BS EN 12516-2:2014

BSI Standards Publication

Industrial valves — Shell design strength

Part 2: Calculation method for steel valve shells

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National foreword

This British Standard is the UK implementation of EN 12516-2:2014. It supersedes BS EN 12516-2:2004 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee PSE/18/1, Industrial valves, steam traps, actuators and safety devices against excessive pressure - Valves - Basic standards.

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

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Contents

Page

Foreword

This document (EN 12516-2:2014) 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of an identical text or by endorsement, at the latest by April 2015, and conflicting national standards shall be withdrawn at the latest by April 2015.

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

This document supersedes EN 12516-2:2004.

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 97/23/EC (Pressure Equipment Directive).

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

In comparison with the previous version, the following significant changes have been made:

a) the normative references were updated;

b) all formulae and figures have been renumbered; in particular 10.6 “Design temperature” became 10.5 “Calculation of the bolt diameter”;

c) some formulae were changed:

   1) Formulae (3) to (6) for calculated wall thickness have been added;

   2) Formulae (9) and (10) for calculation of e_C in case of d_o / d_i > 1,7 have been added;

   3) Formulae (17) and (20) for conical bodies or branches have been added;

d) the figures were changed and/or updated:

   1) a new Figure 1 “Composition of section thickness and tolerance allowances” has been added;

   2) Figure 2 “Cone calculation coefficient” has been over-worked;

   3) former Figures 6a and 6b are now combined in Figure 7 “Calculation coefficient B_n for rectangular cross-sections”;

   4) Figures 23, 24, and 25 used to establish the calculation coefficients C_X, C_Y and C_Z were moved to 8.2.1;

   5) the new Figure 46 “Types of flange connections” has been added;

e) tables were updated:

   1) Table 1 giving the symbols characteristics and units has been revised;

   2) a column for test conditions in Table 2 “Nominal design stresses (allowable stresses)” has been added;

3) Table 5 “Flat circular plates and annular plates — Bending moments as a function of load cases and clamping conditions” has been revised;

4) Table 7 “Lever arms of the forces in the moment formulae” has been revised;

f) Clause 6 “Nominal design stresses for pressure parts other than bolts” now contains references to PED 97/23/EC;

g) Clause 7 “Calculation methods for the wall thickness of valve bodies” has been restructured; and 7.1 now contains information on calculation of the surface-comparison;

h) Subclauses 8.2.2 and 8.2.3 now draw a distinction between “direct loading” and “not subjected to direct loading”; and 8.2.3 now contains a warning regarding the mean support diameter d_{mA};

i) there is a new Subclause 8.3.3.5 regarding the diameter of centre of gravity;

j) Clause 10 “Calculation methods for flanges” has been over-worked;

k) the former informative Annex A “Allowable stresses” has been deleted;

l) the Annex “Characteristic values of gaskets and joints” has been over-worked;

m) Annex ZA has been updated.

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* (the present document);

— *Part 3: Experimental method*;

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

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

Introduction

EN 12516, *Industrial valves — Shell design strength*, is composed of four parts. EN 12516-1 and EN 12516-2 specify methods for determining the thickness of steel valve shells by tabulation and calculation methods respectively. EN 12516-3 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. EN 12516-4 specifies methods for calculating the thickness for valve shells in metallic materials other than steel.

The calculation method, EN 12516-2, is similar in approach to the former DIN 3840 where the designer is required to calculate the wall thickness for each point on the pressure temperature curve using the allowable stress at that 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 EN 12516-2. The formulae in EN 12516-2 consider the valve as a pressure vessel and ensure that there will be no excessive deformation or plastic instability.

The tabulation method, EN 12516-1, is similar in approach to ASME B16.34 (see Bibliography, reference [2]) in that the designer can look up the required minimum wall thickness dimension of the valve body from a table. The internal diameter of the inlet bore of the valve gives the reference dimension from which the tabulated wall thickness of the body is calculated.

The tabulated thicknesses in EN 12516-1 are calculated using the thin cylinder formula that is also used in EN 12516-2. The allowable stress used in the formula is equal to 120,7 MPa and the operating pressure, p_c, in MPa, varies for each PN and Class designation. EN 12516-1 gives these p_c values for all the tabulated PN and Class designations.

EN 12516-1 specifies PN, Standard Class and Special Class pressure temperature ratings for valve shells with 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 EN 12516-1.

Each tabulated pressure temperature rating is given a reference pressure designation to identify it.

The tabulation method gives one thickness for the body for each PN (see EN 12516-1:2014, 3.1 PN (Body)) or Class designation depending only on the inside diameter, D_i, of the body at the point where the thickness is to be determined.

The calculated pressure is limited by the ceiling pressure which sets up an upper boundary for high strength materials and limits the deflection.

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 group varies proportionally to the selected stresses of the material group to which the material belongs, using the simple rules above.

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.

The two methods are based on different assumptions, and as a consequence the detail of the analysis is different (see Bibliography, reference [3]). Both methods offer a safe and proven method of designing pressure-bearing components for valve shells.

1 Scope

This European Standard specifies the method for the strength calculation of the shell with respect to internal pressure of the valve.

2 Normative references

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

EN 19:2002, *Industrial valves — Marking of metallic valves*

EN 1092-1:2007+A1:2013, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 1: Steel flanges*

EN 1591-1:2013, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation*

EN 10269:2013, *Steels and nickel alloys for fasteners with specified elevated and/or low temperature properties*

EN 12266-1:2012, *Industrial valves — Testing of metallic valves — Part 1: Pressure tests, test procedures and acceptance criteria - Mandatory requirements*

EN 12266-2:2012, *Industrial valves — Testing of metallic valves — Part 2: Tests, test procedures and acceptance criteria - Supplementary requirements*

EN 13445-3:2014, *Unfired pressure vessels — Part 3: Design*

EN ISO 3506-1:2009, *Mechanical properties of corrosion-resistant stainless steel fasteners — Part 1: Bolts, screws and studs (ISO 3506-1)*

3 Symbols and units

The following symbols are used:

Table 1 — Symbols characteristics and units

Symbol	Unit	Characteristic
a_H	mm	lever arm for horizontal force
a_S	mm	lever arm for bolt force
a_V	mm	lever arm for vertical force
B	—	calculation coefficient to determine the thickness of the flange
$B_{01...03}$	—	calculation coefficient for oval and rectangular cross-sections
$B_{1...3}$	—	calculation coefficient for oval and rectangular cross-sections
B_5	—	correction factor for oval flanges
B_{FI}, B_{FII}	—	calculation coefficient for flat circular plates
B_h	—	calculation coefficient to determine the thickness of the flange
B_{MI}, B_{MII}	—	calculation coefficient for flat circular plates
B_{PI}, B_{PII}	—	calculation coefficient for flat circular plates

b	mm	double flange width
b_1	mm	minor width in oval and rectangular cross section
b_2	mm	major width in oval and rectangular cross section
b_{D1}, b_{D2}	mm	width of the seal
b'_1	mm	width in oval and rectangular cross section
b_D	mm	width of the seal
b_s	mm	effective width for reinforcement
C_x, C_y, C_z	—	calculation coefficient for covers made of flat plates
C	—	calculation coefficient for lens-shaped gaskets
c	mm	design allowance for bolts
c_1	mm	fabrication tolerance
c_2	mm	standardized corrosion and erosion allowance
d_o	mm	outside diameter
d_0, d'_0	mm	diameter in base body
d_{01}, d_{02}	mm	diameter for self-sealing closure
d_1	mm	diameter in branch
d_2	mm	diameter in further branch
d_4	mm	outside diameter of collar flange
d_A	mm	outside diameter of the plate/cover
d_a	mm	outside flange diameter
d_i	mm	inside diameter
d_f	mm	diameter of the biggest inscribed circle
d_k	mm	diameter in knuckle
d_K	mm	diameter in corner welds
d_L	mm	hole diameter
d'_L	mm	reduced bolt hole diameter
d_m	mm	mean diameter of the plate/cover
d_{mA}	mm	mean diameter of the face (see Figure 28)
d'_m	mm	mean diameter
d_D	mm	mean diameter of the seal
d_s	mm	required bolt diameter
d_t	mm	bold circle diameter / reference circle diameter
d_p	mm	diameter of centre of gravity
d_{ast}	mm	stuffing box outside diameter
d_{ist}	mm	stuffing box inside diameter
d_{S0}	mm	calculated bolt diameter without design allowance
d_V	mm	diameter of the vertical force at the cone
E	MPa	modulus of elasticity

E_D	MPa	modulus of elasticity for material of the seal
e_n	mm	wall thickness
e_{an}	mm	wall thickness (final / actual)
e_{acn}	mm	actual wall thickness less c1 and c2
e_{acF}	mm	thickness of flange neck
e_{cn}	mm	calculated theoretical minimum wall thickness, without c_1 and c_2
F_{DV}	N	minimum bolt force for the assembly condition
F_F	N	flange force
F_H	N	horizontal component force
F_S	N	bolt force for operating conditions
F_{SB}	N	minimum bolt force
F_{S0}	N	bolt force for assembly conditions
F_T	N	tensile force
F_V	N	vertical force at the cone
F_Z	N	additional force
f	MPa	nominal design stress
f_d	MPa	maximum value of the nominal design stress for normal operating load cases
$f_{d/t}$	MPa	nominal design stress for design conditions at temperature t °C
g_1, g_2	mm	welding throat depth
h	mm	plate thickness
h_0	mm	minimum height for the seating shoulder
h_1	mm	minimum height of the inserted ring
h_D	mm	minimum depth of the sealing ledge
h_r	mm	plate thickness
h_A	mm	height of flange hub
h_c	mm	plate thickness
h_F	mm	thickness of flange
h_N	mm	reduced plate thickness
k_c	—	welding factor
l	mm	length
$l_{0...3}$	mm	effective length for cylindrical bodies
l'	mm	length which is influenced by the entry nozzle
l'_0	mm	length for calculating body shapes in cross section II
∩l3	mm	length for calculating body shapes in cross section II
M	Nm	external moment
M_i	Nm	summary of moments M_P, M_F, M_M
M_a	Nm	external moment
M_{a0}	Nm	moment for assembly condition

M_{aB}	Nm	moment for operation condition
M_F	Nm	single force (point force)
M_i	Nm	moment
M_{max}	Nm	maximum bending moment
M_M	Nm	rim moment
M_P	Nm	resulting moment from internal pressure
M_r	Nm	bending moment in radial direction
M_t	Nm	bending moment in tangential direction
m	—	gasket coefficient
n	—	number of bolts
n_1	—	load carrying factor
p	MPa	pressure
p_c	MPa	calculation pressure
p_d	MPa	design pressure
p_F	MPa	contact pressure
P_s		centre of gravity
PS	MPa	maximum allowable pressure
R	mm	radius for calculating load cases
R_{eH}	MPa	upper yield strength
$R_{eH/t}$	MPa	upper yield strength at temperature t °C
R_i	mm	inner Radius of spherical cap
R_m	MPa	tensile strength
$R_{m/t}$	MPa	tensile strength at temperature t °C
$R_{m/T/t}$	MPa	creep rupture strength for T hours at temperature t °C
$R_{p0,2}$	MPa	0,2 % - proof strength
$R_{p0,2/t}$	MPa	0,2 % - proof strength at temperature t °C
$R_{p0,2/t\ Test}$	MPa	0,2 % - proof strength at test temperature t °C
$R_{p1,0/t\ Test}$	MPa	1,0 % - proof strength at test temperature t °C
$R_{p1,0}$	MPa	1,0 % - proof strength
$R_{p1,0/t}$	MPa	1,0 % - proof strength at temperature t °C
$R_{p1,0/T/t}$	MPa	1,0 % - creep proof strength for T hours at temperature t °C
r	mm	radius
r_0	mm	radius for calculating load cases
r_1	mm	radius for calculating load cases
r_o	mm	outside radius
r_i	mm	inside radius
r_D	mm	radius to the middle of the support plate for the seal
r_F	mm	radius to F_1

S_D	—	safety factor for gasket value
SF	—	safety factor
s	mm	distance of the centre of gravity of the half circular ring from the centreline
s_N	mm	thickness of weld
S_S	mm	centre of gravity
S_1, S_2	—	centre of gravity
s_1, s_2	mm	distance of the centre of gravity or distance
s_3	mm	distance
T	h	time
t	°C	temperature
t_d	°C	design temperature
U_D	mm	mean circumference
V	—	correction factor of bolt hole diameter
W, W_I, W_{II}, W_{III}	mm^3	flange resistance
W_{avI}, W_{avII}	mm^3	flange resistance in cross-section
W_{req1}	mm^3	flange resistance in operating condition
W_{req2}	mm^3	flange resistance in assembly condition
X	mm	distance variable
Y	mm	distance variable
Z		coefficient
Z_1	mm^3	coefficient
α	°	angle of lenticular gasket
α	—	form factor
β	—	calculation factor = α / δ
η	—	machining quality factor
μ	—	Poisson's ratio
δ	—	ratio of bolt forces against pressure forces
δ_1	—	proof stress ratio
φ	°	angle for corner welds
φ_k	°	angle in knuckle area
Φ	°	angle of body branch
Φ_A	°	angle for valve bodies with oblique branch
χ	—	calculation factor depending on the gasket material
σ	MPa	stress in the cross sections or branches
σ_{VU}	MPa	minimum sealing constant assembly state
σ_{VO}	MPa	maximum sealing constant assembly state
σ_{BO}	MPa	maximum sealing constant operating state

σ_{I}, σ_{II}, σ_{III}	MPa	stress in the cross section I, II, III

4 General conditions for strength calculation

Formulae (1) and (2) apply to mainly static internal pressure stressing. The extent to which these formulae 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:

$$e_0 = e_{c0} + c_1 + c_2 \quad (1)$$

$$e_1 = e_{c1} + c_1 + c_2 \quad (2)$$

$$e_{a0} \geq e_0 \quad (3)$$

$$e_{a1} \geq e_1 \quad (4)$$

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 1a and 2);

c_1 is a manufacturer tolerance allowance;

c_2 is a standardized corrosion and erosion allowance.

The values of the corrosion allowance are:

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

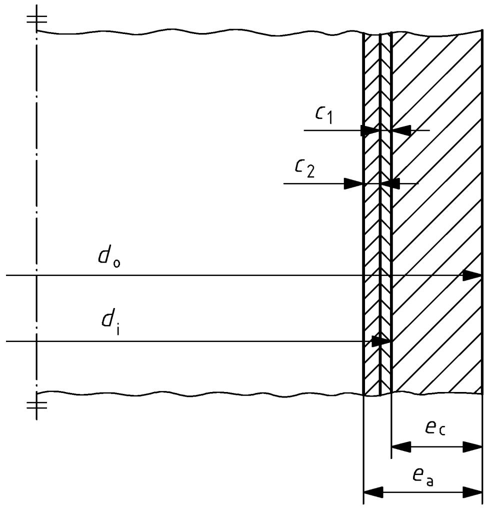
c_2 = 0 mm for all other steels;

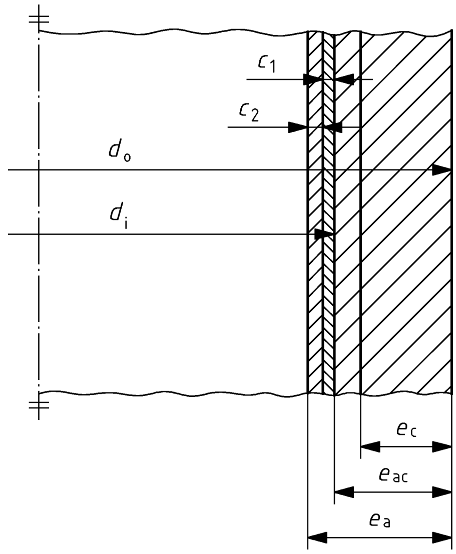
c_2 = 0 mm if $e_{c0} \geq 30$ mm or if $e_{c1} \geq 30$ mm.

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

$$e_{ac0} = e_{a0} - c_1 - c_2 \quad (5)$$

$$e_{ac1} = e_{a1} - c_1 - c_2 \quad (6)$$





a) for new design **b) for verification of existing wall thickness**

Key

e	wall thickness
d	diameter
a	actual
c	calculated
i	inner
o	outer
c_1	fabrication tolerance
c_2	corrosion allowance
e_c	calculated wall thickness
e_{ac}	actual minimum wall thickness less c_1 and c_2
e_a	actual wall thickness

$e_{ac} \geq e_c$

$e_{ac} \geq e_c + c_1 + c_2$

Figure 1 — Composition of section thickness and tolerance allowances

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 not be less than the maximum allowable pressure PS. In formulae, p_d is shortly written as p.

6 Nominal design stresses for pressure parts other than bolts

6.1 General

The nominal design stresses (allowable stresses) for steels (see PED 97/23/EC) with a minimum rupture elongation of ≥ 14 % and a minimum impact energy measured on a Charpy-V-notch impact test specimen of ≥ 27 J shall be calculated in accordance with Table 2.

The calculation of the test conditions is optional.

Table 2 — Nominal design stresses (allowable stresses)

Material	Design conditions	Creep conditions	Test conditions [b]
Steel as defined in 6.2	$f = \min (R_{p0,2/t} / 1,5 ; R_{m/20} / 2,4)$	$f = R_{m/100\,000/t} / 1,5$	$f = R_{p0,2/t_{Test}} / 1,05$
Austenitic steel and austenitic cast steel as defined in 6.2	$f = \min (R_{p1,0/t} / 1,5 ; R_{m/20} / 2,4)$	$f = R_{m/100\,000/t} / 1,5$	$f = R_{p1,0/t_{Test}} / 1,05$
Austenitic steel and austenitic cast steel as defined in 6.3 with rupture elongation ≥ 30 %	$f = R_{p1,0/t} / 1,5$	$f = R_{m/100\,000/t} / 1,5$	$f = R_{p1,0/t_{Test}} / 1,05$
Austenitic steel and austenitic cast steel as defined in 6.4 with rupture elongation ≥ 35 %	$f = \max [R_{p1,0/t} / 1,5; \min (R_{p1,0/t} / 1,2; R_{m/t} / 3,0)]$	$f = R_{m/100\,000/t} / 1,5$	$f = R_{p1,0/t_{Test}} / 1,05$
Cast steel as defined in 6.5	$f = \min (R_{p0,2/t} / 1,9; R_{m/20} / 3,0)$	$f = R_{m/100\,000/t} / 1,9$	$f = R_{p0,2/t_{Test}} / 1,33$
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]	$f = R_{m/100\,000/t} / 1,5$	$f = R_{p0,2/t_{Test}} / 1,05$

[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 $\cdot e_c$, however = 50 mm min. and the angle of the transition ≤ 30°.

[b] For the calculation of the test pressure, EN 12266–1 or EN 12266–2 shall be used.

Materials with lower elongation values and/or 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.

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

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 be 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.

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:

a) 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;

b) the smaller of the two values:

   1) 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;

   2) the minimum tensile strength $R_{m/t}$ at calculation temperature divided by the safety factor SF = 3,0.

6.5 Ferritic and martensitic 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 h value.

The nominal design stress calculated in 6.2 to 6.5 shall 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 h 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 h.

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 9 a) and Figure 9 b).

For the calculation of the surface-comparison, the completed wall thickness shall apply, c_1 and c_2 shall be stripped from the side in contact with the operating medium according to Figure 1 b).

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 Wall thickness of bodies and branches outside crotch area

7.2.1 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 formulae 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 = inside diameter

— r_i = inside radius

— d_o = outside diameter

— r_o = outside radius

7.2.2 Cylindrical bodies or branches

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

If this condition is not met, the procedure shall be according to 7.4.5, Figure 20.

$$e_c = \frac{d_i \cdot p}{(2 \cdot f - p) \cdot k_c} \qquad (7)$$

or

$$e_c = \frac{d_o \cdot p}{(2 \cdot f - p) \cdot k_c + 2 \cdot p} \qquad (8)$$

$d_o / d_i > 1{,}7$

$$e_c = \frac{d_i}{2}\left(\sqrt{\frac{f \cdot k_c + p}{f \cdot k_c - p}} - 1\right) \tag{9}$$

or

$$e_c = \frac{d_o}{2}\left(1 - \sqrt{\frac{f \cdot k_c - p}{f \cdot k_c + p}}\right) \tag{10}$$

Both formulae above are equivalent when $d_i = d_o - 2 \cdot e_c$

7.2.3 Spherical bodies or branches

d_o **/** d_i **≤ 1,2**

$$e_c = \frac{r_i \cdot p}{(2 \cdot f - p) \cdot k_c} \tag{11}$$

or

$$e_c = \frac{r_o \cdot p}{(2 \cdot f - p) \cdot k_c + p} \tag{12}$$

1,2 < d_o **/** d_i **≤ 1,5**

$$e_c = r_i \cdot \left[\sqrt{1 + \frac{2 \cdot p}{(2 \cdot f - p) \cdot k_c}} - 1\right] \tag{13}$$

or

$$e_c = r_o \cdot \frac{\sqrt{1 + \frac{2 \cdot p}{(2 \cdot f - p) \cdot k_c}} - 1}{\sqrt{1 + \frac{2 \cdot p}{(2 \cdot f - p) \cdot k_c}}} \tag{14}$$

The formulae above are equivalent when $r_i = r_o - e_c$

7.2.4 Conical bodies or branches

$$e_a / d_o > 0{,}005 \tag{15}$$

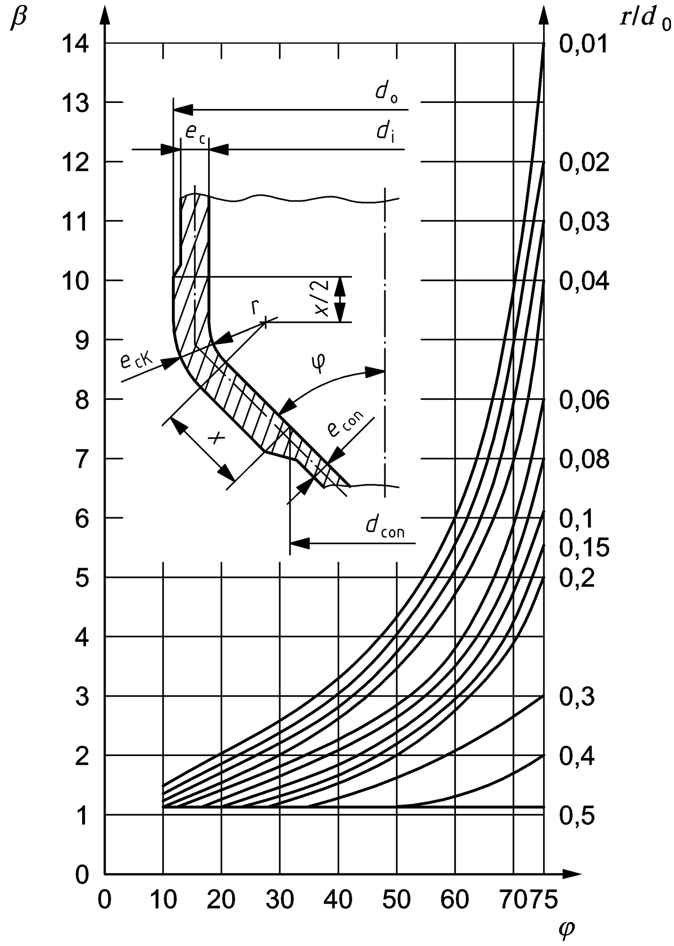


Figure 2 — Cone calculation coefficient

$$e_{\text{con}} = \frac{p \cdot d_{\text{con}}}{(2 \cdot f - p) \cdot k_{\text{c}}} \cdot \frac{1}{cos(\varphi)} \tag{16}$$

$$d_{\text{con}} = d_{\text{o}} - 2(e_{\text{cK}} + r(1 - \cos\varphi) + x \sin\varphi) \tag{17}$$

Wall thickness in the knuckle or in a corner weld:

$$e_{\text{cK}} = \frac{d_{\text{o}} \cdot p \cdot \beta}{4 \cdot f \cdot k_{\text{c}}} \tag{18}$$

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

$$x = \sqrt{d_{\text{o}} \cdot e_{\text{ac}}} \tag{19}$$

$$e_{\text{ac}} \leq e_{\text{a}} - c_1 - c_2 \tag{20}$$

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 $\varphi \leq 30°$, $e_{\text{cK}} \leq 20$ mm and double joint weld, β shall be read off Figure 2 by taking for the ratio $r / d_{\text{o}} = 0{,}01$.

For corner welds, diameter d_{con} is equal to the inside diameter of the wide end.

In case of flat cones with a knuckle and φ > 70°:

$$e_{\text{con}} = 0{,}3 \cdot (d_{\text{o}} - r) \cdot \frac{\varphi}{90} \cdot \sqrt{\frac{p}{f \cdot k_{\text{c}}}} \quad (21)$$

Table 3 — Cone calculation coefficient

Angle	***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**	φ
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 value of β.

