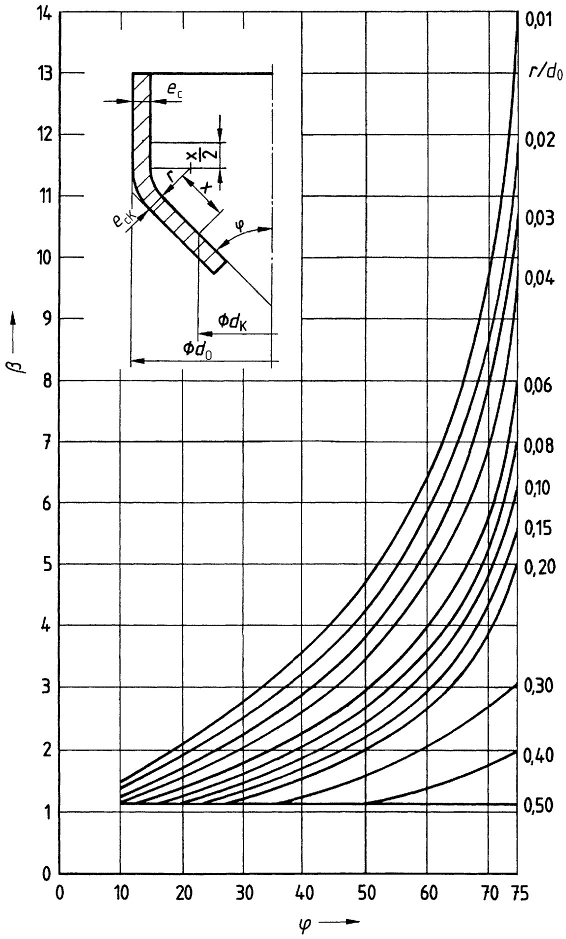


Figure 1 — Cone calculation coefficient

$$e_c = \frac{p \times d_K}{2 \times f \times k_c - p} \times \frac{1}{\cos(\varphi)} \tag{9}$$

Wall thickness in the knuckle or in a corner weld:

$$e_{cK} = \frac{d_o \times p \times \beta}{4 \times f \times k_c} \tag{10}$$

e_{cK} is also required in the zone x and $\frac{x}{2}$

$$x = \sqrt{d_o \times e_c} \tag{11}$$

k_c is now a factor for a weld situated in the knuckle or in the influence zone of the knuckle running in meridian direction.

In cases of corner welds which are admissible for angles $\varphi \leq 30°$, $e_{cK} \leq 20$ mm and double joint weld, β shall be read off Figure 1 by taking for the ratio r / d_o = 0,01.

For corner welds, diameter d_K is equal to the inside diameter of the wide end.

In case of flat cones with a knuckle and $\varphi > 70°$:

$$e_c = 0{,}3 \times (d_o - r) \times \frac{\varphi}{90} \times \sqrt{\frac{p}{f \times k_c}} \qquad (12)$$

Table 3 — Cone calculation coefficient

Angle φ	β for the ratio r / d_o												cos φ
	0,01	**0,02**	**0,03**	**0,04**	**0,06**	**0,08**	**0,10**	**0,15**	**0,20**	**0,30**	**0,40**	**0,50**	
10	1,4	1,3	1,2	1,2	1,1	1,1	1,1	1,1	1,1	1,1	1,1	1,1	0,985
20	2,0	1,8	1,7	1,6	1,4	1,3	1,2	1,1	1,1	1,1	1,1	1,1	0,940
30	2,7	2,4	2,2	2,0	1,8	1,7	1,6	1,4	1,3	1,1	1,1	1,1	0,866
45	4,1	3,7	3,3	3,0	2,6	2,4	2,2	1,9	1,8	1,4	1,1	1,1	0,707
60	6,4	5,7	5,1	4,7	4,0	3,5	3,2	2,8	2,5	2,0	1,4	1,1	0,500
75	13,6	11,7	10,7	9,5	7,7	7,0	6,3	5,4	4,8	3,1	2,0	1,1	0,259

If two conical shells with different taper angles are joined together, the angle φ arising between the conical portion with the more pronounced taper and that with the less pronounced taper shall be determined for the determination of β.

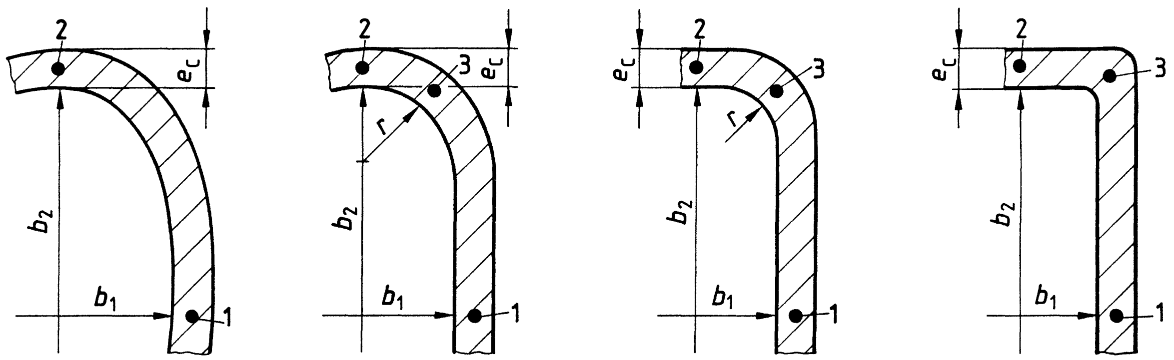
7.2.2.6 Bodies or branches with oval or rectangular cross-sections

7.2.2.6.1 General

The following calculation rules apply to oval or rectangular valve bodies with a wall thickness/diameter ratio $e_c / b_2 \leq 0{,}15$ and a ratio $b_1 / b_2 \geq 0{,}4$.

For ratios $e_c / b_2 \leq 0{,}06$, these rules are applicable for $b_1 / b_2 \geq 0{,}25$ (see Bibliography, reference [3]).

7.2.2.6.2 In the case of oval shaped cross-sections (see Figure 2a)) and of rectangular shapes with or without radiusing of the corners (see Figures 2b) to 2d)), the additional bending stresses, which arise in the walls or in the corners, shall be taken into consideration.



a) oval-shaped

b) rectangular radiused on one side

c) rectangular, with radiused corner

d) rectangular, corner not radiused

Figure 2 — Cross-sections

The theoretical minimum wall thickness of such bodies under internal pressure stressing can be calculated by means of the equation below, without any allowance for edge effects:

$$e_{c0} = \frac{p \times b_2}{2f} \times \sqrt{B_0^2 + \frac{4f}{p} \times B_n} \tag{13}$$

7.2.2.6.3 The calculation shall be carried out in respect of locations 1 and 2 (designated in Figure 2a for oval-shaped cross-sections), and in respect of locations 1 and 3 (designated in Figures 2b to 2d for rectangular cross-sections), because the bending moments, which have a predominant influence on the strength behaviour, exhibit their maximum values at the above locations. In exceptional cases (e.g. a low b_1 / b_2 ratio) a check calculation for location 2 may also be necessary for square cross-sections.

7.2.2.6.4 The calculation coefficient B_0, which is a function of the normal forces, shall be:

$B_0 = b_1 / b_2$ for location 1

$B_0 = 1$ for location 2

For location 3, B_0 can be obtained from Figure 3 as a function of the sides ratio b_1 / b_2 and of the corner radii ratio r / b_2, or it can be calculated in accordance with equation (14):

$$B_0 = \left[1 - \frac{2r}{b_2}(1 - \sin\varphi_k)\right] \sin\varphi_k + \left[\frac{b_1}{b_2} - \frac{2r}{b_2}(1 - \cos\varphi_k)\right] \cos\varphi_k \tag{14}$$

$$\text{with } \tan\varphi_k = \frac{1 - 2r/b_2}{\frac{b_1}{b_2} - \frac{2r}{b_2}} \tag{15}$$

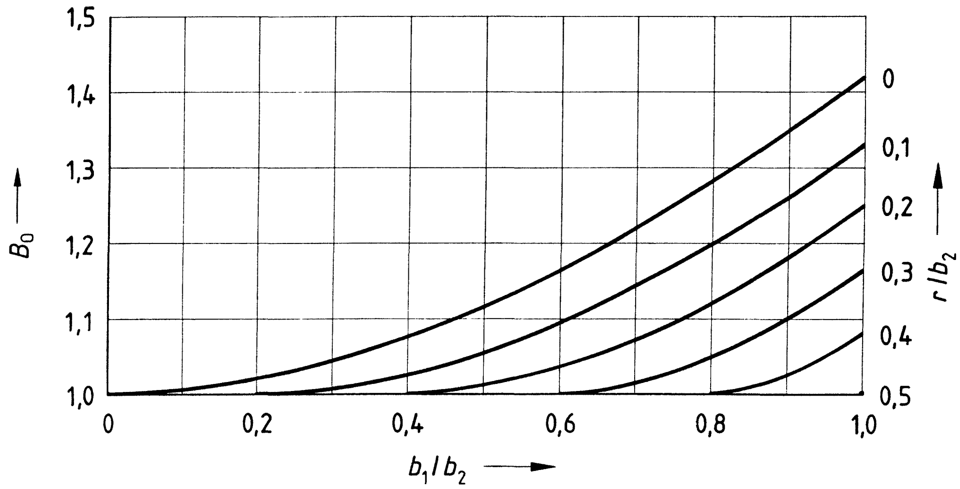


Figure 3 — Calculation coefficient B_0 for location 3

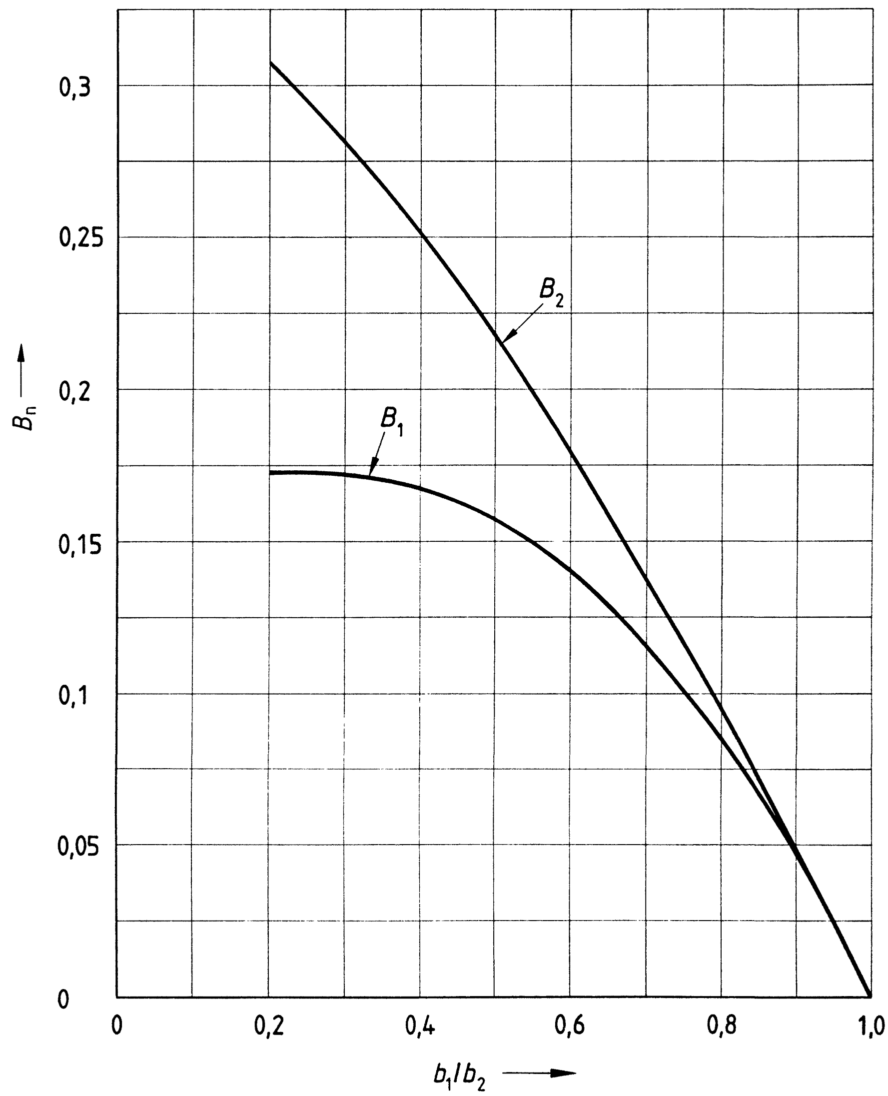
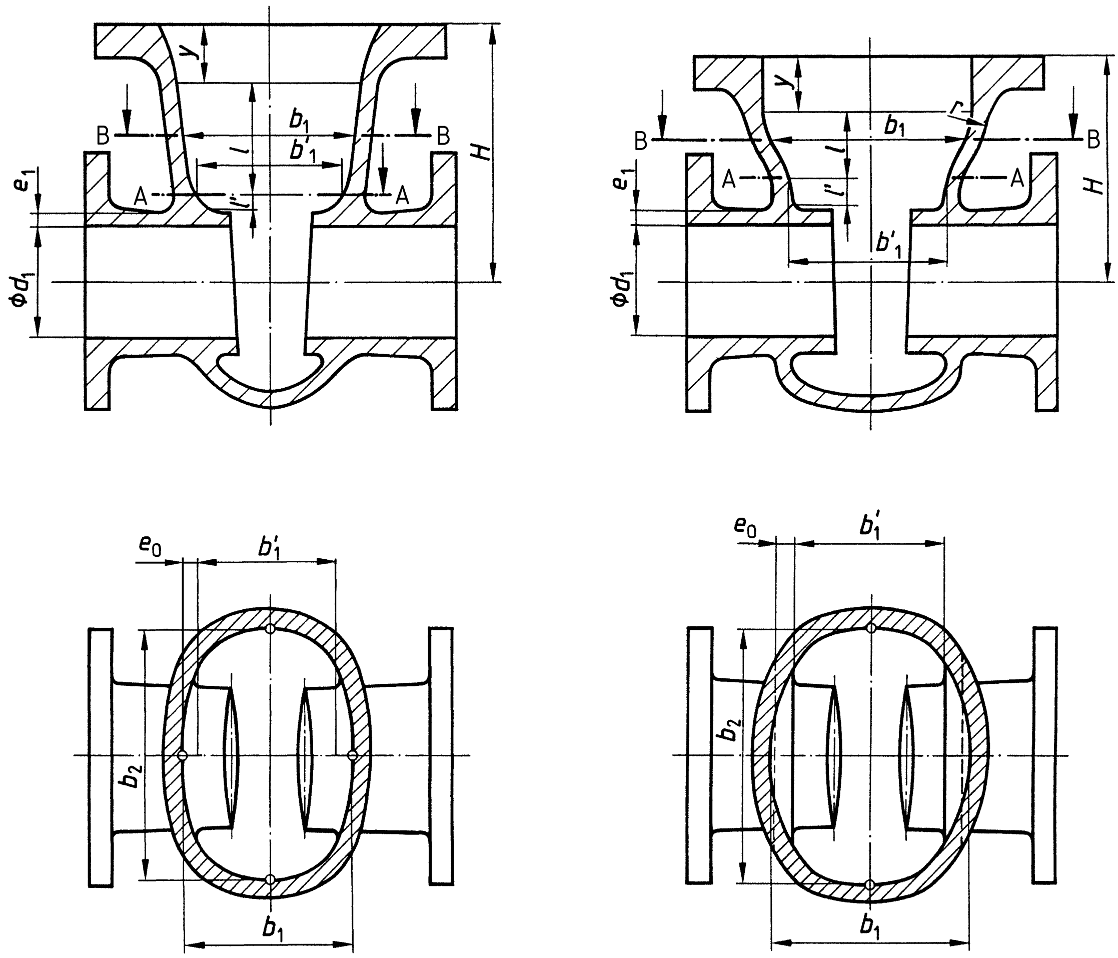


Figure 4 — Calculation coefficient B_n for oval-shaped cross-sections



Design a) **Design b)**

Figure 5 — Examples of changes in cross-section in oval basic bodies

7.2.2.6.5 The calculation coefficients B_n which are dependent on the bending moments are plotted in Figure 4 as a function of ratio b_1 / b_2 for oval-shaped cross-sections for locations 1 and 2. These curves correspond to the equations below:

$$B_1 = \frac{1-k_E^2}{6} \times \frac{K'}{E'} - \frac{1-2k_E^2}{6} \tag{16}$$

$$B_2 = \frac{1+k_E^2}{6} - \frac{1-k_E^2}{6} \times \frac{K'}{E'} \tag{17}$$

$$\text{with } k_E^2 = 1 - \left(\frac{b_1}{b_2}\right)^2 \tag{18}$$

These values result from the analytical solution of the equations of equilibrium for a curved shaped beam. The values of K', E', are explained in reference [4] of the Bibliography.

For determination of the calculation coefficients the following approximation equations may also be used for b_1 / $b_2 \geq 0{,}5$:

$$B_1 = \left(1 - \frac{b_1}{b_2}\right)\left(0{,}625 - 0{,}435 \times \sqrt{1 - \frac{b_1}{b_2}}\right) \tag{19}$$

$$B_2 = \left(1 - \frac{b_1}{b_2}\right)\left[0,5 - 0,125 \times \left(1 - \frac{b_1}{b_2}\right)\right] \quad (20)$$

The calculation coefficients are also valid for changes in cross-section in oval basic bodies (e.g. for gate valves in accordance with Figure 5, design a) and design b)), on these valves, the lateral length b_1 increases from the apex zone of the entry nozzle (flattened oval) to value length b_2 (circular shape) over length l. In this case, value b_1 in cross-section B-B up to $l/2$ is determining for the determination of B_n. l is obtained from:

$$l = H - y - \left(\frac{d_1}{2} + e_1\right) - l' \quad (21)$$

and length l' which is influenced by the entry nozzle is obtained from:

$$l' = 1,25 \times \sqrt{d'_m \times e_0} \quad (22)$$

with $d'_m = \frac{b'_1 + b_2}{2}$ (23)

where b'_1 and b_2 shall be determined at the cross-section A–A at a distance l' from the entry nozzle. The wall thickness at that location is e_0.

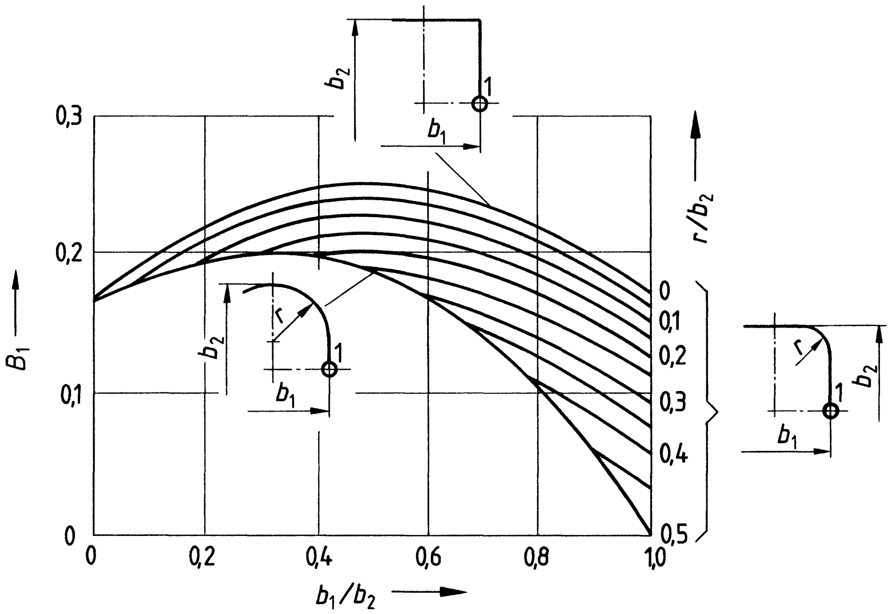


Figure 6a — Calculation coefficient B_1 for rectangular cross-sections

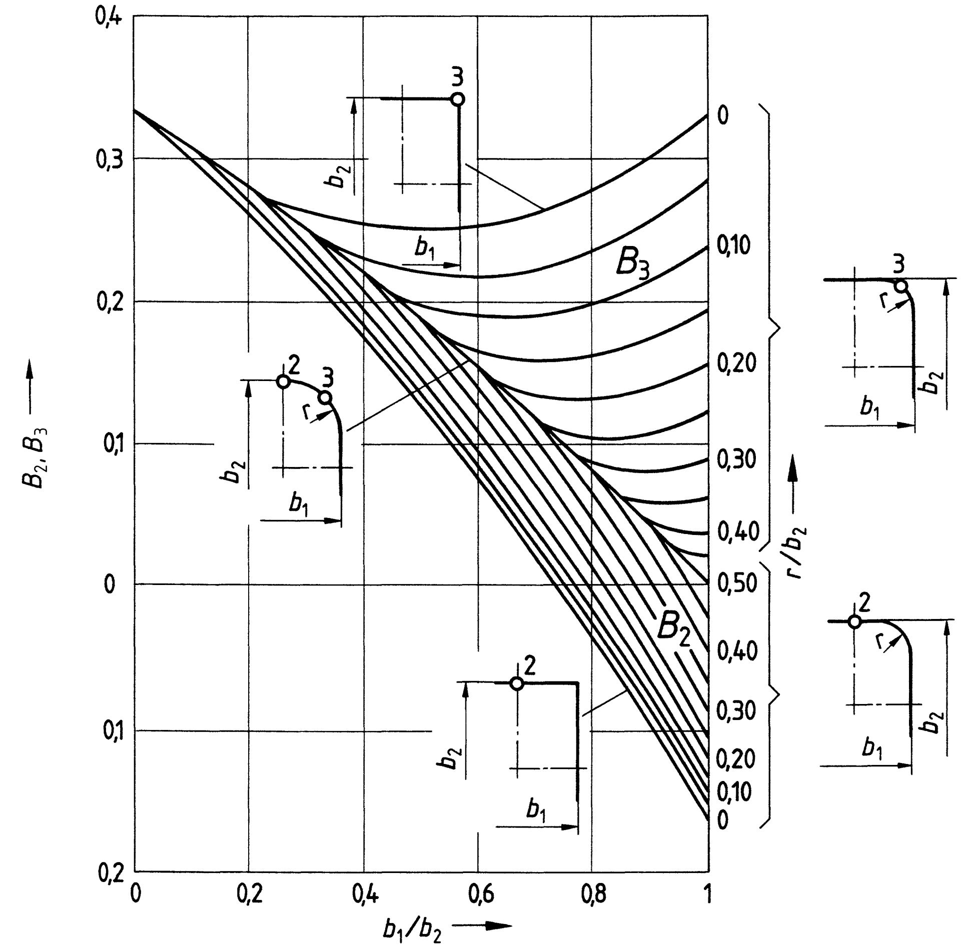


Figure 6b — Calculation coefficients B_2 and B_3 for rectangular cross-sections

Figure 6 — Calculation coefficients

7.2.2.6.6 The calculation coefficients B_n for square cross-sections are plotted in Figure 6 as a function of the ratio b_1 / b_2 for locations 1 to 3 under consideration. The calculation coefficients B_n shall always be entered in equation (13) as positive values.

