BS EN 1591-1:2013

BSI Standards Publication

Flanges and their joints — Design rules for gasketed circular flange connections

Part 1: Calculation

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National foreword

This British Standard is the UK implementation of EN 1591-1:2013. It supersedes BS EN 1591-1:2001+A1:2009 which is withdrawn.

The UK participation in its preparation was entrusted to Technical Committee PSE/15/2, Flanges - Jointing materials and compounds.

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

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Brides et leurs assemblages - Règles de calcul des assemblages à brides circulaires avec joint - Partie 1: Méthode de calcul

Flansche und ihre Verbindungen - Regeln für die Auslegung von Flanschverbindungen mit runden Flanschen - Teil 1: Berechnung

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Foreword

This document (EN 1591-1:2013) has been prepared by Technical Committee CEN/TC 74 “Flanges and their joints”, the secretariat of which is held by DIN.

This European Standard shall be given the status of a national standard, either by publication of an identical text or by endorsement, at the latest by June 2014, and conflicting national standards shall be withdrawn at the latest by June 2014.

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 1591-1:2001+A1:2009.

The major changes in comparison with the previous edition include:

— correction of load ratio calculation for blind flanges;

— integration of spacers (washers);

— modification of bolt load ratio calculation;

— integration of lateral forces and torsion moments applied on the bolted joint;

— integration of an alternative calculation method (more precise) for the determination of the gasket effective width (informative annex);

— integration of the possibility to handle gasket creep/relaxation behaviour through additional deflection;

— integration of an informative annex concerning leakage rates conversions;

— integration of the possibility to check a bolted flange connection for a specified initial bolt load value;

— integration of the possibility to perform a calculation even when no tightness requirement is defined through basic gasket parameters (Annex G).

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

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

EN 1591 consists of several parts:

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

— EN 1591-2, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 2: Gasket parameters*

— CEN/TS 1591-3, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 3: Calculation method for metal to metal contact type flanged joint*

— EN 1591-4, *Flanges and their joints — Part 4: Qualification of personnel competency in the assembly of the bolted connections of critical service pressurized systems*

- CEN/TR 1591-5, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 5: Calculation method for full face gasketed joints*

The calculation method satisfies both leak tightness and strength criteria. The behaviour of the complete flanges-bolts-gasket system is considered. Parameters taken into account include not only basic ones such as:

- fluid pressure;
- material strength values of flanges, bolts and gaskets;
- gasket compression factors;
- nominal bolt load;

but also:

- possible scatter due to bolting up procedure;
- changes in gasket force due to deformation of all components of the joint;
- influence of connected shell or pipe;
- effect of external axial and lateral forces and torsion and bending moments;
- effect of temperature difference between bolts and flange ring.

The use of this calculation method is particularly useful for joints where the bolt load is monitored when bolting up. The greater the precision of this, the more benefit can be gained from application of the calculation method.

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.

1 Scope

This European Standard defines a calculation method for bolted, gasketed, circular flange joints. Its purpose is to ensure structural integrity and control of leak tightness. It uses gasket parameters based on definitions and test methods specified in EN 13555.

The calculation method is not applicable to joints with a metallic contact out of the sealing face or to joints whose rigidity varies appreciably across gasket width. For gaskets in incompressible materials, which permit large deformations, the results given by the calculation method can be excessively conservative (i.e. required bolting load too high, allowable pressure of the fluid too low, required flange thickness too large, etc.).

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 13555:2004, *Flanges and their joints — Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connections*

3 Notation

3.1 Use of figures

Figure 1 to Figure 14 illustrate the notation corresponding to the geometric parameters. They only show principles and are not intended to be practical designs. They do not illustrate all possible flange types for which the calculation method is valid.

NOTE For standard flange types, e.g as shown in EN 1092 or EN 1759, the relevant figures are the following:

Type 01	Figure 10
Type 02	Figure 12
Type 04	Figure 12
Type 05	Figure 11
Type 07	Figure 12
Type 11	Figure 6
Type 12	Figure 13
Type 13	Figure 14
Type 21	Figures 6 to 9

3.2 Subscripts and special marks

3.2.1 Subscripts

A – Additional (F_A, M_A)

B – Bolt

C – Creep of gasket (Δe_{Gc})

D – Equivalent cylinder (tapered hub + connected shell) for load limit calculation

E – Equivalent cylinder (tapered hub + connected shell) for flexibility calculation

F – Flange

G – Gasket

H – Hub

I – Load condition identifier (taking values 0, 1, 2 ...)

L – Loose flange, Lateral (F_{LI})

M – Moment

N – Nut

P – Fluid pressure

Q – Net axial force due to pressure

R – Net axial force due to external force

S – Shell, shear

T – Shell, modified

TG – Torsion (M_{TG})

X – Flange weakest cross section

W – Washer

Δ – Symbol for change or difference

av – average

c – calculated

d – design

e – effective

i – Interim value

max – maximum

min – minimum

nom – nominal

opt – optimal

req – required

s – non-threaded part of bolt

specified – refers to the case of calculation performed for a given (specified) initial bolt load

t – theoretical, torque, thread

0 – initial bolt-up condition (I = 0, see subscript I)

3.2.2 Special marks

~ – Accent placed above symbols of flange parameters that refers to the second flange of the joint, possibly different from the first.

3.3 Symbols

Where units are applicable, they are shown in brackets. Where units are not applicable, no indication is given.