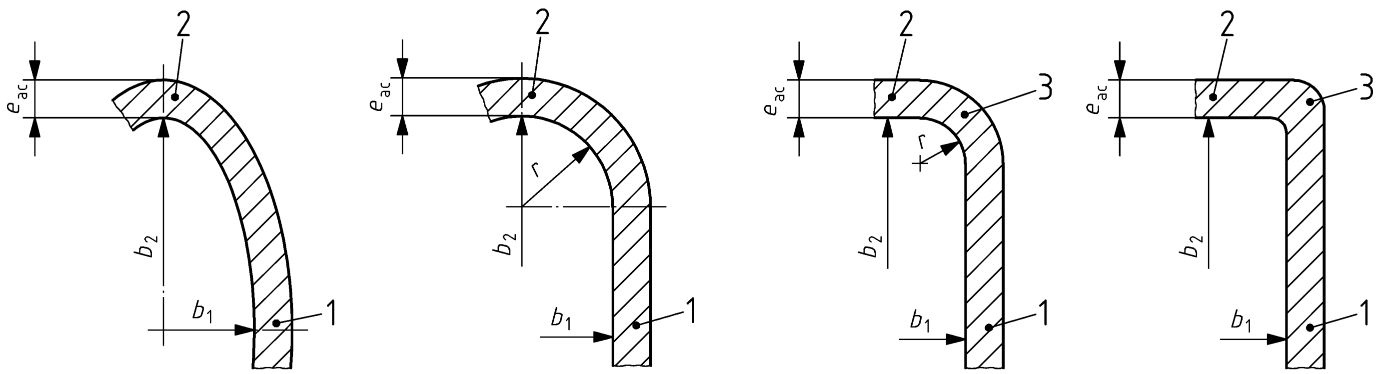
7.2.5 Bodies or branches with oval or rectangular cross-sections

7.2.5.1 General

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

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

In the case of oval shaped cross-sections (see Figure 3 a) and Figure 3 b)) and of rectangular shapes with or without radiusing of the corners (see Figure 3 c) and Figure 3 d)), 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 for $r > e_{\text{ac}}$** **d) Rectangular, corner not radiused**

Figure 3 — Cross-sections

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

$$e_{c0} = \frac{p \cdot b_2}{2f} \cdot \sqrt{B_{0n}^2 + \frac{4 \cdot k_c \cdot f}{p}} \cdot B_n \quad (22)$$

The calculation shall be carried out in respect of locations 1 and 2 (designated in Figure 3 a) and Figure 3 b) for oval-shaped cross-sections), and in respect of locations 1 and 3 (designated in Figure 3 c) and Figure 3 d) 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.

The calculation coefficient B_{0n}, which is a function of the normal forces, shall be:

$B_{01} = b_1 / b_2$ for location 1 (23)

$B_{02} = 1$ for location 2 (24)

For Location 3, B_{03} can be obtained from Figure 4 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 Formula (25):

$$B_{03} = \left[1 - \frac{2r}{b_2}\,(1 - \sin\varphi_k)\right] \sin\varphi_k + \left[\frac{b_1}{b_2} - \frac{2r}{b_2}\,(1 - \cos\varphi_k)\right] \cos\varphi_k \quad (25)$$

with $\tan\varphi_k = \dfrac{1 - 2r/b_2}{\dfrac{b_1}{b_2} - \dfrac{2r}{b_2}}$ (26)

NOTE In case of Figure 3 b) ($b_1 = 2r$): $\varphi_k = 90°$ and $B_{03} = 1$.

7.2.5.2 Oval-shaped cross sections

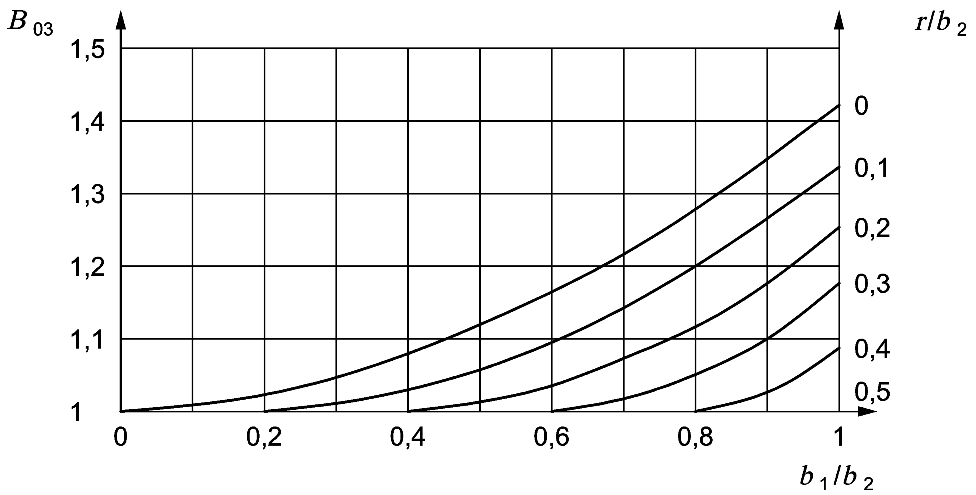


Figure 4 — Calculation coefficient B_{03} for location 3

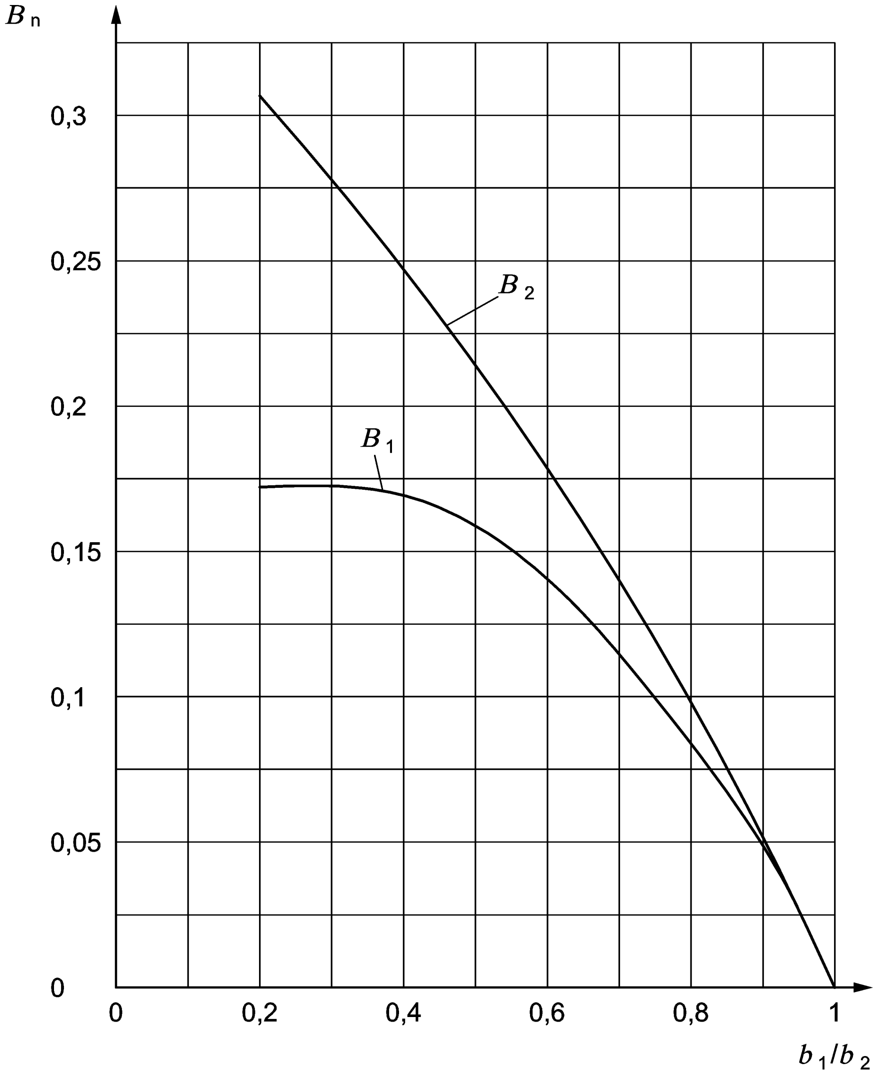
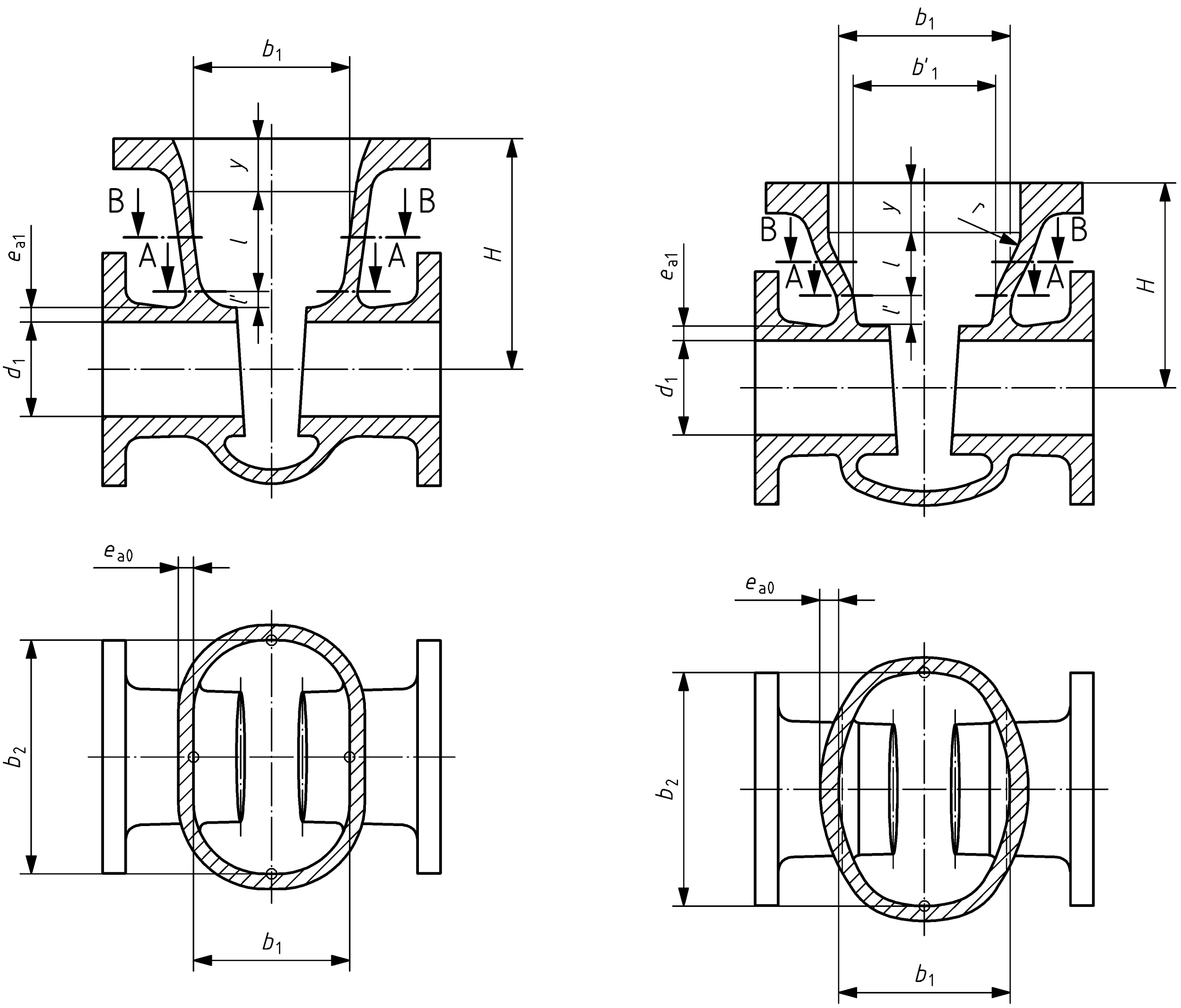


Figure 5 — Calculation coefficient B_n for oval-shaped cross-sections



a) Design A **b) Design B**

Figure 6 — Examples of changes in cross-section B - B in oval basic bodies

The calculation coefficients B_n which are dependent on the bending moments plotted in Figure 5 as a function of ratio b_1 / b_2 for oval-shaped cross-sections for locations 1 and 2. These curves correspond to the formulae below:

$$B_1 = \frac{1-k_E^2}{6} \cdot \frac{K'}{E'} - \frac{1-2\,k_E^2}{6} \qquad (27)$$

$$B_2 = \frac{1+k_E^2}{6} - \frac{1-k_E^2}{6} \cdot \frac{K'}{E'} \qquad (28)$$

with $k_E^2 = 1 - \left(\frac{b_1}{b_2}\right)^2 \qquad (29)$

These values result from the analytical solution of the formulae 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 formulae may also be used for $b_1 / b_2 \geq 0{,}5$:

$$B_1 = \left(1 - \frac{b_1}{b_2}\right)\left(0{,}625 - 0{,}435 \cdot \sqrt{1 - \frac{b_1}{b_2}}\right) \quad (30)$$

$$B_2 = \left(1 - \frac{b_1}{b_2}\right)\left[0{,}5 - 0{,}125 \cdot \left(1 - \frac{b_1}{b_2}\right)\right] \quad (31)$$

The calculation coefficients are also valid for changes in cross-section in oval basic bodies (e.g. for gate valves in accordance with Figure 6, 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:

$$l = H - y - \left(\frac{d_1}{2} + e_{ac1}\right) - l' \quad (32)$$

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

$$l' = 1{,}25 \cdot \sqrt{d'_m \cdot e_{ac0}} \quad (33)$$

with $$d'_m = \frac{b'_1 + b_2}{2} \quad (34)$$

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_{a0}.

7.2.5.3 Rectangular cross sections

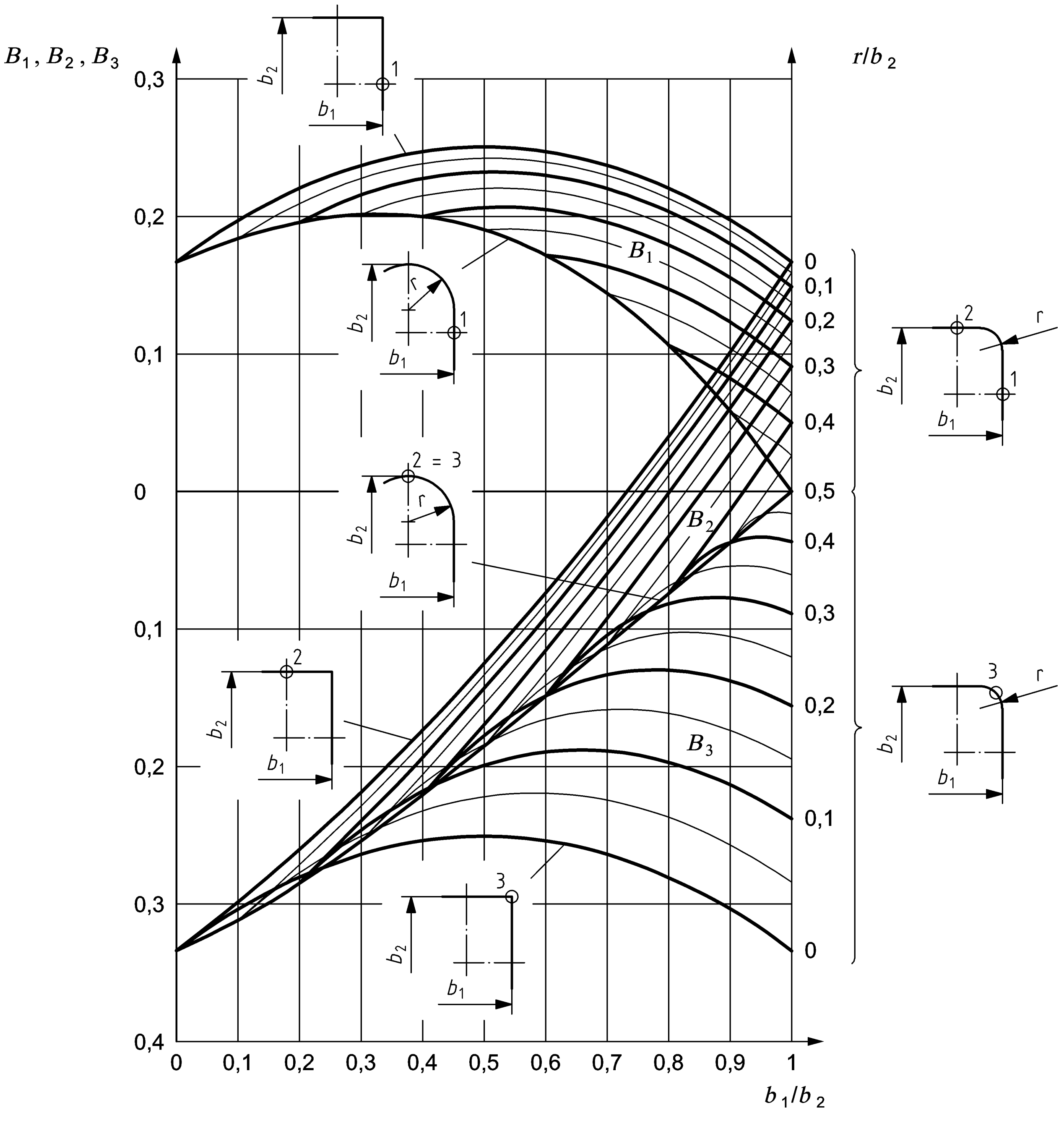


Figure 7 — Calculation coefficient B_n for rectangular cross-sections

The calculation coefficients B_n for rectangular cross-sections are plotted in Figure 7 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 Formula (22) as positive values.

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

$$B_1 = \frac{1}{6} \cdot \frac{1 - 2 \cdot \left(\frac{b_1}{b_2}\right)^3 + 3 \cdot \left(\frac{b_1}{b_2}\right) - 3 \cdot \frac{2r}{b_2} \cdot \left(2 - \frac{\pi}{2}\right) - 3 \cdot \left(\frac{2r}{b_2}\right)^2 \cdot \left(1 + \frac{b_1}{b_2}\right) \cdot (\pi - 3) + \left(\frac{2r}{b_2}\right)^3 \cdot \left(\frac{9}{2}\pi - 14\right)}{1 + \frac{b_1}{b_2} - \frac{2r}{b_2} \cdot \left(2 - \frac{\pi}{2}\right)} \quad (35)$$

$$B_2 = B_1 - \frac{1}{2}\left[1-\left(\frac{b_1}{b_2}\right)^2\right] \quad (36)$$

$$B_3 = \frac{1}{2}\left[1-2\frac{2r}{b_2}\cdot(1-\sin\varphi_k)+\frac{4r^2}{b_2^2}\cdot(3-2\sin\varphi_k-2\cos\varphi_k)-2\cdot\frac{b_1}{b_2}\cdot\frac{2r}{b_2}\cdot(1-\cos\varphi_k)\right]-B_1 \quad (37)$$

where Formula (26) shall apply to the angle of the maximum moment.

For short valve bodies (e.g. design A or B of Figure 6) 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 Formula (22) becomes:

$$e_c = e_{c0}\cdot k \quad (38)$$

The correction factor k is obtained from Formula (40) below, by analogy to the damping behaviour of the stresses in cylindrical shells, taking into consideration the experimental investigation results on non-circular casings:

$$k = 0{,}48\cdot\sqrt[3]{\frac{l^2}{d_m\cdot e_{c0}}} \quad (39)$$

with $0{,}6 \le k \le 1$ (40)

This function is plotted in Figure 8 as a function of $l^2 / d_m \cdot e_{c0}$.

d_m shall be entered in the formula at a value $d_m = (b_1 + b_2) / 2$, and e_{c0} corresponds to Formula (22). In the case of changes in cross-section over length l, (e.g. in accordance with Figure 6, 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.

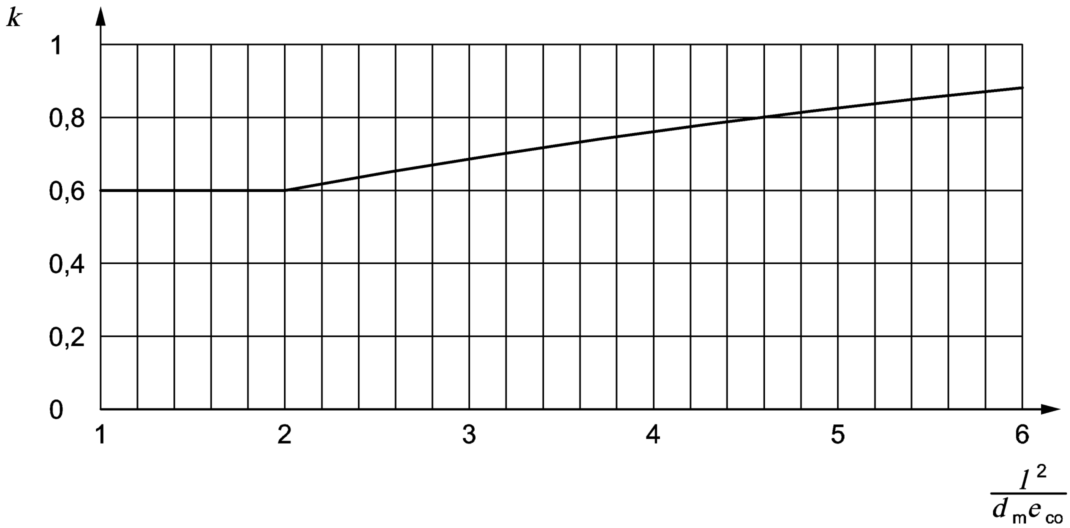


Figure 8 — Correction factor k for short casing bodies

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 Formula (22) or Formula (38), 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.

7.3 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 9 to 20.

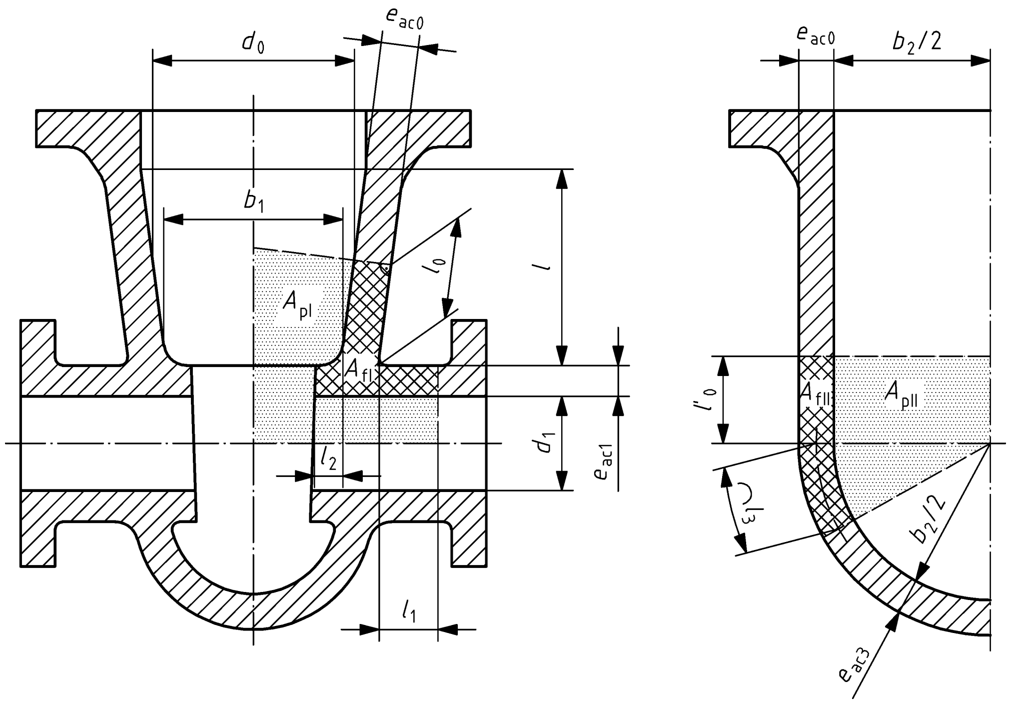


Figure 9 — Calculation procedure in the crotch area

According to Figure 9 the equilibrium of forces corresponds to the formula:

$$p \cdot A_p = f \cdot A_f \cdot k_c \tag{41}$$

where

$p \cdot A_{p\,I}$ or $p \cdot A_{p\,II}$	is the pressure loading area;
$f \cdot A_{f\,I}$ or $f \cdot A_{f\,II}$	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 Figures 9 to 20.

For the allowable value of *f*, see Clause 6.

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

Condition: e_{a0} body ≥ e_{a1} branches, if this is not possible: e_{a1} branches = e_{a0} body in the whole A_f region including the distances *l*.