The curves for B_n can also be determined analytically with the aid of the following equations:

$$B_1 = \frac{1}{6} \times \frac{1 - 2 \times \left(\frac{b_1}{b_2}\right)^3 + 3 \times \left(\frac{b_1}{b_2}\right) - 3 \times \frac{2r}{b_2} \times \left(2 - \frac{\pi}{2}\right) - 3 \times \left(\frac{2r}{b_2}\right)^2 \left(1 + \frac{b_1}{b_2}\right) (\pi - 3) + \left(\frac{2r}{b_2}\right)^3 \left(\frac{9}{2}\pi - 14\right)}{1 + \frac{b_1}{b_2} - \frac{2r}{b_2} \times \left(2 - \frac{\pi}{2}\right)} \quad (24)$$

$$B_2 = B_1 - \frac{1}{2}\left[1 - \left(\frac{b_1}{b_2}\right)^2\right] \quad (25)$$

$$B_3 = \frac{1}{2}\left[1 - 2\,\frac{2r}{b_2} \times (1 - \sin\varphi_k) + \frac{4\,r^2}{b_2^2} \times (3 - 2\,\sin\varphi_k - 2\cos\varphi_k) - 2 \times \frac{b_1}{b_2} \times \frac{2r}{b_2} \times (1 - \cos\varphi_k)\right] - B_1 \quad (26)$$

where equation (15) shall apply to the angle of the maximum moment.

7.2.2.6.7 For short valve bodies (e.g. design a) or b) of Figure 5) with the undisturbed length l corresponding to the calculation geometry, the supporting action of the components adjoining the ends (e.g. flanges, bottoms, covers) can be taken into account in the calculation. In this case the required minimum wall thickness in accordance with equation (13) becomes:

$$e_c = e_{c0} \times k \tag{27}$$

The correction factor k is obtained from equation (28) below, by analogy to the damping behaviour of the stresses in cylindrical shells, taking into consideration the experimental investigation results on non-circular casings:

$$k = 0{,}48 \times \sqrt[3]{\frac{l^2}{d_m \times e_{c0}}} \tag{28}$$

with $0{,}6 \le k \le 1$

This function is plotted in Figure 7 as a function of $l^2 / d_m \times e_{c0}$.

d_m shall be entered in the equation at a value $d_m = (b_1 + b_2) / 2$, and e_{c0} corresponds equation (13). In the case of changes in cross-section over length l, (e.g. in accordance with Figure 5 design a) or b)), dimensions b_1 and b_2 shall be taken at cross-section B–B (for $l / 2$). Local deviations from the shape of the casing body, whether they be of convex or concave nature, can as a general rule be ignored.

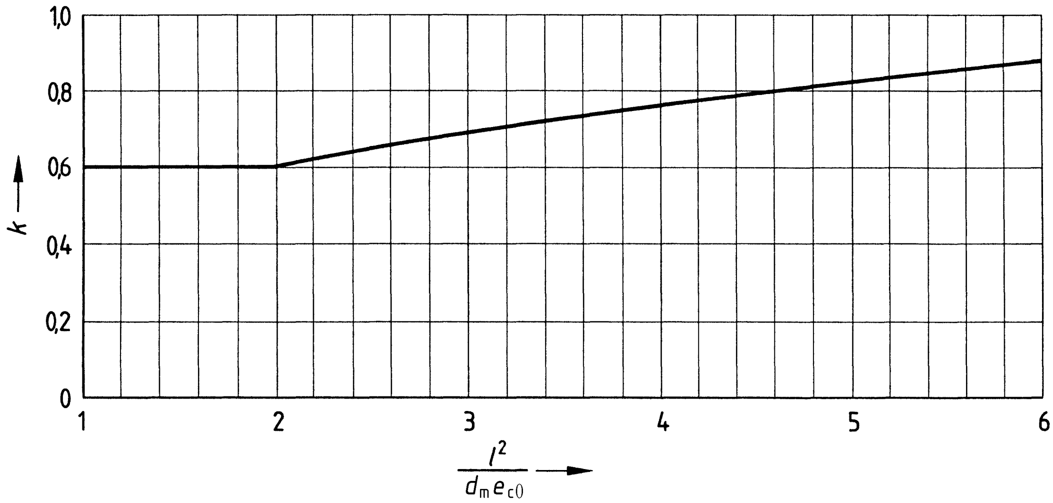


Figure 7 — Correction factor k for short casing bodies

7.2.2.6.8 The strength conditions can be deemed to be satisfied if the required wall thickness is attained locally, on the precondition that wall thickness transitions are gradual and gentle.

Should a finished design not meet the strength condition in accordance with equation (13) or (27), a local reinforcement, e.g. in the form of ribs, may be provided, and this will require a separate verification of the strength for the design, or alternatively the strength shall be verified by means of some other approved procedure.

7.2.3 Wall thickness in the crotch area

A direct calculation of the wall thickness in this area is not possible.

As a first step a wall thickness in this area shall be assumed; this assumption can also be derived from the wall thickness calculation in 7.2.2.

This assumed wall thickness shall be checked by considering the equilibrium of forces. The crotch area here is limited by the distances l, see Figures 8 to 20.

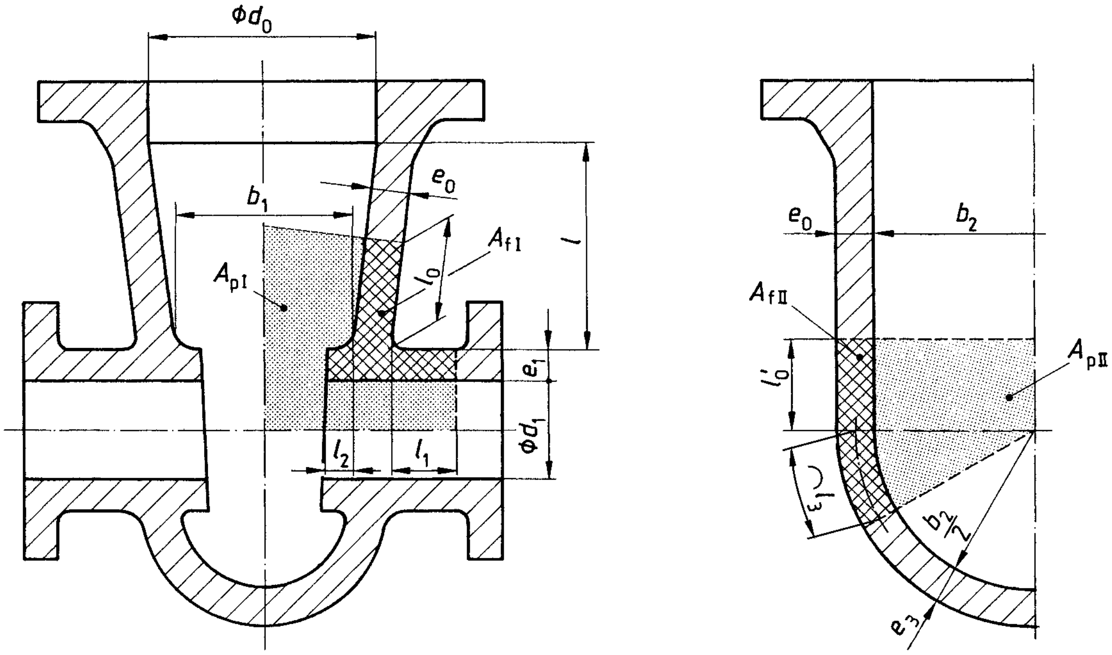


Figure 8a

Figure 8b

Figure 8 — Calculation procedure in the crotch area

According to Figure 8 the equilibrium of forces corresponds to the equation:

$$p \times A_p = f \times A_f \times k_c \tag{29}$$

where

$p \times A_{p\,I}$ or $p \times A_{p\,II}$	is the pressure loading area;
$f \times A_{f\,I}$ or $f \times A_{f\,II}$	is the metal cross-sectional area effective as compensation;
k_c	is the calculation coefficient depending on the welding process.

The areas A_p and A_f are determined by the centrelines of the bonnet and the flow passage and by the distances l, see Figure 8 to 20.

For the allowable value of f, see clause 6.

Table 4 shows the equations for f, depending on the body shape.

Condition: e_c body ≥ e_c branches, if this is not possible: e_c branches = e_c body in the whole A_f region including the distances l.

Table 4 — Calculation equations

circular cross-section	Diameter ratio $d_1 / d_0 < 0{,}7$ $p \times \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \times k_{\mathrm{c}}} + \frac{1}{2} \right] \le f$	(30)
non-circular cross-section	$p \times \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \times k_{\mathrm{c}}} + \frac{1}{2} \right] \le f / 1{,}2$ $p \times \left[\frac{A_{\mathrm{p\,II}}}{A_{\mathrm{f\,II}} \times k_{\mathrm{c}}} + \frac{1}{2} \right] \le f / 1{,}2$	(31) (32)
circular cross-section	Diameter ratio $d_1 / d_0 \ge 0{,}7$ $p \times \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \times k_{\mathrm{c}}} + \frac{1}{2} \right] \le f$ $p \times \left[\frac{A_{\mathrm{p\,II}}}{A_{\mathrm{f\,II}} \times k_{\mathrm{c}}} + \frac{1}{2} \right] \le f$	(33) (34)
circular cross-section	Diameter ratio $d_1 / d_0 \ge 0{,}7$ and $e_{\mathrm{c1}} / e_{\mathrm{c0}} < d_1 / d_0$ — Equation (33) — Equation (35) — additional for Section II: (Figure 8, right) $p \times \left[\frac{d_0 + e_{\mathrm{c0}}}{2 \times e_{\mathrm{c0}}} + 0{,}2 \times \frac{d_1 + e_{\mathrm{c1}}}{e_{\mathrm{c1}}} \times \sqrt{\frac{d_0 + e_{\mathrm{c0}}}{e_{\mathrm{c0}}}} \right] \le 1{,}5 \times f$	(35)
non-circular cross-section	— Equation (31) — Equation (32) — additional for Section II: $p \times \left[\frac{b_2 + e_{\mathrm{c0}}}{2 \times e_{\mathrm{c0}}} + 0{,}25 \times \frac{d_1 + e_{\mathrm{c1}}}{e_{\mathrm{c1}}} \times \sqrt{\frac{b_2 + e_{\mathrm{c0}}}{e_{\mathrm{c0}}}} \right] \le 1{,}5 \times f$	(36)

7.2.4 Examples of pressure-loaded areas A_{p} and metallic cross-sectional areas A_{f}

7.2.4.1 General

The pressure-containing areas A_{p} and the effective cross-sectional areas A_{f} shall be determined by calculation or from CAD drawings. The effective lengths shall be determined from the following relations.

For determination of the pressure-containing area A_{p}, the limitation inside the valve body is circumscribed by the geometrical centrelines of the basic body and the branch (see Figures 9 to 18). Reduced seats such as shown in Figure 9 shall not be taken into account. Due to the complicated geometrical shapes of bodies in accordance with Figures 9 to 13, the effective lengths l_0 and l_1 are indicated in the drawing as running parallel to the outside contour of the casing starting from the tangent point of the normal to the contour to the circle formed by the transition radius between the basic body and the branch (see examples in Figure 9). For small transition radii, it is sufficient to start from the intersection of the linearly extended contours of the bodies (see Figure 13). At the terminal point, the perpendicular is drawn to the relevant centreline.

Any material of the basic body or branch protruding inwards can be included in the effective cross-sectional area A_f up to a maximum length of l_0 / 2 or and l_1 / 2 with the limitation thus determined representing also the boundary of the pressure-loaded area (see for example Figures 8, 9 and 16). For penetration welds which can be tested, welded-in seat rings inside the valve body can be included in the calculation.

Abrupt wall thickness transitions shall be avoided (chamfer angle ≤ 30°).

For branch/valve body diameter ratios d_1 / d_0 > 0,8, the factor preceding the square root shall be 1 in all subsequent equations for effective lengths.

For all valve body shapes in cross-section II (Figure 8b) it is:

$$l'_0 = 1{,}25 \times \sqrt{(b_2 + e_0) \times e_0} \tag{37}$$

$$l_3 = \sqrt{(b_2 + e_3) \times e_3} \tag{38}$$

7.2.4.2 Cylindrical valve bodies

The effective lengths for cylindrical bodies e.g. in accordance with Figure 9 are:

$$l_0 = \sqrt{(d_0 + e_0) \times e_0} \tag{39}$$

$$l_1 = 1{,}25 \times \sqrt{(d_1 + e_1) \times e_1} \tag{40}$$

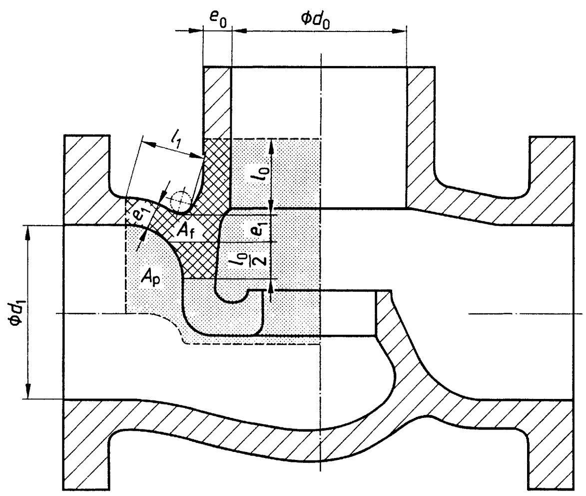


Figure 9a

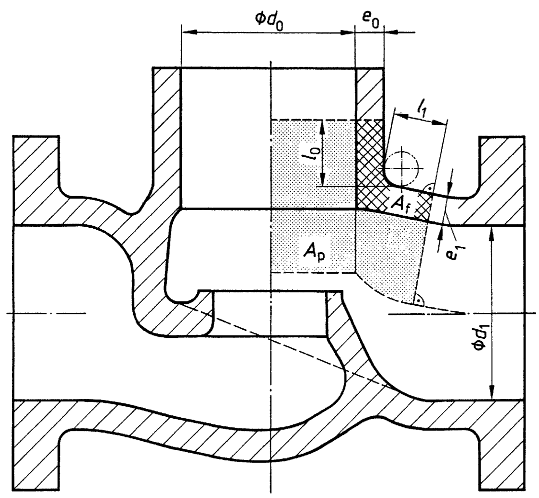


Figure 9b

Figure 9 — Cylindrical valve body

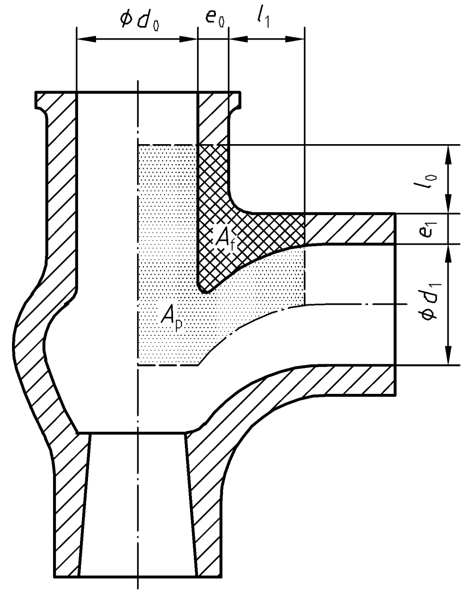


Figure 10a

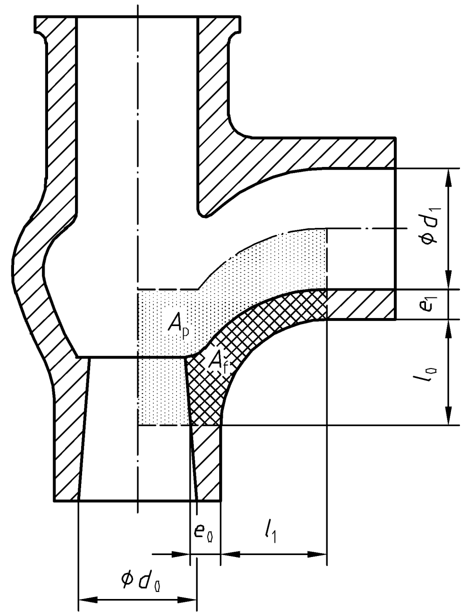


Figure 10b

Figure 10 — Angle valve

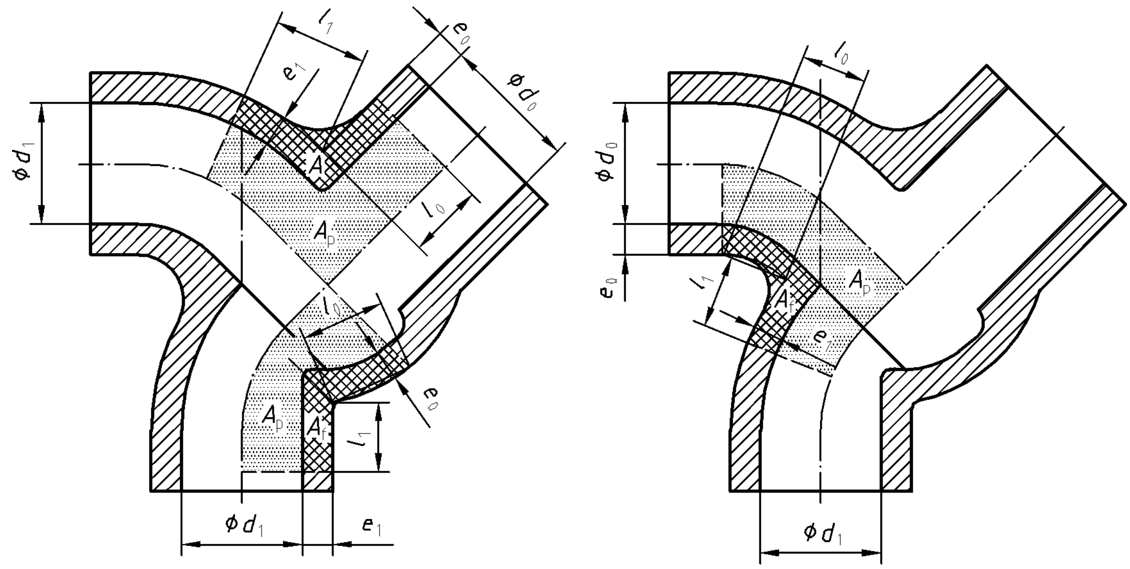


Figure 11 — Angle screw-down valve

For cylindrical valve bodies with oblique basic body or branch (e.g. in accordance with Figure 12) with $\phi_A \geq 45°$, instead of equation (39) the following equation shall be used for l_0 :

$$l_0 = \left(1 + 0{,}25\,\frac{\phi_A}{90°}\right)\sqrt{(d_0 + e_0) \times e_0} \tag{41}$$

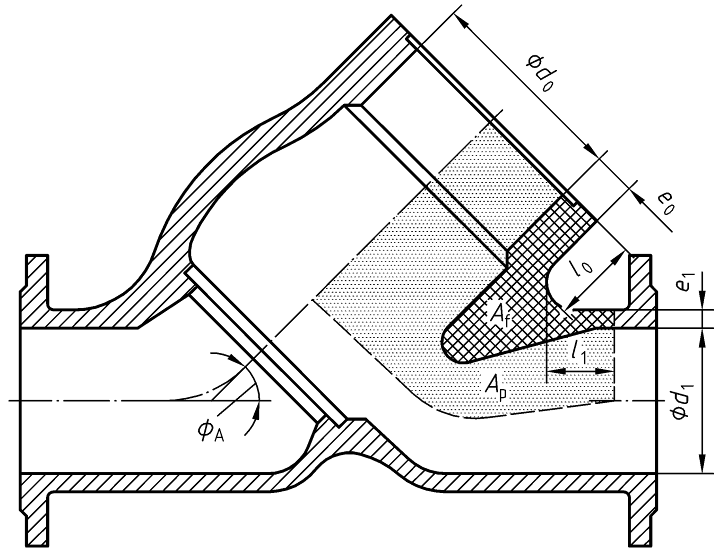


Figure 12 — Cylindrical valve body with oblique branch

For tapered basic bodies or branches the smallest diameters prevailing at the opening shall be taken in each case for d_0 and d_1 (see Figure 10 b) and Figure 13).

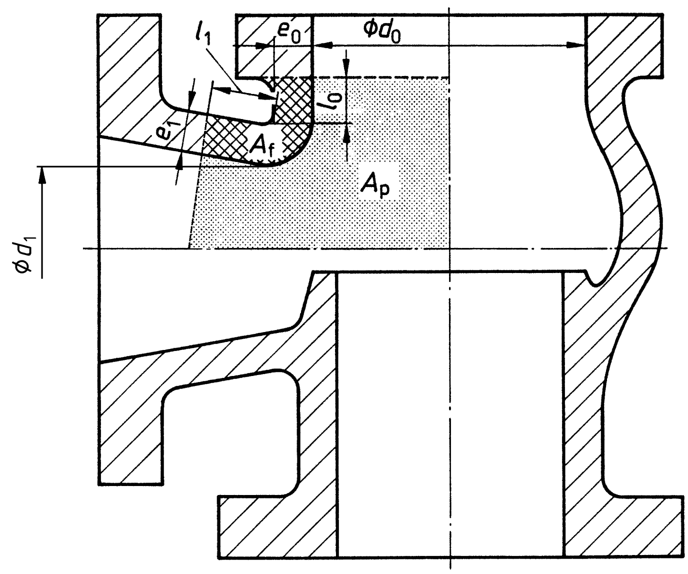


Figure 13 — Angle pattern valve body

7.2.4.3 Spherical valve bodies

For branches in spherical valve bodies with d_1 / d_0 or with $d_2 / d_0 \leq 0{,}5$, the effective lengths l_0 shall be determined using equation (39) on condition that

$l_0 \leq 0{,}5 \times l_3$

(see Figure 14, type a))

The corresponding length in the branch is:

$$l_1 = \sqrt{(d_1 + e_1) \times e_1} \tag{42}$$

In cases, where

$d_1 / d_0 > 0{,}5$ and

$d_2 / d_0 > 0{,}5$

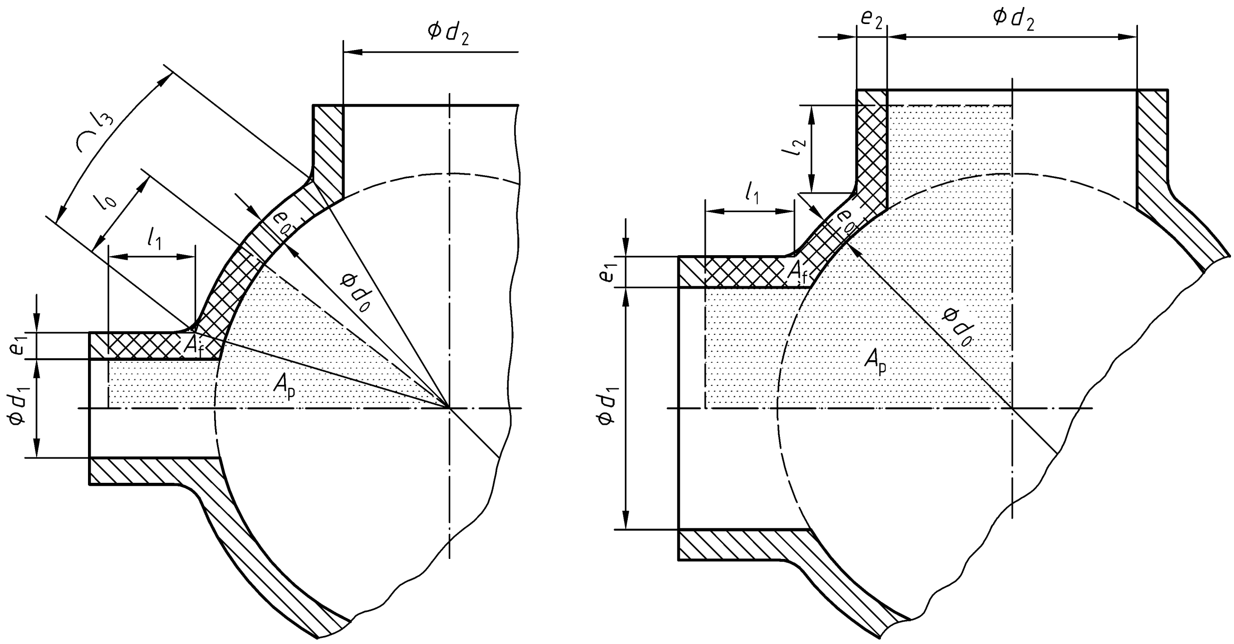
the pressure-containing area A_p and the stress area A_f shall be determined for both branches together in accordance with Figure 14, type b).