Table 4 — Calculation formulae

circular cross-section	Diameter ratio $d_1 / d_0 < 0,7$ $p \cdot \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \cdot k_{\mathrm{c}}} + \frac{1}{2} \right] \leq f$	(42)
non-circular cross-section	$p \cdot \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \cdot k_{\mathrm{c}}} + \frac{1}{2} \right] \leq f / 1,2$ $p \cdot \left[\frac{A_{\mathrm{p\,II}}}{A_{\mathrm{f\,II}} \cdot k_{\mathrm{c}}} + \frac{1}{2} \right] \leq f / 1,2$	(43) (44)
circular cross-section	Diameter ratio $d_1 / d_0 \geq 0,7$ $p \cdot \left[\frac{A_{\mathrm{p\,I}}}{A_{\mathrm{f\,I}} \cdot k_{\mathrm{c}}} + \frac{1}{2} \right] \leq f$ $p \cdot \left[\frac{A_{\mathrm{p\,II}}}{A_{\mathrm{f\,II}} \cdot k_{\mathrm{c}}} + \frac{1}{2} \right] \leq f$	(45) (46)
circular cross-section	Diameter ratio $d_1 / d_0 \geq 0,7$ and $e_{\mathrm{a1}} / e_{\mathrm{a0}} < d_1 / d_0$ — Formula (45) — Formula (46) — additional for Section II: (Figure 9b) $p \cdot \left[\frac{d_0 + e_{\mathrm{ac0}}}{2 \cdot e_{\mathrm{ac0}}} + 0,2 \cdot \frac{d_1 + e_{\mathrm{ac1}}}{e_{\mathrm{ac1}}} \cdot \sqrt{\frac{d_0 + e_{\mathrm{ac0}}}{e_{\mathrm{ac0}}}} \right] \leq 1,5 \cdot f$	(47)
non-circular cross-section	— Formula (43) — Formula (44) — additional for Section II: $p \cdot \left[\frac{b_2 + e_{\mathrm{ac0}}}{2 \cdot e_{\mathrm{ac0}}} + 0,25 \cdot \frac{d_1 + e_{\mathrm{ac1}}}{e_{\mathrm{ac1}}} \cdot \sqrt{\frac{b_2 + e_{\mathrm{ac0}}}{e_{\mathrm{ac0}}}} \right] \leq 1,5 \cdot f$	(48)

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

7.4.1 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 10 to 20). Reduced seats such as shown in Figure 10 shall not be taken into account. Due to the complicated geometrical shapes of bodies in accordance with Figures 10 to 14, 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 10). For small transition radii, it is sufficient to start from the intersection of the linearly extended contours of the bodies (see Figure 14). 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 9, 10 and 17). 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 formulae for effective lengths.

For all valve body shapes in cross-section II (Figure 9 b)) it is:

$$l'_0 = 1{,}25 \cdot \sqrt{(b_2 + e_{ac0}) \cdot e_{ac0}} \tag{49}$$

$$l_3 = \sqrt{(b_2 + e_{ac3}) \cdot e_{ac3}} \tag{50}$$

7.4.2 Cylindrical valve bodies

The effective lengths for cylindrical bodies e.g. in accordance with Figure 10 are:

$$l_0 = \sqrt{(d_0 + e_{ac0}) \cdot e_{ac0}} \tag{51}$$

$$l_1 = 1{,}25 \cdot \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \tag{52}$$

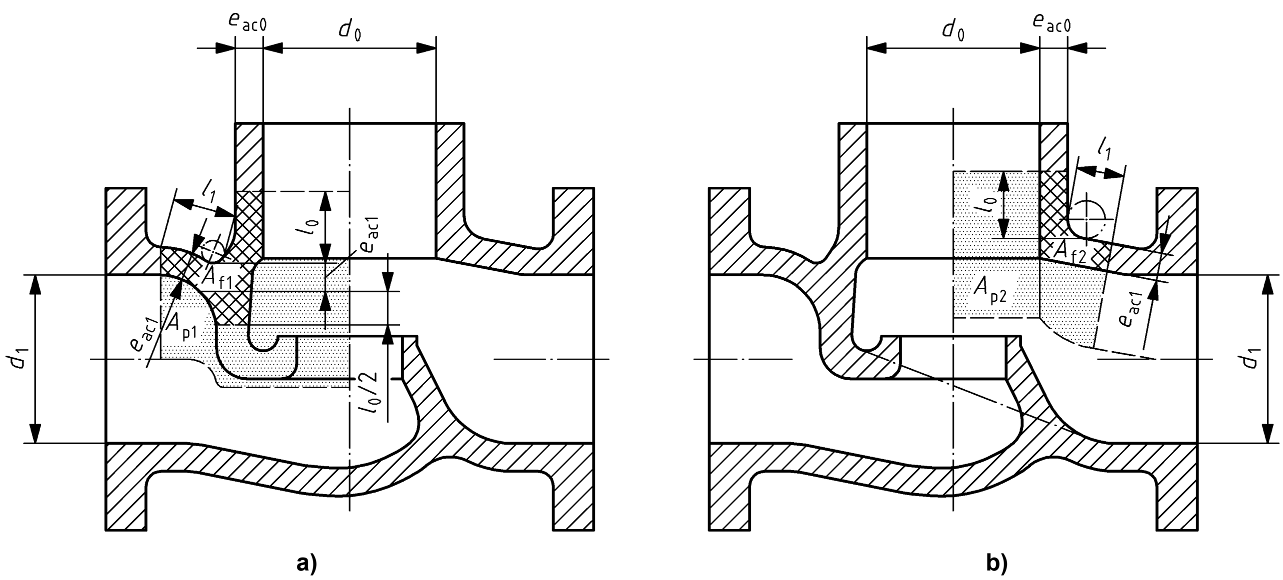
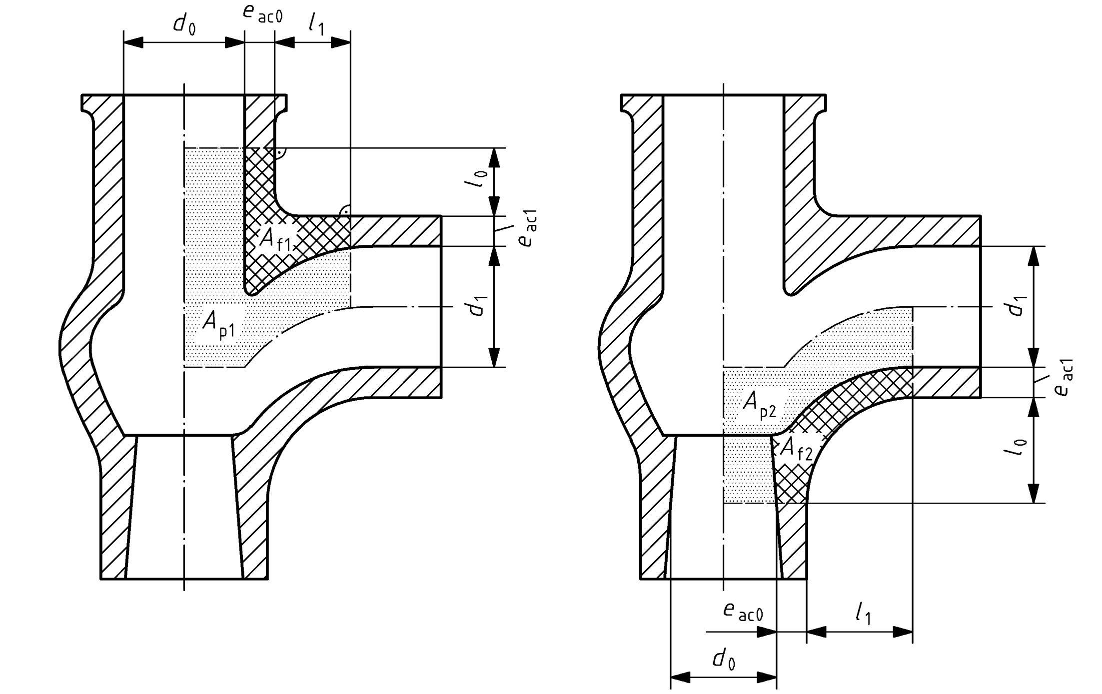
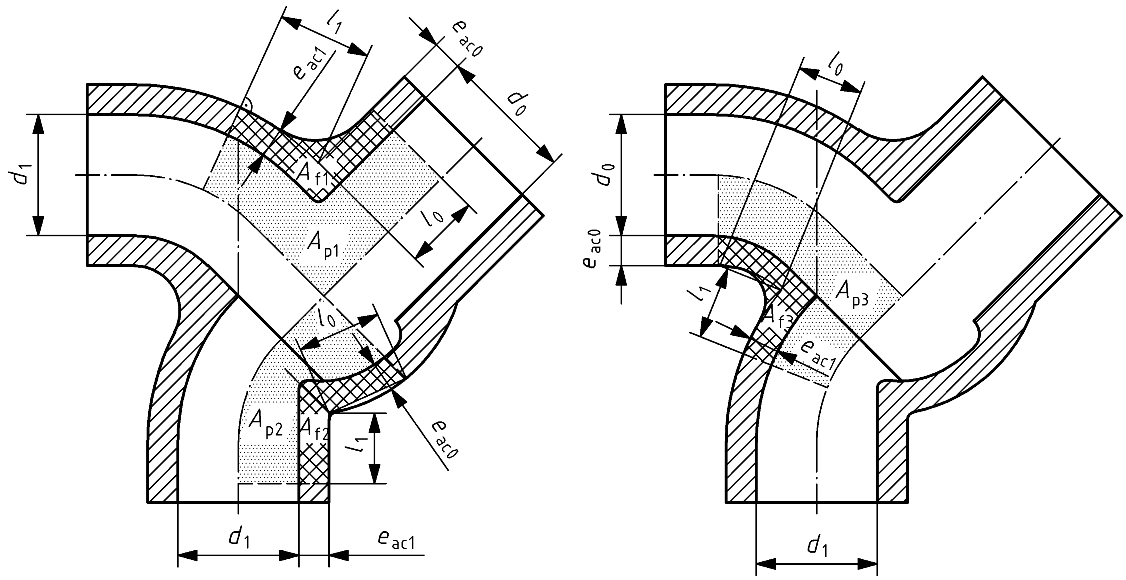


Figure 10 — Cylindrical valve body



a) b)

Figure 11 — Angle valve



a) b)

Figure 12 — Angle screw-down valve

For cylindrical valve bodies with oblique basic body or branch (e.g. in accordance with Figure 13) with $\phi_A \geq 45°$, instead of Formula (51) the following formula shall be used for l_0:

$$l_0 = \left(1 + 0,25 \cdot \frac{\phi_A}{90°}\right) \cdot \sqrt{(d_0 + e_{ac0}) \cdot e_{ac0}} \tag{53}$$

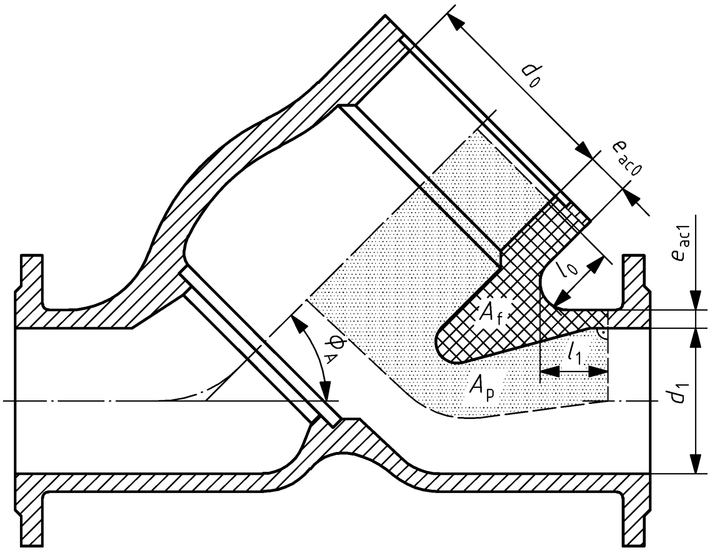


Figure 13 — Cylindrical valve body with oblique branch

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 11 b) and Figure 14).

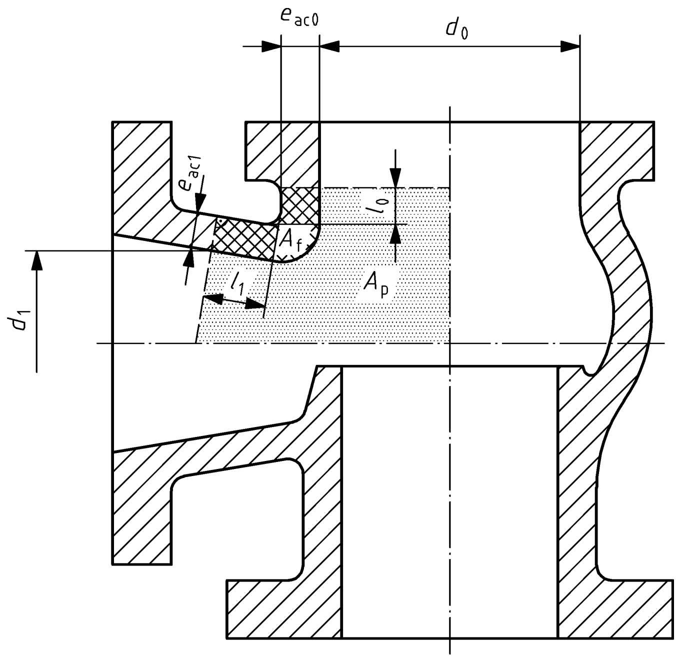


Figure 14 — Angle pattern valve body

7.4.3 Spherical valve bodies

For branches in spherical valve bodies with d_1 / d_0 or with d_2 / d_0 ≤ 0,5, the effective lengths l_0 shall be determined using Formula (51) on condition that

$$l_0 \leq 0{,}5 \cdot l_3 \qquad (54)$$

(see Figure 15 a))

The corresponding length in the branch is:

$$l_1 = \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \quad (55)$$

In cases, where

$$d_1 / d_0 > 0{,}5 \text{ and} \quad (56)$$

$$d_2 / d_0 > 0{,}5 \quad (57)$$

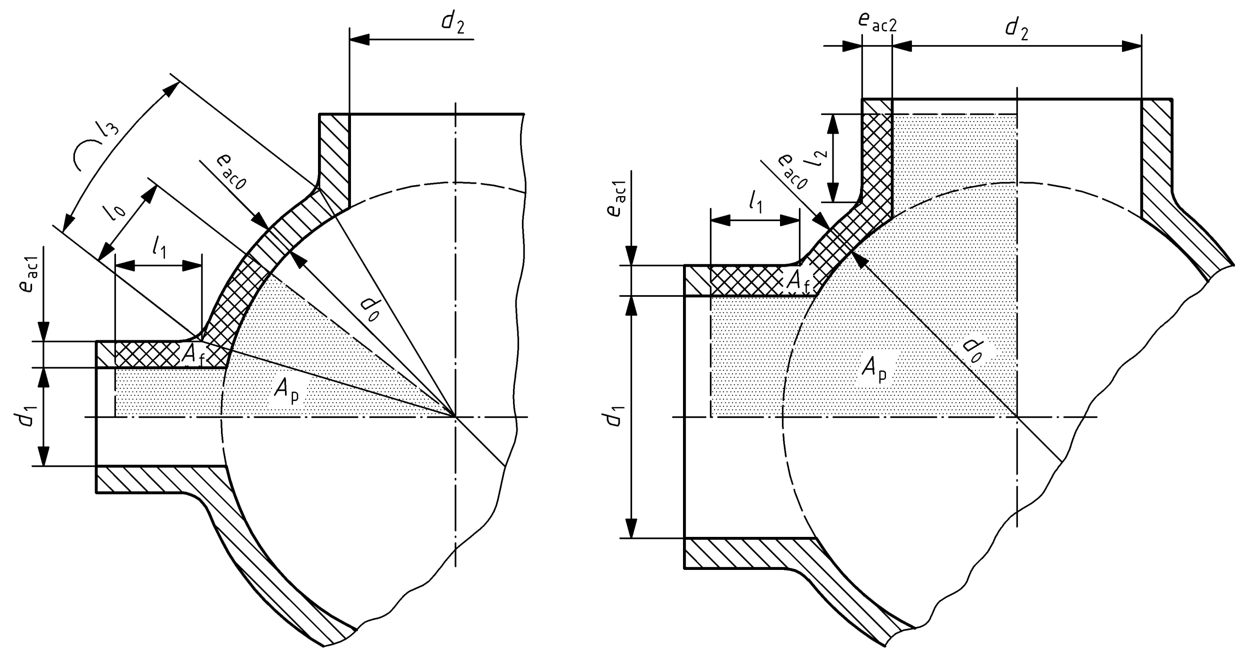
the pressure-containing area A_p and the stress area A_f shall be determined for both branches together in accordance with Figure 15, type b).

The effective lengths shall be determined as follows:

$$l_1 = \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \quad (58)$$

$$l_2 = \sqrt{(d_2 + e_{ac2}) \cdot e_{ac2}} \quad (59)$$

l_0 corresponds to the effective length between the branches.



a) **b)**

Figure 15 — Spherical valve body

7.4.4 Oval and rectangular cross-sections

For valve bodies with oval or rectangular cross-sections (see Figure 3), the lengths in accordance with Figure 9 shall be determined as follows:

$$l_0 = \sqrt{(b_1 + e_{ac0}) \cdot e_{ac0}} \quad (60)$$

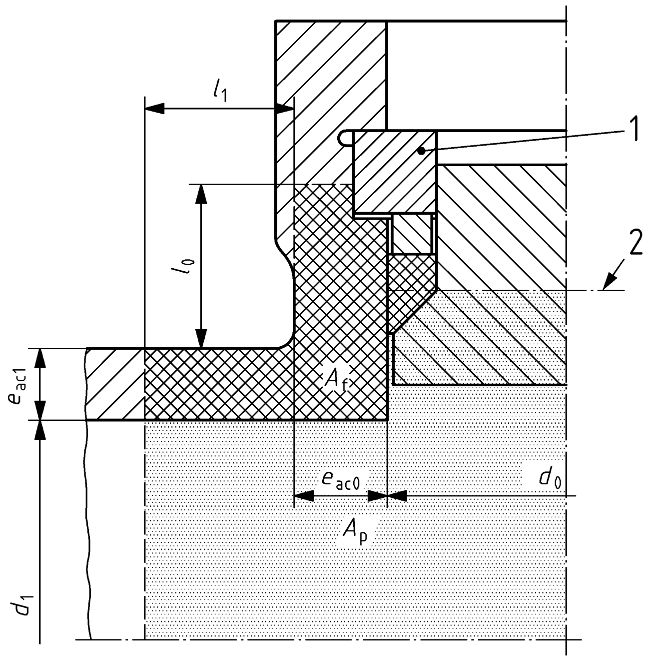
$$l_1 = 1{,}25 \cdot \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \quad (61)$$

7.4.5 Details

For design types with recesses (e.g. Figure 16), 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 16 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 16. 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 16 — Example of a closure

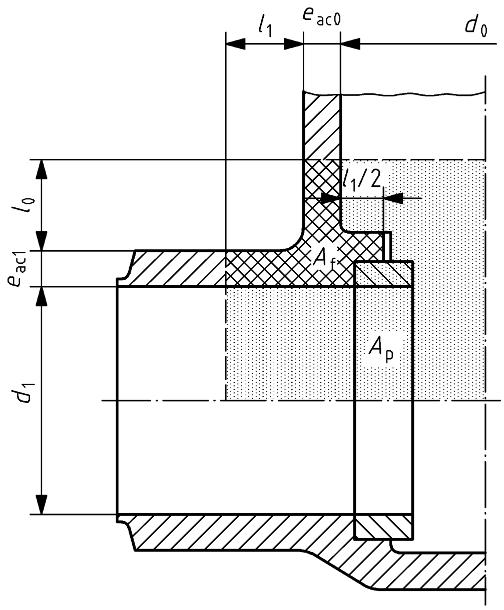


Figure 17 — Example of an end connection

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

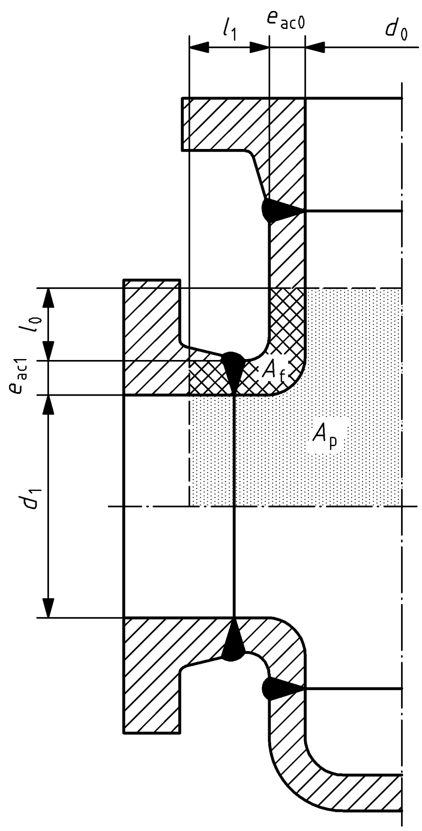
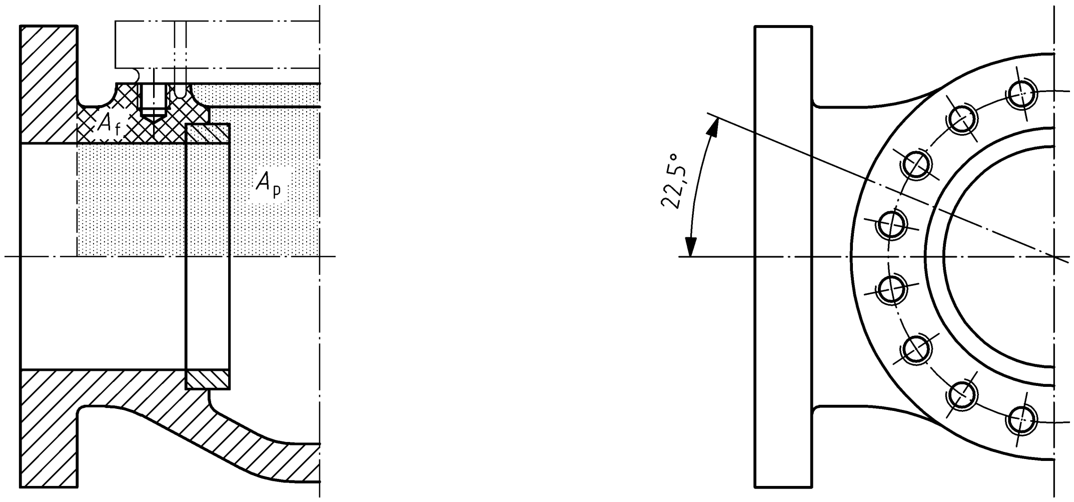


Figure 18 — Example of an end connection

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 19).

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



a) Calculated cross-section **b) Top view**

Figure 19 — Example of a flanged connection with blind holes

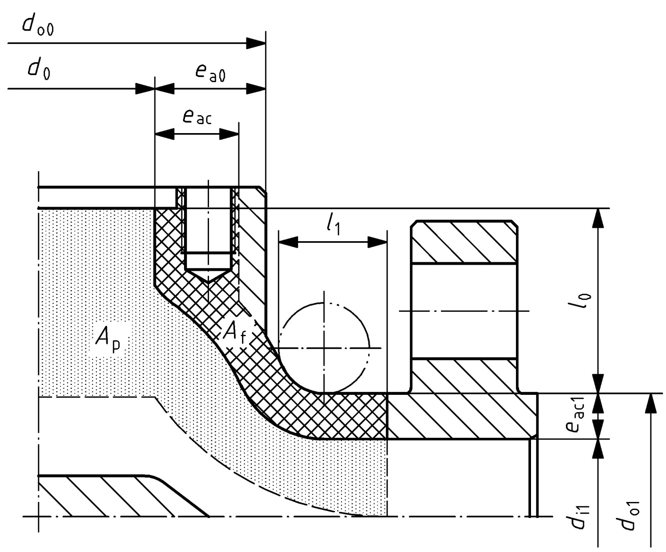


Figure 20 — Example of a thick-walled body

If the ratio of diameters is d_o / d_i > 1,7, the wall thickness for the surface comparison shall be applied with $e_{ac} = \left(\left(d_i \cdot 1{,}7\right) - d_i\right) / 2$ at the most.

In case of high mean wall temperature for the shell (more than 250 °C) or in the presence of severe temperature gradients through the shell, the use of reinforcing plates (see Figure 21 and Figure 22) shall be avoided; if it is necessary then the material of the reinforcing plate shall be of the same quality of shell material, and special measures and warnings shall be taken to avoid thermal stress concentrations. 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:

$$b_s = n_1 \cdot \sqrt{(d_0 + e_{ac0}) \cdot e_{ac0}} \quad (62)$$

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 22, design A where n_1 = 0,8 may be used for calculation.

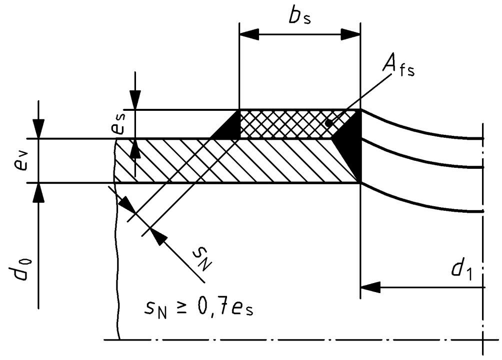
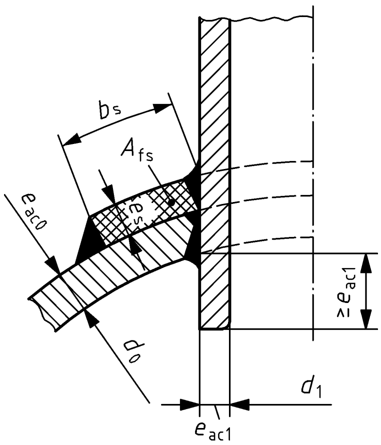
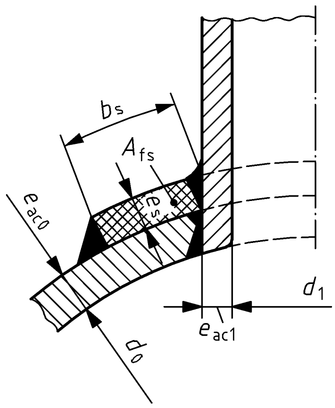


Figure 21 — Example of opening reinforcement



a) Design A



b) Design B

Figure 22 — Examples of opening reinforcement

8 Calculation methods for 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;

— dished heads.

8.2 Covers made of flat plates

8.2.1 General

The following formulae apply to plates with a plate thickness h_c / diameter d_D ratio ≤ 1/4.

The plate thickness h_c is calculated in accordance with Formula (63):

$$h_c = C_x \cdot C_y \cdot C_z \cdot d_D \cdot \sqrt{\frac{p}{f}} + c_1 + c_2 \qquad (63)$$

C_x, C_y, C_z are calculation coefficients depending on

— different diameter ratios,

— the ratio δ of bolt forces against pressure forces.

$$\delta = 1 + 4 \cdot \frac{m \cdot b_D \cdot S_D}{d_D} \qquad (64)$$

where

S_D is equal to 1,2 for operating conditions;

m is the gasket coefficient, see Annex A.

To find out the coefficients C_x, C_y, C_z and the diameter d_D use the Figures 23 to 25 and the indications listed in these figures.

In cases of gaskets not subjected to direct loading, the diameter d_D shall be replaced by d_{mA}.

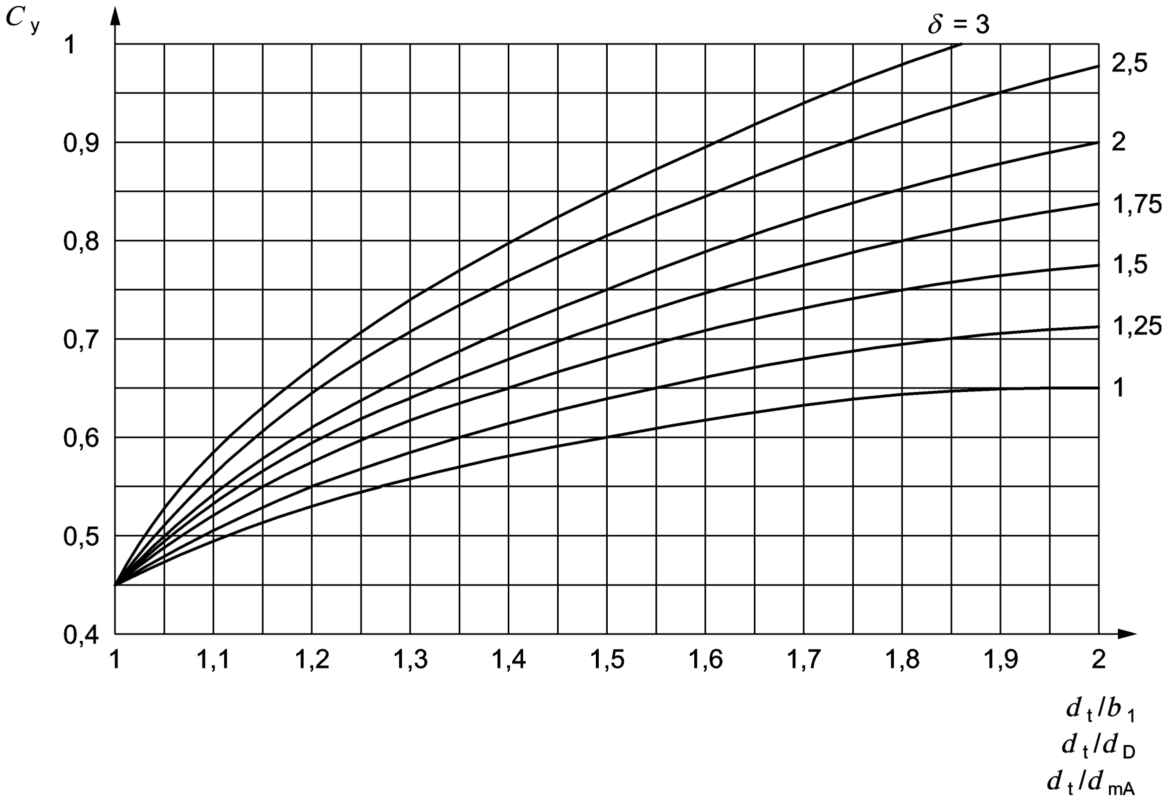


Figure 23 — Calculation coefficient C_y for flat plates with supplementary marginal moment acting in the same sense as the pressure load

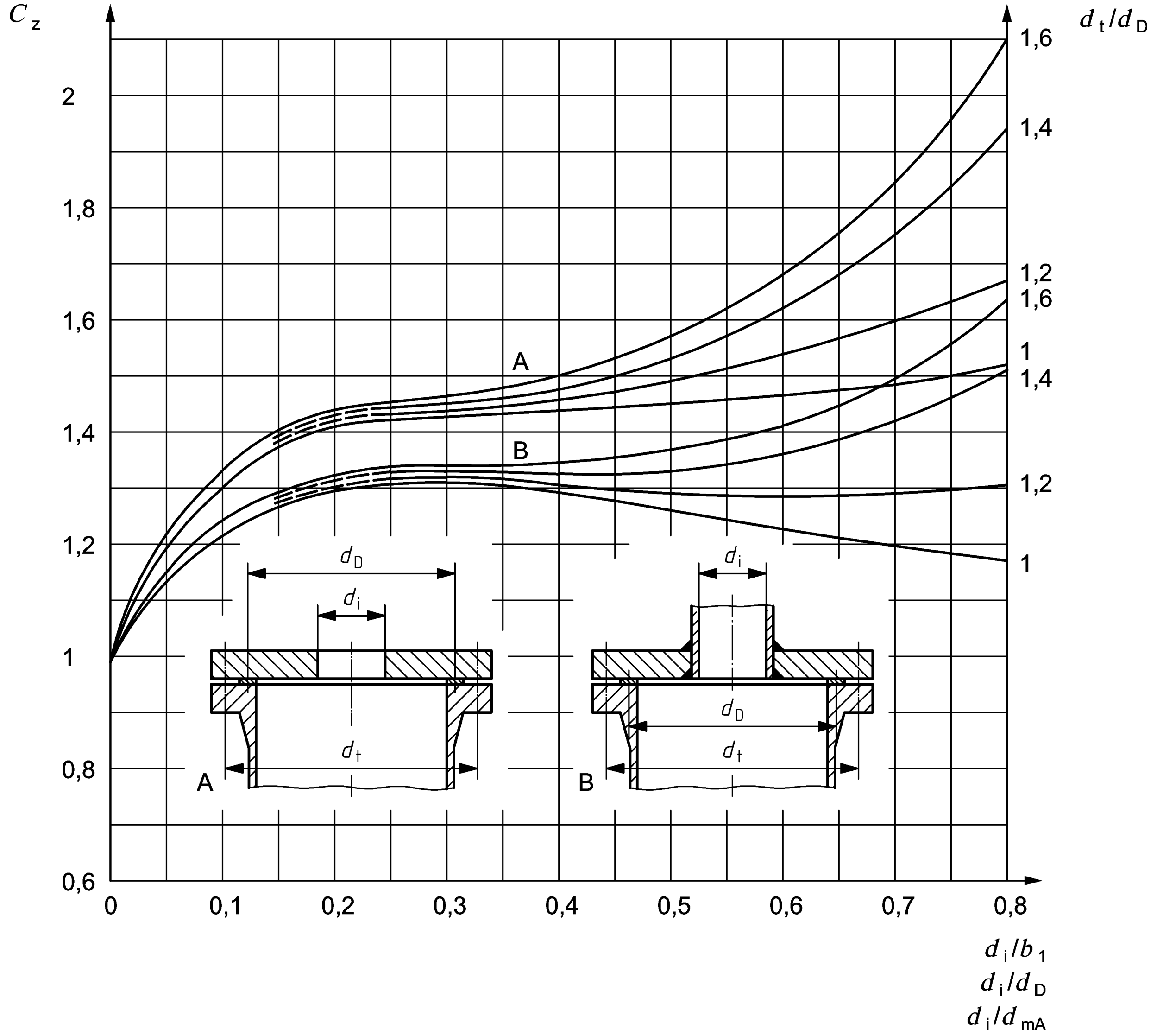


Figure 24 — Opening factor C_z for flat plates with additional marginal moment

In case of force shunt, d_D shall be replaced by d_{mA} (see Figures 28 and 30).

Type A

d_i inside diameter of opening

d_t pitch circle diameter

d_D mean gasket diameter or support diameter d_{mA}

b_1 short side of an elliptical end

$$C_z = \begin{cases} \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij}\cdot\left(\frac{d_i}{d_D}\right)^{i-1}\cdot\left(\frac{d_t}{d_D}\right)^{j-1} & \left| \begin{array}{l} 0 \quad < \left(\frac{d_i}{d_D}\right) \le 0,8 \\ 1,0 \quad \le \left(\frac{d_t}{d_D}\right) \le 1,6 \end{array} \right. \\ \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij}\cdot\left(\frac{d_i}{b_1}\right)^{i-1}\cdot\left(\frac{d_t}{b_1}\right)^{j-1} & \left| \begin{array}{l} 0 \quad < \left(\frac{d_i}{d_D}\right) \le 0,8 \\ 1,0 \quad \le \left(\frac{d_t}{d_D}\right) \le 1,6 \end{array} \right. \end{cases} \qquad (65)$$

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_i inside diameter of opening

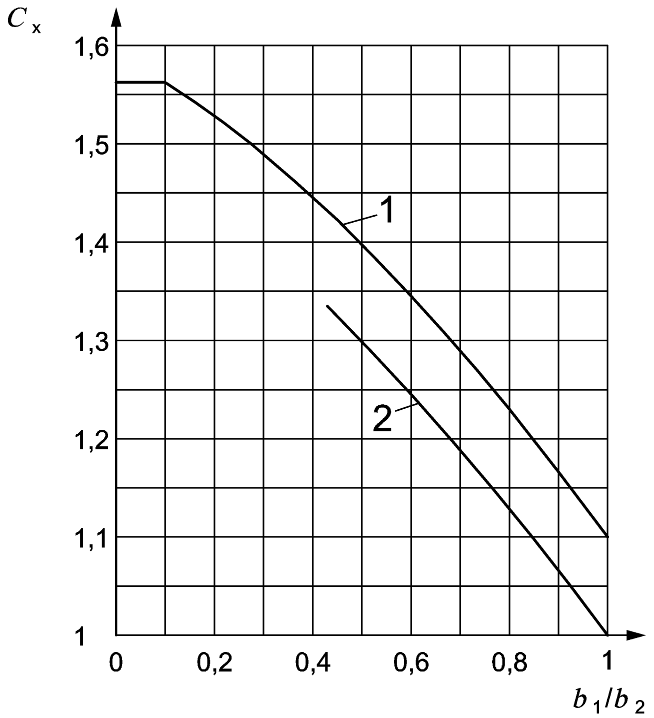
d_t pitch circle diameter

d_D mean gasket diameter or support diameter d_{mA}

b_1 short side of an elliptical end

$$C_z = \begin{cases} \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij}\cdot\left(\frac{d_i}{d_D}\right)^{i-1}\cdot\left(\frac{d_t}{d_D}\right)^{j-1} & \begin{matrix} 0 < \left(\frac{d_i}{d_D}\right) \le 0{,}8 \\ 1{,}0 \le \left(\frac{d_t}{d_D}\right) \le 1{,}6 \end{matrix} \\ \sum_{i=1}^{6}\sum_{j=1}^{4} A_{ij}\cdot\left(\frac{d_i}{b_1}\right)^{i-1}\cdot\left(\frac{d_t}{b_1}\right)^{j-1} & \begin{matrix} 0 < \left(\frac{d_i}{d_D}\right) \le 0{,}8 \\ 1{,}0 \le \left(\frac{d_t}{d_D}\right) \le 1{,}6 \end{matrix} \end{cases} \quad (66)$$

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;



Key

1 rectangular plates

2 elliptical flat plates

Figure 25 — Calculation coefficient C_x for rectangular (1) or elliptical flat plates (2)

Rectangular plates

b_1 short side of the rectangular plate

b_2 long side of the rectangular plate

$$C_x = \begin{cases} \sum_{i=1}^{4} A_i \cdot \left(\frac{b_1}{b_2}\right)^{i-1} & \left| 0{,}1 < \left(\frac{b_1}{b_2}\right) \leq 1{,}0 \right. \\ 1{,}562 & \left| 0 < \left(\frac{b_1}{b_2}\right) \leq 0{,}1 \right. \end{cases} \qquad (67)$$

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

b_1 short side of the elliptical plate

b_2 long side of the elliptical plate

$$C_X = \left\{ \sum_{i=1}^{4} A_i \cdot \left(\frac{b_1}{b_2}\right)^{i-1} \mid 0{,}43 \leq \left(\frac{b_1}{b_2}\right) \leq 1{,}0 \right\} \qquad (68)$$

A_1 = + 1,489 146 00

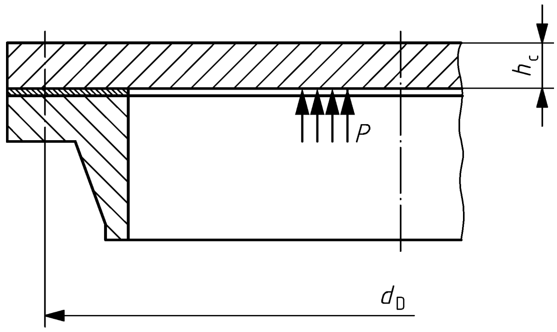
A_2 = – 0,239 349 90

A_3 = – 0,335 179 80

A_4 = + 0,085 211 76

8.2.2 Circular cover without opening, with

a) full face gasket:



C_x 1,0

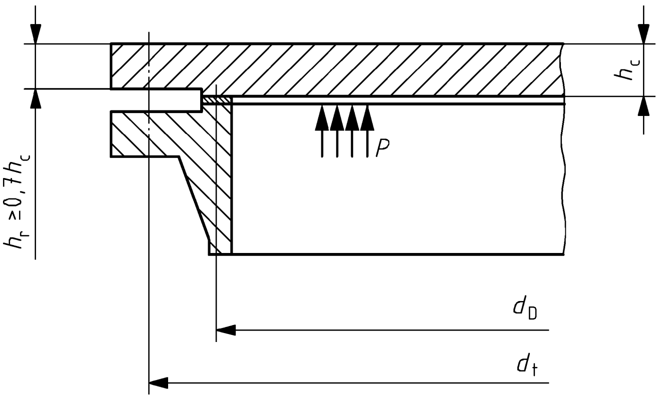
C_y 1,0

C_z 0,35

δ see Formula (64)

Figure 26 — Cover (direct loading) with full face gasket

b) gasket entirely within the bolt circle:



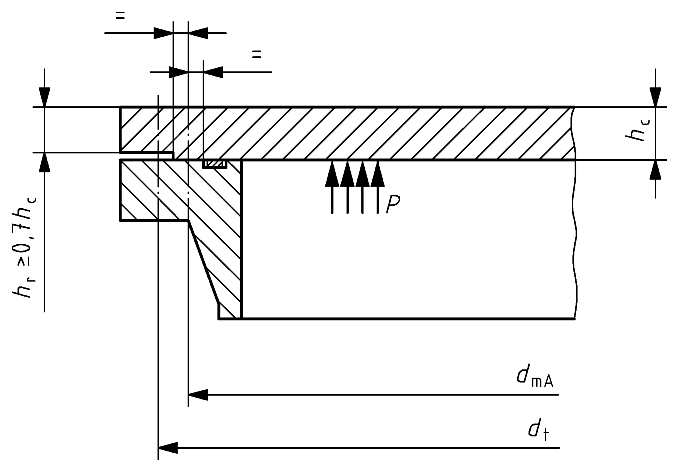
C_x 1,0

C_y see Figure 23

C_z 1,0

δ see Formula (64)

Figure 27 — Cover with gasket (direct loading) entirely within the bolt circle

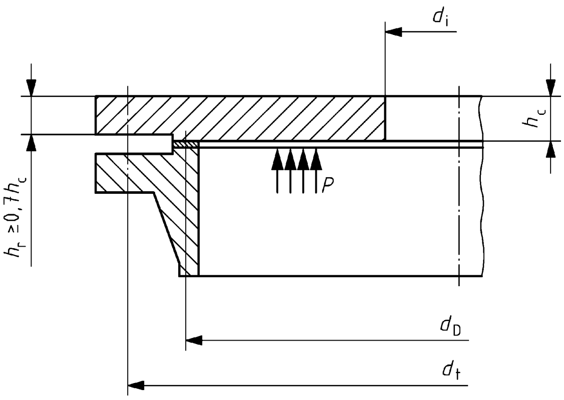


C_x 1,0
C_y see Figure 23
C_z 1,0
δ see Formula (64)

Figure 28 — Cover with gasket (not subjected to direct loading) entirely within the bolt circle

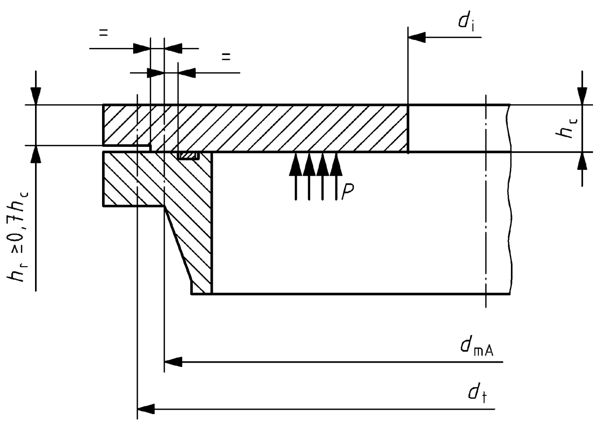
8.2.3 Circular covers with concentric circular opening, with

a) gasket entirely within the bolt circle:



C_x 1,0
C_y see Figure 23
C_z see Figure 24
δ see Formula (64)

Figure 29 — Cover with gasket (direct loading) entirely within the bolt circle

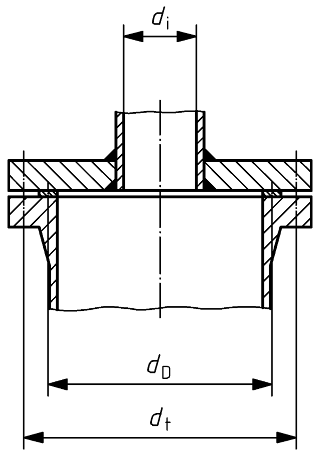


C_x 1,0
C_y see Figure 23
C_z see Figure 24
δ see Formula (64)

Figure 30 — Cover with gasket (not subjected to direct loading) entirely within the bolt circle

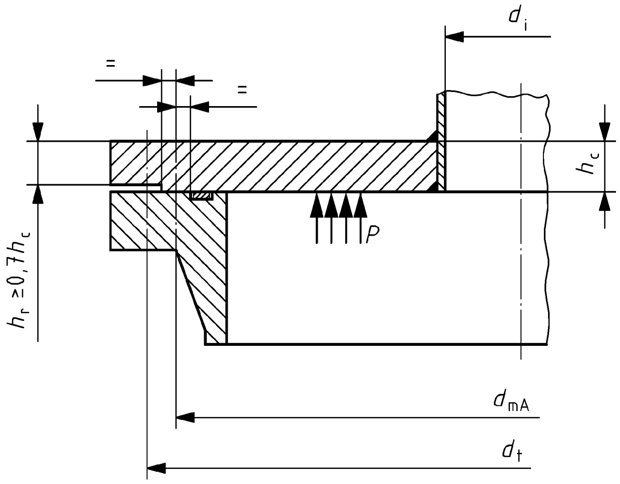
WARNING — The mean support diameter d_{mA} shall be considered in the figure. The calculation coefficients shall be taken from Figures 23 and 24.

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



C_x 1,0
C_y see Figure 23
C_z see Figure 24
δ see Formula (64)

Figure 31 — Cover with central nozzle gasket (direct loading) entirely within the bolt circle



C_x 1,0
C_y see Figure 23
C_z see Figure 24
δ see Formula (64)

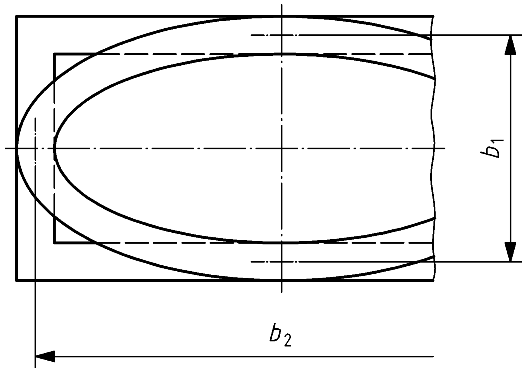
Figure 32 — Cover with central nozzle gasket (not subjected to direct loading) entirely within the bolt circle

8.2.4 Non-circular covers (elliptical or rectangular)

For non-circular covers the plate thickness is also calculated according to Formulae (63) and (64).

The calculation coefficients C_x, C_y, C_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 b_1 in the following Figure 33.



C_x see Figure 25
C_y see Figure 23
C_z see Figure 24
δ see Formula (64)

Figure 33 — 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 8.2.5.

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. The designations for the maximum moments and their centre points are also listed and these are sufficient for checking the strength.

The strength condition is:

$$\frac{6 \cdot M_i}{h^2} \leq 1{,}5 \cdot f \qquad (69)$$

with M_i equal to 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_{DB}. 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 34.