The effective lengths shall be determined as follows:

$$l_1 = \sqrt{(d_1 + e_1) \times e_1} \tag{43}$$

$$l_2 = \sqrt{(d_2 + e_2) \times e_2} \tag{44}$$

l_0 corresponds to the effective length between the branches.



Type a) **Type b)**

Figure 14 — Spherical valve body

7.2.4.4 Oval and rectangular cross-sections

For valve bodies with oval or rectangular cross-sections (see Figure 2), the lengths in accordance with Figure 8 shall be determined as follows:

$$l_0 = \sqrt{(b_1 + e_0) \times e_0} \tag{45}$$

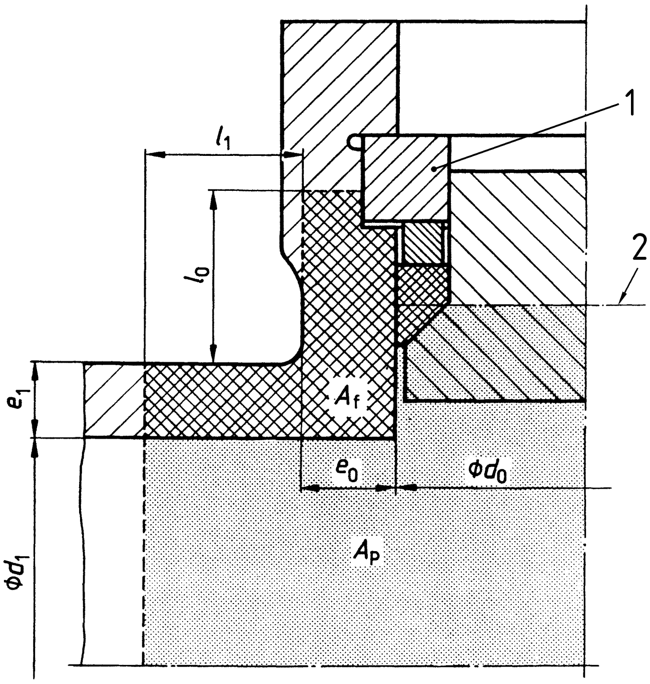
$$l_1 = 1{,}25 \times \sqrt{(d_1 + e_1) \times e_1} \tag{46}$$

7.2.4.5 Details

For design types with recesses (e.g. Figure 15), the recessed wall thickness shall be entered as wall thickness e_0 for the determination of the load-bearing cross-sectional area A_f. Increases in wall thickness beyond the recessed area shall not be allowed in the calculation.

For design types in accordance with Figure 15 where by the provision of a gasket it is ensured that the pressure-containing area A_p is smaller than the area corresponding to the effective length l_0 or l_1, the centreline of the gasket can be taken as boundary of the area A_p, whereas the stress area A_f is limited by the calculated length l_0 or l_1.

The following limitation is only for pressure sealed bonnet designs in accordance with Figure 15. In the case where the segmented split ring is arranged within the effective length, l_0 or l_1, the stress area A_f shall be determined only by taking into account the value of l_0 or l_1 to the centreline of the segmented ring. This is to ensure that the radial forces introduced by the gasket and the bending stresses acting at the bottom of the groove are limited.



Key

1 Segmented ring
2 Centreline of gasket

Figure 15 — Example of a closure

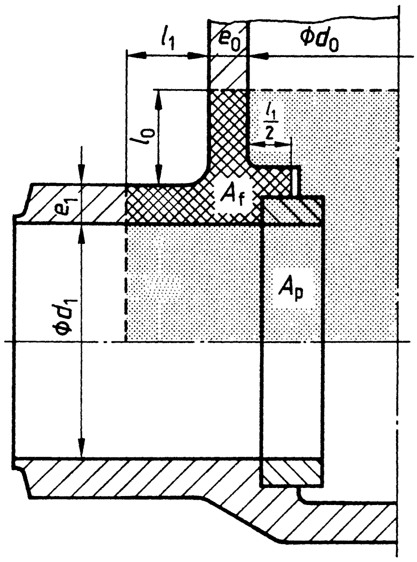


Figure 16 — Example of an end connection

Flanges shall not to be taken into account for the calculation. The chamfer of the end taper shall also not be taken into account.

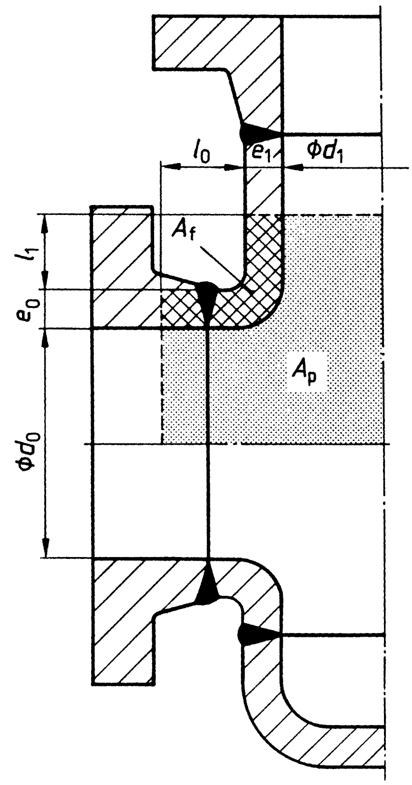


Figure 17 — Example of an end connection

In the case of very short flanged ends, occasionally blind bolt holes can extend into the zone of A_f. In such cases the area of the blind hole shall be deducted (from A_f). This applies to bolt holes within a zone of ± 22,5° of the calculated cross-section viewed from the top (Figure 18).

The influence of the flange load on the valve body has to be taken into account.

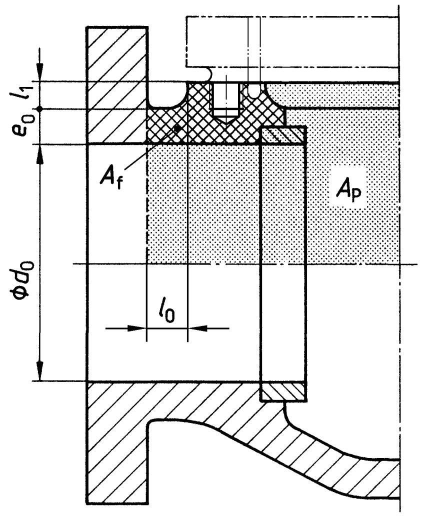


Figure 18a — Calculated cross-section

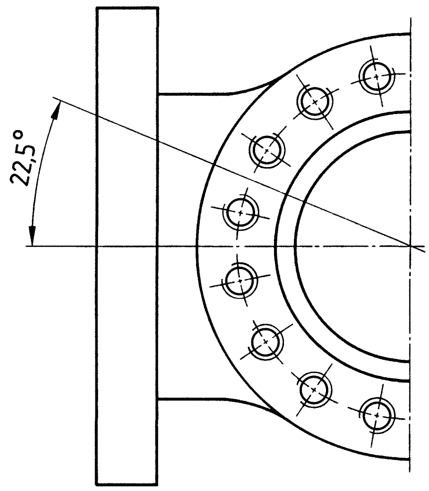


Figure 18b — Top view

Figure 18 — Example of a flanged connection with blind holes

Disc-shaped reinforcements of the basic body shown in Figures 19 and 20 may only be used for calculation temperatures ≤ 250 °C. For the determination of the additional metal cross-sectional area A_{fs}, the effective width b_s may be considered only as a value not exceeding:

$$b_s = n_1 \times \sqrt{(d_0 + e_0) \times e_0} \tag{47}$$

The disc thickness e_s may be considered in the calculation only as a value not exceeding the actual wall thickness of the basic body. The load carrying factor is generally n_1 = 0,7 except for designs with tubular reinforcement and an internal projecting length of the branch according to Figure 20 Design A where n_1 = 0,8 may be used for calculation.

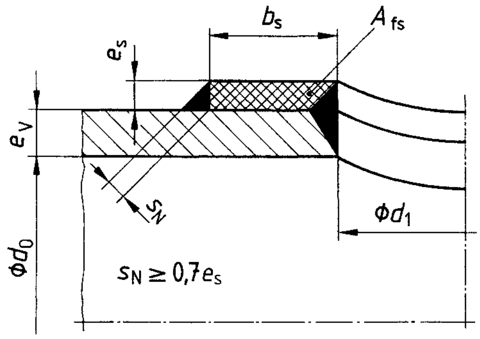


Figure 19 — Example of opening reinforcement

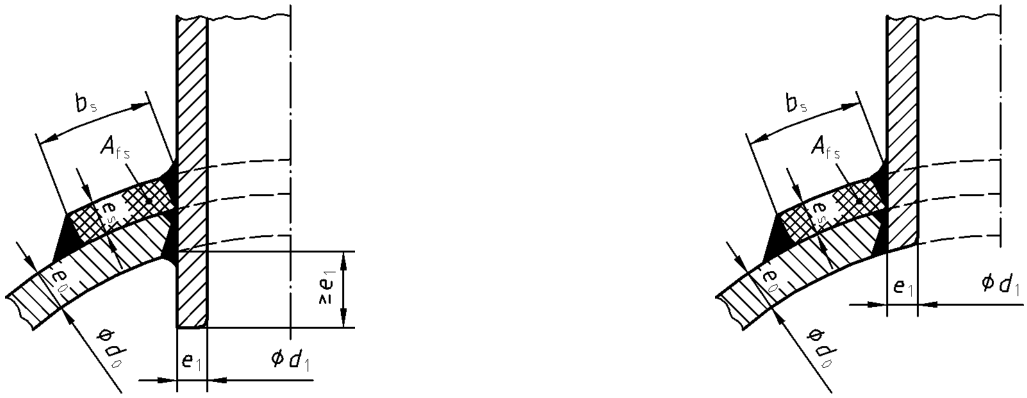


Figure 20 — Example of opening reinforcement

8 Calculation methods for bolted bonnets and covers

8.1 General

Bonnets used as closures of valve bodies are subdivided into three standard bonnets or covers:

— covers made of flat plates;

— covers consisting of a hemispherical shell and an adjoining flanged ring;

— pressure seal bonnets.

8.2 Covers made of flat plates

8.2.1 General

The following two equations apply to plates with a plate thickness/diameter ratio ≤ ¼.

The plate thickness h_c is calculated in accordance with equation (48a):

$$h_c = C_x \times C_y \times C_z \times d_D \times \sqrt{\frac{p}{f}} + c_1 + c_2 \tag{48a}$$

$C_{x,\,y,\,z}$ are calculation coefficients depending on

— different diameter ratios;

— the ratio δ of bolt forces against pressure forces (see Figure 28).

$$\delta = 1 + 4 \times \frac{m \times b_D \times S_D}{d_D} \quad (48b)$$

where

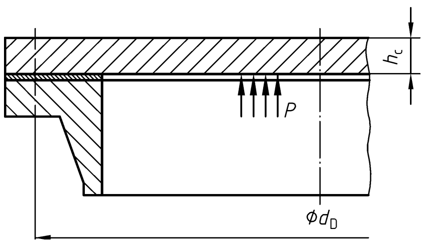
S_D is equal to 1,2 for operating conditions;

m is the gasket coefficient, see annex B.

To find out the coefficients $C_{x,\ y,\ z}$ and the diameter d_D use the Figures 21 to 25 and the indications listed in these figures.

8.2.2 Circular cover without opening, with:

a) full face gasket



C_x 1,0

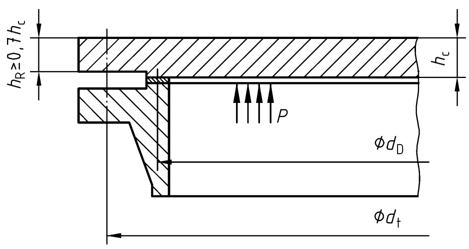
C_y 1,0

C_z 0,35

δ see equation (48b)

Figure 21 — Cover with full face gasket

b) gasket entirely within the bolt circle



C_x 1,0

C_y see Figure 28

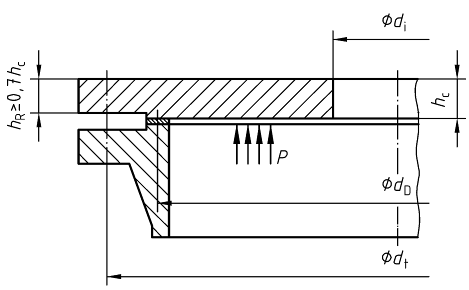
C_z 1,0

δ see equation (48b)

Figure 22 — Cover with gasket entirely within the bolt circle

8.2.3 Circular covers with concentric circular opening

a) gasket entirely within the bolt circle



C_x 1,0

C_y see Figure 28

C_z see Figure 29

δ see equation (48b)

Figure 23 — Cover with gasket entirely within the bolt circle

b) cover with gasket entirely within the bolt and with central nozzle

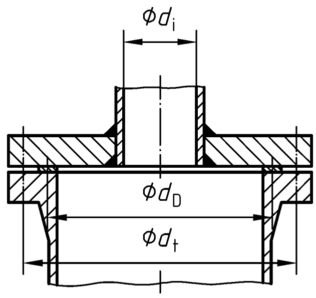


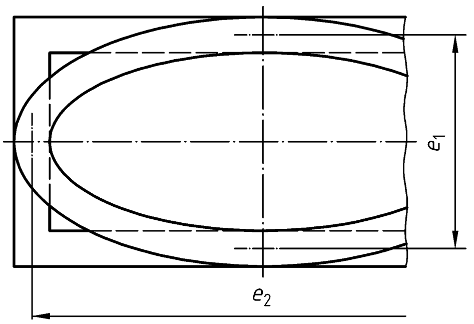
Figure 24 — Cover with central nozzle

8.2.4 Non-circular covers (elliptical or rectangular)

For non-circular covers the plate thickness is also calculated according to the equations (47) and (48).

The calculation coefficients $C_{x,\,y,\,z}$ are the same as used in 8.2.2 and 8.2.3.

The diameter d_D is now to be substituted by the small distance e_1 in the following figure.



C_x see Figure 30

$\left.\begin{array}{l} C_y \\ C_z \end{array}\right\}$ according to 8.2.2 and 8.2.3

Figure 25 — Diameter of non-circular covers

8.2.5 Special covers made of flat circular plates for specific load and clamping conditions

Covers, closures and ends in the form of flat plates are often adopted as external and internal closures of valve bodies. In most cases flat circular plates and flat annular plates are considered such as those illustrated in Table 5. Other shapes of plates (e.g. rectangular or elliptical) represent special cases, which are not part of 8.2.4.

The most used designs are illustrated in Table 5 for various load cases and clamping conditions. The bending moments M_r in radial direction and M_t in tangential directions in correlation to a distance variable x are listed in the table for the individual cases. Also the designations for the maximum moments and their centre points are listed and these are sufficient for checking the strength.

The strength condition is:

$$\frac{6 \times M_i}{h^2} \le 1{,}5 \times f \qquad (49)$$

With M_i equal M_{max}, M_r, M_t calculated in accordance with Table 5 or in accordance with the moment determined from a composite load case.

Superimposed load cases can arise in valves composed of the internal pressure loading and additional forces e.g. gasket force F_g. These load cases can be reduced back to the individual loadings featured in Table 5 and can be determined by summation of the moments. It shall, however, be taken into account that the maximum moments of the individual loadings do not in every case give the maximum total moment. In such cases, the location and magnitude of the maximum shall be determined from the pattern of the moments.

Examples of circular plates with centre holes:

— non-reinforced = type I, with r_o/r_D

— reinforced at the rim = type II, with r_F/r_D

are shown in Table 6. For the calculation coefficients B_P, B_F and B_M see Figure 26.

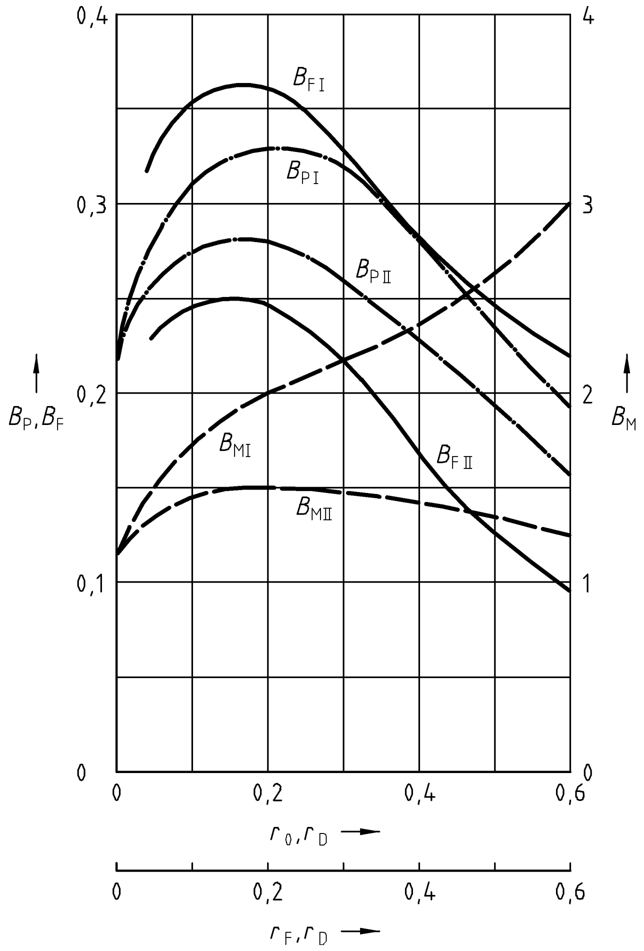


Figure 26 — Calculation coefficients B_P, B_F and B_M

The gasket force F_g is assumed with 25 % of the force resultant from the internal pressure, see Figure 27:

$$F_g = 0{,}25 \times p \times \frac{\pi \times d_i^2}{4} \tag{50}$$

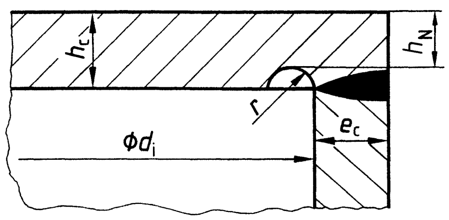


Figure 27 — Flat plate with annular groove

For flat plate the plate thickness of the cover h_c is give by:

$$h_c = 0{,}4 \times d_i \times \sqrt{\frac{p}{f}} \tag{51}$$

with the following conditions:

$$h_N \geq p \times \left(\frac{d_i}{2} - r \right) \times \frac{1{,}3}{f} \text{ and } h_N = \leq 0{,}77 \times e_c \tag{52}$$

h_N shall be not less than 5 mm.

r shall be $\geq 0{,}2 \times h_c$ but not less than 5 mm.

Table 5 — Flat circular plates and annular plates — Bending moments as a function of load cases and clamping conditions

P specific load in N/mm^2 — M_r specific bending moment in radial direction in N mm/mm

F annular force in N — M_t specific bending moment in tangential direction in N mm/mm

h plate thickness in mm — r, r_0, r_1, R (see load cases) in mm

μ Poison's ratio (for steel ≈ 0,3) — Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5f \quad M_i \mid M_r; M_t; M_{max}$

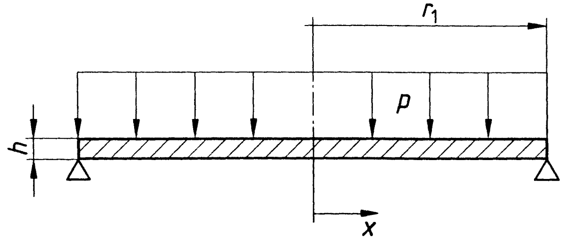
Load case	Load diagram	Specific bending moment
1	Freely supported at the rim. 	$M_r = \frac{P \times r_1^2}{16} \times (3+\mu) \times \left(1 - \frac{x^2}{r_1^2}\right)$ $M_t = \frac{P \times r_1^2}{16} \times \left[(3+\mu) - (1+3\mu) \times \frac{x^2}{r_1^2}\right]$ max. moment for x = 0 (centre of plate) $M_{max} = M_r = M_t = \frac{P \times r_1^2}{16} \times (3+\mu)$

Table 5 — (*continued*)

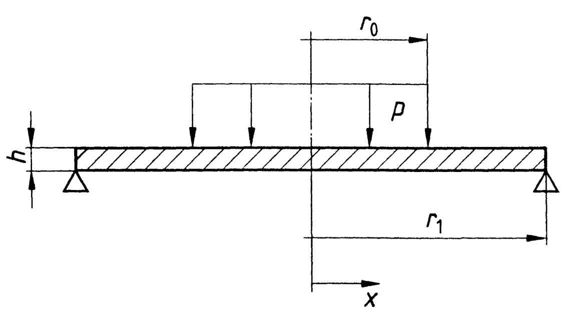
<table>
<tr><th>Load case</th><th>Load diagram</th><th>Specific bending moment</th></tr>
<tr><td>2</td><td>Freely supported at the outer rim.