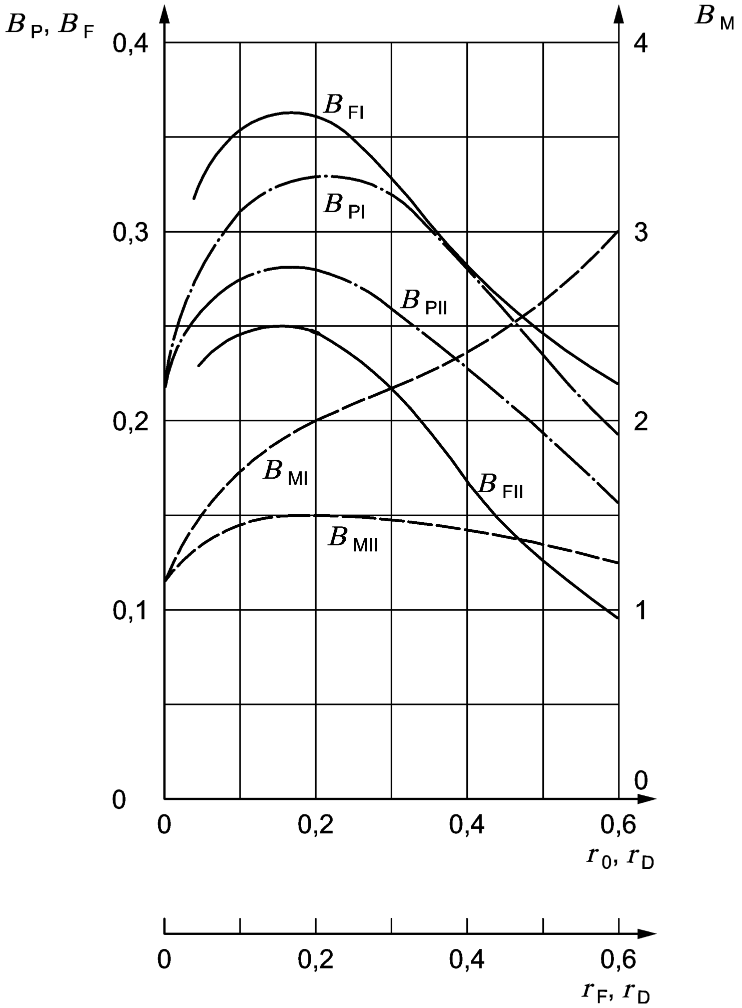


Figure 34 — Calculation coefficients B_P, B_F and B_M

The additional force F_{add} is assumed with 25 % of the force resultant from the internal pressure, see Table 6.

$$F_{add} = 0,25 \cdot p \cdot \frac{\pi \cdot d_i^2}{4} \tag{70}$$

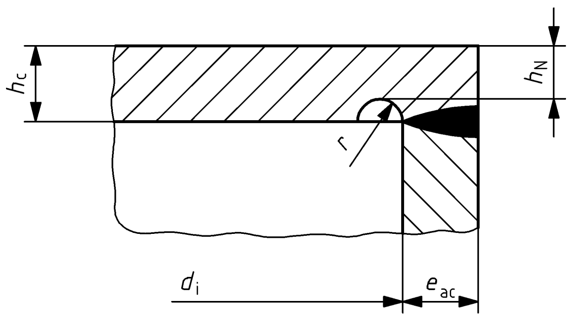


Figure 35 — Flat plate with annular groove

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

$$h_c = 0,4 \cdot d_i \cdot \sqrt{\frac{p}{f}} \tag{71}$$

with the following conditions:

$$h_N \geq p \cdot \left(\frac{d_i}{2} - r \right) \cdot \frac{1{,}3}{f} \text{ and } h_N \leq 0{,}77\, e_{ac} \qquad (72)$$

h_N shall be not less than 5 mm.

r shall be ≥ 0,2 × 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 MPa

F annular force in N

h plate thickness in mm

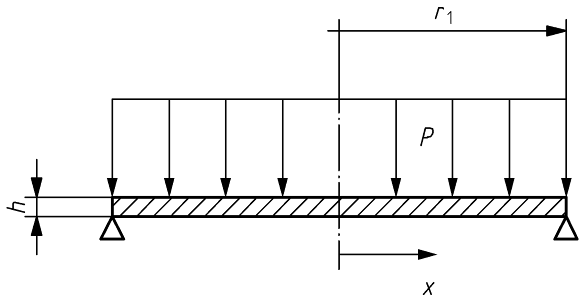
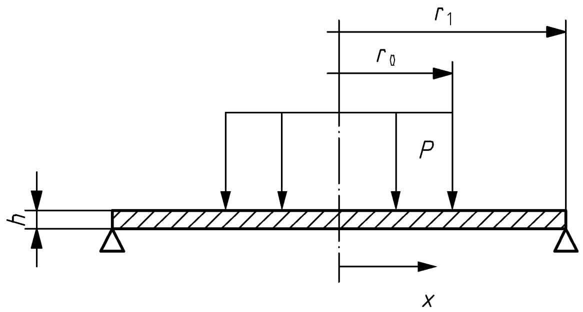
μ Poisson's ratio (for steel approximately 0,3)

M_r specific bending moment in radial direction in N mm/mm

M_t specific bending moment in tangential direction in N mm/mm

r, r_0, r_1, R (see load cases) in mm

Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5 f$ M_i equivalent to M_r; M_t; M_{max} (same as (69))

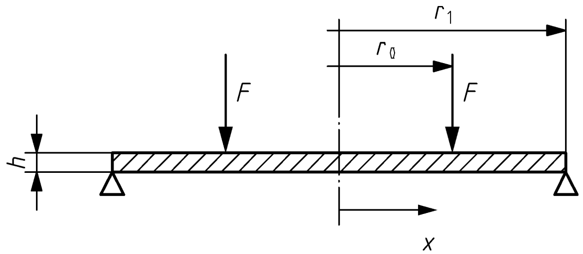
Load case	Load diagram	Specific bending moment
1	Freely supported at the rim:  	$M_r = \frac{P \cdot r_1^2}{16} \cdot (3+\mu) \cdot \left(1 - \frac{x^2}{r_1^2}\right)$ (73) $M_t = \frac{P \cdot r_1^2}{16} \cdot \left[(3+\mu) - (1+3\mu) \cdot \frac{x^2}{r_1^2}\right]$ (74) max. moment for x = 0 (centre of plate): $M_{max} = M_r = M_t = \frac{P \cdot r_1^2}{16} \cdot (3+\mu)$ (75)
2	Freely supported at the outer rim:  	for $x \leq r_0$: $M_r = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2} - (3+\mu) \cdot \frac{x^2}{r_0^2}\right]$ (76) $M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2} - (1+3\mu) \cdot \frac{x^2}{r_0^2}\right]$ (77) for $x > r_0$: $M_r = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} + (1-\mu)\left(\frac{r_0^2}{x^2} - \frac{r_0^2}{r_1^2}\right)\right]$ (78)

P specific load in MPa — 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 Poisson's ratio (for steel approximately 0,3) — Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5 f$ — M_i equivalent to M_r; M_t; M_{max} (same as (69))

Load case	Load diagram	Specific bending moment
		$M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} + 4(1-\mu) - (1-\mu)\left(\frac{r_0^2}{x^2} - \frac{r_0^2}{r_1^2}\right)\right]$ (79) max. moment for x = 0 (centre of plate) $M_{max} = M_r = M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + 4 - (1-\mu)\frac{r_0^2}{r_1^2}\right]$ (80)
3	Freely supported at the outer rim:  	for $x \leq r_0$: $M_r = M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{r_0} + (1-\mu)\left(1 - \frac{r_0^2}{r_1^2}\right)\right]$ (81) for $x > r_0$: $M_r = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} + (1-\mu)\left(\frac{r_0^2}{x^2} - \frac{r_0^2}{r_1^2}\right)\right]$ (82) $M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{x} + 2(1-\mu) - (1-\mu)\left(\frac{r_0^2}{x^2} + \frac{r_0^2}{r_1^2}\right)\right]$ (83) max. moment for $0 \leq x \leq r_0$: $M_{max} = M_r = M_t = \frac{F}{8\pi}\left[2(1+\mu)\ln\frac{r_1}{r_0} + (1-\mu)\left(1 - \frac{r_0^2}{r_1^2}\right)\right]$ (84) max. moment for $x = r_1$ (outer rim):

P specific load in MPa

F annular force in N

h plate thickness in mm

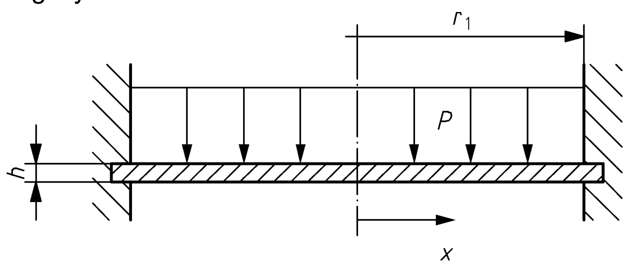
μ Poisson's ratio (for steel approximately 0,3)

M_r specific bending moment in radial direction in N mm/mm

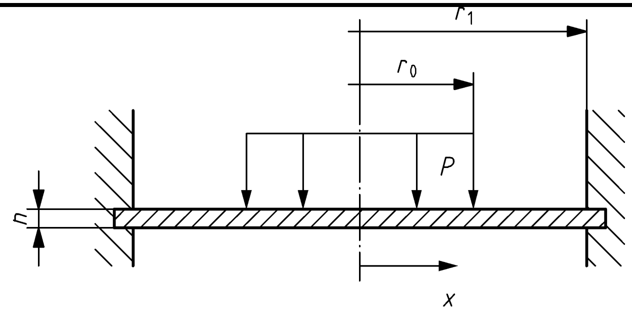
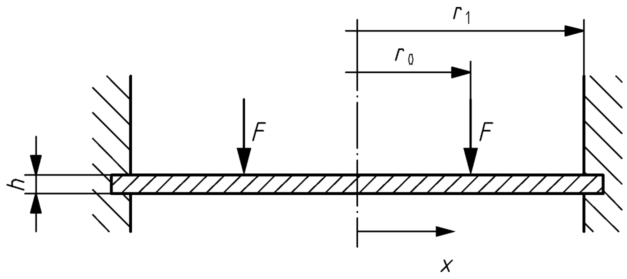
M_t specific bending moment in tangential direction in N mm/mm

r, r_0, r_1, R (see load cases) in mm

Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5f$ M_i equivalent to M_r; M_t; M_{max} (same as (69))

Load case	Load diagram	Specific bending moment
		$M_{max} = M_t = \frac{F}{8\pi}\left[2(1-\mu) - (1-\mu)\frac{2r_0^2}{r_1^2}\right]$ (85)
4	Rigidly constrained outer rim:  	$M_r = \frac{P \cdot r_1^2}{16}\left[(1+\mu) - (3+\mu)\frac{x^2}{r_1^2}\right]$ (86) $M_t = \frac{P \cdot r_1^2}{16}\left[(1+\mu) - (1+3\mu)\frac{x^2}{r_1^2}\right]$ (87) max. moment for x = 0 (centre of plate): $M_{max} = M_r = M_t = \frac{P \cdot r_1^2}{16}(1+\mu)$ (88) max. moment for $x = r_1$ (outer rim): $M_{max} = M_r = -\frac{P \cdot r_1^2}{8}$ (89)
5	Rigidly constrained outer rim:	for $x \leq r_0$: $M_r = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2} - (3+\mu)\frac{x^2}{r_0^2}\right]$ (90) $M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2} - (1+3\mu)\frac{x^2}{r_0^2}\right]$ (91)

P specific load in MPa	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 Poisson's ratio (for steel approximately 0,3)	Condition: $\sigma_i = \frac{6M_i}{h^2} \le 1{,}5f$ M_i equivalent to M_r; M_t; M_{max} (same as (69))

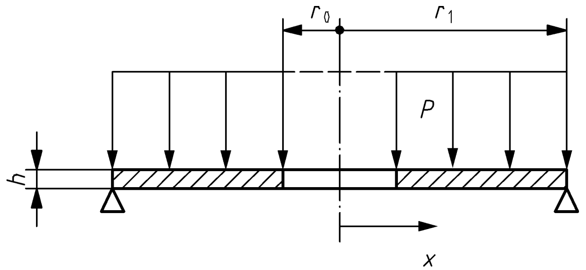
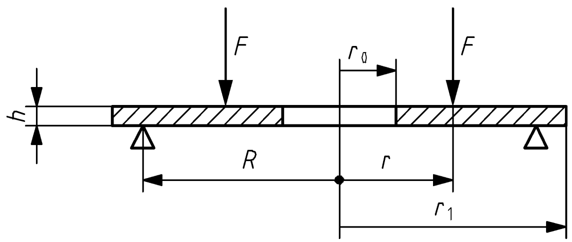
Load case	Load diagram	Specific bending moment
		for $x > r_0$: $M_r = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} - 4 + (1+\mu)\frac{r_0^2}{r_1^2} + (1-\mu)\frac{r_0^2}{x^2}\right]$ (92) $M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{x} - 4\mu + (1+\mu)\frac{r_0^2}{r_1^2} - (1-\mu)\frac{r_0^2}{x^2}\right]$ (93) max. moment for x = 0 (centre of plate): $M_{max} = M_r = M_t = \frac{P \cdot r_0^2}{16}\left[4(1+\mu)\ln\frac{r_1}{r_0} + (1+\mu)\frac{r_0^2}{r_1^2}\right]$ (94)
6	Rigidly constrained outer rim: 	for $x \le r_0$: $M_r = M_t = \frac{F}{8\pi}\left[(1+\mu)\left(2\ \ln\frac{r_1}{r_0} - 1 + \frac{r_0^2}{r_1^2}\right)\right]$ (95) for $x > r_0$: $M_r = \frac{F}{8\pi}\left[2(1+\mu)\ \ln\frac{r_1}{x} - (1+\mu)\left(1-\frac{r_0^2}{r_1^2}\right) - (1-\mu)\left(1-\frac{r_0^2}{x^2}\right)\right]$ (96) $M_t = \frac{F}{8\pi}\left[2(1+\mu)\ \ln\frac{r_1}{x} - (1+\mu)\left(1-\frac{r_0^2}{r_1^2}\right) + (1-\mu)\left(1-\frac{r_0^2}{x^2}\right)\right]$ (97) max. moment at the outer rim:

P specific load in MPa — 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 Poisson's ratio (for steel approximately 0,3) — Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5 f$ — M_i equivalent to M_r; M_t; M_{max} (same as (69))

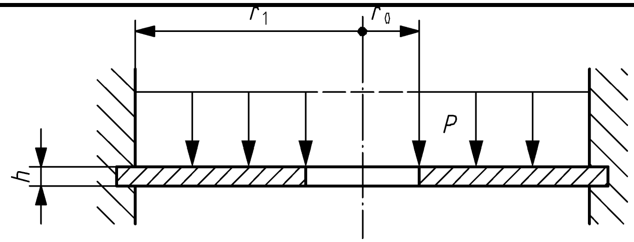
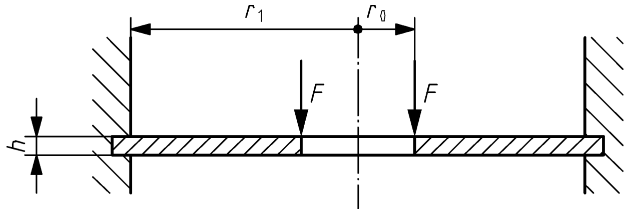
Load case	Load diagram	Specific bending moment
		$M_{max} = M_r = -\frac{F}{4\pi}\left(1 - \frac{r_0^2}{r_1^2}\right)$ (98)
7	Freely supported outer rim: 	max. moment at the inner rim: $M_{max} = M_t = \frac{P}{8\left(r_1^2 - r_0^2\right)}\left[r_1^4(3+\mu) + r_0^4(1-\mu) - 4\,(1+\mu)\,r_1^2\,r_0^2\,\ln\frac{r_1}{r_0} - 4\,r_1^2\,r_0^2\right]$ (99)
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: $M_{max} = M_t = \frac{F}{4\pi}\left[\frac{2\,r_1^2(1+\mu)}{r_1^2 - r_0^2}\,\ln\frac{R}{r} + (1-\mu)\,\frac{R^2 - r^2}{r_1^2 - r_0^2}\right]$ (100)
9	Rigidly constrained outer rim:	max. moment at the outer rim:

P specific load in MPa — 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 Poisson's ratio (for steel approximately 0,3) — Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5 f$ — M_i equivalent to M_r; M_t; M_{max} (same as (69))

Load case	Load diagram	Specific bending moment
		$$M_{max} = M_r = \frac{P}{8}\left[r_1^2 - 2r_0^2 + \frac{r_0^4(1-\mu) - 4r_0^4(1+\mu)\ln\frac{r_1}{r_0} + r_1^2 r_0^2(1+\mu)}{r_1^2(1-\mu) + r_0^2(1+\mu)}\right] \quad (101)$$ max. moment at the inner rim: $$M_{max} = M_t = \frac{P}{8}\left[\frac{(1-\mu^2)\left(r_1^4 - r_0^4 - 4r_1^2 r_0^2 \ln\frac{r_1}{r_0}\right)}{r_1^2(1-\mu) + r_0^2(1+\mu)}\right] \quad (102)$$
10	Rigidly constrained outer rim: 	max. moment at the outer rim: $$M_{max} = M_r = -\frac{F}{4\pi}\left[\frac{2(1+\mu)r_0^2\ln\frac{r_1}{r_0} + (1-\mu)(r_1^2 - r_0^2)}{(1-\mu)r_1^2 + (1+\mu)r_0^2}\right] \quad (103)$$ max moment at the inner rim: $$M_{max} = M_t = \frac{F}{4\pi}\left[\frac{(1-\mu^2)\left[r_1^2\left(2\ln\frac{r_1}{r_0} - 1\right) + r_0^2\right]}{(1-\mu)r_1^2 + (1+\mu)r_0^2}\right] \quad (104)$$

P specific load in MPa

F annular force in N

h plate thickness in mm

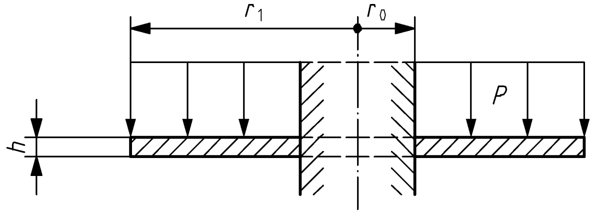
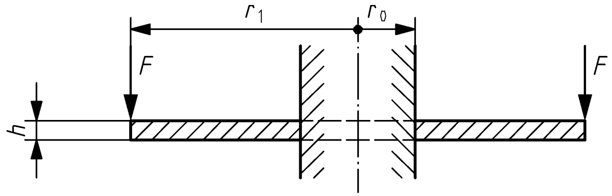
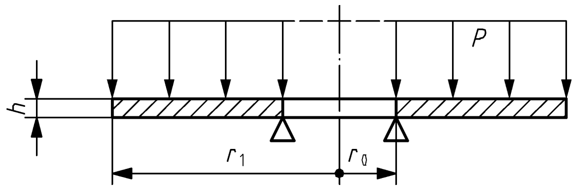
μ Poisson's ratio (for steel approximately 0,3)

M_r specific bending moment in radial direction in N mm/mm

M_t specific bending moment in tangential direction in N mm/mm

r, r_0, r_1, R (see load cases) in mm

Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5 f$ M_i equivalent to M_r; M_t; M_{max} (same as (69))

Load case	Load diagram	Specific bending moment
11	Rigidly constrained inner rim: 	max. moment at the inner rim: $M_{max} = M_r = \frac{P}{8}\left[\frac{4 r_1^4 (1+\mu) \ln \frac{r_1}{r_0} - r_1^4 (1+3\mu) + r_0^4 (1-\mu) + 4 r_0^2 r_1^2 \mu}{r_1^2 (1+\mu) + r_0^2 (1-\mu)}\right]$ (105)
12	Rigidly constrained inner rim: 	max. moment at the inner rim: $M_{max} = M_r = \frac{F}{4\pi}\left[\frac{2 r_1^2 (1+\mu) \ln \frac{r_1}{r_0} + r_1^2 (1-\mu) - r_0^2 (1-\mu)}{r_1^2 (1+\mu) + r_0^2 (1-\mu)}\right]$ (106)
13	Freely supported inner rim: 	max. moment at the inner rim: $M_{max} = M_t = \frac{P}{8\left(r_1^2 - r_0^2\right)}\left[4 r_1^4 (1+\mu) \ln \frac{r_1}{r_0} + 4 r_1^2 r_0^2 \mu + r_0^4 (1-\mu) - r_1^4 (1+3\mu)\right]$ (107)
14	Constrained but free to move at the centre, freely supported at the outer rim:	max. moment at the inner rim:

P specific load in MPa

F annular force in N

h plate thickness in mm

μ Poisson's ratio (for steel approximately 0,3)

M_r specific bending moment in radial direction in N mm/mm

M_t specific bending moment in tangential direction in N mm/mm

r, r_0, r_1, R (see load cases) in mm

Condition: $\sigma_i = \frac{6M_i}{h^2} \leq 1{,}5f$ M_i equivalent to M_r; M_t; M_{max} (same as (69))

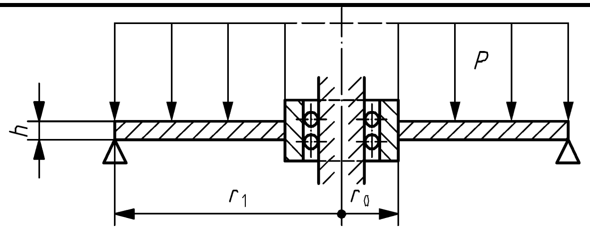
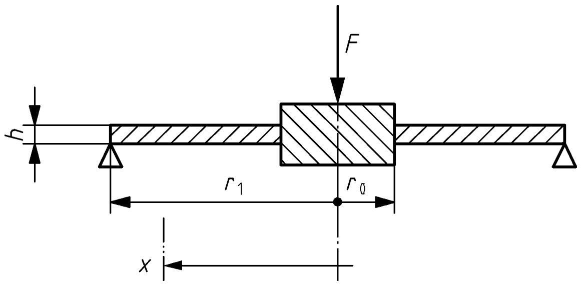
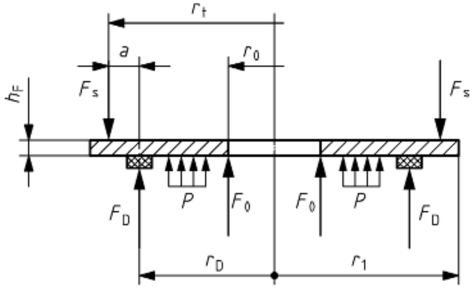
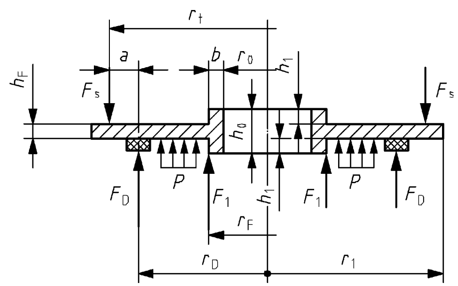
Load case	Load diagram	Specific bending moment
	 	$M_{max} = M_r = \frac{P}{8}\left[\frac{r_1^4(3+\mu)+r_0^4(1-\mu)-4r_1^2r_0^2\left[1+(1+\mu)\ln\frac{r_1}{r_0}\right]}{r_1^2(1+\mu)+r_0^2(1-\mu)}\right]$ (108)
15	Freely supported at the outer rim rigid centre plate:  	$M_r = \frac{F}{4\pi}\left\{\frac{(1+\mu)+(1-\mu)\,r_0^2/x^2}{(1+\mu)+(1-\mu)\,r_0^2/r_1^2}\left[(1+\mu)\ \ln\frac{r_1}{r_0}+1\right]-(1+\mu)\ \ln\frac{x}{r_0}-1\right\}$ (109) max. moment at the inner rim with $x = r_0$: $M_{max} = M_r = \frac{F}{4\pi}\left\{\frac{2}{(1+\mu)+(1-\mu)\,r_0^2/r_1^2}\left[(1+\mu)\ \ln\frac{r_1}{r_0}+1\right]-1\right\}$ (110)
	$r_0 \leq x \leq r_1$	max. moment at the outer rim with $x = r_1$: $M_{max} = M_t = \frac{F\,(1-\mu^2)}{4\pi}\left[\frac{r_1^2-r_0^2\left(1+2\ \ln\frac{r_1}{r_0}\right)}{r_1^2\,(1+\mu)+r_0^2\,(1-\mu)}\right]$ (111)

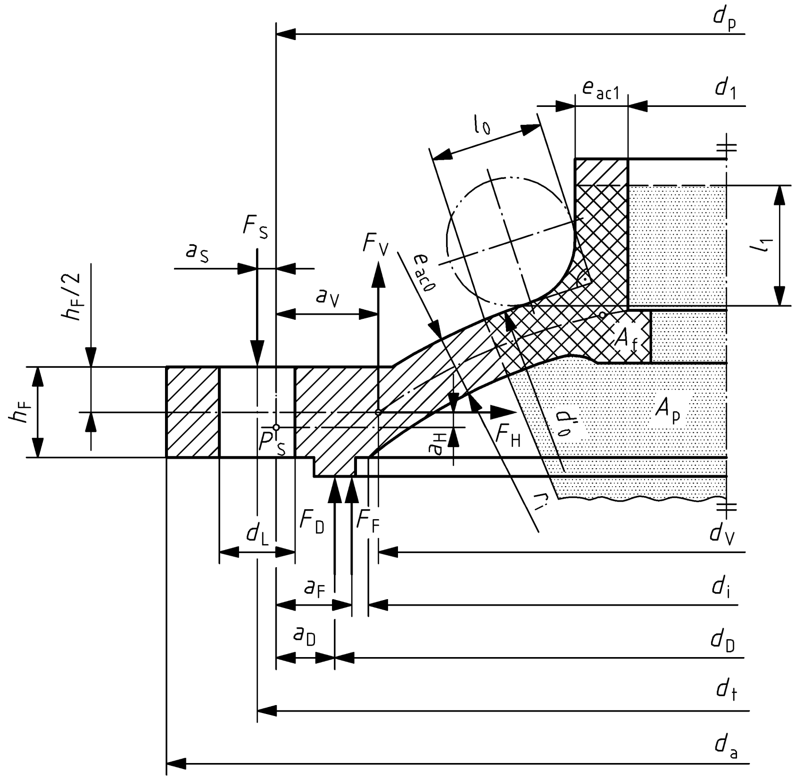
Table 6 — Application cases of circular plates with non-reinforced centre hole and with reinforced centre hole

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	$M_{PI} = p\ r_D^2 \left[B_{PI} - 0,044 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (112)	
		resulting from a single force (point force)	$M_{FI} = F_0 \left[B_{FI} - 0,028 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (113)	$F_0 = \pi r_0^2 \cdot p + F_{add}$ (114)
		resulting from a rim moment	$M_{MI} = \frac{F_{DB} 2a}{\pi\ r_D} \left[B_{MI} - 0,35 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (115)	F_{DB} according to Formula (133)
				$M_{iI} = M_{PI} + M_{FI} + M_{MI}$ (116)
II	centre hole reinforced inner rim rigid $h_1 \leq 1,25 \sqrt{(2r_0 + b) \cdot b}$ (117) 	resulting from internal pressure	$M_{PII} = p\ r_D^2 \left[B_{PII} - 0,044 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (118)	
		resulting from a single force (point force)	$M_{FII} = F_1 \left[B_{FII} - 0,028 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (119)	$F_1 = \pi r_F^2 \cdot p + F_{add}$ (120)
		resulting from a rim moment	$M_{MII} = \frac{F_{DB} 2a}{\pi\ r_D} \left[B_{MII} - 0,35 \left(1 - \frac{r_D^2}{r_1^2} \right) \right]$ (121)	F_{DB} according to Formula (133)
				$M_{iII} = M_{PII} + M_{FII} + M_{MII}$ (122)
		The calculation coefficients B_p, B_F and B_M shall be taken from Figure 34. F_{add} are the additional forces of the stem (according to Formula (70)).		