</td><td>for $x \le r_0$:
$$M_r = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2} - (3+\mu) \times \frac{x^2}{r_0^2}\right]$$
$$M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2} - (1+3\mu) \times \frac{x^2}{r_0^2}\right]$$
for $x > r_0$:
$$M_r = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} + (1-\mu)\left(\frac{r_0^2}{x^2} - \frac{r_0^2}{r_1^2}\right)\right]$$
$$M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} + 4\,(1-\mu) - (1-\mu)\left(\frac{r_0^2}{x^2} + \frac{r_0^2}{r_1^2}\right)\right]$$
max. moment for $x = 0$ (centre of plate)
$$M_{max} = M_r = M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2}\right]$$</td></tr>
</table>

Table 5 — (*continued*)

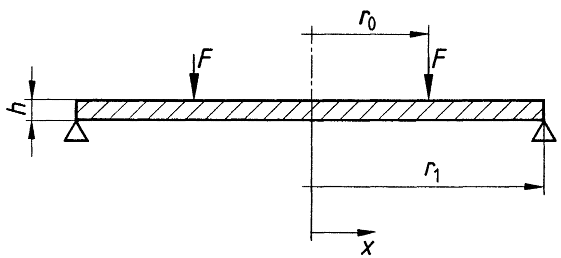
Load case	Load diagram	Specific bending moment
3	Freely supported at the outer rim.  	for $x \le r_0$: $M_r = M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{r_0} + (1-\mu)\left(1-\frac{r_0^2}{r_1^2}\right)\right]$ for $x > r_0$: $M_r = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} + (1-\mu)\left(\frac{r_0^2}{x^2}-\frac{r_0^2}{r_1^2}\right)\right]$ $M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} + 2(1-\mu) - (1-\mu)\left(\frac{r_0^2}{x^2}+\frac{r_0^2}{r_1^2}\right)\right]$ max. moment for $0 \le x \le r_0$ $M_{max} = M_r = M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{r_0} + (1-\mu)\left(1-\frac{r_0^2}{r_1^2}\right)\right]$ max. moment for $x = r_1$ (outer rim) $M_{max} = M_t = \frac{F}{8\pi}\left[2(1-\mu) - (1-\mu)\frac{2\ r_0^2}{r_1^2}\right]$

Table 5 — *(continued)*

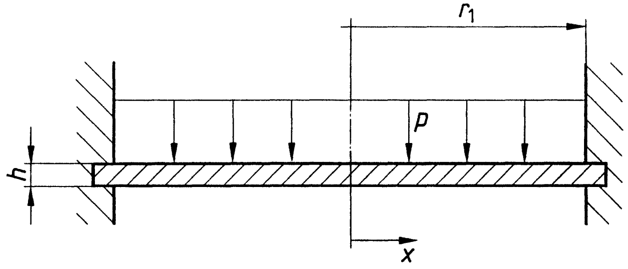
Load case	Load diagram	Specific bending moment
4	Rigidly constrained outer rim. 	$M_r = \frac{P \times r_1^2}{16}\left[(1+\mu) - (3+\mu)\frac{x^2}{r_1^2}\right]$ $M_t = \frac{P \times r_1^2}{16}\left[(1+\mu) - (1+3\mu)\frac{x^2}{r_1^2}\right]$ max. moment for $x = 0$ (centre of plate) $M_{max} = M_r = M_t = \frac{P \times r_1^2}{16}(1+\mu)$ max. moment for $x = r_1$ (outer rim) $M_{max} = M_r = -\frac{P \times r_1^2}{8}$

Table 5 — (*continued*)

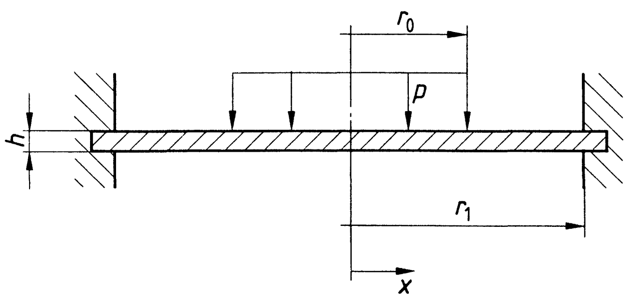
Load case	Load diagram	Specific bending moment
5	Rigidly constrained outer rim. 	for $x \le r_0$: $M_r = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2} - (3+\mu)\frac{x^2}{r_0^2}\right]$ $M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2} - (1+3\mu)\frac{x^2}{r_0^2}\right]$ for $x > r_0$: $M_r = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} - 4 + (1+\mu)\frac{r_0^2}{r_1^2} + (1-\mu)\frac{r_0^2}{x^2}\right]$ $M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} - 4\mu + (1+\mu)\frac{r_0^2}{r_1^2} - (1-\mu)\frac{r_0^2}{x^2}\right]$ max. moment for $x = 0$ (centre of plate) $M_{max} = M_r = M_t = \frac{P \times r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2}\right]$

Table 5 — (*continued*)

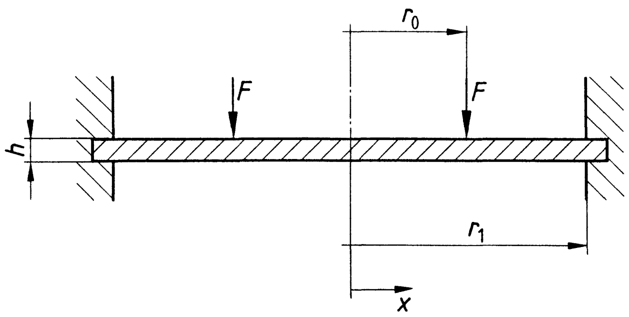
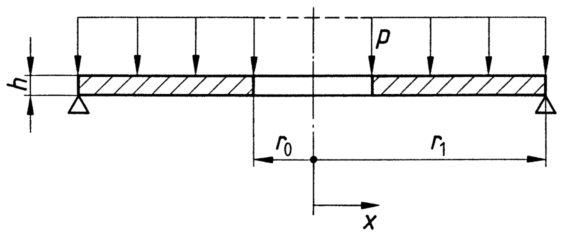
<table>
<tr><th>Load case</th><th>Load diagram</th><th>Specific bending moment</th></tr>
<tr><td>6</td><td>Rigidly constrained outer rim.

</td><td>for $x \le r_0$:
$$M_r = M_t = \frac{F}{8\pi}\left[(1+\mu)\left(2\ln\frac{r_1}{r_0} - 1 + \frac{r_0^2}{r_1^2}\right)\right]$$
for $x > r_0$:
$$M_r = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} - (1+\mu)\left(1-\frac{r_0^2}{r_1^2}\right) - (1-\mu)\left(1-\frac{r_0^2}{x^2}\right)\right]$$
$$M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} - (1+\mu)\left(1-\frac{r_0^2}{r_1^2}\right) + (1-\mu)\left(1-\frac{r_0^2}{x^2}\right)\right]$$
max. moment at the outer rim:
$$M_{max} = M_r = -\frac{F}{4\pi}\left(1-\frac{r_0^2}{r_1^2}\right)$$</td></tr>
<tr><td>7</td><td>Freely supported outer rim.

</td><td>max. moment at the inner rim:
$$M_{max} = M_t = \frac{P}{8(r_1^2 - r_0^2)}\left[r_1^4(3+\mu) + r_0^4(1-\mu) - 4(1+\mu)r_1^2 r_0^2 \ln\frac{r_1}{r_0} - 4\, r_1^2 r_0^2\right]$$</td></tr>
</table>

Table 5 — (*continued*)

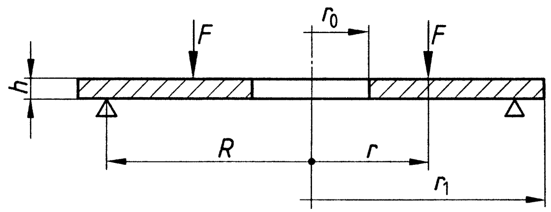
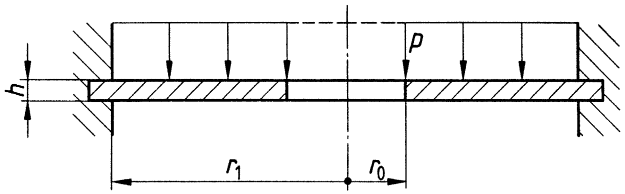
Load case	Load diagram	Specific bending moment
8	Freely supported in the vicinity of the outer rim. Total load F distributed as line load in the vicinity of the inner rim. 	max. moment at the inner rim: $M_{\max} = M_{\mathrm{t}} = \frac{F}{4\pi}\left[\frac{2\ r_1^2(1+\mu)}{r_1^2 - r_0^2}\ln\frac{R}{r} + (1-\mu)\frac{R^2 - r^2}{r_1^2 - r_0^2}\right]$
9	Rigidly constrained outer rim. 	max. moment at the outer rim: $M_{\max} = M_{\mathrm{r}} = \frac{P}{8}\left[r_1^2 - 2r_0^2 + \frac{r_0^4(1-\mu) - 4r_0^4(1+\mu)\ln\frac{r_1}{r_0} + r_1^2 r_0^2(1+\mu)}{r_1^2(1-\mu) + r_0^2(1+\mu)}\right]$ max. moment at the inner rim: $M_{\max} = M_{\mathrm{t}} = \frac{P}{8}\left[\frac{(1-\mu^2)\left(r_1^4 - r_0^4 - 4r_1^2 r_0^2\ln\frac{r_1}{r_0}\right)}{r_1^2(1-\mu) + r_0^2(1+\mu)}\right]$

Table 5 — (*continued*)

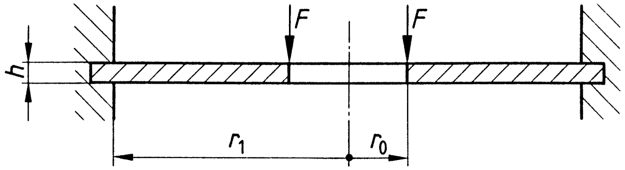
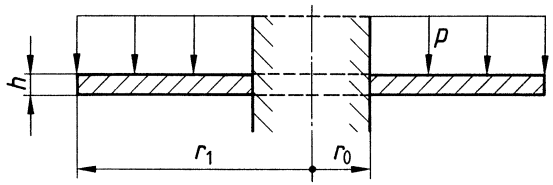
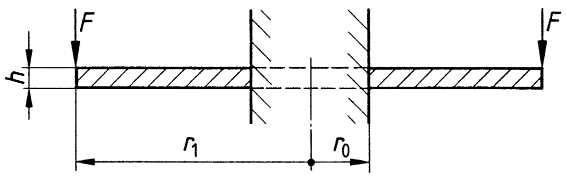
Load case	Load diagram	Specific bending moment
10	Rigidly constrained outer rim. 	max. moment at the outer rim: $M_{\max} = M_{\mathrm{r}} = -\frac{F}{4\pi}\left[\frac{2(1+\mu)r_0^2 \ln\frac{r_1}{r_0} + (1-\mu)(r_1^2 - r_0^2)}{(1-\mu)r_1^2 + (1+\mu)r_0^2}\right]$ max moment at the inner rim: $M_{\max} = M_{\mathrm{t}} = \frac{F}{4\pi}\left[\frac{(1-\mu^2)\left[r_1^2\left(2\ln\frac{r_1}{r_0} - 1\right) + r_0^2\right]}{(1-\mu)r_1^2 + (1+\mu)r_0^2}\right]$
11	Rigidly constrained inner rim. 	max. moment at the inner rim: $M_{\max} = M_{\mathrm{r}} = \frac{P}{8}\left[\frac{4r_1^4(1+\mu)\ln\frac{r_1}{r_0} - r_1^4(1+3\mu) + r_0^4(1-\mu) + 4r_0^2 r_1^2 \mu}{r_1^2(1+\mu) + r_0^2(1-\mu)}\right]$
12	Rigidly constrained inner rim. 	max. moment at the inner rim: $M_{\max} = M_{\mathrm{r}} = \frac{F}{4\pi}\left[\frac{2\ r_1^2(1+\mu)\ln\frac{r_1}{r_0} + r_1^2(1-\mu) - r_0^2(1-\mu)}{r_1^2(1+\mu) + r_0^2(1-\mu)}\right]$

Table 5 — (*concluded*)

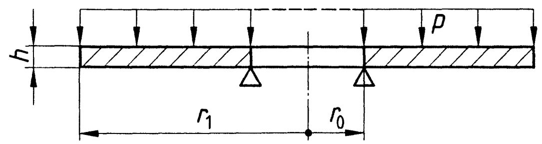
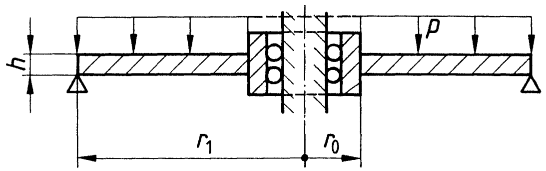
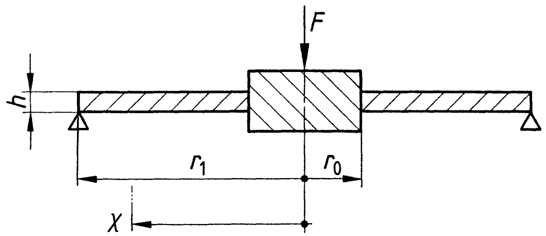
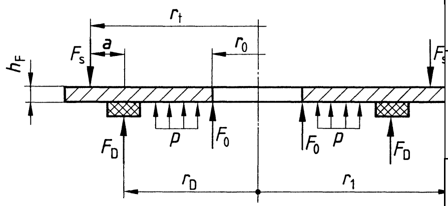
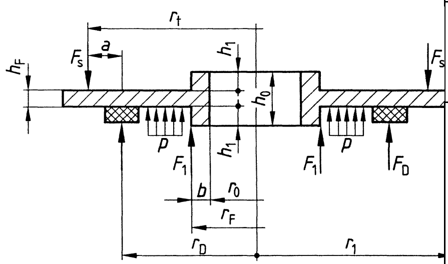
Load case	Load diagram	Specific bending moment
13	Freely supported inner rim. 	max. moment at the inner rim: $M_{max} = M_t = \frac{P}{8\,(r_1^2 - r_0^2)}\left[4\,r_1^4(1+\mu)\ln\frac{r_1}{r_0} + 4\,r_1^2 r_0^2 \mu + r_0^4(1-\mu) - r_1^4(1+3\mu)\right]$
14	Constrained but free to move at the centre, freely supported at the outer rim. 	max. moment at the inner rim: $M_{max} = M_r = \frac{P}{8}\left[\frac{r_1^4(3+\mu) + r_0^4(1-\mu) - 4\,r_1^2 r_0^2\left[1+(1+\mu)\ln\frac{r_1}{r_0}\right]}{r_1^2(1+\mu) + r_0^2(1-\mu)}\right]$
15	Freely supported at the outer rim rigid centre plate.  $r_0 \le x \le r_1$	$M_r = \frac{F}{4\pi}\left\{\frac{(1+\mu)+(1-\mu)r_0^2/x^2}{(1+\mu)+(1-\mu)r_0^2/r_1^2}\left[(1+\mu)\ln\frac{r_1}{r_0}+1\right] - (1+\mu)\ln\frac{x}{r_0} - 1\right\}$ max. moment at the inner rim: $M_{max} = M_r = \frac{F}{4\pi}\left\{\frac{2}{(1+\mu)+(1-\mu)r_0^2/r_1^2}\left[(1+\mu)\ln\frac{r_1}{r_0}+1\right] - 1\right\}$
		max. moment at the outer rim: $M_{max} = M_r = \frac{F(1-\mu^2)}{4\pi}\left[\frac{r_1^2 - r_0^2\left(1+2\ln\frac{r_1}{r_0}\right)}{r_1^2(1+\mu) + r_0^2(1-\mu)}\right]$

Table 6 — Application cases of circular plates with non-reinforced centre hole and with reinforced centre hole

Application case	Load diagram		Specific single moments	Resulting bending moments M_i and point forces F_0, F_1
I	non-reinforced centre hole inner rim free to move 	resulting from internal pressure	$M_{PI} = p\, r_D^2 \left[B_{PI} - 0{,}044 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	
		resulting from a single force (point force)	$M_{FI} = F_0 \left[B_{FI} - 0{,}028 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	$F_0 = \pi\, r_0^2 \times p$
		resulting from a rim moment	$M_{MI} = \frac{F_D\, 2a}{\pi\, r_D} \left[B_{MI} - 0{,}35 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	
				$M_{iI} = M_{PI} + M_{FI} + M_{MI}$
II	centre hole reinforced inner rim rigid $h_1 \le 1{,}25\sqrt{(2\, r_0 + b) \times b}$ 	resulting from internal pressure	$M_{PII} = p\, r_D^2 \left[B_{PII} - 0{,}044 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	
		resulting from a single force (point force)	$M_{FII} = F_1 \left[B_{FII} - 0{,}028 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	$F_1 = \pi\, r_F^2 \times p$
		resulting from a rim moment	$M_{MII} = \frac{F_D\, 2a}{\pi\, r_D} \left[B_{MII} - 0{,}35 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$	
				$M_{iII} = M_{PII} + M_{FII} + M_{MII}$
		The calculation coefficients B_p, B_F and B_M shall be taken from Figure 26.		

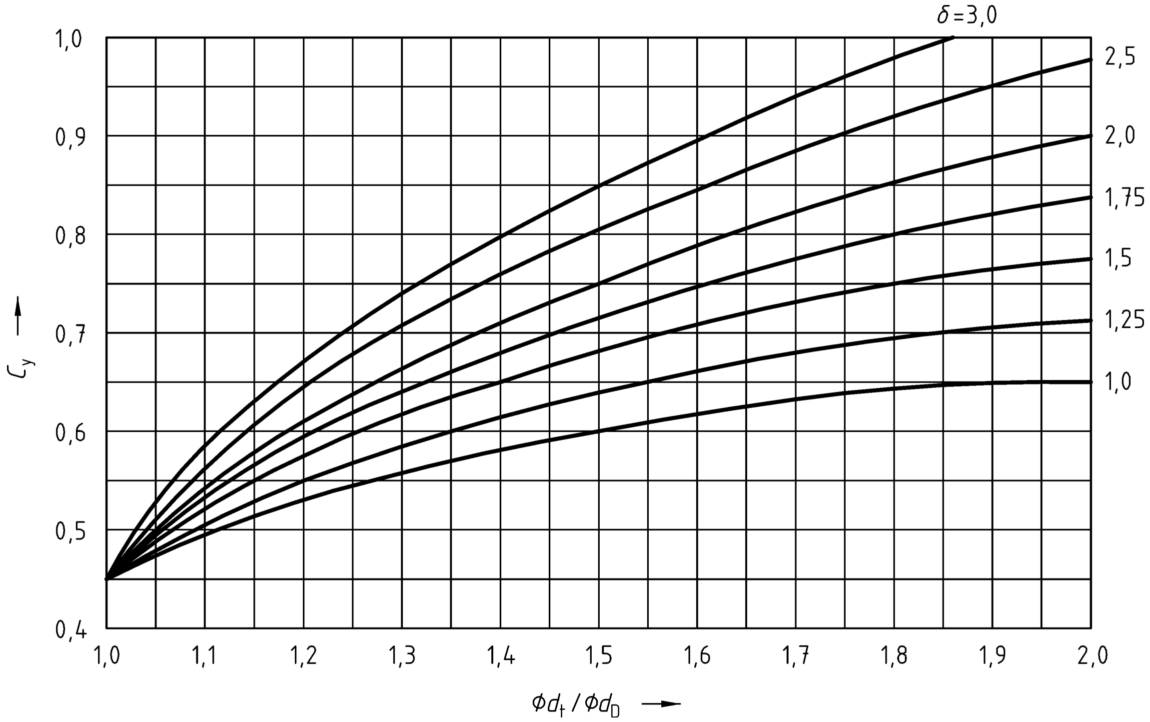


Figure 28 — Calculation coefficient C_y for flat plates with supplementary marginal moment acting in the same sense as the pressure load

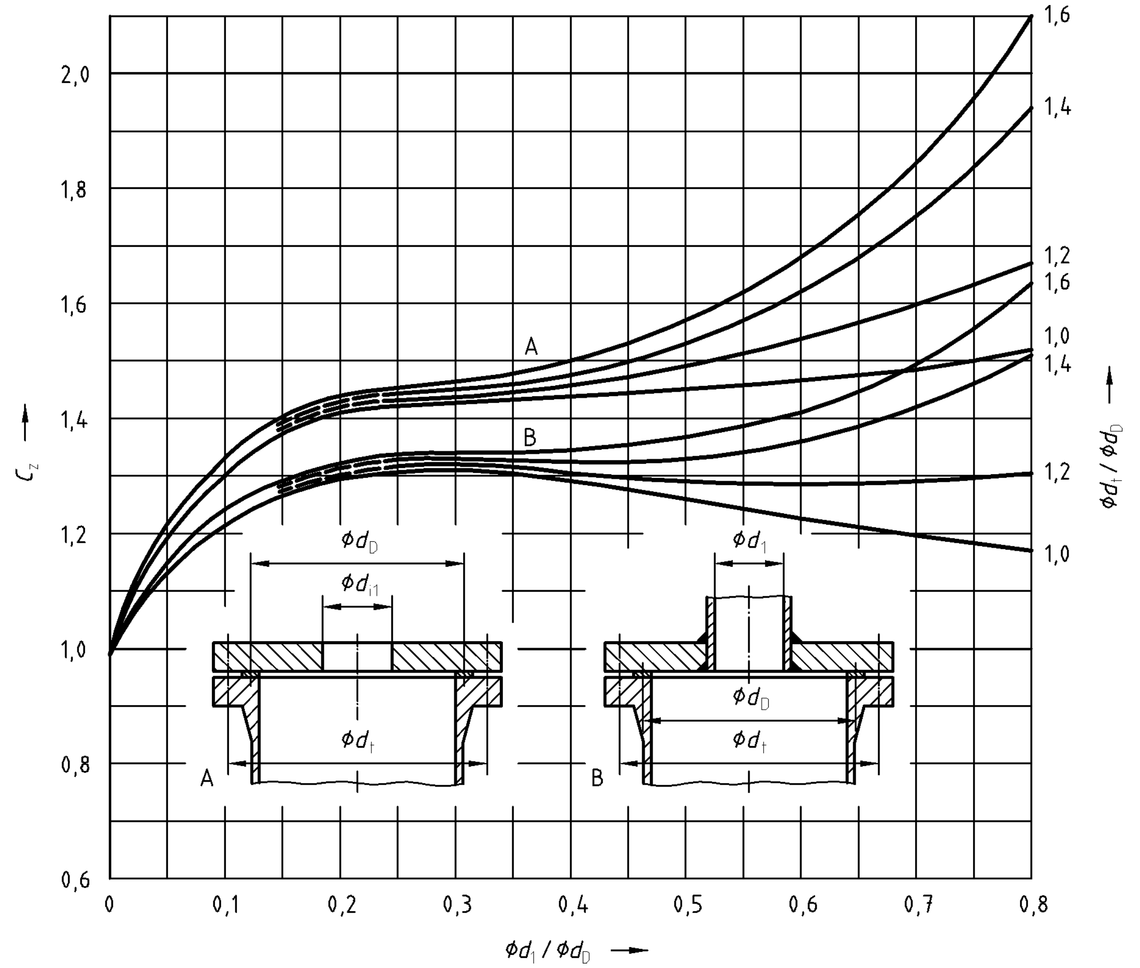


Figure 29 — Opening factor C_z for flat plates with additional marginal moment

Type A

d_1 inside diameter of opening
d_t pitch circle diameter
d_D mean gasket diameter
e_1 short side of an elliptical end

$$C_z = \begin{cases} \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij} \times \left(\dfrac{d_1}{d_D}\right)^{i-1} \times \left(\dfrac{\mathrm{d_t}}{\mathrm{d_D}}\right)^{j-1} & \begin{array}{lcl} 0 & < & \left(\dfrac{d_1}{d_D}\right) \le 0{,}8 \\ 1{,}0 & \le & \left(\dfrac{d_t}{d_D}\right) \le 1{,}6 \end{array} \\ \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij} \times \left(\dfrac{d_1}{e_1}\right)^{i-1} \times \left(\dfrac{d_t}{e_1}\right)^{j-1} & \begin{array}{lcl} 0 & < & \left(\dfrac{d_1}{d_D}\right) \le 0{,}8 \\ 1{,}0 & \le & \left(\dfrac{d_t}{d_D}\right) \le 1{,}6 \end{array} \end{cases}$$

A_{11} = 0,783 610 00;	A_{12} = 0,576 489 80;	A_{13} = – 0,501 335 00;	A_{14} = 0,143 743 30;
A_{21} = – 6,176 575 00;	A_{22} = 25,974 130 00;	A_{23} = – 20,204 770 00;	A_{24} = 5,251 153 00;
A_{31} = 55,155 200 00;	A_{32} = – 187,501 200 00;	A_{33} = 151,229 800 00;	A_{34} = – 40,465 850 00;
A_{41} = – 102,762 800 00;	A_{42} = 385,656 200 00;	A_{43} = – 328,177 400 00;	A_{44} = 92,130 280 00;
A_{51} = 17,634 760 00;	A_{52} = – 218,652 200 00;	A_{53} = 223,865 800 00;	A_{54} = – 71,600 250 00;
A_{61} = 76,137 990 00;	A_{62} = – 99,252 910 00;	A_{63} = 46,208 960 00;	A_{64} = – 3,458 830 00;

Type B

d_1 inside diameter of opening
d_t pitch circle diameter
d_D mean gasket diameter
e_1 short side of an elliptical end

$$C_z = \begin{cases} \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij} \times \left(\dfrac{d_1}{d_D}\right)^{i-1} \times \left(\dfrac{d_t}{d_D}\right)^{j-1} & \begin{array}{lcl} 0 & < & \left(\dfrac{d_1}{d_D}\right) \le 0{,}8 \\ 1{,}0 & \le & \left(\dfrac{d_t}{d_D}\right) \le 1{,}6 \end{array} \\ \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij} \times \left(\dfrac{d_1}{e_1}\right)^{i-1} \times \left(\dfrac{d_t}{e_1}\right)^{j-1} & \begin{array}{lcl} 0 & < & \left(\dfrac{d_1}{d_D}\right) \le 0{,}8 \\ 1{,}0 & \le & \left(\dfrac{d_t}{d_D}\right) \le 1{,}6 \end{array} \end{cases}$$

A_{11} = 1,007 489 00;	A_{12} = – 0,024 092 78;	A_{13} = 0,021 445 46;	A_{14} = – 0,004 895 828;
A_{21} = 3,208 035 00;	A_{22} = – 1,091 489 00;	A_{23} = 1,553 827 00;	A_{24} = – 0,423 889 000;
A_{31} = – 13,191 820 00;	A_{32} = 10,651 000 00;	A_{33} = – 13,276 560 00;	A_{34} = 3,535 713 000;
A_{41} = 30,588 180 00;	A_{42} = – 44,899 680 00;	A_{43} = 47,627 930 00;	A_{44} = – 11,935 440 000;
A_{51} = – 43,361 780 00;	A_{52} = 79,567 940 00;	A_{53} = – 71,673 550 00;	A_{54} = 16,794 650 000;
A_{61} = 42,253 490 00;	A_{62} = – 92,644 660 00;	A_{63} = 74,767 170 00;	A_{64} = – 17,856 930 000;

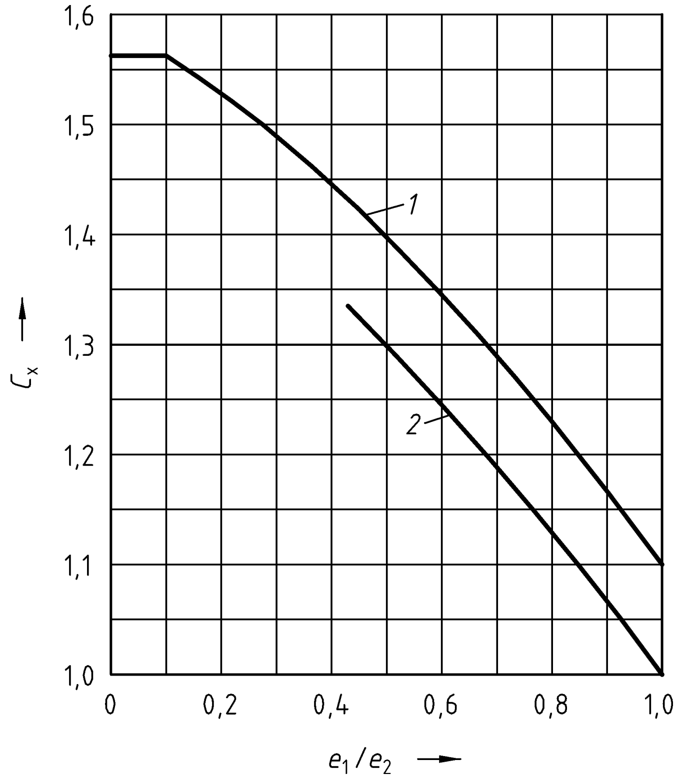


Figure 30 — Calculation coefficient C_x for rectangular (1) or elliptical flat plates (2)

Rectangular plates

e_1 short side of the rectangular plate

e_2 long side of the rectangular plate

$$C_x = \begin{cases} \sum_{i=1}^{4} A_i \times \left(\frac{e_1}{e_2}\right)^{i-1} & \left| 0{,}1 < \left(\frac{e_1}{e_2}\right) \le 1{,}0 \right. \\ 1{,}562 & \left| 0 < \left(\frac{e_1}{e_2}\right) \le 0{,}1 \right. \end{cases}$$

A_1 = + 1,589 146 00

A_2 = – 0,239 349 90

A_3 = – 0,335 179 80

A_4 = + 0,085 211 76

Elliptical plates

e_1 short side of the elliptical plate

e_2 long side of the elliptical plate

$$C_x = \left\{ \sum_{i=1}^{4} A_i \times \left(\frac{e_1}{e_2}\right)^{i-1} \; \left| 0{,}43 \le \left(\frac{e_1}{e_2}\right) \le 1{,}0 \right. \right\}$$

A_1 = + 1,489 146 00

A_2 = – 0,239 349 90

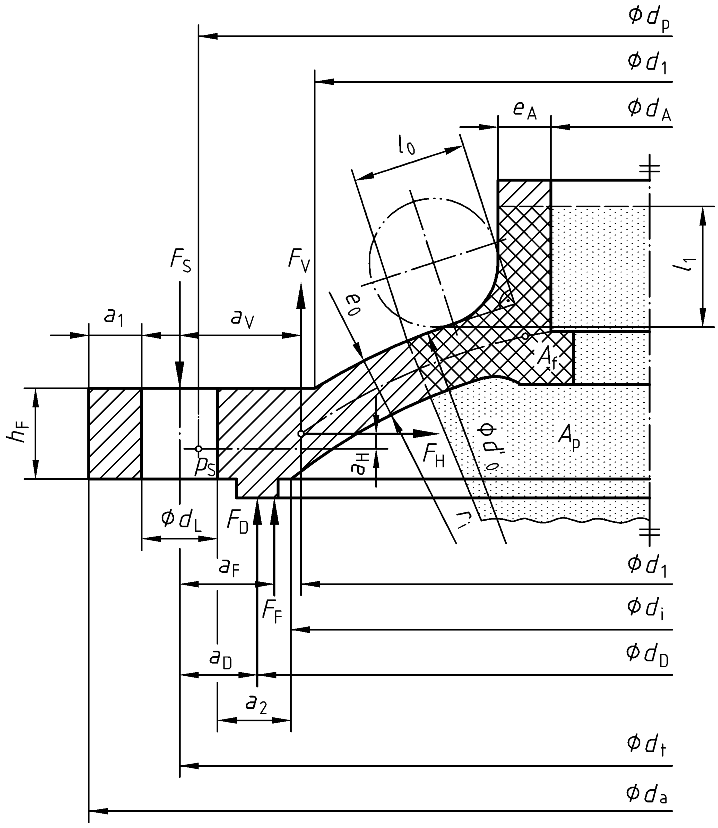
A_3 = – 0,335 179 80

A_4 = + 0,085 211 76

8.3 Covers consisting of a spherically domed end and an adjoining flanged ring

8.3.1 General

The strength calculation consists of the strength calculation of the flanged ring and the strength calculation of the spherically domed end. Depending on the geometric relationships a distinction is made between two types:

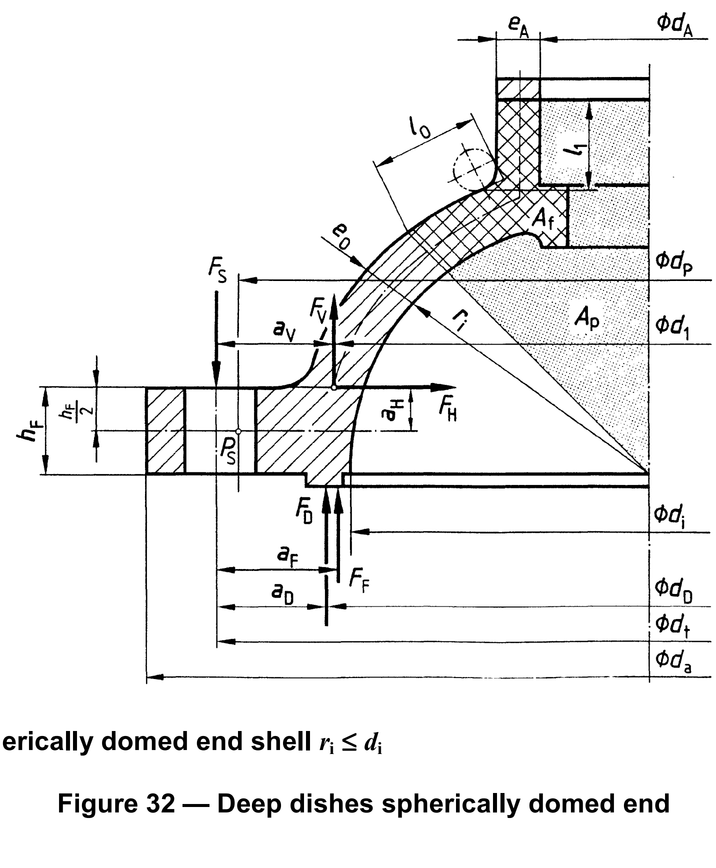


Ø d_1 diameter of intersection flange surface/spherical segment

Type I: Spherically domed end $r_i > d_i$

Detail Type I

Figure 31 — Spherically domed end



Type II: Deep dishes spherically domed end shell $r_i \le d_i$

Figure 32 — Deep dishes spherically domed end

8.3.2 Wall thickness and strength calculation of the spherical segment

The wall thickness e_C, excluding allowances, is calculated from:

— for the ratio $(r_i + e_O) / r_i \leq 1{,}2$

$$e_c = \frac{r_i p}{(2f - p) \times k_c} \tag{53}$$

— for the ratio $1{,}2 < (r_1 + e_O) / r_1 \leq 1{,}5$

$$e_c = r_i \left[\sqrt{1 + \frac{2p}{(2f - p) \times k_c}} - 1 \right] \tag{54}$$

At the transition zone between the flange and the spherical segment the wall thickness is:

$$e'_c = e_c \times \beta \tag{55}$$

β takes into account the fact that due to large percentage of bending stresses there is an increase of the load bearing capacity. Starting from the proof stress ratio δ_1 of dished heads which characterizes the load bearing capacity we enter with $\beta = 3{,}5$. This is for flanges with internal gaskets according to Figures 31 and 32 is and an approximation for $\beta = \frac{\alpha}{\delta_1}$ in Figure 33.

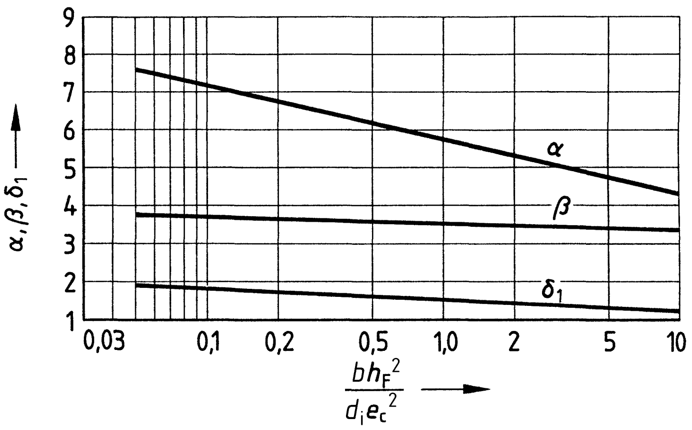


Figure 33 — Calculation coefficient

e'_c not to be thicker than it would be as a result of the calculation of flat plate cover in accordance with 8.3.

8.3.3 Calculation of the flanged ring

8.3.3.1 The strength condition is as follows:

$$\frac{F_H}{2 \times \pi \times b \times h_F} \leq f \tag{56}$$

$$\frac{M_a}{2\pi\left[\frac{b}{6} \times h_F^2 + \frac{d_1}{12} \times (e'^2_c - e_0^2)\right]} + \frac{F_H}{2 \times \pi \times b \times h_F} \leq 1{,}5 \times f \tag{57}$$

The moments rotating in a clockwise direction shall be entered with minus sign in the equations.

Also the strength condition equations shall be calculated with the two moments M_{aB} and $M_{aO} \rightarrow e_C = 0$ for the assembly conditions.

The moments, forces, lever arms and other geometrical dimensions of the strength condition equations are written in 8.3.3.2 to 8.3.3.5.

8.3.3.2 Forces and moments of equation (56) and (57)

The horizontal component of end force:

$$F_H = p \times \frac{\pi}{2} \times d_i \times \sqrt{r_i^2 - \frac{d_i^2}{4}} \tag{58}$$

For operation conditions:

$$M_{aB} = F_V \times a_V + F_F \times a_F + F_{DB} \times a_D + F_H \times a_H \tag{59}$$

For assembly conditions:

$$M_{aO} = F_{SO} \times a_D \tag{60}$$

8.3.3.3 Forces in the moment equations (59) and (60)

$$F_V = p \times \frac{\pi}{4} d_i^2 \tag{61}$$

$$F_F = p \times \frac{\pi}{4} (d_D^2 - d_i^2) \tag{62}$$

$$F_{DB} = p \times \pi \times d_D \times m \times b_D \times S_D ; \; S_D = 1{,}2 \tag{63}$$

m see Table B.1

F_H see equation (58)

F_S, the bolt force for operation conditions is:

$$F_S = F_V + F_F + F_{DB} \tag{64}$$

F_{SO}, the bolt force for assembly condition is the larger value of the following:

$F_S \times K$ with

K = 1,1 general

K = 1,2 for soft gaskets

or

$$F_{DV} = \pi \times d_D \times \sigma_{vu} \times b_D \tag{65}$$

σ_{vu} see Table B.1

8.3.3.4 Lever arms in the moment equation (59 and 60)

Table 7 — Lever arms of the forces in the moment equations

Lever arm	Bonnet	
	Type I	Type II
a_V =	$0{,}5 \times (d_t - d_1)$	
a_D =	$0{,}5 \times (d_t - d_D)$	
a_H =	to be determined diagrammatically	$0{,}5 \times h_F$
a_F =	$a_D + 0{,}25 \times (d_D - d_i)$	

8.3.3.5 Other geometrical dimensions in the equations (56 and 57)

b – the load bearing width of the flange

$$b = 0{,}5 \times (d_a - d_i - 2d'_L) \tag{66}$$

with

$d'_L = V \times d_L \rightarrow d_i \geq 500 \rightarrow V = 0{,}5$

$d_i < 500 \rightarrow V = 1 - 0{,}001 \times d_i$

8.3.4 Reinforcement of the stuffing box area

The calculation is in accordance with the method for the calculation of crotch areas, see 7.2.2, i.e. the comparison of pressure loaded areas A_p with metal cross-sectional areas A_f .

$$p \times \left[\frac{A_p}{A_f} + \frac{1}{2} \right] \leq f \tag{67}$$

The stuffing box area is limited by the effective lengths:

$$l_0 = \sqrt{(2 \times r + e_0) \times e_0} \tag{68}$$

$$l_1 = \sqrt{(d_A + e_A) \times e_A} \tag{69}$$

For the areas A_p, A_f, the length l_0 and l_1 — see Figures 31 and 32.

8.4 Dished heads

8.4.1 General remarks

The calculation rules shall apply to dished heads consisting of a spherical shell, a knuckle and a cylindrical rim as solid dished heads (see Figure 34), and to dished heads with cut-outs (see Figure 36) or with branches (see Figure 37).

As a general rule, the following conditions shall be applicable to dished heads with a bottom diameter d_o:

Inner radius of spherical cap $R_i \leq d_o$

knuckle radius $r \geq 0{,}1 \times d_o$

Related wall thickness $0{,}001 \leq e_c / d_o \leq 0{,}10$

In particular, the following values shall apply:

to torispherical heads $R_i = d_o$; $r = 0{,}1 \times d_o$, $h_w = 0{,}1935 \times d_o - 0{,}455 \times e_K$

to semi-ellipsoidal heads $R_i = 0{,}8 \times d_o$; $r = 0{,}154 \times d'_o$, $h_w = 0{,}255 \times d_o - 0{,}635 \times e_K$

to hemispherical heads $d_o / d_i \leq 1{,}2$

In cases where dished heads are fabricated from a spherical cap welded to a knuckle portion, the welded joint shall be situated at a sufficiently great distance from the knuckle. Such a distance is given by the following relationships:

a) if the wall thickness of the knuckle portion and the spherical cap are different:

$$x = 0{,}5 \times \sqrt{R_i \times e_{cK}} \tag{70}$$

b) if the wall thicknesses of the knuckle portion equals the spherical cap:

for torispherical heads $x = 3{,}5 \times e_c$

for semi-ellipsoidal heads $x = 3{,}0 \times e_c$

but x shall be equal to 100 mm at least.

For welded heads, k_c can be entered at $k_c = 1{,}0$ if the weld intersects the apex zone at $0{,}6 \times d_o$. In other cases $k_c = 0{,}85$.

8.4.2 Solid dished heads

The required wall thickness without allowances in the spherical cap shall be obtained from equations (71) to (77).

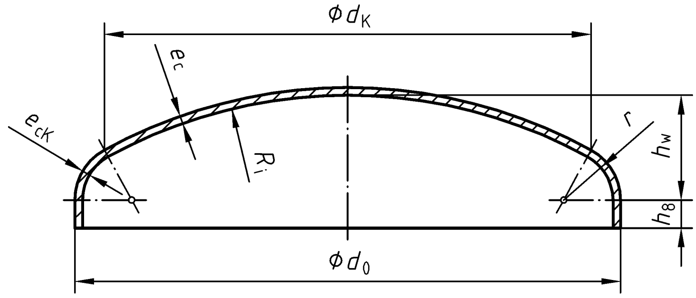


Figure 34 — Solid dished head

For hemispherical heads, the wall thickness determined in accordance with equations (71) to (77) shall be multiplied by a factor of 1,1 in the zone of the welded joint.

The required wall thickness without allowances in the knuckle zone shall be:

$$e_{cK} = \frac{p \times d_o \times \beta}{4 \times f \times k_c} \tag{71}$$

The calculation coefficient β can be obtained from Figure 35 as a function of e_c / d_o for torispherical heads (1) and semi-ellipsoidal (2) heads.

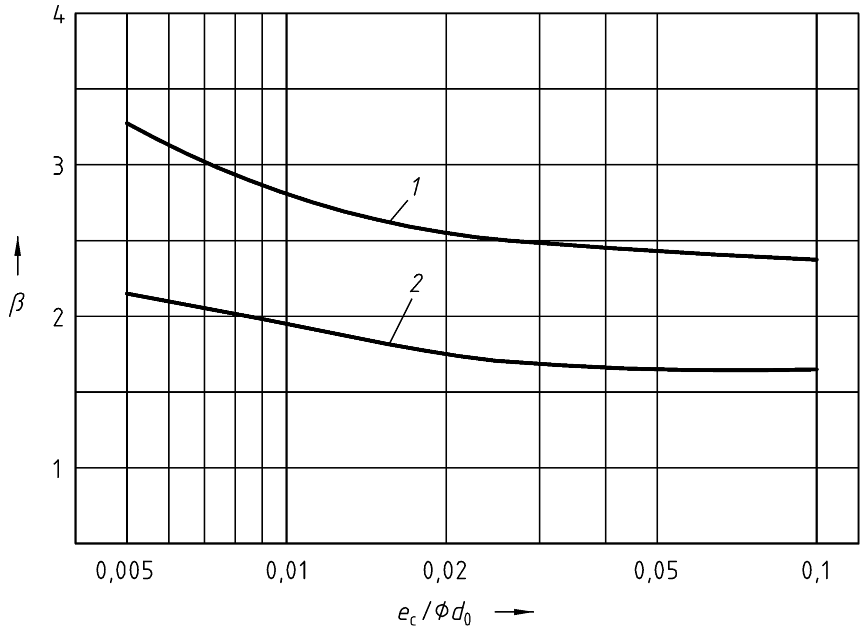


Figure 35 — Calculation coefficient β

8.4.3 Dished heads with opening

In the cases of dished heads with opening subjected to internal pressure, the highest stress may occur either in the knuckle or in the zone of the opening, depending on the circumstances in each case, and consequently the calculation shall be carried out for both of these locations.

In the case of opening in the apex zone $0{,}6 \times d_0$ of torispherical heads and semi-ellipsoidal heads, and in the entire zone of hemispherical heads, the weakening of the basic body can be countered by the following measures.

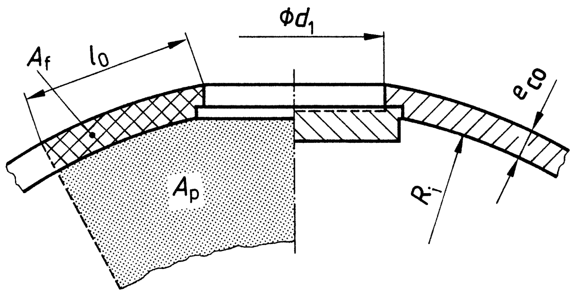


Figure 36 — Dished head with opening

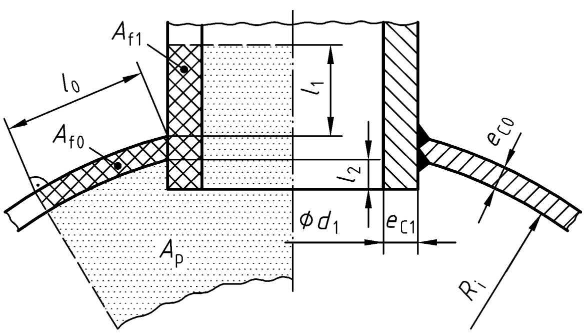


Figure 37 — Dished head with branch (welded-in tubular reinforcement)

a) by means of an increased wall thickness as compared with the thickness of the non-weakened bottom.

This increased thickness shall extend at least as far as the length:

$$l_0 = \sqrt{(2R_i + e_{c0}) \times e_{c0}} \text{ (see Figure 36)} \tag{72}$$

b) by means of tubular reinforcements, either without, or combined with an increase in the wall thickness of the basic body.

If a portion of the branch protrudes inwards, only a portion of the length:

$$l_2 \leq 0{,}5\sqrt{(d_1 + e_{c1}) \times e_{c1}} \tag{73}$$

can be included as load bearing in the calculation.

If $e_{c1} > e_{c0}$, it shall be calculated with $e_{c1} = e_{c0}$.

c) by means of neckings in conjunction with an increase in the wall thickness of the basic body.

If the areas A_p subjected to pressure and the effective cross-sectional areas A_f are determined in this case in the same way as in the case of tubular reinforcements, i.e. without taking the necking radii and the losses of cross-section into consideration, then the value $A_f'' = 0{,}9 \times A_f$ shall be entered in the calculation for A_f (see Figure 38).

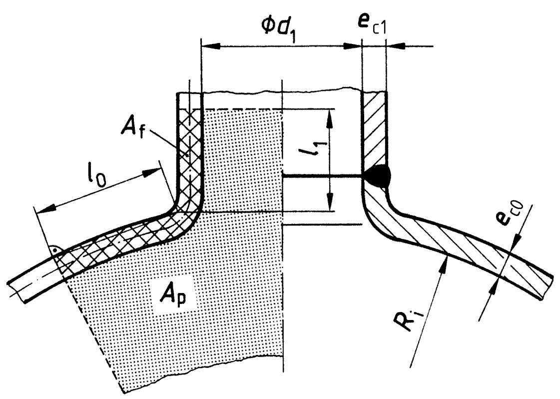


Figure 38 — Dished head with necked opening

d) by means of disc-shaped reinforcements (see Figure 39) for temperatures used for calculation ≤ 250 °C.

These reinforcements shall be designated to fit closely the basic body. Their effective length b_s shall not be entered at a value exceeding l_0. Their thickness e_s shall not exceed the actual wall thickness e_{c0} of the basic body.

Reinforcement of the opening by means of discs welded on the inside is not permitted.

Disc shaped reinforcements shall be taken into account in the calculation with a valuation factor k_s = 0,7.