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

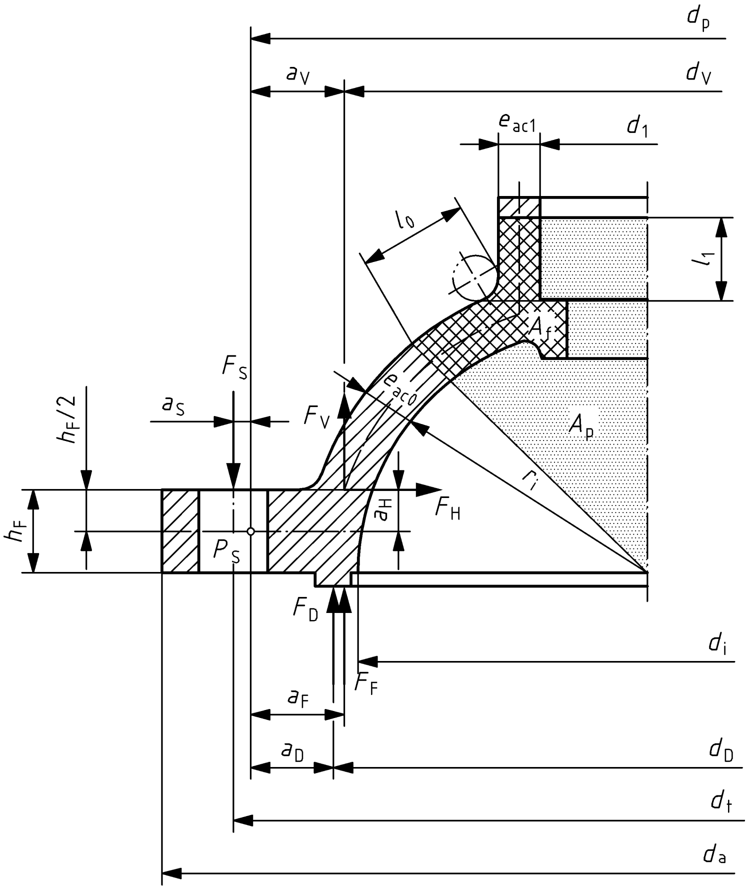
8.3.1 General

The strength calculation consists of the strength calculation of the flanged ring and the strength calculation of the spherically domed end. For strength calculation of the spherically domed end the calculation method in accordance with EN 13445-3 can alternatively be used. Depending on the geometric relationships a distinction is made between two types:



NOTE Detail Type I: spherically domed end $r_i > d_i/2$.

Figure 36 — Spherically domed end



NOTE Detail Type II: deep dishes spherically domed end shell $r_i = d_i/2$.

Figure 37 — 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 $(r_i + e_{a0}) / r_i \leq 1{,}2$

$$e_c = \frac{r_i p}{(2f - p) \cdot k_c} \tag{123}$$

— for the ratio $1{,}2 < (r_i + e_{a0}) / r_i \leq 1{,}5$

$$e_c = r_i \left[\sqrt{1 + \frac{2p}{(2f - p) \cdot k_c}} - 1 \right] \tag{124}$$

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

$$e'_c = e_c \cdot \beta \tag{125}$$

β takes into account the fact that due to the 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 36 and 37 and an approximation for $\beta = \frac{\alpha}{\delta_1}$ in Figure 38.

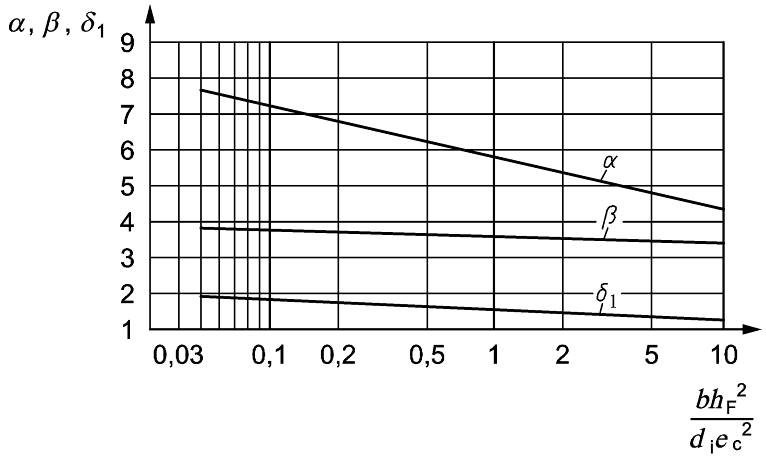


Figure 38 — Calculation coefficient

e'_c is not to be thicker than it would be as a result of the calculation of flat plate cover in accordance with 8.2.5.

8.3.3 Calculation of the flanged ring

8.3.3.1 Strength condition

The strength condition is as follows:

$$\frac{F_H}{2 \cdot \pi \cdot b \cdot h_F} \leq f \tag{126}$$

$$\frac{M_a}{2\pi\left[\frac{b}{6} \cdot h_F^2 + \frac{d_V}{12} \cdot (e'^2_c - e_c^2)\right]} + \frac{F_H}{2 \cdot \pi \cdot b \cdot h_F} \leq 1,5 \cdot f \tag{127}$$

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

The strength condition formulae 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 formulae are given in 8.3.3.2 to 8.3.3.5.

8.3.3.2 Forces and moments of Formulae (126) and (127)

The horizontal component of end force:

$$F_H = p \cdot \frac{\pi}{2} \cdot d_i \cdot \sqrt{r_i^2 - \frac{d_i^2}{4}} \tag{128}$$

The moment M_a of the external forces related to the centre of gravity P_S of the flange shall be:

$$M_{aB} = F_V \cdot a_V + F_F \cdot a_F + F_{DB} \cdot a_D + F_H \cdot a_H + F_S \cdot a_S \tag{129}$$

for operation conditions and:

$$M_{aO} = F_{SO} \cdot (a_D + a_S) \tag{130}$$

for assembly conditions.

8.3.3.3 Forces in the moment Formulae (129) and (130)

$$F_V = p \cdot \frac{\pi}{4} d_i^2 \tag{131}$$

$$F_F = p \cdot \frac{\pi}{4}\left(d_D^2 - d_i^2\right) \tag{132}$$

$$F_{DB} = p \cdot \pi \cdot d_D \cdot m \cdot b_D \cdot S_D; \quad S_D = 1{,}2 \tag{133}$$

For F_H see Formula (128).

For F_S, the bolt force for operation conditions is:

$$F_S = F_V + F_F + F_{DB} \tag{134}$$

The minimum bolt force for gasket pre-load F_{DV} results from:

$$F_{DV} = \pi \cdot d_D \cdot \sigma_{VU} \cdot b_D \tag{135}$$

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

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

$$F_{S0} = \max(\chi \cdot F_S; F_{DV}) \tag{136}$$

with

χ = 1,1 for general cases;

χ = 1,2 for soft gaskets.

$$F_{S0} \leq \pi \cdot d_D \cdot \sigma_{vo} \cdot b_D \tag{137}$$

For σ_{vo} see Table A.1.

Lever arms in the moment Formulae (129) and (130):

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

Lever arm	Bonnet	
	Type I	Type II
a_V =	0,5 · $(d_p - d_V)$	
a_D =	0,5 · $(d_p - d_D)$	
a_H =	to be determined graphically	0,5 · h_F
a_F =	a_D + 0,25 · $(d_D - d_i)$	
a_S =	0,5 · $(d_t - d_p)$	

8.3.3.4 Other geometrical dimensions in the Formulae (126) and (127)

b – the load bearing width of the flange

$$b = 0{,}5 \cdot (d_a - d_i - 2d'_L) \tag{138}$$

with

$$d'_L = v \cdot d_L \rightarrow d_i > 500 \rightarrow v = 0{,}5 \tag{139}$$

$$d_i \leq 500 \rightarrow v = 1 - 0{,}001 \cdot d_i \tag{140}$$

8.3.3.5 d_p Diameter of centre of gravity

$$d_p = d_a - 2 \cdot S_S \text{ with} \tag{141}$$

$$S_S = \frac{0{,}5 \cdot a_1^2 + a_2 \cdot (a_1 + d_L + 0{,}5 \cdot a_2)}{a_1 + a_2} \tag{142}$$

$$a_1 = 0{,}5 \cdot (d_a - d_t - d_L) \tag{143}$$

$$a_2 = 0{,}5 \cdot (d_t - d_i - d_L) \tag{144}$$

8.3.4 Reinforcement of the stuffing box area

The calculation is in accordance with the method for the calculation of crotch areas, see 7.3, i.e. the comparison of pressure loaded areas A_p with metal cross-sectional areas A_f .

$$p \cdot \left[\frac{A_p}{A_f} + \frac{1}{2} \right] \leq f \tag{145}$$

The stuffing box area is limited by the effective lengths:

$$l_0 = \sqrt{(2 \cdot r_i + e_{ac0}) \cdot e_{ac0}} \tag{146}$$

$$l_1 = \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \tag{147}$$

For the areas A_p, A_f, the length l_0 and l_1, see Figures 36 and 37.

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 39), and to dished heads with cut-outs (see Figure 41) or with branches (see Figure 42).

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	$r \geq 0{,}1 \cdot d_o$
related wall thickness	$0{,}001 \leq e_{ac} / d_o \leq 0{,}10$

In particular, the following values shall apply:

to torispherical heads $R_i = d_o$; $r = 0,1 \cdot d_o$, $h_w = 0,1935 \cdot d_o - 0,455 \cdot e_{ac}$

to semi-ellipsoidal heads $R_i = 0,8 \cdot d_o$; $r = 0,154 \cdot d_o$, $h_w = 0,255 \cdot d_o - 0,635 \cdot e_{ac}$

to hemispherical heads $d_o / d_i \leq 1,2$ (see 7.2.3)

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:

$$x = 0,5 \cdot \sqrt{R_i \cdot e_{cK}} \tag{148}$$

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

for torispherical heads $x = 3,5 \cdot e_{cK}$

for semi-ellipsoidal heads $x = 3,0 \cdot e_{cK}$

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 $0,6 \cdot d_o$. 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 Formulae (149) to (155).

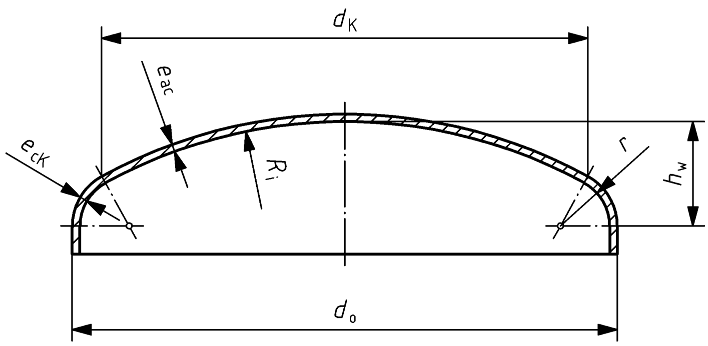


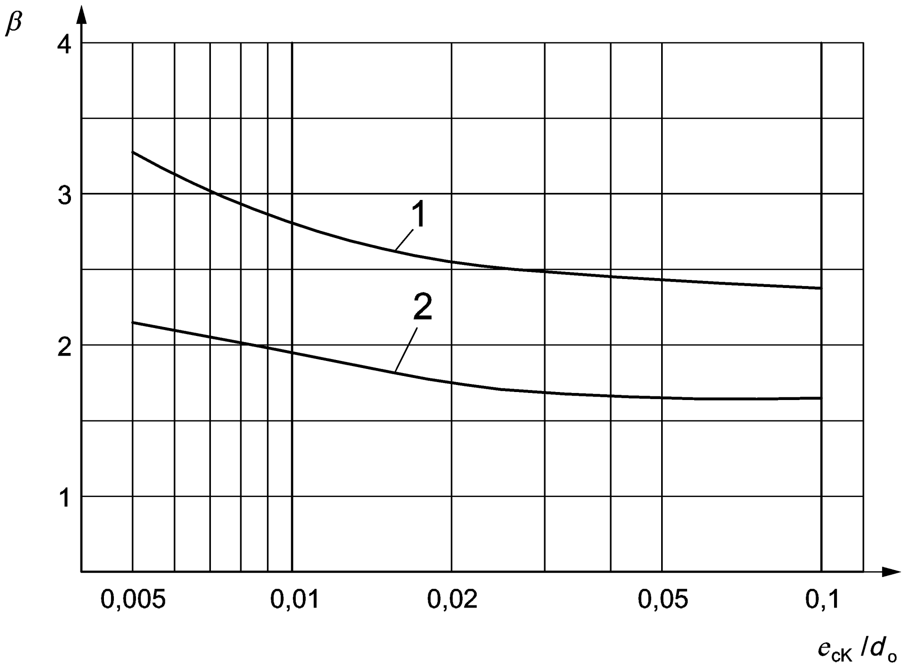
Figure 39 — Solid dished head

For hemispherical heads, the wall thickness determined in accordance with Formulae (149) to (155) 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:

$$e_{cK} = \frac{p \cdot d_0 \cdot \beta}{4 \cdot f \cdot k_c} \tag{149}$$

The calculation coefficient β can be obtained from Figure 40 as a function of e_{cK} / d_o for torispherical heads (heads 1) and semi-ellipsoidal heads (heads 2).



Key

1 curve for torispherical head

2 curve for semi-ellipsoidal head

Figure 40 — 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 $0,6 \cdot d_o$ 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.

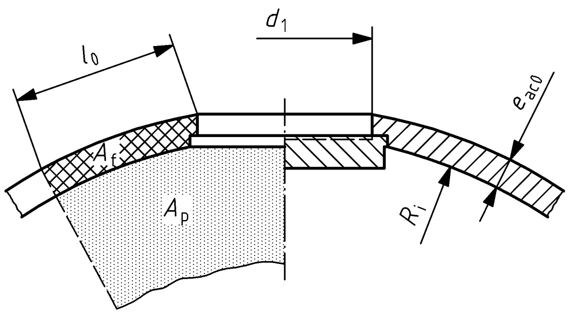


Figure 41 — Dished head with opening

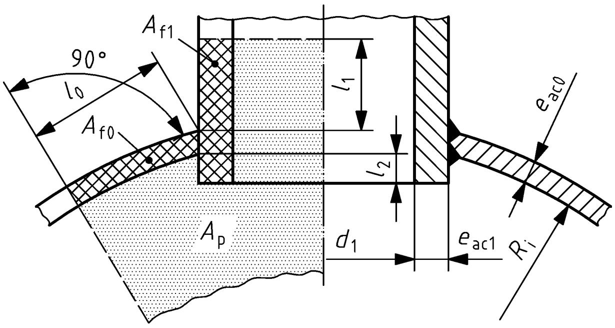


Figure 42 — Dished head with branch (welded-in tubular reinforcement)

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:

$$l_0 = \sqrt{(2R_i + e_{ac0}) \cdot e_{ac0}} \text{ (see Figure 41)} \quad (150)$$

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:

$$l_2 \leq 0,5\sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \quad (151)$$

can be included as load bearing in the calculation.

If $e_{ac1} > e_{ac0}$, it shall be calculated with $e_{ac1} = e_{ac0}$.

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 $A_f^{''} = 0,9 \cdot A_f$ shall be entered in the calculation for A_f (see Figure 43).

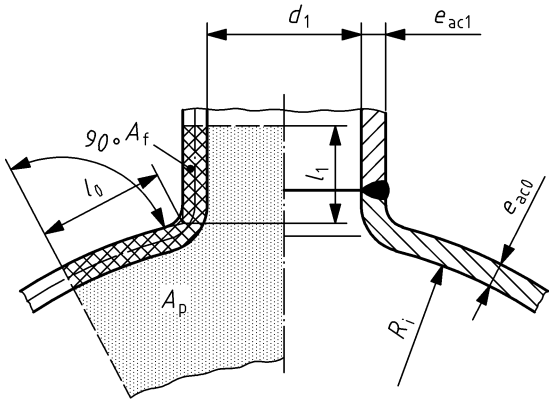


Figure 43 — Dished head with necked opening

d) By means of disc-shaped reinforcements (see Figure 44) 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_c = 0,7.

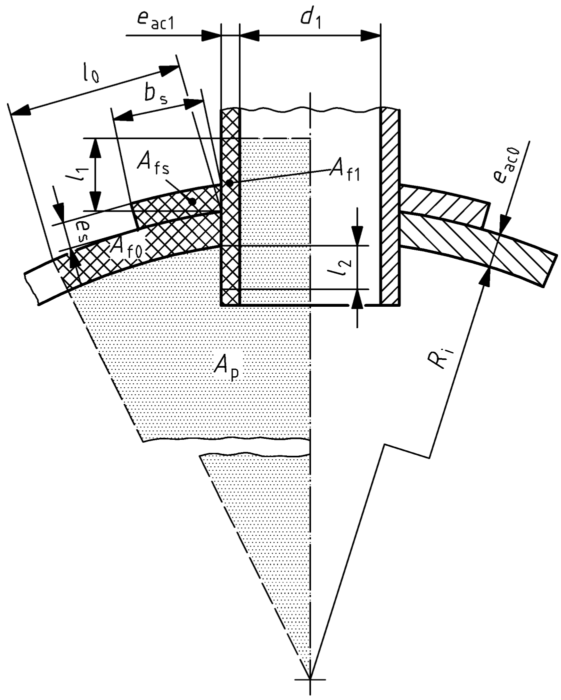


Figure 44 — Dished head with disc-shaped reinforcement

The strength condition for cut-outs in the apex zone 0,6 · d_o is given by the Formula (152), with the area A_p subjected to pressure, and with the effective cross-sectional areas A_{f0}, A_{f1} and A_{fS}:

$$p \cdot \left(\frac{A_p}{A_{f0} + A_{f1} + k_c \cdot A_{fs}} + \frac{1}{2} \right) \leq f \tag{152}$$

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

$$l_0 = \sqrt{(2R_i + e_{ac0}) \cdot e_{ac0}} \tag{153}$$

and the lengths for the branch shall not exceed:

$$l_1 = \sqrt{(d_1 + e_{ac1}) \cdot e_{ac1}} \tag{154}$$

$$l_2 \leq \frac{l_1}{2} \tag{155}$$

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 45). At the same time, the most important main dimensions of the closure shall be calculated in accordance with elemental procedures.

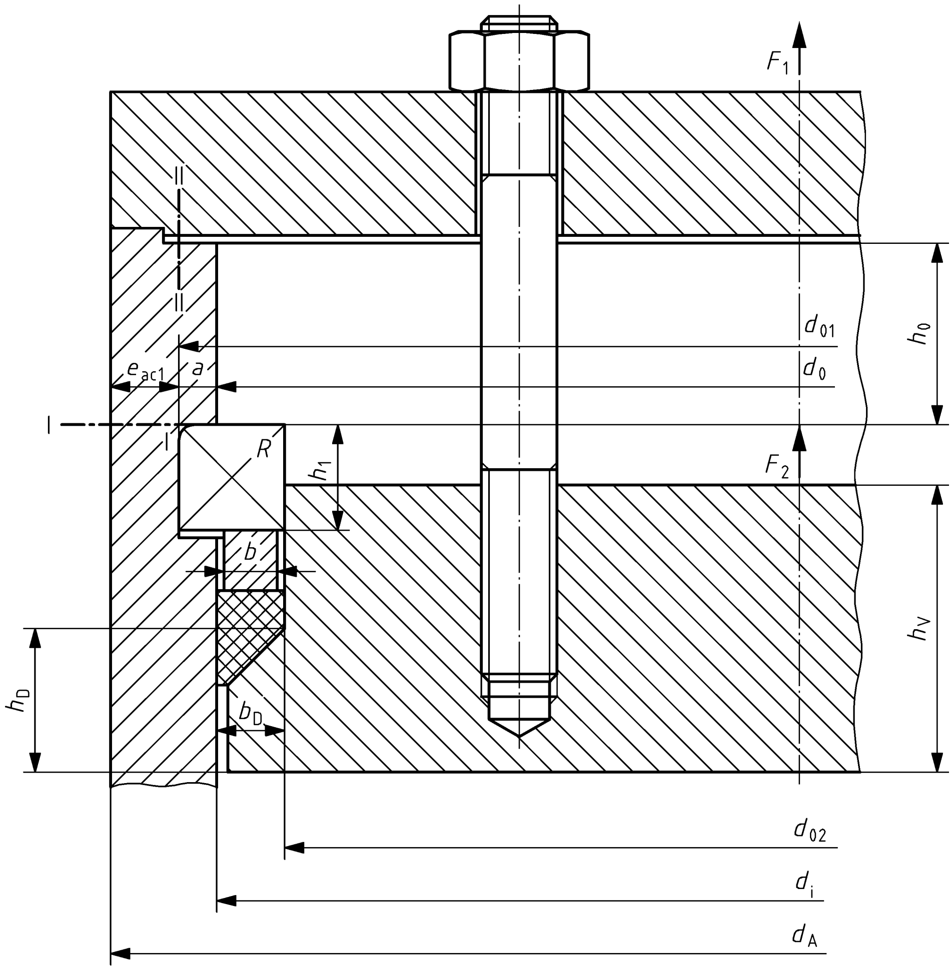


Figure 45 — Self-sealing closure

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

$$F_B = p \cdot \frac{\pi}{4} d_i^2 \tag{156}$$

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 Formulae (157) and (158), taking the friction conditions and the gasket requirement into account:

$$a = 2 \cdot \frac{F_B + F_2}{\pi \cdot (d_{01} + d_0) \cdot 1{,}5 \cdot f} \tag{157}$$

$$b = 2 \cdot \frac{F_B + F_2}{\pi \cdot (d_i + d_{02}) \cdot 1{,}5 \cdot f} \tag{158}$$

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:

$$h_1 = \frac{2(F_\mathrm{B} + F_2)}{\pi \cdot d_\mathrm{i} \cdot f} \quad (159)$$

For bending:

$$h_1 = 1{,}38 \cdot \sqrt{\frac{(F_\mathrm{B} + F_2) \cdot (d_{01} - d_{02})}{4 \cdot d_\mathrm{i} \cdot f}} \quad (160)$$

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:

$$h_0 = \frac{2 \cdot (F_\mathrm{B} + F_2)}{\pi \cdot d_{01} \cdot f} \quad (161)$$

For bending:

$$h_0 = 1{,}13 \cdot \sqrt{\frac{(F_\mathrm{B} + F_2) \cdot a}{d_{01} \cdot f}} \text{ with } a = \frac{d_{01} - d_0}{2} \quad (162)$$

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:

$$h_\mathrm{D} = \frac{2 \cdot (F_\mathrm{B} + F_2)}{\pi \cdot d_{02} \cdot f} \quad (163)$$

For bending:

$$h_D = 1{,}13 \sqrt{\frac{(F_\mathrm{B} + F_2) \cdot b_\mathrm{D}}{2 \cdot d_{02} \cdot f}} \quad (164)$$

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

Strength condition for cross-section I-I:

$$(F_\mathrm{B} + F_2)\left(a + \frac{e_{ac1}}{2}\right) \leq \frac{\pi}{4}\left[h_0^2(d_\mathrm{A} - d_{01}) + (d_\mathrm{A} - e_{ac1}) \cdot \left(e_{ac1}^2 - e_{c2}^2\right)\right] \cdot f \quad (165)$$

where $e_{c2} = \dfrac{F_\mathrm{B} + F_1}{\pi \cdot (d_\mathrm{A} - e_{ac1}) \cdot f}$ (166)

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 EN 13445-3. The calculation can however also be carried out in accordance with the formulae featured below, which are solved with respect to the thickness of the flange plate h_F as a simplification.