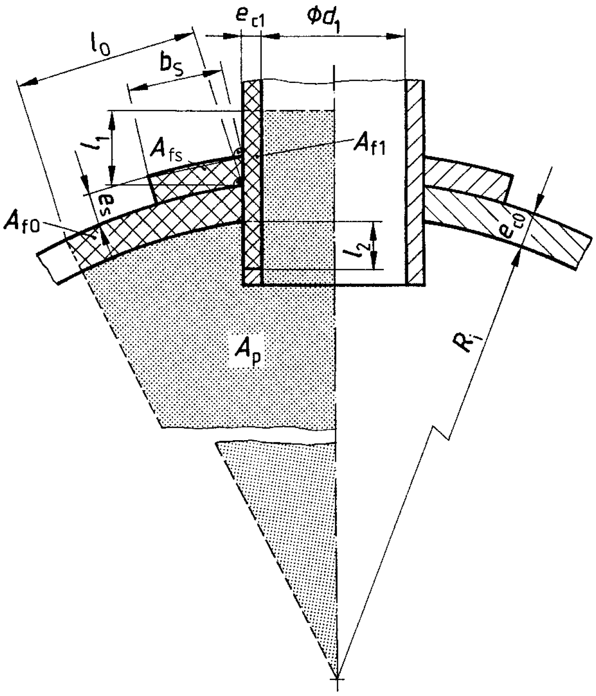


Figure 39 — Dished head with disc-shaped reinforcement

The strength condition for cut-outs in the apex zone $0{,}6 \times d_o$ is given by the equation (74), with the area A_p subjected to pressure, and with the effective cross-sectional areas A_{f0}, A_{f1} and A_{fS}:

$$p \times \left(\frac{A_p}{A_{f0} + A_{f1} + k_s \times A_{fS}} + \frac{1}{2} \right) \leq f \tag{74}$$

The effective lengths for the spherical cap shall not be entered at a value exceeding:

$$l_0 = \sqrt{(2R_i + e_{c0}) \times e_{c0}} \tag{75}$$

and the lengths for the branch shall not exceed:

$$l_1 = \sqrt{(d_1 + e_{c1}) \times e_{c1}} \tag{76}$$

$$l_2 \leq \frac{l_1}{2} \tag{77}$$

8.4.4 Allowances on the wall thickness

In addition to the remarks in clause 4, any reductions in wall thickness due to manufacturing reasons (e.g. in the case of cast or deep drawn dished heads) shall be taken into consideration for the determination of the allowance c_1.

9 Calculation method for pressure sealed bonnets and covers

The object of the strength calculation is to investigate the weakest cross-section (section I-I or II-II of Figure 40). At the same time, the most important main dimensions of the closure shall be calculated in accordance with elemental procedures.

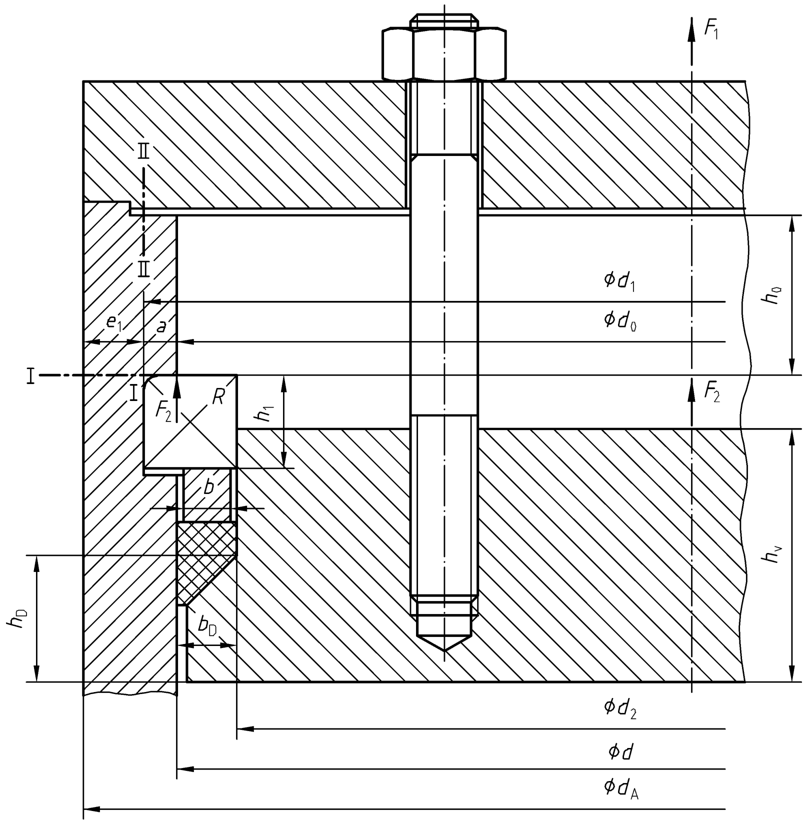


Figure 40 — Self-sealing closure

The axial force uniformly distributed across the circumference shall be calculated from:

$$F_B = p \times \frac{\pi}{4} d^2 \tag{78}$$

F_1 and F_2 are axial forces, which may occur due to actuating processes or pre-tightening of bolts.

The minimum widths of the pressure faces at the seating face and at the distance ring shall be given by the equation (79), taking the friction conditions and the gasket requirement into account:

$$a = \frac{F_B + F_2}{\pi \times d_a \times 1{,}5 \times f} \qquad d_a = (d_1 + d_0) / 2$$

$$b = \frac{F_B + F_2}{\pi \times d_b \times 1{,}5 \times f} \qquad d_b = (d + d_2) / 2 \tag{79}$$

where f is the nominal design stress of the materials in question.

The minimum height h_1 of the inserted ring R can be obtained from calculations with respect of shearing off and bending. The greater of the two values obtained from these calculations shall be adopted.

For shear:

$$h_1 = \frac{2(F_B + F_2)}{\pi \times d \times f} \tag{80}$$

For bending:

$$h_1 = 1{,}38 \times \sqrt{\frac{(F_B + F_2) \times (d_1 - d_2)}{4 \times d \times f}} \tag{81}$$

The minimum height h_0 for the seating shoulder (cross-section II-II) can be obtained from the calculation in respect of shearing off and of bending. The greater of the two values obtained from these calculations shall be adopted.

For shear:

$$h_0 = \frac{2(F_B + F_2)}{\pi \times d_1 \times f} \tag{82}$$

For bending:

$$h_0 = 1{,}13 \times \sqrt{\frac{(F_B + F_2) \times a}{d_1 \times f}} \quad \text{with} \quad a = \frac{d_1 - d_0}{2} \tag{83}$$

The minimum depth of the sealing ledge can be obtained from the calculation with respect to shearing off and bending. The greater of the two values obtained from these calculations shall be adopted.

For shear:

$$h_D = \frac{2(F_B + F_2)}{\pi \times d_2 \times f} \tag{84}$$

For bending:

$$h_D = 1{,}13 \times \sqrt{\frac{(F_B + F_2) \times b_D}{2 \times d_2 \times f}} \tag{85}$$

For flat designs, the closure cover shall be verified according to the equations for flat circular or annular plates.

Strength condition for cross-section I-I:

$$(F_B + F_2)\left(a + \frac{e_1}{2}\right) \le \frac{\pi}{4}\left[h_0^2 (d_A - d_1) + (d_A - e_1)\left(e_1^2 - e_2^2\right)\right] \times f \tag{86}$$

where $e_2 = \dfrac{F_B + F_1}{\pi \times (d_A - e_1) \times f}$ (87)

10 Calculation methods for flanges

10.1 General

The calculation of the flanges shall be carried out in accordance with, or on the lines of, the specifications laid down in EN 1591-1: or in the EN 13445-3. The calculation can however also be carried out in accordance with the equations featured below, which are solved with respect to the thickness of the flange plate h_F as a simplification.

For piping flanges in accordance with EN 1092-1, up to DN 600 (included) a check calculation will not be required on condition that the permissible pressures, temperatures and materials to be used for the flanges, bolts, and gaskets are in accordance with the flange standard.

The flanged joint shall be designed in such a way that it can absorb the forces which arise during assembly (initial deformation of the gasket) and during operation.

10.2 Circular flanges

10.2.1 The decisive factor for the flange design is the maximum flange resistance W required, resulting from equations (88) and (89).

$$W = \frac{F_{SB} \times a}{1{,}5 \times f} \tag{88}$$

$$W = \frac{F_{SO} \times a_D}{1{,}5 \times f} \tag{89}$$

The minimum bolt force F_{SB} for the operating condition is obtained from the pipe force F_p resulting from internal pressure and the gasket force during operation F_{DB}:

$$F_{SB} = F_p + F_{DB} = \frac{\pi d_D^2}{4} \times p + p \times \pi \times d_D \times m \times b_D \times S_D \tag{90}$$

S_D = 1,2 for non-metallic gaskets and metallic envelope gaskets;

S_D = 1,0 for metallic gaskets.

The minimum bolt force F_{DV} for the assembly condition results from:

$$F_{DV} = \pi \times d_D \times \sigma_{VU} \times b_D \tag{91}$$

Characteristic values for the gaskets, m and σ_{VU} are given in Table B.1 as a function of gasket shapes and of the condition of the medium.

Additional forces shall be taken into account by adding them to F_{SB} and F_{DV}.

The minimum bolt force F_{SO} for the assembly condition results from:

$$F_{SO} = \max(\chi \times F_{SB}; F_{DV}) \tag{92}$$

with

χ = 1,1 for general cases;

χ = 1,2 for non-metallic gaskets and metallic envelope gaskets.

10.2.2 Flanges with tapered neck

10.2.2.1 Flanges with tapered neck according to Figure 41 shall be subjected to a check calculation in the cross-sections I-I and II-II.

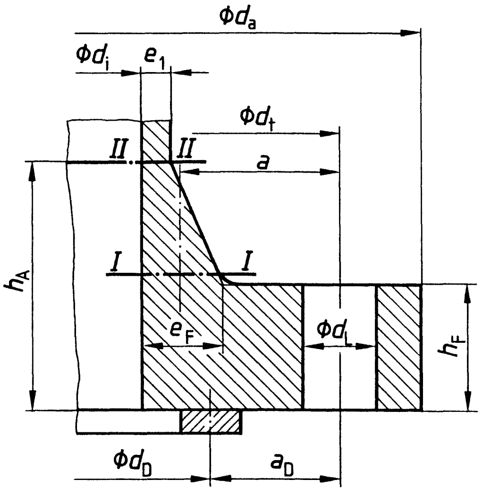


Figure 41 — Flange with tapered neck

10.2.2.2 Cross-section I-I

The required thickness of the flange h_F results from:

$$h_F = \sqrt{\frac{1{,}91 \times W - Z}{b}} \tag{93}$$

The lever of the bolt force in equations (88) and (89) for the operating condition result from:

$$a = \frac{d_t - d_i - e_F}{2} \tag{94}$$

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \tag{95}$$

The calculated double flange width b results from:

$$b = d_a - d_i - 2\,d'_L \tag{96}$$

The reduced bolt hole diameter d'_L

$$d'_L = v \times d_L \tag{97}$$

with $v = 1 - 0{,}001 \times d_i$ for $d_i \leq 500$ mm

and $v = 0{,}5$ for $d_i > 500$ mm

The coefficient Z results from:

$$Z = (d_i + e_F) \times e_F^2 \tag{98}$$

The thickness of the flange neck e_F shall not be entered in equations (94) and (98) at any value exceeding $^1/_3\, h_F$.

10.2.2.3 Cross-section II-II

The following equations apply within the limits:

$$0{,}5 \leq \frac{h_A - h_F}{h_F} \leq 1 \text{ and} \tag{99}$$

$$0{,}1 \leq \frac{e_1 + e_F}{b} \leq 0{,}3 \tag{100}$$

All other cases shall be calculated in accordance with EN 1591-1 or EN 13445-3.

The required flange thickness h_F results from:

$$h_F = B \times \sqrt{\frac{1{,}91 \times W - Z_1}{b}} \tag{101}$$

The coefficient Z_1 results from:

$$Z_1 = \frac{3}{4}(d_i + e_1)\,e_1^2 \tag{102}$$

The calculation coefficient B results from:

$$B = \frac{1 + \frac{2\,e_m}{b} \times B_h}{1 + \frac{2\,e_m}{b}(B_h^2 + 2\,B_h)} \tag{103}$$

where $e_m = \frac{e_F + e_1}{2}$ (104)

and $B_h = \frac{h_A - h_F}{h_F}$ (105)

The lever of the bolt force for the operating condition result from:

$$a = \frac{d_t - d_i - e_1}{2} \tag{106}$$

and for the assembly condition a_D from equation (95).

10.2.3 Flanges greater than DN 1 000

If the neck depth $h_A - h_F$ is at least $0{,}6 \times h_F$ the neck thickness $e_F - e_1$ is at least $0{,}25\ h_F$ equations (107) and (108) below may be used for cross-sections I-I and II-II in lieu of equations (93) and (101); these will result in smaller dimensions:

Cross-section I-I:

$$h_F = \sqrt{\frac{1{,}59W - 0{,}8Z}{b} + \left(\frac{0{,}05Z}{b \times e_F}\right)^2} - \frac{0{,}05Z}{b \times e_F} \tag{107}$$

with Z in accordance with equation (98).

Cross-section II-II:

$$h_F = B \times \sqrt{\frac{1{,}59W - 2Z_1}{b}} \tag{108}$$

with Z_1 in accordance with equation (102).

10.2.4 Welding neck with tapered neck according to Figure 42

The calculation shall be carried out in accordance with equations (93) to (108), using the value for d_a instead of d_t and $d'_L = 0$

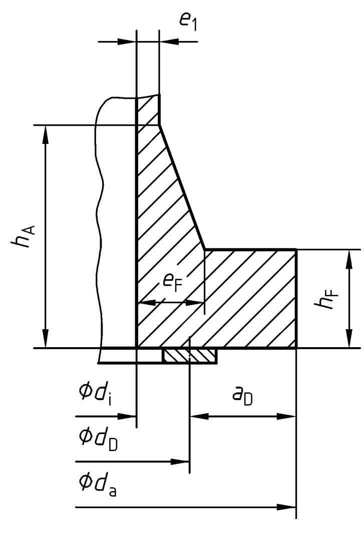


Figure 42 — Welding neck with tapered neck

10.2.5 Weld-on flanges

10.2.5.1 Weld-on flanges in accordance with Figure 43, design a) and design b), and integral flange in accordance with Figure 44.

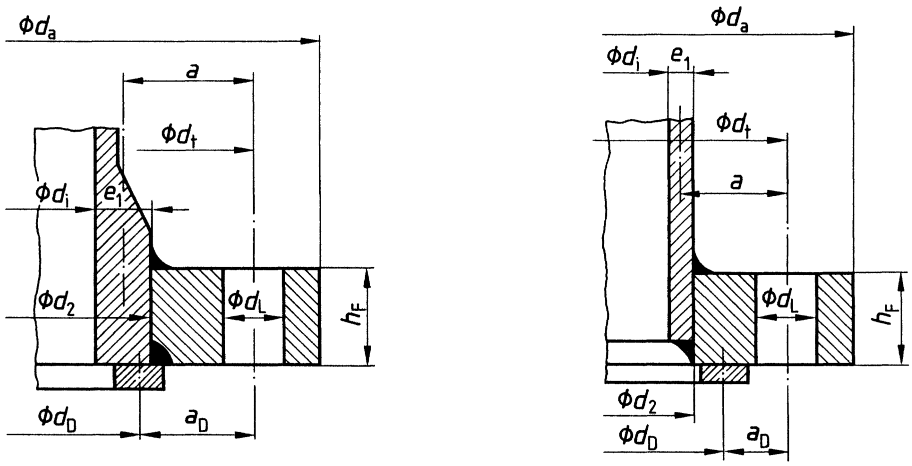


Figure 43 — Weld-on flanges

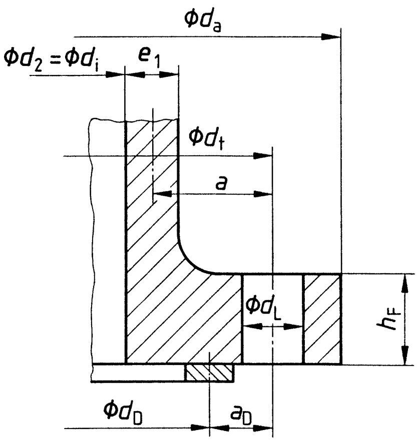


Figure 44 — Integral flange

The required flange thickness h_F is:

$$h_F = \sqrt{\frac{2{,}13W - Z}{b}} \tag{109}$$

The calculated double flange width b results from:

$$b = d_a - d_2 - 2d'_L \tag{110}$$

with d'_L in accordance with equation (97).

In lieu of d_2, d_i can be entered in the above equation, if the welds correspond to types 4 or 5 of Table 8. The same shall apply to integrally cast or integrally forged integral flanges in accordance with Figure 44.

The coefficient Z results from:

$$Z = (d_1 + e_1) \times e_1^2 \tag{111}$$

where e_1 shall not be entered at any value exceeding ½ × h_F.

The lever of the bolt force for the operating conditions result from:

$$a = \frac{d_t - d_i - e_1}{2} \tag{112}$$

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \tag{113}$$

10.2.5.2 Welded-on collars in accordance with Figure 45

The calculation shall be carried out in accordance with equations (109) to (113), using the value for d_a instead of d_t and $d'_L = 0$.

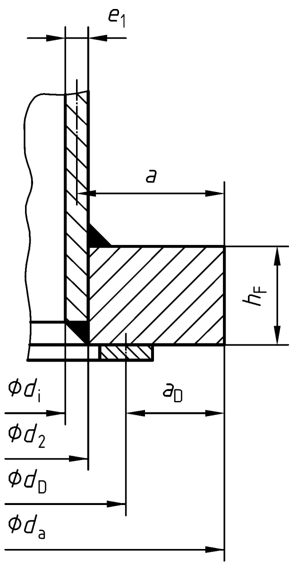
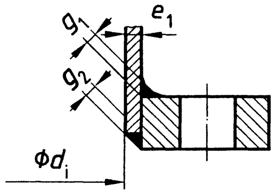
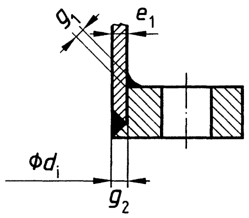
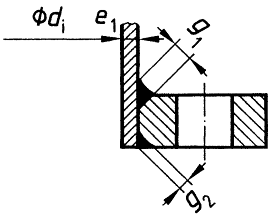
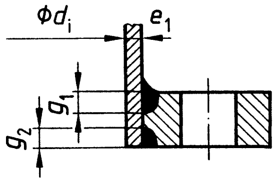
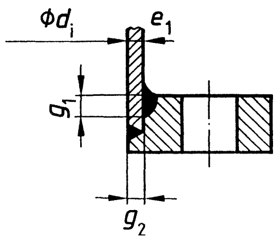


Figure 45 — Welded-on collar

Table 8 — Field of application of various weld-on flanges

Weld type	**Weld thickness**	**Limitation $d_i \times p$** mm × bar
	$g_1 + g_2 \geq 1{,}4 \times e_1$	10 000
	$g_1 + g_2 \geq 1{,}4 \times e_1$	10 000
	$g_1 + g_2 \geq 2 \times e_1$	20 000
	$g_1 + g_2 \geq 2 \times e_1$	—
	$g_1 + g_2 \geq 2 \times e_1$	—
The difference between g_1 and g_2 shall not exceed 25 %.		

10.2.6 Reverse flanges

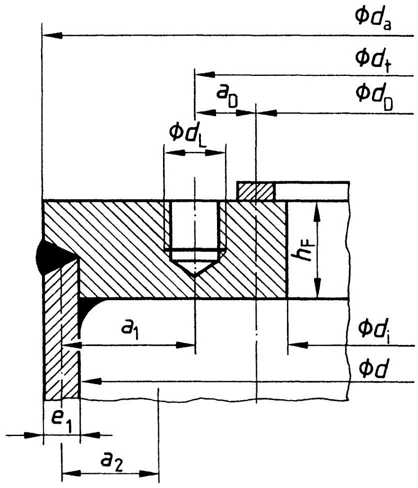


Figure 46 — Reverse flange

The required flange thickness h_F results from equation (109). The calculated double flange width b results from:

$$b = d - d_i - 2d_L' \tag{114}$$

The coefficient Z results from:

$$Z = (d + e_1)e_1^2 \tag{115}$$

The lever of the bolt force for the operating conditions result from:

$$a = a_1 + a_2\left(\frac{d^2}{d_D^2} - 1\right) \tag{116}$$

with $a_1 = \dfrac{d - d_t + e_1}{2}$ (117)

and $a_2 = \dfrac{d - d_D + 2e_1}{4}$ (118)

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \tag{119}$$

10.2.7 Loose flanges

The required flange thickness h_F results from:

$$h_F = \sqrt{1{,}91 \times \frac{W}{b}} \tag{120}$$

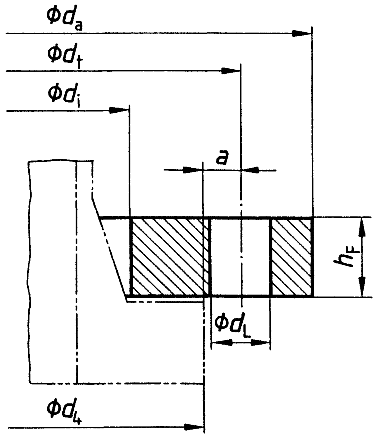


Figure 47 — Loose flange

The calculated double flange width b results from:

$$b = d_a - d_i - 2d'_L \tag{121}$$

with d'_L in accordance with equation (97).

The lever of the bolt force for the operating and assembly conditions result from:

$$a = a_D = \frac{d_t - d_4}{2} \tag{122}$$

The contact pressure p_F between the collar and the loose flange results from:

$$P_F = 1{,}27 \times \frac{F_{SB}}{d_4^2 - d_i^2} \leq 1{,}5 \times f \tag{123}$$

In the case of split loose flanges, the bolt forces shall be doubled; if the splitting is staggered by 90°, the forces may be increased by 50 % only.

10.3 Oval flanges

10.3.1 Oval flanges in accordance with Figure 48

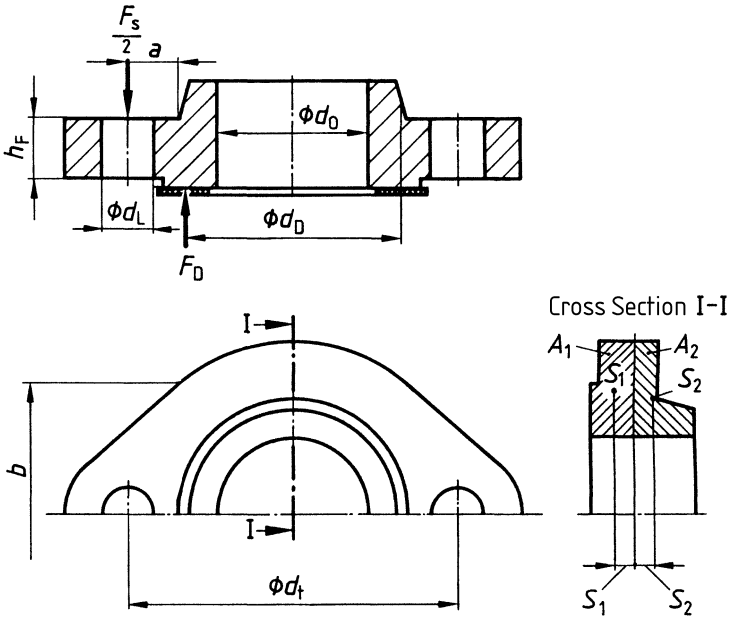


Figure 48 — Oval flange with two bolts

The most highly stressed cross-sections I-I and II-II shall be examined.

The decisive factor for the flange design is the maximum flange resistance W required. It applies to the operating condition $W_{\text{req1}} = \dfrac{M}{1{,}5 \times f_{\text{d/t}}}$ and for the assembly condition $W_{\text{req2}} = \dfrac{M}{2{,}2 \times f_{\text{d}}}$

The minimum bolt force F_{SB} for the operating condition is obtained from the pipe force F_{p} and the gasket operation force F_{DB}:

$$F_{\text{SB}} = F_{\text{p}} + F_{\text{DB}} = p \times \frac{\pi \times d_{\text{D}}^2}{4} + p \times \pi \times d_{\text{D}} \times m \times b_{\text{D}} \times S_{\text{D}} \tag{124}$$

with S_{D} = 1,2 for non-metallic gaskets and metallic envelope gaskets;

S_{D} = 1,0 for metallic gaskets.

The minimum bolt force F_{DV} for the gasket seating results from:

$$F_{\text{DV}} = \pi \times d_{\text{D}} \times \sigma_{\text{VU}} \times b_{\text{D}} \tag{125}$$

The characteristic factors for the gaskets, m and σ_{VU} are given in Table B.1.

Supplementary forces shall be taken into account by adding them to F_{SB} and F_{DV}.

The minimum bolt force F_{SO} for the assembly condition results from:

$$F_{\text{SO}} = \max\,(\chi \times F_{\text{SB}}; F_{\text{DV}}) \tag{126}$$

with χ = 1,1 for general applications;

χ = 1,2 for non-metallic gaskets and combined seals.

The external moment M in the cross-section I-I under calculation results from:

$$M = \frac{F_S}{4} \times d_t \tag{127}$$

with $F_S = F_{SB}$ or F_{SO}.

The cross-sectional area A in the cross-section I-I shall be subdivided in such a way that $A_1 = A_2 = A/2$. s_1 and s_2 are the two distances of the centres of gravity of the partial areas A_1 and A_2 from the centreline. Consequently, the flange resistance in this cross-section results from:

$$W_{avI} = 2A_1(s_1 + s_2) \tag{128}$$

In cross-section II-II the external moment M results from:

$$M = \frac{F_s}{2} \times a \tag{129}$$

The flange resistance W results from:

$$W_{avII} = \frac{b \times h_F^2}{6} \tag{130}$$

10.3.2 Oval flanges in accordance with Figure 49

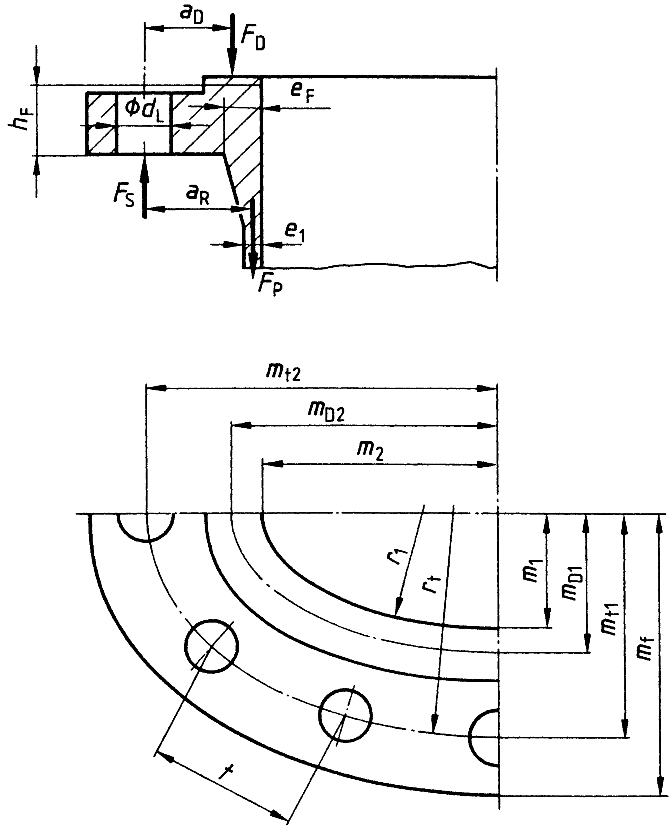


Figure 49 — Oval flange with more than two bolts

The minimum bolt force F_{SB} for the operating condition results from:

$$F_{SB} = F_p + F_{DB} = p \times \pi \times m_{D1} \times m_{D2} + p \times U_D \times m \times b_D \times S_D \tag{131}$$

with S_D = 1,2 for non-metallic gaskets and combined seals;

S_D = 1,0 for metallic gaskets.

The minimum bolt force for the assembly condition results from:

$$F_{DV} = U_D \times \sigma_{VU} \times b_D \tag{132}$$

The mean circumference U_D of the gasket results from:

$$U_D = \pi \left[3(m_{D1} + m_{D2}) - \sqrt{(3m_{D1} + m_{D2})(3m_{D2} + m_{D1})} \right] \tag{133}$$

Characteristic factors for the gaskets m and σ_{VU} are given in Table B.1.

The minimum bolt force F_{SO} for the assembly condition results from:

$$F_{SO} = \max(\chi \times F_{SB}; F_{DV}) \tag{134}$$

with χ = 1,1 for general applications;

χ = 1,2 for non-metallic gaskets and combined seals.

Consequently, the strength condition will be:

$$\frac{F_p \times a_R + F_D \times a_D}{f} \leq \left[\frac{\pi}{2}(m_f - m_1 - d'_L)h_F^2 + \frac{1}{4}U_D\left(e_F^2 - \frac{e_1^2}{4}\right)\right] \times \frac{1}{2}\left[1 + \left(\frac{m_{t1}}{m_{t2}}\right)^2\right] \times \frac{1}{B_5} \tag{135}$$

with B_5 [5] according to Figure 50.

$$B_5 = \begin{cases} 2 - 50 \times \dfrac{e_1}{m_2}; & 0{,}01 < \dfrac{e_1}{m_2} \leq 0{,}02 \\ 1; & 0{,}02 < \dfrac{e_1}{m_2} \leq 0{,}05 \\ \dfrac{2}{3} + 6\,\dfrac{2}{3} \times \dfrac{e_1}{m_2}; & 0{,}05 < \dfrac{e_1}{m_2} \leq 0{,}25 \end{cases}$$

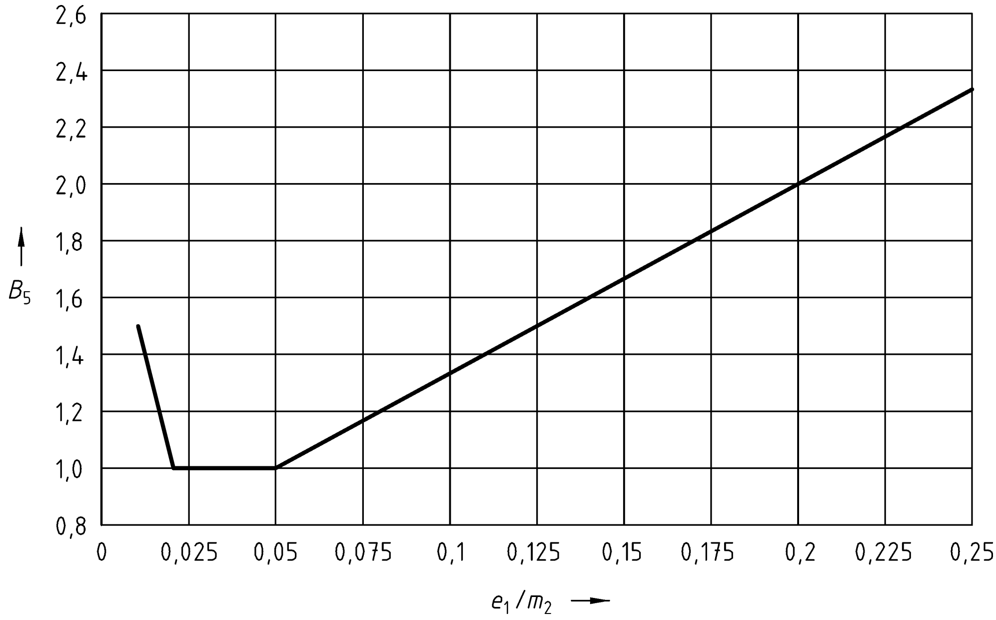


Figure 50 — Correction factor B_5 of equation (135)

10.4 Rectangular or square flanges

10.4.1 Rectangular or square flanges in accordance with Figure 51

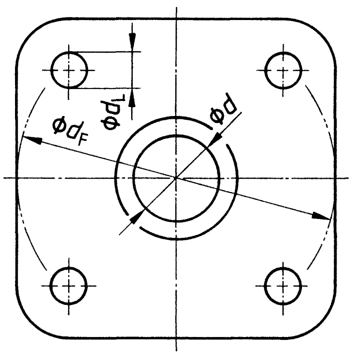


Figure 51 — Rectangular or square flange

The calculation shall be carried out in accordance with 10.2. The flange diameter d_a entered in the equation is the diameter d_F of the largest inscribed circle.

10.4.2 Rectangular slip-on flanges in accordance with Figure 52

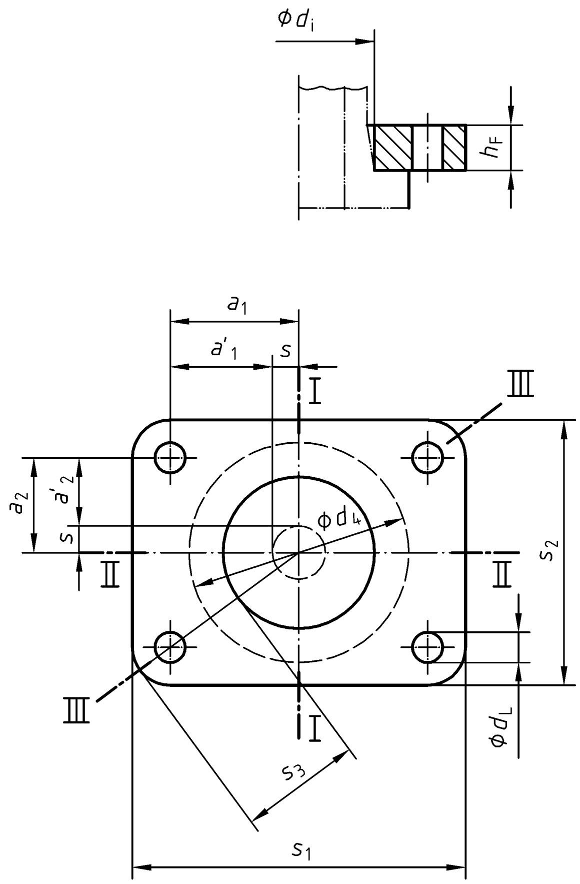


Figure 52 — Rectangular or square slip-on flange

The calculation shall be carried out for the cross-sections I-I, II-II and III-III. With the dimensions according to Figure 52, and the minimum bolt force F_S according to the equations (124) and (134), the moments in the cross-sectional planes shall be calculated from:

$$M_{\mathrm{I}} = \frac{F_s}{2} \times a_1' = \frac{F_s}{2} \times (a_1 - s) \tag{136}$$

$$M_{\mathrm{II}} = \frac{F_s}{2} \times a_2' = \frac{F_s}{2} \times (a_2 - s) \tag{137}$$

$$M_{\mathrm{III}} = \frac{F_s}{4} \left(\sqrt{a_1^2 + a_2^2} - s \right) \tag{138}$$

s is the distance of the centre of gravity of the half circular ring from the centreline:

$$s = \frac{2}{3\pi} \times \frac{d_4^3 - d_i^3}{d_4^2 - d_i^2} \tag{139}$$

The flange resistance can be calculated analogously from:

$$W_{\mathrm{I}} = \frac{h_F^2}{6} (s_2 - d_i) \tag{140}$$

$$W_{II} = \frac{h_F^2}{6}(s_1 - d_i) \tag{141}$$

$$W_{III} = \frac{h_F^2}{6}(s_3 - d_L^{'}) \times 2 \tag{142}$$

with d'_L according to equation (97).

The strength condition in accordance with equations (88) and (89) shall be satisfied for the three cross-sections.

It means $W \geq M$ / (1,5*f*).

10.5 Calculation of the bolt diameter

10.5.1 Diameter of the nominal tensile stress

The required diameter d_S of the nominal tensile stress area of a bolt shall be calculated as follows:

$$d_S = \sqrt{\frac{4}{\pi\, n} \times \frac{F_s}{f \eta}} + c \tag{143}$$

where

F_s is the tensile force of the connection per load case;

n is the number of bolts;

f is the allowable bolt stress;

η is the machining quality factor;

c is the design allowance.

10.5.2 Load cases

The diameter shall be determined for the following two load cases:

a) for the operating condition at the permissible design pressure p_d and at the design temperature t_d;

b) for the assembly condition before pressure application at ambient temperature.

10.5.3 The allowable bolt stress f is obtained from the strength parameter (see EN 1515-1) divided by the safety factor SF in accordance with Table 9.

Table 9 — Safety factors

Condition	Necked-down bolts	Other bolts
Operation	1,5	1,8
Assembly	1,1	1,3

10.5.4 For support faces created by machining with chip removal, or by some other manufacturing process which can be regarded as equivalent, a quality factor η = 1,0 can be assumed. For unmachined surfaces, eyebolts, and hinged bolts, a factor η = 0,75 shall be assumed.

10.5.5 In the case of rigid bolts, the following values shall be entered for the design allowance c in equation (143) for the operating condition:

c = 3 mm for $d_S \leq 20$ mm

c = 1 mm for $d_S \geq 50$ mm

In the intermediate size range, c can be obtained by linear interpolation in accordance with:

$c = (65 - d_S) / 15$

For necked-down bolts c = 0 mm.

For assembly conditions c shall be also c = 0 mm.

10.5.6 For standardized piping flanges, the requirements relating to the bolts shall be regarded as having been complied with on condition that the diameters and numbers of these bolts are selected in accordance with the corresponding piping standards, and that the permissible temperature used for calculation for the flanges is not exceeded.

10.6 Design temperature

The design temperature of the bolts is a function of the type of bolted connection and the heat insulation. If no particular temperature verification is carried out and if the bolts are not directly exposed to a medium having a temperature > 50 °C, the design temperature for bolted connections of non-insulated flanges can be assumed to remain under the maximum temperature of the medium conveyed by the following values:

a) loose flange and loose flange 30 °C;

b) integral flange and loose flange 25 °C;

c) integral flange and integral flange 15 °C.

11 Calculation methods for glands

11.1 Loads

The components of the gland bolting are loaded by the internal pressure F_p and eventually, by the additional forces F_z.

$$F_{SB} = F_p + F_z \tag{144}$$

The pressure force on the ring surface area of the gland is:

$$F_p = \left(d_a^2 - d_i^2\right) \times \frac{\pi}{4} \times p \tag{145}$$

where

d_a = gland outside diameter;

d_i = gland inside diameter.

Additionally forces F_z occurring as a function of the design are to be taken into account, as appropriate.

11.2 Gland bolts

Gland bolts shall be verified by means of the force F_{SB} in accordance with equation (144); allowance c may be ignored in this case.

11.3 Gland flanges

Gland flanges shall be verified by means of the force F_{SB} according to 11.1 in consideration of the actual design.

11.4 Other components

Other components, which form part of the gland design and which are subjected to stress shall be verified in accordance with sound engineering practice by means of the appropriate force F. The nominal design stresses in accordance with clause 6 and, in the case of bolts, Table 9, shall be considered.

12 Fatigue

In the case of alternating stresses, verification shall be in accordance with EN 13445-3.

13 Marking

Valve shells designed for a specified pressure and associated temperature or for a range of specified pressures at associated temperatures shall be marked in accordance with EN 19.

Annex A
(informative)

Allowable stresses

Tables A.1 to A.3 contain allowable stresses for the three different types of material; cast steel, flat products and forgings.

RT in column three means a temperature range from – 10 °C to + 20 °C.

These tables are only for information and are currently under review.

Table A.1 — Allowable stresses for cast steel

Material	Mat.-Group	Allowable stresses f in N/mm^2 at temperature t in °C																				
		RT	50	100	150	200	250	300	350	400	450	500	510	520	530	540	550	560	570	580	590	600
1.0619	3E0	126,3	120,4	110,5	101,3	92,1	84,2	76,3	71,1	68,4	43,7	—	—	—	—	—	—	—	—	—	—	—
1.0621	2E0	126,3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.1131	7E0	126,3	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.4308	11E0	133,3	123,3	106,7	95,0	83,3	78,3	73,3	68,3	63,3	58,3	53,3	—	—	—	—	—	—	—	—	—	—
1.4309	10E0	140,0	128,7	110,0	98,3	86,7	80,0	73,3	66,7	60,0	53,3	46,7	—	—	—	—	—	—	—	—	—	—
1.4408	14E0	140,0	130,0	113,3	101,7	90,0	83,3	76,7	73,3	70,0	68,0	67,0	66,8	66,5	66,3	66,1	65,9	65,7	63,4	58,6	53,7	49,3
1.4409	13E0	146,7	135,4	116,7	106,7	96,7	86,7	76,7	73,3	70,0	66,7	63,3	—	—	—	—	—	—	—	—	—	—
1.4552	12E0	133,3	124,5	110,0	103,3	96,7	91,7	86,7	84,4	80,0	76,7	73,3	72,0	70,7	69,3	68,0	66,7	—	—	—	—	—
1.4581	15E0	140,0	133,7	123,3	115,0	106,7	101,7	96,7	91,7	86,7	83,3	80,0	79,3	78,7	78,0	77,3	76,7	—	—	—	—	—
1.4931	9E0	308,3	308,3	308,3	308,3	300,0	293,3	286,7	273,3	260,0	206,0	138,0	126,1	114,3	102,4	90,5	78,7	69,5	60,3	51,1	41,9	32,7
1.5419	4E0	128,9	124,0	116,1	108,0	100,0	93,4	86,8	81,6	78,9	76,3	44,7	—	—	—	—	—	—	—	—	—	—
1.5638	7E1	166,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.6220	7E0	157,8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.7357	5E0	163,3	160,0	150,6	141,1	131,6	126,3	121,1	113,2	105,3	100,0	61,6	55,1	48,5	42,0	—	—	—	—	—	—	—
1.7365	6E1	262,5	262,5	262,5	262,5	260,0	256,7	253,3	250,0	246,7	225,0	70,7	—	—	—	—	—	—	—	—	—	—
1.7379	6E0	196,7	196,7	196,7	193,4	186,8	184,2	181,6	173,7	165,8	114,7	71,6	64,2	56,8	49,5	42,1	34,7	30,7	26,7	22,7	—	—

NOTE
Material strength values are chosen from EN material standards.
The allowable stresses apply to wall thickness values up to 40 mm.
Values shown in shaded areas are applicable only if carbon content is 0,04 % or higher.

Table A.2 — Allowable stresses for flat products

Material	Mat.-Group	Allowable stresses f in N/mm^2 at temperature t in °C																				
		RT	50	100	150	200	250	300	350	400	450	500	510	520	530	540	550	560	570	580	590	600
1.0037	1E0	141,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0038	1E1	141,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0425	3E0	170,0	156,0	143,3	136,7	130,0	116,7	103,3	93,3	86,7	46,0	—	—	—	—	—	—	—	—	—	—	—
1.0481	3E1	191,7	181,3	166,7	156,7	150,0	136,7	123,3	113,3	103,3	—	—	—	—	—	—	—	—	—	—	—	—
1.0486	8E0	162,5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0487	8E2	162,5	162,5	156,7	144,0	130,7	118,0	98,0	84,7	72,0	—	—	—	—	—	—	—	—	—	—	—	—
1.0488	7E0	162,5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0562	8E1	204,2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0565	8E3	204,2	204,2	196,0	183,3	163,3	150,7	144,0	130,7	111,3	—	—	—	—	—	—	—	—	—	—	—	—
1.0566	7E1	204,2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.1104	7E0	162,5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.1106	7E1	204,2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.4301	11E0	173,3	164,5	150,0	140,0	130,8	120,8	112,5	107,5	104,2	101,7	100,0	100,0	100,0	100,0	100,0	100,0	—	—	—	—	—
1.4306	10E0	173,3	159,5	136,7	126,7	120,0	114,2	105,8	100,8	96,7	93,3	90,8	90,7	90,5	90,3	90,2	90,0	—	—	—	—	—
1.4311	10E1	183,3	175,8	163,3	153,3	143,3	140,0	136,7	134,2	130,0	126,7	124,2	—	—	—	—	—	—	—	—	—	—
1.4401	14E0	173,3	162,0	143,3	136,7	130,0	128,3	126,7	125,0	120,0	117,5	115,8	115,4	115,1	114,7	114,4	114,1	113,7	105,3	96,0	86,7	78,7
1.4404	13E0	173,3	162,0	143,3	136,7	130,0	128,3	120,8	114,2	112,5	108,3	106,7	106,5	106,3	106,1	105,9	105,8	—	—	—	—	—
1.4541	12E0	166,7	158,5	145,0	133,3	123,3	116,7	113,3	111,7	110,0	106,7	103,3	102,6	101,9	101,3	98,2	94,7	86,0	78,7	71,3	64,0	57,3
1.4550	12E0	166,7	158,5	145,0	133,3	123,3	116,7	113,3	111,7	110,0	106,7	103,3	102,6	101,9	101,3	98,2	94,7	86,0	78,7	71,3	64,0	57,3
1.4571	15E0	173,3	163,3	146,7	136,7	130,0	128,3	125,0	125,0	125,0	123,3	120,0	120,0	120,0	116,6	113,3	110,0	106,7	102,7	94,0	85,3	77,3
1.4580	15E0	173,3	163,3	146,7	136,7	130,0	128,3	125,0	125,0	125,0	123,3	120,0	120,0	120,0	116,6	113,3	110,0	106,7	102,7	94,0	85,3	77,3
1.5415	4E0	180,0	173,8	163,7	153,5	143,3	133,3	113,3	106,7	100,0	96,7	67,3	—	—	—	—	—	—	—	—	—	—
1.5637	7E1	204,2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.5662	7E2	266,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.5680	7E1	220,8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.6212	7E0	175,0	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.6228	7E1	204,2	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.7335	5E0	187,5	187,5	177,4	165,3	153,3	146,7	136,7	126,7	120,0	113,3	91,3	77,3	62,7	52,0	—	—	—	—	—	—	—
1.7380	6E0	200,0	193,8	183,7	173,5	163,3	153,3	146,7	140,0	133,3	126,7	90,0	78,7	68,7	60,0	—	—	—	—	—	—	—

NOTE
Material strength values are chosen from EN material standards.
The allowable stresses apply to wall thickness values up to 40 mm.
Values shown in shaded areas are applicable only if carbon content is 0,04 % or higher.