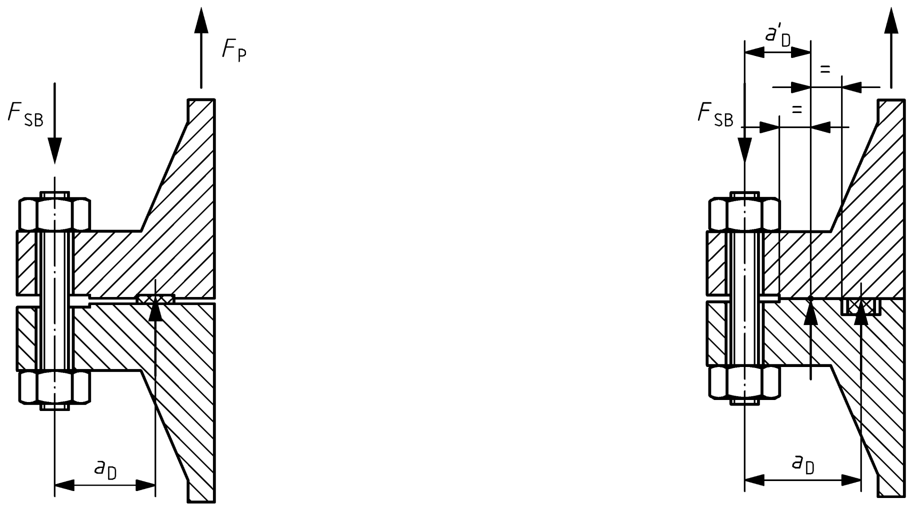
For piping flanges in accordance with EN 1092-1, up to DN 600 (included) a check calculation will not be required on condition that the permissible pressures, temperatures and materials to be used for the flanges, bolts, and gaskets are in accordance with the flange standard and if there are no additional loads specified (for example pipe loads).

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.

The subsequent requirements apply for gaskets in the primarily direct loading (non-positive connections) in accordance with Figure 46 a). Flanges with gaskets not subject to direct loading (metal to metal contact type) in accordance with Figure 46 b) can also be calculated according to this standard. In that case, the mean diameter of the metal contact surface shall be taken into account for calculating the lever arms instead of the mean gasket diameter. Characteristic gasket parameters shall be taken from the gasket data documented by the gasket manufacturer or, if unavailable, from Annex A.

Flanges with gaskets not in the load path can also be calculated according to this standard. Instead of the middle gasket diameter the middle metal contact diameter for the calculation of the moment arms shall be used. Characteristic gasket parameters shall be taken from the documented gasket data from the gasket manufacturer or, if unavailable, from Annex A.

On flat gaskets with direct loading, only characteristic factors of gaskets which were obtained on the basis of testing standard EN 13555 (or the former standard DIN 28090-1) may be taken into account.



a) Direct loading **b) Not subject to direct loading**

Figure 46 — Types of flange connections

10.2 Circular flanges

10.2.1 General

The decisive factor for the flange design is the maximum flange resistance W required, resulting from Formulae (167) and (168).

$$W = \frac{F_{SB} \cdot a}{1,5 \cdot f} \quad (167)$$

$$W = \frac{F_{S0} \cdot a_D}{1,5 \cdot f} \quad (168)$$

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

$$F_{SB} = F_p + F_{DB} = \frac{\pi \cdot d_D^2}{4} \cdot p + p \cdot \pi \cdot d_D \cdot m \cdot b_D \cdot S_D \quad (169)$$

where

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:

$$F_{DV} = \pi \cdot d_D \cdot \sigma_{VU} \cdot b_D \quad (170)$$

Characteristic values for the gaskets, m, σ_{VO} and σ_{VU} are given in Table A.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_{S0} for the assembly condition results from:

$$F_{S0} = \max(\chi \cdot F_{SB};\ F_{DV}) \quad (171)$$

where

χ = 1,1 for general cases;

χ = 1,2 for soft gaskets.

$$F_{S0} \le \pi \cdot d_D \cdot \sigma_{VO} \cdot b_D \quad (172)$$

10.2.2 Flanges with tapered neck

10.2.2.1 Flanges with tapered neck according to Figure 47 shall be subjected to a check calculation in the cross-sections I-I and II-II.

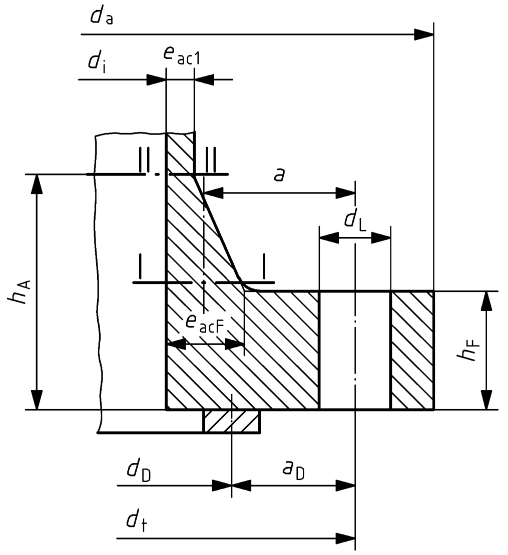


Figure 47 — Flange with tapered neck

Cross-section I-I

The required thickness of the flange h_F results from:

$$h_F = \sqrt{\frac{1{,}91 \cdot W - Z}{b}} \quad (173)$$

The lever of the bolt force in Formulae (167) and (168) for the operating condition result from:

$$a = \frac{d_t - d_i - e_{acF}}{2} \quad (174)$$

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \quad (175)$$

The thickness of the flange neck e_{acF} shall not be entered in Formulae (174) and (178) at any value exceeding 1/3 h_F.

The calculated double flange width b results from:

$$b = d_a - d_i - 2\,d'_L \quad (176)$$

The reduced bolt hole diameter d'_L

$$d'_L = v \cdot d_L \quad (177)$$

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

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

The coefficient Z results from:

$$Z = (d_i + e_{acF}) \cdot e_{acF}^2 \quad (178)$$

Cross-section II-II

The following formulae apply within the limits:

$$0{,}5 \leq \frac{h_A - h_F}{h_F} \leq 1 \text{ and} \tag{179}$$

$$0{,}1 \leq \frac{e_{ac1} + e_{acF}}{b} \leq 0{,}3 \tag{180}$$

For fulfilling this condition the flange calculation could alternatively be done with reduced values for ***e*** and ***h*.**

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

The required flange thickness h_F results from:

$$h_F = B \cdot \sqrt{\frac{1{,}91 \cdot W - Z_1}{b}} \tag{181}$$

The coefficient Z_1 results from:

$$Z_1 = \frac{3}{4}(d_i + e_{ac1}) \cdot e_{ac1}^2 \tag{182}$$

The calculation coefficient B results from:

$$B = \frac{1 + \frac{2e_{cm}}{b} \cdot B_h}{1 + \frac{2e_{cm}}{b}(B_h^2 + 2B_h)} \tag{183}$$

where $e_{cm} = \frac{e_{acF} + e_{ac1}}{2}$ (184)

and $B_h = \frac{h_A - h_F}{h_F}$ (185)

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

$$a = \frac{d_t - d_i - e_{ac1}}{2} \tag{186}$$

and for the assembly condition a_D from Formula (175).

10.2.3 Flanges greater than DN 1 000

If the neck depth $h_A - h_F$ is at least $0{,}6 \cdot h_F$ the neck thickness $e_{acF} - e_{ac1}$ is at least $0{,}25\ h_F$ Formulae (187) and (188) below may be used for cross-sections I-I and II-II in lieu of Formulae (173) and (181); these will result in smaller dimensions:

Cross-section I-I:

$$h_F = \sqrt{\frac{1{,}59W - 0{,}8Z}{b} + \left(\frac{0{,}05Z}{b \cdot e_{acF}}\right)^2} - \frac{0{,}05Z}{b \cdot e_{acF}} \tag{187}$$

where Z is in accordance with Formula (178).

Cross-section II-II:

$$h_F = B \cdot \sqrt{\frac{1{,}59W - 2Z_1}{b}} \tag{188}$$

where Z_1 is in accordance with Formula (183).

10.2.4 Welding neck with tapered neck according to Figure 48

The calculation shall be carried out in accordance with Formulae (173) to (188), using the value for d_a instead of d_t and $d'_L = 0$

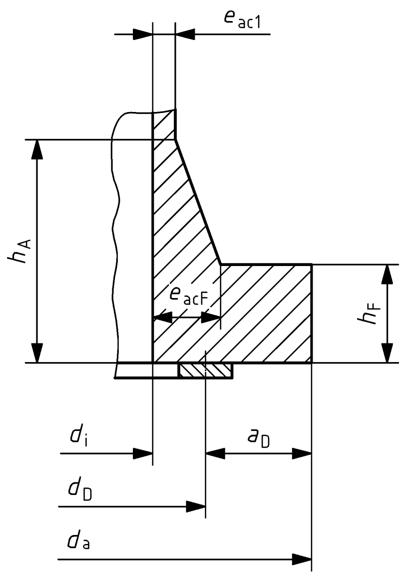
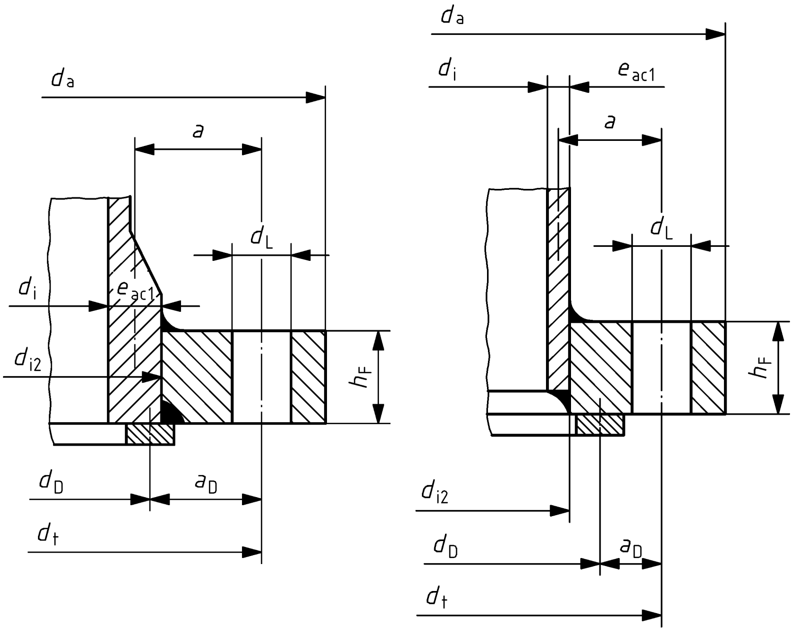


Figure 48 — Welding neck with tapered neck

10.2.5 Weld-on flanges

10.2.5.1 Weld-on flanges in accordance with Figure 49, design A and design B, and integral flange in accordance with Figure 50



a) Design A **b) Design B**

Figure 49 — Weld-on flanges

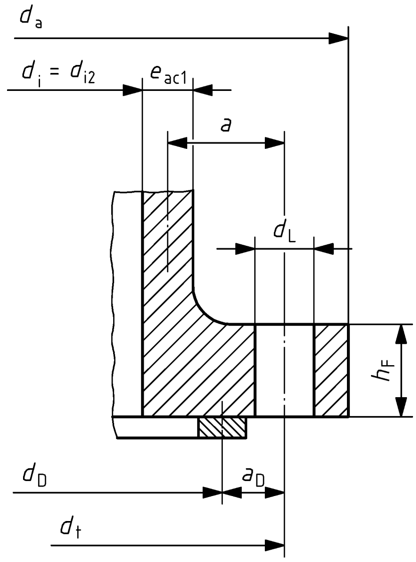


Figure 50 — Integral flange

The required flange thickness h_F is:

$$h_F = \sqrt{\frac{2{,}13W - Z}{b}} \tag{189}$$

The calculated double flange width b results from:

$$b = d_a - d_{i2} - 2d'_L \qquad (190)$$

where d'_L is in accordance with Formula (177).

In lieu of d_{i2}, d_i can be entered in the above formula 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 50.

The coefficient Z results from:

$$Z = (d_i + e_{ac1}) \cdot e_{ac1}^2 \qquad (191)$$

In Formulae (191) and (192) e_{ac1} shall not be higher than 1/2 h_F.

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

$$a = \frac{d_t - d_i - e_{ac1}}{2} \qquad (192)$$

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \qquad (193)$$

10.2.5.2 Welded-on collars in accordance with Figure 51

The calculation shall be carried out in accordance with Formulae (189) to (193), using the value for d_a instead of d_t and d'_L = 0.

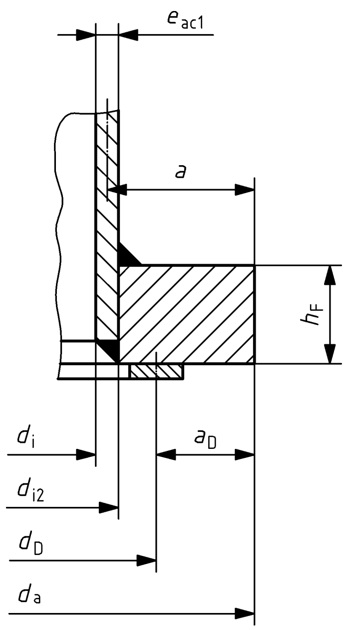
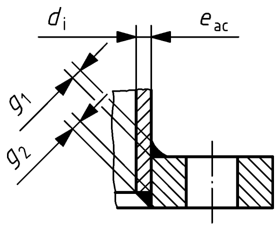
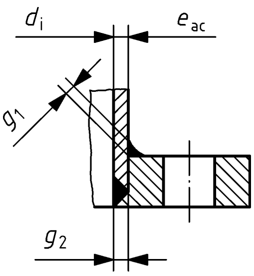
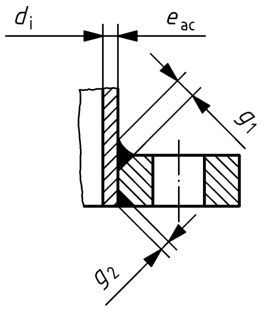
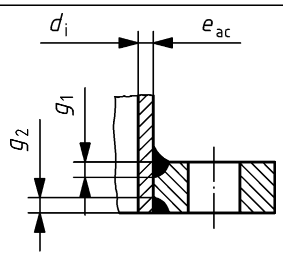
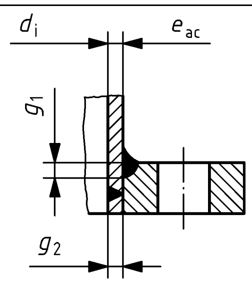


Figure 51 — Welded-on collar

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

Weld type	**Weld thickness**	**Limitation $d_i \cdot p$** mm × bar
 Type 1	$g_1 + g_2 \geq 1{,}4 \cdot e_{ac1}$	10 000
 Type 2	$g_1 + g_2 \geq 1{,}4 \cdot e_{ac1}$	10 000
 Type 3	$g_1 + g_2 \geq 2 \cdot e_{ac1}$	20 000
 Type 4	$g_1 + g_2 \geq 2 \cdot e_{ac1}$	—
 Type 5	$g_1 + g_2 \geq 2 \cdot e_{ac1}$	—
The difference between g_1 and g_2 shall not exceed 25 %.		

10.2.6 Reverse flanges

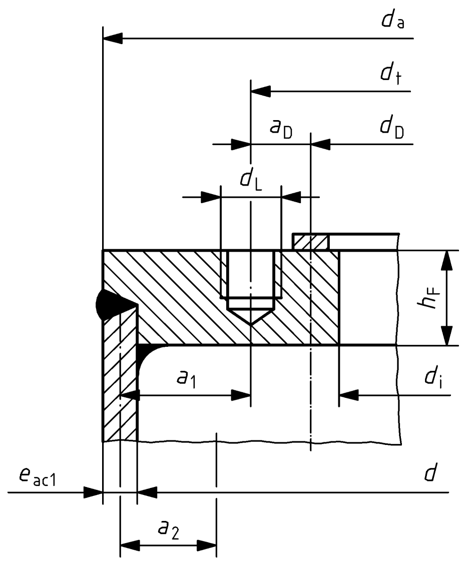


Figure 52 — Reverse flange

The required flange thickness h_F results from Formula (189). The calculated double flange width b results from:

$$b = d - d_i - 2d'_L \tag{194}$$

The coefficient Z results from:

$$Z = (d + e_{ac1}) \cdot e_{ac1}^2 \tag{195}$$

where d'_L is in accordance with Formula (177).

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

$$a = a_1 + a_2\left(\frac{d^2}{d_D^2} - 1\right) \tag{196}$$

with $a_1 = \dfrac{d - d_t + e_{ac1}}{2}$ (197)

and $a_2 = \dfrac{d - d_D + 2e_{ac1}}{4}$ (198)

and for the assembly condition from:

$$a_D = \frac{d_t - d_D}{2} \tag{199}$$

10.2.7 Loose flanges

The required flange thickness h_F results from:

$$h_F = \sqrt{1{,}91 \cdot \frac{W}{b}} \tag{200}$$

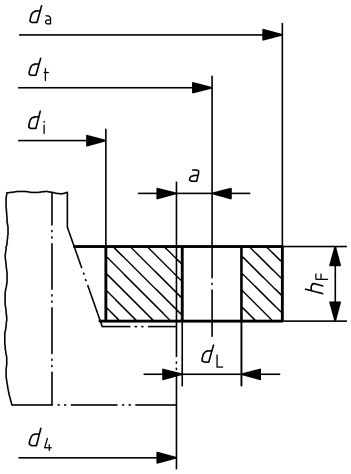


Figure 53 — Loose flange

The calculated double flange width b results from:

$$b = d_a - d_i - 2d'_L \tag{201}$$

where d'_L is in accordance with Formula (177).

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

$$a = a_D = \frac{d_t - d_4}{2} \tag{202}$$

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

$$p_F = 1{,}27 \cdot \frac{F_{SB}}{d_4^2 - d_i^2} \leq 1{,}5 \cdot f \tag{203}$$

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 54

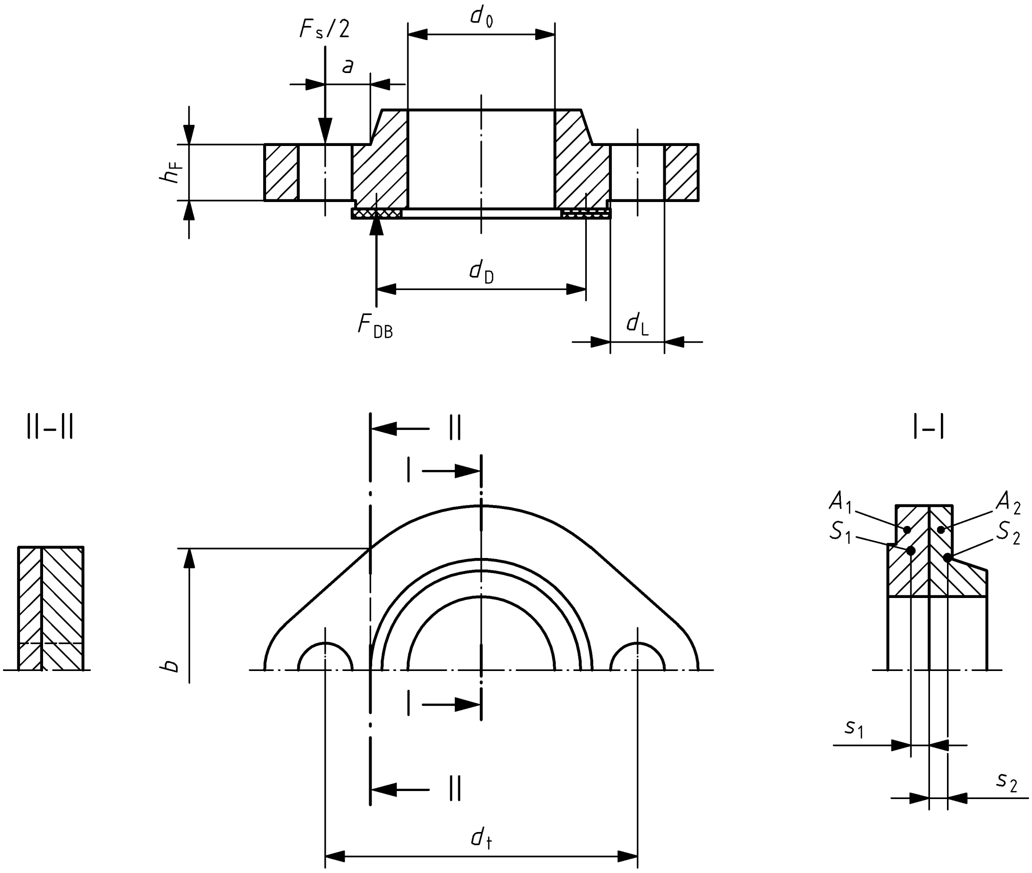


Figure 54 — Oval flange with two bolts

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 $W_{\text{req1}} = \dfrac{M}{1{,}5 \cdot f}$ and for the assembly condition $W_{\text{req2}} = \dfrac{M}{2{,}2 \cdot f}$.