Table A.3 — Allowable stresses for forgings

Material	Mat.-Group	Allowable stresses f in N/mm^2 at temperature t in °C																				
		RT	50	100	150	200	250	300	350	400	450	500	510	520	530	540	550	560	570	580	590	600
1.0037	1E0	141,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0038	1E1	141,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.0352	3E0	146,7	140,4	130,0	123,3	116,7	106,7	96,7	90,0	83,3	46,0	—	—	—	—	—	—	—	—	—	—	—
1.0426	3E1	186,7	179,1	166,7	156,7	150,0	136,7	123,3	113,3	103,3	—	—	—	—	—	—	—	—	—	—	—	—
1.0460	3E0	166,7	163,4	158,0	144,0	126,7	113,3	100,0	86,7	73,3	46,0	—	—	—	—	—	—	—	—	—	—	—
1.0477	8E2	162,5	162,5	162,5	156,7	137,3	124,0	104,7	91,3	78,7	—	—	—	—	—	—	—	—	—	—	—	—
1.0565	8E3	204,2	204,2	202,7	189,3	170,0	156,7	144,0	130,7	111,3	—	—	—	—	—		—	—	—	—	—	—
1.4301	11E0	166,7	160,4	150,0	140,0	130,8	120,8	112,5	107,5	104,2	102,1	100,0	92,2	84,5	76,7	69,0	61,3	—	—	—	—	—
1.4306	10E0	153,3	147,0	136,7	126,7	120,0	114,2	105,8	100,8	96,7	93,3	90,8	90,7	90,5	90,3	90,2	90,0	—	—	—	—	—
1.4311	10E1	166,7	165,4	163,3	153,3	143,3	140,0	136,7	133,3	130,0	127,1	124,2	—	—	—	—	—	—	—	—	—	—
1.4401	14E0	170,0	160,0	143,3	136,7	130,0	128,3	126,7	125,0	120,0	117,9	115,8	115,4	115,1	114,7	114,4	114,1	113,7	105,3	96,0	86,7	78,7
1.4404	13E0	163,3	155,8	143,3	136,7	130,0	128,3	120,8	115,8	112,5	109,6	106,7	106,5	106,3	106,1	105,9	105,8	—	—	—	—	—
1.4406	13E1	193,3	185,8	173,3	163,3	153,3	150,0	145,8	140,8	136,7	134,2	131,7	131,5	131,3	131,1	130,9	130,8	—	—	—	—	—
1.4541	12E0	156,7	149,9	138,7	130,7	124,0	118,0	111,3	107,3	104,0	101,7	99,3	99,0	98,7	98,5	98,2	94,7	86,0	78,7	71,3	64,0	57,3
1.4571	15E0	150,0	148,7	146,7	136,7	130,0	128,3	125,0	125,0	125,0	122,3	120,0	119,5	119,0	118,5	118,0	117,6	111,3	102,7	94,0	85,3	77,3
1.4922	9E0	291,7	291,7	291,7	291,7	286,7	276,7	260,0	253,3	240,0	220,0	157,3	141,3	125,3	111,3	98,0	85,3	74,0	63,3	54,0	46,0	39,3
1.5415	4E0	183,3	183,3	176,0	163,3	150,0	136,7	120,0	113,3	106,7	103,3	62,0	49,3	39,3	31,3	—	—	—	—	—	—	—
1.5637	7E1	195,8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.5662	7E2	266,7	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.5680	7E1	212,5	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.6217	7E0	175,0	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.6228	7E1	195,8	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
1.7335	5E0	183,3	183,3	173,3	163,3	160,0	153,3	143,3	133,3	126,7	120,0	91,3	77,3	62,7	52,0	40,7	32,7	26,7	22,0	—	—	—
1.7366	6E1	266,7	261,2	230,0	223,3	218,0	215,3	214,7	210,7	204,0	184,0	75,3	—	—	—	—	—	—	—	—	—	—
1.7383	6E0	206,7	195,4	176,7	166,7	156,7	153,3	146,7	136,7	130,0	123,3	90,0	78,7	68,7	60,0	52,0	45,3	38,7	34,0	29,3	25,3	22,7

NOTE
Material strength values are chosen from EN material standards.
The allowable stresses apply to wall thickness values up to 40 mm.
Values shown in shaded areas are applicable only if carbon content is 0,04 % or higher.

Annex B
(informative)

Characteristic values of gaskets and joints

Tables B.1 and B.2 contain characteristic values for the calculation of flanged joints in accordance with this standard.

The use of asbestos is in accordance with national laws and directives.

The definition "metallic envelope gasket" means combined seals.

The characteristic factors for the gaskets σ_{VU} , σ_{VO} in column 3 and 4 of Table B.1 mean the minimum required and the maximum recommended gasket stress for assembly conditions. The factor m is a scale-factor used in clause 10.

Table B.1 — Characteristic values of gaskets and joints

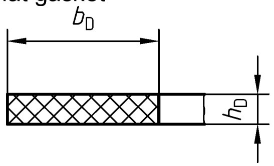
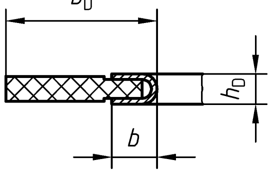
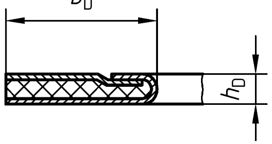
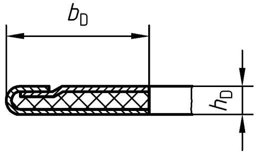
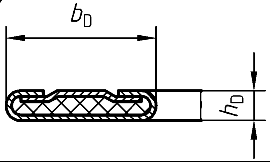
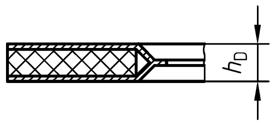
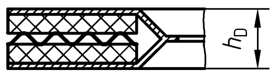
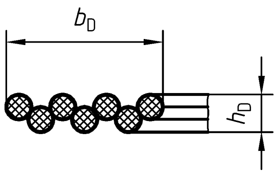
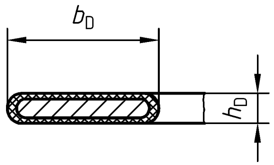
Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
		σ_{VU} N/mm²	σ_{VO} N/mm²	m	t °C: 20	100	200	300	400	500	600	
					σ_{BO} N/mm² [a]							
Flat gasket 	Rubber, general nitrile rubber chloroprene rubber	2	10	1,3	10	6	—	—	—	—	—	—
	Fluorine rubber	2	10	1,3	10	7	—	—	—	—	—	—
	PTFE h_D = 0,5	10	90	1,1	90	40	25	—	—	—	—	Precondition for non-enclosed gasket $b_D/h_D = 20$
	PTFE h_D = 1		70		70							
	PTFE h_D = 2		50		50							
Flat gasket b_D/h_D < 5	It	40	200	1,3	200	190	180	170	160	—	—	If b_D/h_D < 5 the gasket should be enclosed
b_D/h_D ≥ 5	It except It S h_D = 0,5	30	200		200	190	180	170	160	—	—	
	It except It S h_D = 1		180		180	160	150	140	130	—	—	
	It except It S h_D = 1,5		175		175	120	110	100	90	—	—	
	It except It S h_D = 2		165		165	86	80	75	65	—	—	
	It except It S h_D = 3 up to 4		135		135	56	52	48	48	—	—	
	It S h_D = 0,5		150		150	—	—	—	—	—	—	
	It S h_D = 1		135		135	50	50	—	—	—	—	
	It S h_D = 1,5		120		120	50	50	—	—	—	—	
	It S h_D = 2		105		105	43	40	—	—	—	—	
	It S h_D = 3 up to 4		90		90	28	26	—	—	—	—	
b_D/h_D ≥ 20	Graphite [b] non-reinforced	15	150		150	150	150	130	120	120	—	—
b_D/h_D 15 up to < 20			120		120	120	120	105	95	95	—	
b_D/h_D 10 up to < 15			100		100	100	100	85	80	80	—	
b_D/h_D 5 up to < 10			80		80	80	80	70	65	65	—	
b_D/h_D ≥ 15	Graphite [b] reinforced		180		180	180	180	155	145	145	—	—
b_D/h_D 10 up to < 15			150		150	150	150	130	120	120	—	
b_D/h_D 7,5 up to < 10			120		120	120	120	105	95	95	—	
b_D/h_D 5 up to < 7,5			100		100	100	100	85	80	80	—	

Table B.1 — (*continued*)

Metallic envelope gaskets													
Shape	Material	Assembly condition σ_{VU} N/mm²	Assembly condition σ_{VO} N/mm²	Operating condition m	t °C 20	100	200	300	400	500	600	σ_{BO} N/mm² [a]	Remarks
It-gasket restrained at inside rim 	It except It S with 0,25 mm 1.4541 restrained	50	135	1,3	135	66	62	58	55	—	—		$h_D \geq 3$ mm
Fully enveloped It- or asbestos gaskets a) 	Envelope material	-	-	-	-	-	-	-	-	-	-		—
b) 	Al	50	135	1,4	135	120	90	(60)	—	—	—		-
	CuZn alloys (Ms)	60	150	1,6	150	140	130	120	(100)	—	—		
c) 	Fe/Ni	70	180	1,8	180	170	160	150	140	(130)	—		
	CrNi-Steel	100	250	2,0	250	240	220	200	180	(160)	—		

Values in brackets are not sufficiently verified

Table B.1 — (*continued*)

Metallic envelope gaskets												
Shape	Material	Assembly condition			Operating condition							Remarks
		σ_{VU} N/mm²	σ_{VO} N/mm²	m	t °C							
					20	100	200	300	400	500	600	
					σ_{BO} N/mm² [a]							
	PTFE envelope It-gasket	10*	90**	1,1	90	55	45	—	—	—	—	* When used between glass-lined flanges, σ_{VU} shall be increased as a function of waviness or an other gasket shall be chosen ** Precondition: envelope ≤ 0,5 mm
	PTFE envelope corrugated gasket with It covering for glass-lined flanges	10	90	1,0	90	55	45	—	—	—	—	
Corrugated gasket 	Al/Asbestos	30	80	0,6	80	75	70	(60)	—	—	—	Asbestos rope, impregnated
	CuZn alloys (Ms)/ Asbestos	35	110	0,7	110	105	100	90	(80)	—	—	
	St/Asbestos or CrNi-Steel/Asbestos	45	150	1,0	150	145	135	125	105	95	—	
	Flat steel with asbestos envelope (braiding or cloth)	45	150	1,0	150	145	135	125	105	—	—	—

Values in brackets are not sufficiently verified

Table B.1 — (*continued*)

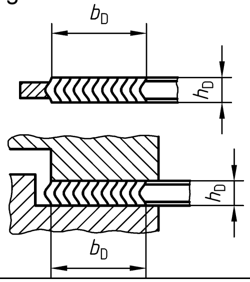
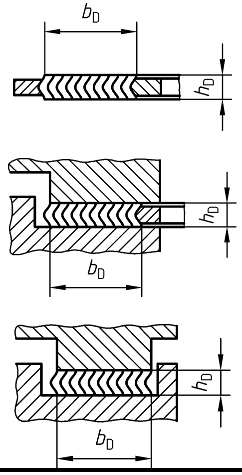
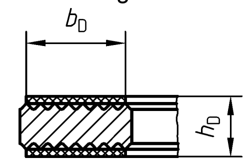
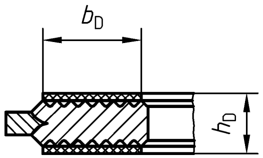
Metallic envelope gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
		σ_{VU} N/mm²	σ_{VO} N/mm²	m	t °C							
					20	100	200	300	400	500	600	
					σ_{BO} N/mm² [a]							
Spiral wound gasket, single enclosure 	PTFE	20	110	1,3	110	110	100	(90)	—	—	—	( ) t_{max} = 250 °C gaskets with double enclosure shall be used, if possible
	Graphite	20	110		110	110	100	90	80	—	—	
	It, Asbestos impregnated	55	150		150	140	—	—	—	—	—	
Spiral wound gasket, double enclosure 	PTFE	20	300		300	170	160	(150)	—	—	—	—
	Graphite	20	300		300	170	160	—	—	—	—	
	It, Asbestos impregnated	55	300		300	170	130	—	—	—	—	
Values in brackets are not sufficiently verified												

Table B.1 — (*continued*)

Metallic envelope gaskets													
Shape	Material	Assembly condition		m	Operating condition								Remarks
		σ_{VU} N/mm^2	σ_{VO} N/mm^2		t °C								
					20	100	200	300	400	500	600		
					σ_{BO} N/mm^2 [a]								
Grooved gasket with a layer of additional gasket material	Grooved/layer 1.0333/PTFE	10	350	1,1	350	320	290	(265)	—	—	—	( ) t_{max} = 250 °C
	1.4541/PTFE		500		500	480	450	(420)	—	—	—	
	1.0333/Graphite	15	350	1,1	350	320	290	265	—	—	—	—
	1.5415/Graphite		450		450	400	360	330	270	220	—	
	1.4541/Graphite		500		500	480	450	420	390	350	—	
	1.4828/Graphite		600		600	570	540	500	460	400	240	
	1.0333/It	65	350	1,3	350	320	290	265	—	—	—	—
	1.5415/It		450		450	400	360	330	270	220	—	
	1.4541/It		500		500	480	450	420	390	350	—	
	1.4828/It		600		600	570	540	500	460	400	240	
	1.5415/Silver	125	450	1,5	450	400	360	330	270	220	—	—
	1.4828/Silver		600		600	570	540	500	460	400	240	

Values in brackets are not sufficiently verified

Table B.1 — (*continued*)

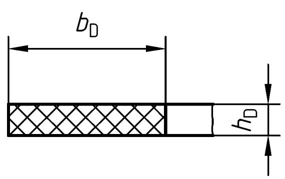
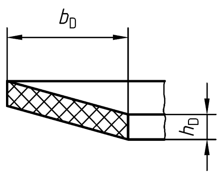
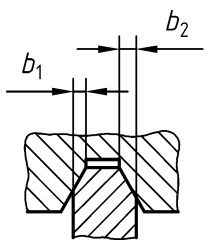
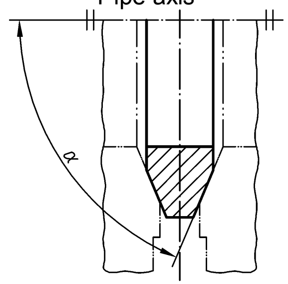
Metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
		σ_{VU} N/mm^2	σ_{VO} N/mm^2	m	t °C: 20	100	200	300	400	500	600	
					σ_{BO} N/mm^2 [a]							
Flat gaskets 	Al	70	140	1,3	140	120	93	—	—	—	—	The effective sealing width in each case is the projection of the sealing face in the direction of load. In the case of solid metallic gaskets, special consideration shall be given to the characteristic value k if no crowned shapes are used. In the case of double contact gaskets, the distance is to be taken into account.
	Cu	135	300		300	270	195	150	—	—	—	
	Fe	235	525		525	465	390	315	260	—	—	
	St 35	265	600		600	570	495	390	300	—	—	
	13 CrMo 44	300	675		675	675	630	585	495	420	—	
	1.4541	335	750		750	720	675	630	585	515	420	
	1.4828	400	900		900	855	810	750	690	600	480	

Table B.1 — (*concluded*)

Metallic gaskets												
Shape	**Material**	**Assembly condition**		**Operating condition**								**Remarks**
		σ_{VU} N/mm²	σ_{VO} N/mm²	**m**	t °C 20	100	200	300	400	500	600	
					σ_{BO} N/mm² [a]							
Lens-shaped gaskets	Al	70	140	1,3	140	120	93	—	—	—	—	The sealing width is calculated as follows: for shapes a) to c) by $b_D = c^2 \times \frac{\sigma}{E_D} \times r$ for shape d) (lenticular gasket, $\alpha = 70°$) $b_D = c^2 \times \frac{\sigma}{E_D} \times r \times \sin\alpha$ for shapes e) to f) (with contact at two faces) $b_D = 2c^2 \times \frac{\sigma}{E_D} \times r \times \sin\alpha$
	Cu	135	300		300	270	195	150	—	—	—	
	Fe	235	525		525	465	390	315	260	—	—	
	St 35	265	600		600	570	495	390	300	—	—	
	13 CrMo 44	300	675		675	675	630	585	495	420	—	
	1.4541	335	750		750	720	675	630	585	515	420	
	1.4828	400	900		900	855	810	750	690	600	480	

The equations given under "Remarks" for the sealing width of metallic lens-shaped gaskets according to Figure B.1a) to B.1f) are only applicable if the characteristic shape of the gasket remains unchanged. That is always the case when the value of the sealing width b_D is small in relation to the characteristic width b or the radius r of the gasket. In these cases only the contact area will be subjected to plastic deformation.
Gaskets, e.g. according to Figure B.1e) of soft plastic materials such as aluminium, copper or silver may also be subjected to full plastic deformation. In this case, the ring volume shall exceed the groove volume by approximately 3 % to achieve a durable connection. This is the case if F_{DVO} according to equation (65) or F_{DBO} according to equation (63) provide with the designated gasket force values in the range $2 \times r$ for b_D.

Angle α, for example on a lenticular gasket



r mm	c
5 to 20	10
> 20 to 80	8
> 80 to 120	6

α = Opening angle of he cone
For the ring-joint gasket shape f), $\alpha = 23°$.
As a function of the condition, σ_{VU}, σ_{VO} or σ_{BO} shall be put in for σ

Values in brackets are not sufficiently verified

[a] Intermediate values to be determined by interpolation.

[b] Exceeding of the characteristic values can cause spontaneous failure of the gasket.

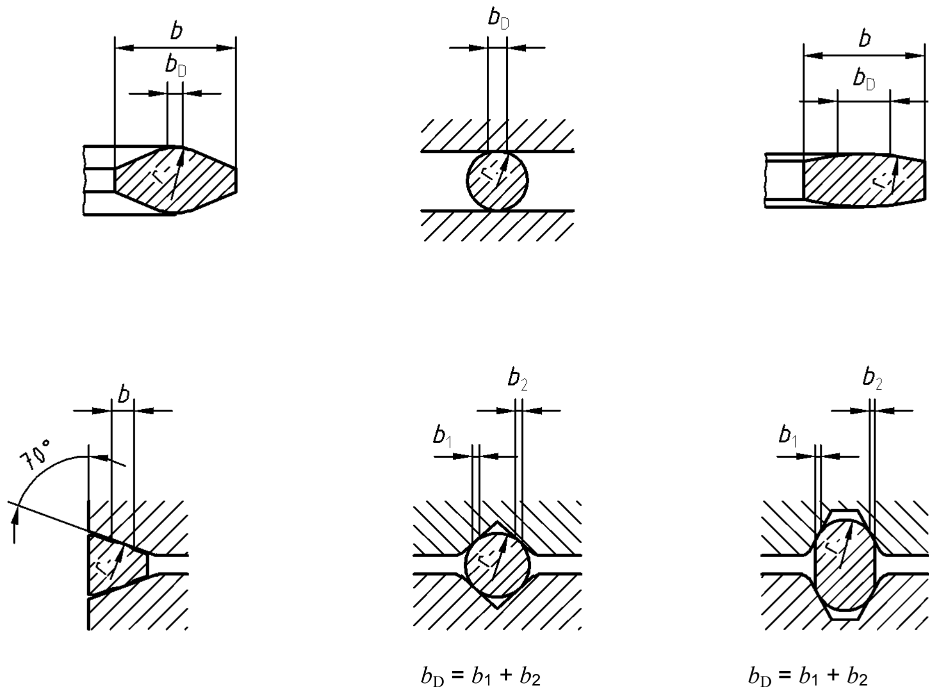


Figure B.1 — Sealing width

Table B.2 — (Equivalent) modulus of elasticity of the gasket materials

Gasket material	(Equivalent) modulus of elasticity E_D in N/mm² at a temperature of				
	20 °C	200 °C	300 °C	400 °C	500 °C
It	1 000 up to 1 500	—	2 200	—	—
Rubber, soft (45 Shore-A)	≈ 30	—	—	—	—
Rubber, hard (80 Shore-A)	≈ 80	—	—	—	—
PTFE	600 647	45 (at 260 °C)	—	—	—
Graphite	$E_{DRT} \cong 5\ 000$	—	—	—	—
Corrugated gasket	8 000	—	—	—	—
Spiral wound gasket	10 000	—	—	—	—
Fully enveloped gasket	12 000	—	—	—	—
Grooved gasket	20 000	—	—	—	—
Soft iron C-Steel Low alloy steel	212 000	200 000	194 000	185 000	176 000
Austenitic CrNi-Steel	200 000	186 000	179 000	172 000	165 000
Al	70 000	63 000	50 000	—	—
Cu	129 000	122 000	—	111 000	105 000

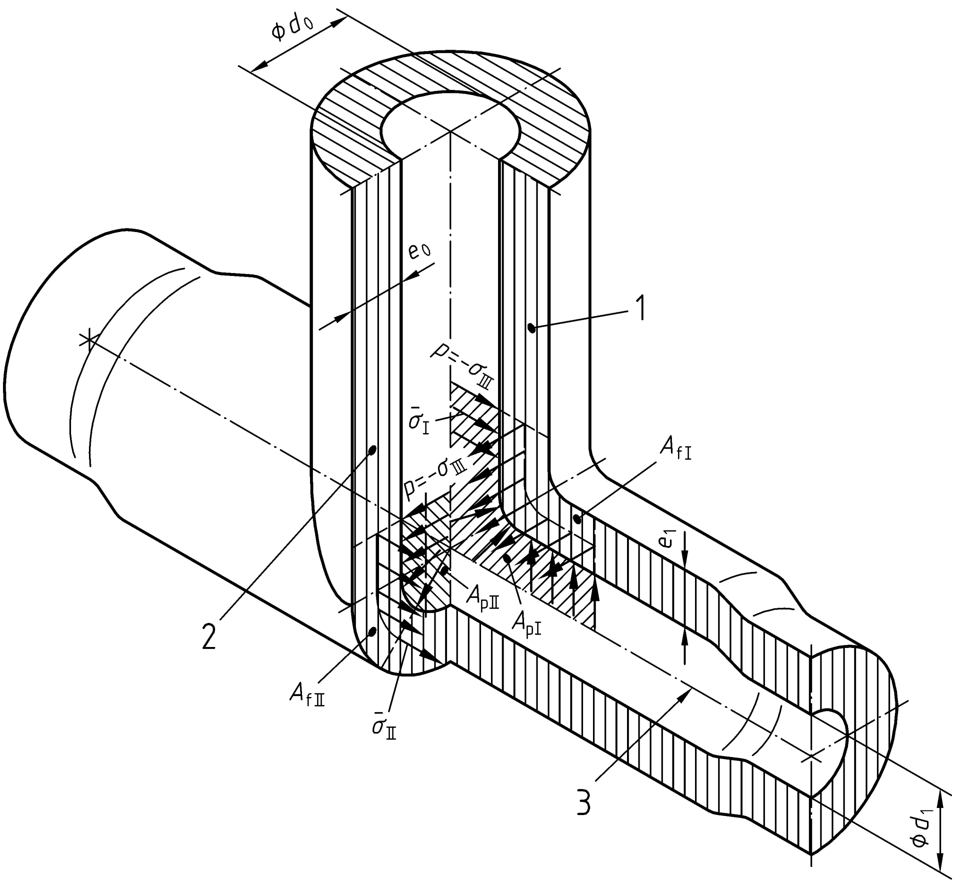
For non-metallic gaskets and metallic-envelope gaskets, the above values should be verified for the appropriate operating condition.

Annex C
(informative)

Calculation procedure

The strength calculation of the valve body with branch is carried out on the basis of an equilibrium consideration between the external and internal forces for the most highly stressed zones. These zones are deemed to be the transitions between the cylindrical, spherical or non-circular basic bodies and the branch. For these calculations, diameter d_0 and wall thickness e_0 are allocated to the basic body, and diameter d_1 and wall thickness e_1 are allocated to the branch. The relationship $d_0 \geq d_1$ applies.

For cylindrical basic bodies as illustrated in Figure C.1, the cross-section I situated in the longitudinal section through the main axis exhibits the highest stress as a general rule, with a mean principal stress $\overline{\sigma}_{\mathrm{I}}$. However, if the ratio of nozzle aperture to basic body aperture is ≥ 0,7, the bending stresses arising in the transverse section to the main axis (cross-section II) are taken into consideration, i.e. this direction is also calculated.



Key
1 Cross-section I
2 Cross-section II
3 Main axis

Figure C.1 — Sections for calculating the strength of valve bodies with branch

In the case of non-circular valve bodies with branches, and generally in the case of additional actions of forces in the direction of the main axis, the greatest stress may often lie in the transverse section with the mean principal stress $\overline{\sigma}_{\mathrm{II}}$ (cross-section II). In such cases, the calculation is also carried out for both the longitudinal section and the transverse section (see also Figure 8).

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 (PED).

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 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 (PED)

Clause(s)/sub-clauses of this European Standard	Annex 1 of PED Essential Safety Requirements	Nature of requirement
4	2.1	General design
4 to 11	2.2.2	Design for adequate strength — calculation method
4	2.6	Corrosion or other chemical attack
13	3.3	Marking
6	4.2 a	Materials
6	7.1.2	Permissible membrane stresses
7.2.1	7.2	Joint coefficients

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

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