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

$$F_{\text{SB}} = F_{\text{p}} + F_{\text{DB}} = p \cdot \frac{\pi \cdot d_{\text{D}}^2}{4} + p \cdot \pi \cdot d_{\text{D}} \cdot m \cdot b_{\text{D}} \cdot S_{\text{D}} \tag{204}$$

where

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:

$$F_{\text{DV}} = \pi \cdot d_{\text{D}} \cdot \sigma_{\text{VU}} \cdot b_{\text{D}} \tag{205}$$

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

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

The minimum bolt force F_{S0} for the assembly condition results from:

$$F_{S0} = \max\ (\chi \cdot F_{SB};\ F_{DV}) \tag{206}$$

where

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:

$$M = \frac{F_S}{4} \cdot d_t \tag{207}$$

where

$$F_S = F_{SB} \text{ or } F_{S0}. \tag{208}$$

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:

$$W_{avI} = 2 \cdot A_1 \cdot (s_1 + s_2) \tag{209}$$

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

$$M = \frac{F_s}{2} \cdot a \tag{210}$$

The flange resistance W results from:

$$W_{avII} = \frac{b \cdot h_F^2}{6} \tag{211}$$

10.3.2 Oval flanges in accordance with Figure 55

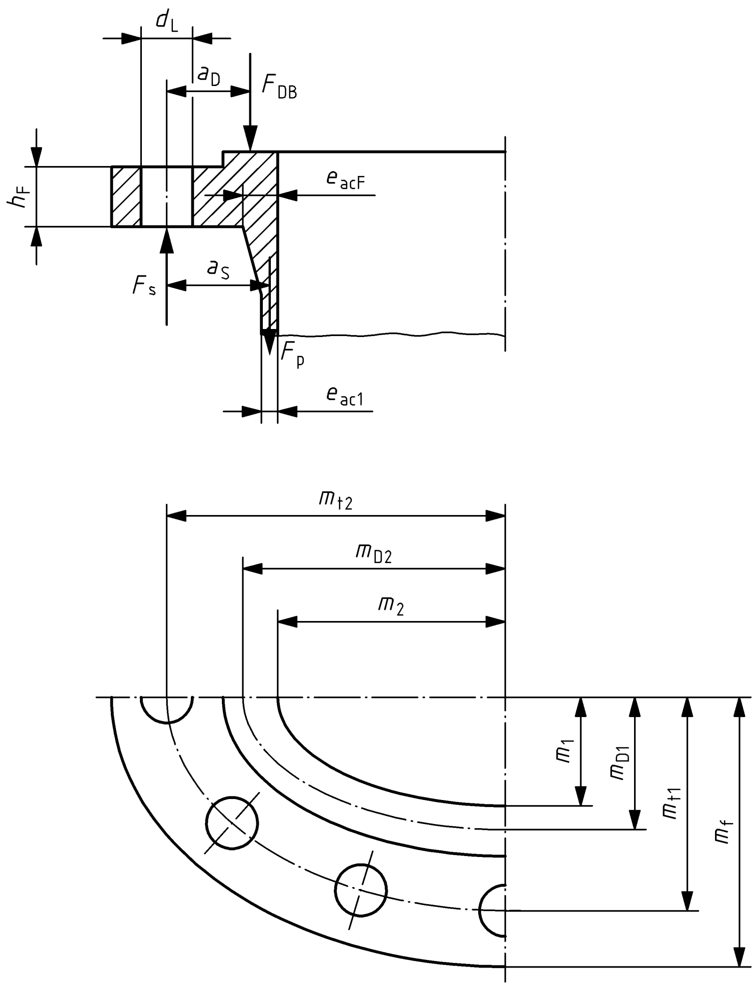


Figure 55 — Oval flange with more than two bolts

The minimum bolt force FSB for the operating condition results from:

$$F_{SB} = F_p + F_{DB} = p \cdot \pi \cdot m_{D1} \cdot m_{D2} + p \cdot U_D \cdot m \cdot b_D \cdot S_D \tag{212}$$

where

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:

$$F_{DV} = U_D \cdot \sigma_{VU} \cdot b_D \tag{213}$$

The mean circumference U_D of the gasket results from:

$$U_D = \pi \left[3(m_{D1} + m_{D2}) - \sqrt{(3m_{D1} + m_{D2}) \cdot (3m_{D2} + m_{D1})} \right] \tag{214}$$

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

The minimum bolt force F_{S0} for the assembly condition results from:

$$F_{S0} = \max\ (\chi \cdot F_{SB}; F_{DV}) \tag{215}$$

where

χ = 1,1 for general applications;

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

Consequently, the strength condition will be:

$$\frac{F_p \cdot a_S + F_{DB} \cdot a_D}{f} \le \left[\frac{\pi}{2}(m_f - m_1 - d'_L) h_F^2 + \frac{1}{4} U_D \left(e_{acF}^2 - \frac{e_{ac1}^2}{4} \right) \right] \cdot \frac{1}{2} \left[1 + \left(\frac{m_{t1}}{m_{t2}} \right)^2 \right] \cdot \frac{1}{B_5} \tag{216}$$

The reduced bolt hole diameter d'_L:

$$d'_L = v \cdot d_L \tag{217}$$

where $v = 1 - 0{,}001 \cdot (m_1 + m_2)/2$ for $(m_1 + m_2)/2 \le 500$ mm,

and $v = 0{,}5$ for $(m_1 + m_2)/2 > 500$ mm,

with B_5 [5] according to Figure 56 or the following formula:

$$B_5 = \begin{cases} 2 - 50 \cdot \dfrac{e_{ac1}}{m_2}; & 0{,}01 < \dfrac{e_{ac1}}{m_2} \le 0{,}02 \\ 1 & ; 0{,}02 < \dfrac{e_{ac1}}{m_2} \le 0{,}05 \\ \dfrac{2}{3} + \left(6 + \dfrac{2}{3} \right) \cdot \dfrac{e_{ac1}}{m_2}; & 0{,}05 < \dfrac{e_{ac1}}{m_2} \le 0{,}25 \end{cases} \tag{218}$$

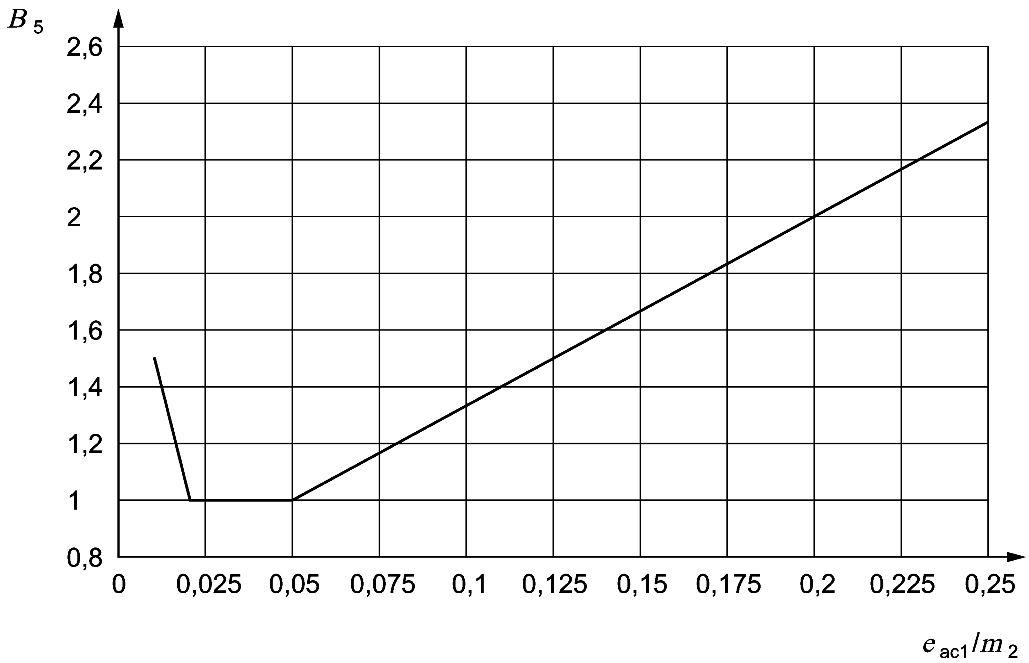


Figure 56 — Correction factor B_5 of Formula (217)

10.4 Rectangular or square flanges

10.4.1 Rectangular or square flanges in accordance with Figure 57

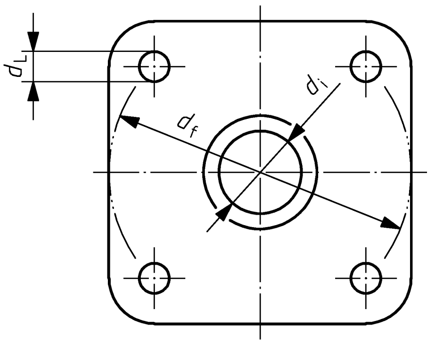


Figure 57 — Rectangular or square flange

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

10.4.2 Rectangular slip-on flanges in accordance with Figure 58

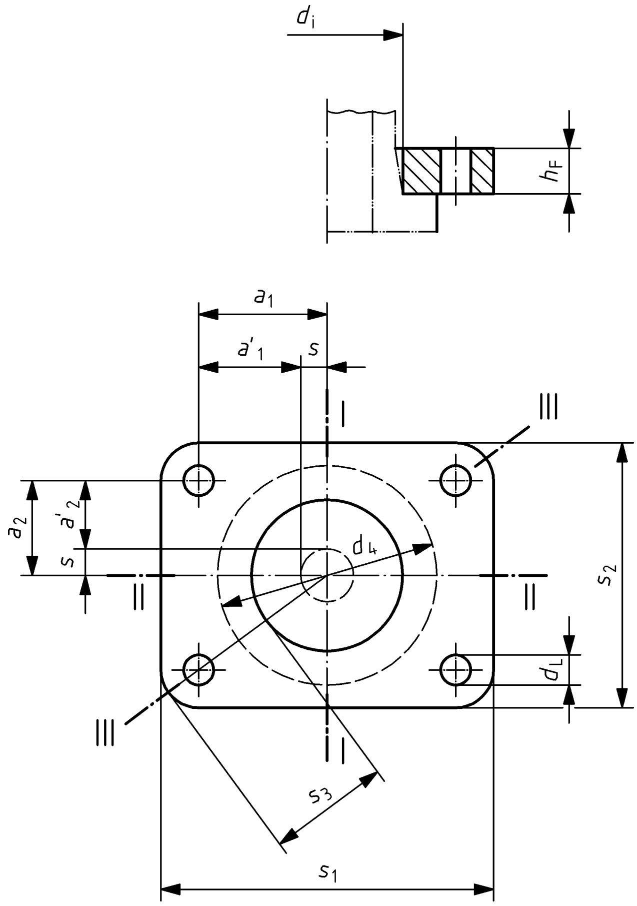


Figure 58 — Rectangular or square slip-on flange

The calculation shall be carried out for the cross-sections **I-I**, **II-II** and **III-III**. With the dimensions according to Figure 58 and the minimum bolt force F_S according to the Formulae (204), (206) and (208), the moments in the cross-sectional planes shall be calculated from:

$$M_{\mathrm{I}} = \frac{F_{\mathrm{s}}}{2} \cdot a_1^{'} = \frac{F_{\mathrm{s}}}{2} \cdot (a_1 - s) \tag{219}$$

$$M_{\mathrm{II}} = \frac{F_{\mathrm{s}}}{2} \cdot a_2^{'} = \frac{F_{\mathrm{s}}}{2} \cdot (a_2 - s) \tag{220}$$

$$M_{\mathrm{III}} = \frac{F_{\mathrm{s}}}{4} \left(\sqrt{a_1^2 + a_2^2} - s \right) \tag{221}$$

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

$$s = \frac{2}{3\pi} \cdot \frac{d_4^3 - d_{\mathrm{i}}^3}{d_4^2 - d_{\mathrm{i}}^2} \tag{222}$$

The flange resistance can be calculated analogously from:

$$W_{\mathrm{I}} = \frac{h_{\mathrm{F}}^2}{6} (s_2 - d_{\mathrm{i}}) \tag{223}$$

$$W_{\mathrm{II}} = \frac{h_{\mathrm{F}}^2}{6} (s_1 - d_{\mathrm{i}}) \tag{224}$$

$$W_{\mathrm{III}} = \frac{h_{\mathrm{F}}^2}{6} (s_3 - d_{\mathrm{L}}^{'}) \cdot 2 \tag{225}$$

with d'_{L} according to Formula (177).

The strength condition in accordance with Formulae (167) and (168) shall be satisfied for the three cross-sections.

It means $W \geq M / (1{,}5f)$. (226)

10.5 Calculation of the bolt diameter

10.5.1 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.

10.5.2 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:

$$d_{\mathrm{S}} = \sqrt{\frac{4}{\pi \cdot n} \cdot \frac{F_{\mathrm{T}}}{\mathrm{f} \cdot \eta}} + c = d_{s0} + c \tag{227}$$

where

F_{T} 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.

10.5.3 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.

10.5.4 Safety factors and allowances

The allowable bolt stress f is obtained from the (yield) strength parameter (see EN 10269 and EN ISO 3506-1) divided by the safety factor SF in accordance with Table 9.

Table 9 — Safety factors

Condition	Necked-down bolts	Rigid bolts
Operation	1,5	1,8
Assembly/Testing	1,1	1,3

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.

In the case of rigid bolts, the following values shall be entered for the design allowance c in Formula (227) for the operating condition:

c = 3 mm for $d_{S0} \leq 20$ mm

c = 1 mm for $d_{S0} \geq 50$ mm

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

$$c = (65 - d_{S0}) / 15 \tag{228}$$

For necked-down bolts c = 0 mm.

For assembly-, test- and incidents conditions c shall be also c = 0 mm.

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.

11 Calculation methods for glands

11.1 Loads

The components of the gland bolting are loaded by the stuffing box force F_{st} and eventually, by the additional forces F_z.

$$F_{SB} = F_{st} + F_z \tag{229}$$

The stuffing box force on the ring surface area of the gland is:

$$F_{\text{st}} = \left(d_{\text{ast}}^2 - d_{\text{ist}}^2\right) \cdot \frac{\pi}{4} \cdot p \qquad (230)$$

where

d_{ast} = stuffing box outside diameter;

d_{ist} = stuffing box 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 Formula (229); allowance c may be ignored in this case.

11.3 Gland flanges

Gland flanges shall be verified by means of the force F_{SB} in accordance with Formula (229) 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)

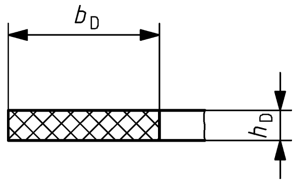
Characteristic values of gaskets and joints

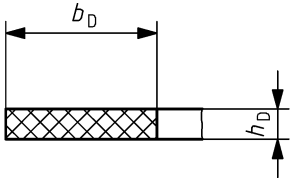
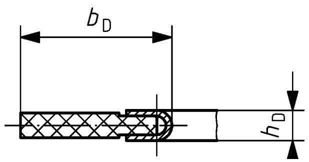
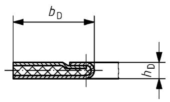
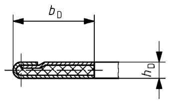
Tables A.1 and A.2 contain characteristic values for the calculation of flanged joints in accordance with this standard.

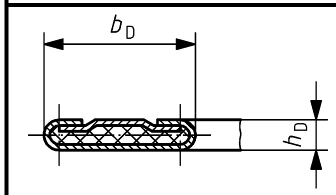
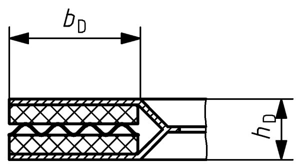
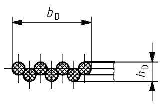
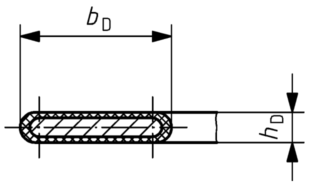
The definition “metallic envelope gasket” means combined seals.

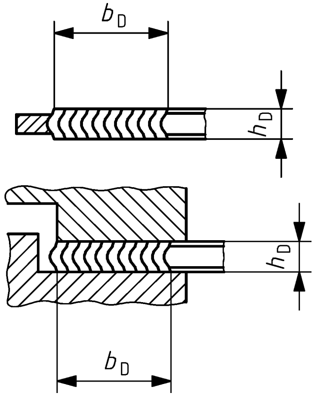
The characteristic factors for the gaskets σ_{VU}, σ_{VO} in Columns 3 and 4 of Table A.1 mean the minimum required and the maximum recommended gasket stress for assembly conditions. The factor m is a scale-factor used in Clause 10.

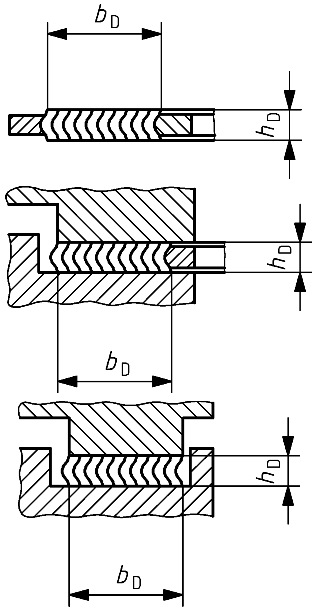
Table A.1 — Characteristic values of gaskets and joints

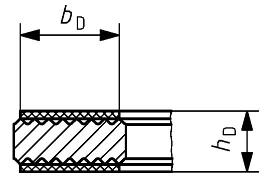
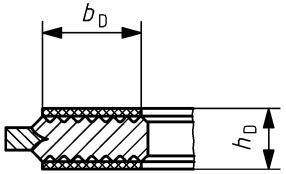
Non-metallic gaskets												
Shape	**Material**	**Assembly condition**		**Operating condition**								**Remarks**
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [a]							
Flat gasket 	Rubber, general nitrile rubber, chloroprene rubber	2	10	1,3	10	6	—	—	—	—	—	—
	Fluorine rubber	2	10	1,3	10	7	—	—	—	—	—	—
	PTFE h_D = 0,5		90		90							
	PTFE h_D = 1	10	70	1,1	70	40	25	—	—	—	—	Precondition for non-enclosed gasket $b_D/h_D = 20$
	PTFE h_D = 2		50		50							
b_D/h_D ≥ 20	Graphite [b] non-reinforced	15	150	1,3	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	—	

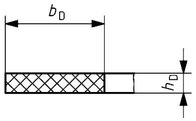
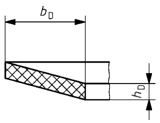
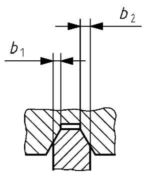
Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [a]							
Fibre -gasket restrained at inside rim 	Fibre, except inside rim with 0,25 mm 1.4541	50	135	1,3	135	66	62	58	55	—	—	h_D ≥ 3 mm
Fully enveloped fibre 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)	—	

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

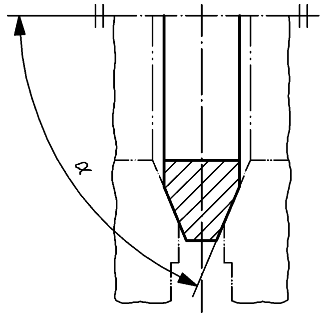
Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [a]							
Spiral wound gasket, single enclosure 	PTFE	20	110	1,3	110	110	100	(90)	—	—	—	() t_{max} = 250 °C gaskets with double enclosure shall be used, if possible
	Graphite	20	110		110	110	100	90	80	—	—	
	Fibre impregnated	55	150		150	140	—	—	—	—	—	

Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [a]							
Spiral wound gasket, double enclosure 	PTFE	20	300	-	300	170	160	(150)	—	—	—	—
	Graphite	20	300		300	170	160	—	—	—	—	
	Fibre impregnated	55	300		300	170	130	—	—	—	—	

Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [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/Fibre	65	350	1,3	350	320	290	265	—	—	—	—
	1.5415/ Fibre		450		450	400	360	330	270	220	—	
	1.4541/ Fibre		500		500	480	450	420	390	350	—	
	1.4828/ Fibre		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	

Non-metallic gaskets													
Shape	Material	Assembly condition		Operating condition									Remarks
				m	t °C								
					20	100	200	300	400	500	600		
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [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	

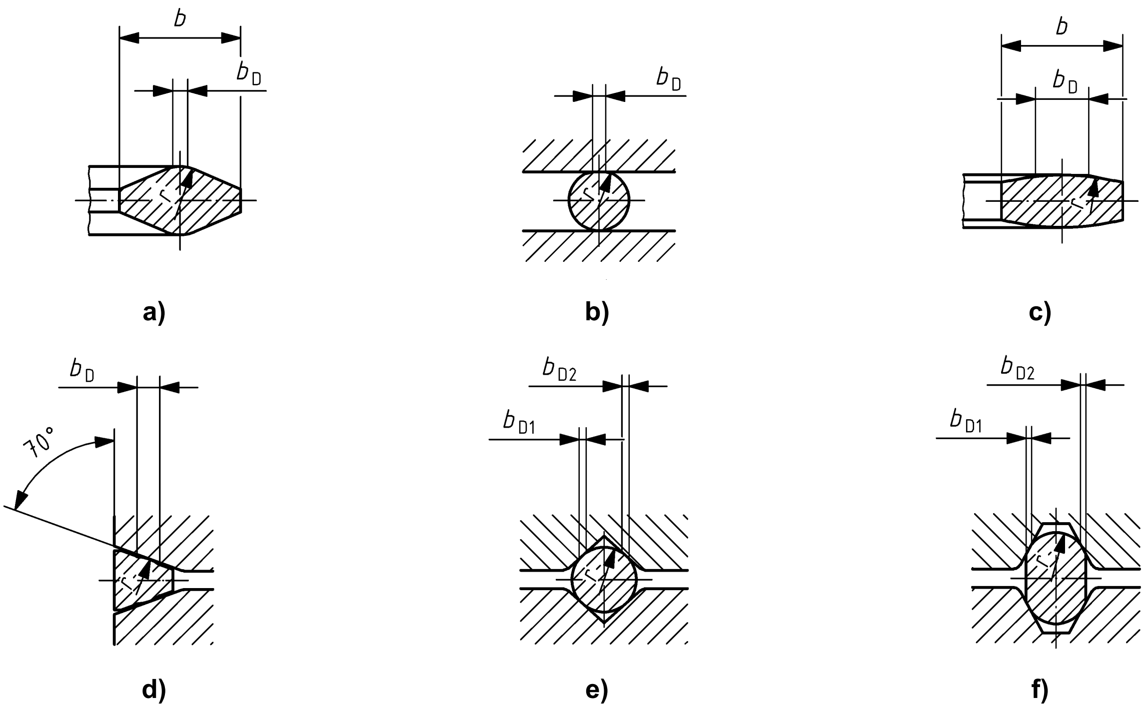
Non-metallic gaskets												
Shape	Material	Assembly condition		Operating condition								Remarks
				m	t °C							
					20	100	200	300	400	500	600	
		σ_{VU} MPa	σ_{VO} MPa		σ_{BO} MPa [a]							
	1.4828	400	900		900	855	810	750	690	600	480	
Lens-shaped gaskets	Al	70	140	1,3	140	120	93	—	—	—	—	The sealing width is calculated as follows: For shapes a) to c) by: $b_D = C^2 \cdot \frac{\sigma}{E_D} \cdot r$ (A.1) For shape d) (lenticular gasket, α = 70°): $b_D = C^2 \cdot \frac{\sigma}{E_D} \cdot r \cdot \sin\alpha$ (A.2) For shapes e) to f) (with contact at two faces) $b_D = 2C^2 \cdot \frac{\sigma}{E_D} \cdot r \cdot \sin\alpha$ (A.3)
	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	

Non-metallic gaskets												
Shape	**Material**	**Assembly condition**		**Operating condition**								**Remarks**
		σ$_{VU}$ MPa	**σ**$_{VO}$ MPa	**m**	t °C							
					20	100	200	300	400	500	600	
					σ$_{BO}$ MPa [a]							
The formulae given under “Remarks” for the sealing width of metallic lens-shaped gaskets according to Figure A.1a) to f) 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 A.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_{DV} according to Formula (135) or F_{DB} according to Formula (133) provide with the designated gasket force values in the range $2 \cdot r$ for b_D.					Angle α, for example on a lenticular gasket Pipe axis  							r mm / C: 5 to 20 — 10 > 20 to 80 — 8 > 80 to 120 — 6 α = Opening angle of the cone. For the ring-joint gasket shape f), α = 23°. As a function of the condition, σ_{VU}, σ_{VO} or σ_{BO} shall be put in for σ.

NOTE 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.



$b_D = b_{D1} + b_{D2}$ $b_D = b_{D1} + b_{D2}$

Figure A.1 — Sealing width

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

Gasket material	(Equivalent) modulus of elasticity E_D in MPa at a temperature of				
	20 °C	200 °C	300 °C	400 °C	500 °C
Fibre	1 000 up to 1 500	—	2 200	—	—
Rubber, soft (45 Shore-A)	approximately 30	—	—	—	—
Rubber, hard (80 Shore-A)	approximately 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

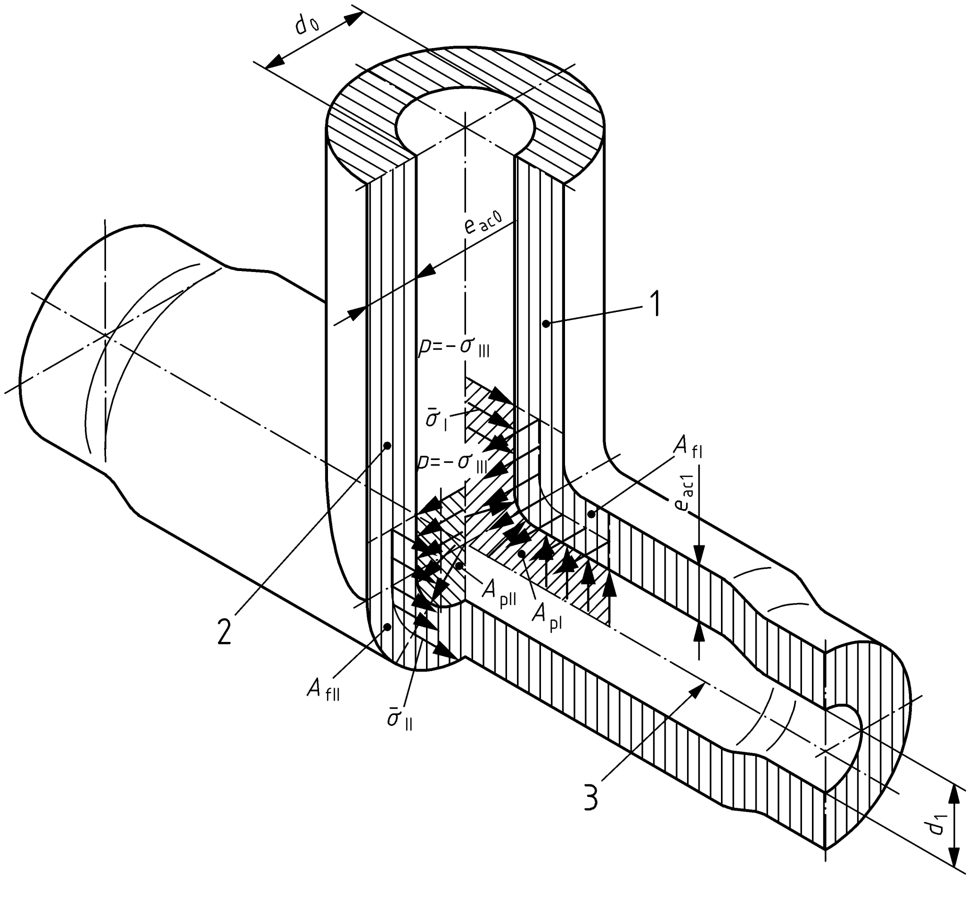
For non-metallic gaskets and metallic-envelope gaskets, the above values should be verified for the appropriate operating condition.

Annex B
(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 B.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 $\overline{\sigma_I}$. 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 B.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 $\overline{\sigma}_{II}$ (cross-section II). In such cases, the calculation is also carried out for both the longitudinal section and the transverse section (see also Figure 9).

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

Table ZA.1 — Correspondence between this European Standard and Directive 97/23/EC (PED)

Clause(s)/sub-clauses of this European Standard	Annex I of PED Essential Safety Requirements	Nature of requirement
4	2.1	General design
4 to 11	2.2.2, 2.2.3	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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