A_B	Effective total cross-section area of all bolts [mm^2], Formula (41)
A_F, A_L	Gross radial cross-section area (including bolt holes) of flange ring, loose flange [mm^2], Formulae (10), (13) and (16)
A_{Ge}, A_{Gt}	Gasket area, effective, theoretical [mm^2], Formulae (56), (53)
A_Q	Effective area for the axial fluid-pressure force [mm^2], Formula (90)
E_B, E_F, E_L E_W	Modulus of elasticity of the part designated by the subscript, at the temperature of the part [MPa]
$E_{G,}$	Modulus of elasticity of the gasket for unloading/reloading at the considered temperature, considering the initial compressed thickness [MPa]
F_A	Additional external axial force [N], tensile force > 0, compressive force < 0, see Figure 1, Formulae (92) and (96)
F_B	Bolt force (sum of all bolts) [N]
F_G	Gasket force [N]
$F_{G\Delta,}$	Minimum gasket force in assembly condition [N] that guarantees, after all load changes, to subsequent conditions the required gasket force, Formulae (105), (106)
F_L	Force resulting from the additional radial forces [N], Formula (93) and (104)
F_Q	Axial fluid-pressure force [N], Formula (91)
F_R	Force resulting from the additional external loads [N], Formula (96)
F_X, F_Y, F_Z	Additional forces along X, Y and Z-axis at gasket interface [N], Formulae (92) and (93)
I	Load condition identifier, for assembly condition I = 0, for subsequent conditions I = 1, 2, 3,...
M_A	Resulting external bending moment [N × mm], Figure 1, Formula (94) and (104)
M_t	Bolt assembly torque [N × mm], Formula (B.4)

$M_{t,B}$	Twisting moment [N × mm] applied to bolt shanks as a result of application of the bolt assembly torque M_t, Formula (B.9)
M_{TG}	Additional external torsion moment due to friction, Formula (95) and (104)
N_R	Number of re-assemblies and re-tightenings during service life of joint, Formulae (119), (2)
P	Pressure of the fluid [MPa], internal pressure > 0, external pressure < 0 (1 bar = 0,1 MPa), Formula (91)

NOTE P in this standard is equal to the maximum allowable pressure PS according to the PED.

P_{QR}	Creep factor which is the ratio of the residual and the original gasket surface pressure at load conditions [-] (Annex F).
Q_G	Mean effective gasket compressive stress [MPa], $Q_G = F_G/A_{Ge}$ (57)
Q_A	Gasket surface pressure at assembly prior to the unloading which is necessary for the validity of the corresponding $Q_{smin\ (L)I}$ in all subsequent conditions [MPa], Formula (103).The lowest acceptable value for Q_A is $Q_{min\ (L)}$ from EN 13555.
$Q_{0,min}$	Gasket surface pressure required at assembly prior to the unloading when no specific leak rate is requested [MPa], replacement of Q_A in Formula (103), Annex G
$Q_{min\ (L)}$	Minimum level of gasket surface pressure required for tightness class L at assembly (on the effective gasket area) from EN 13555 test results [MPa] (see 7.4.2 NOTE 1)
$Q_{smin\ (L)}$	Minimum level of gasket surface pressure required for tightness class L in service conditions (on the effective gasket area) from EN 13555 test results [MPa], Formula (104)
Q_{smax}	Maximum gasket surface pressure that can be safely imposed upon the gasket at the considered temperature without damage [MPa], Formula (65), (70), (75) and (128)
T_B, T_F, T_G, T_L, T_W	Temperature (average) of the part designated by the subscript [°C] or [K], Formula (97)
T_O	Temperature of joint at assembly [°C] or [K] (usually + 20 °C), Formula (97)
U^I	Axial displacement due to thermal effect [mm]; ΔU^T according to Formula (97)
W_F, W_L, W_X	Resistance of the part and/or cross-section designated by the subscript [N × mm], Formulae (130), (146), (150), (148)
X_B, X_G, X_w	Axial flexibility modulus of bolts, gasket, washer [1/ mm], Formulae (42), (63), (43), (49), (50)
Y_B, Y_G, Y_Q, Y_R	Axial compliance of the bolted joint, related to F_B, F_G, F_Q, F_R [mm/N], Formulae (99), (100), (101), (102)
Z_F, Z_L	Rotational flexibility modulus of flange, loose flange [mm $^{-3}$], Formulae (34), (38), (35), (39), (40)
b_0	Width of chamfer (or radius) of a loose flange such that: $d_{7min} = d_6+2\times b_0$ [mm], Figure 12, Formula (85)

b_F, b_L	Effective width of flange, loose flange [mm], Formulae (7) to (14)
b_{Gi}, b_{Ge}, b_{Gt}	Gasket width (radial), interim, effective, theoretical [mm], Formula (51), (55), (64), (65), (69), (70), (72), (74) and (75)
b_{KB}	Contact widths bolt side [mm], Formula (48)
b_W	Width of a washer [mm], Formula (44)
c_A, c_B, c_F, c_M, c_S	Correction factors [-],Formulae (123) to (127), (28), (134), (135)
d_0	Inside diameter of flange ring [mm] and also the outside diameter of central part of blank flange (with thickness e_0), in no case greater than inside diameter of gasket [mm], Figures 6 to 14
d_1	Average diameter of hub, thin end [mm], Figures 6, 7, 13 and 14
d_2	Average diameter of hub, thick end [mm], Figures 6, 7, 13 and 14
d_3, d_{3e}	Bolt circle diameter, real, effective [mm], Figures 6 to 14, Formula (6)
d_4	Outside diameter of flange [mm], Figures 6 to 14
d_5, d_{5t}, d_{5e}	Diameter of bolt hole, pierced, blind, effective [mm], Figures 6 to 14, Formulae (4), (5)
d_6	Inside diameter of loose flange [mm], Figures 12, 14
d_7	Diameter of position of reaction between loose flange and stub or collar [mm], Figure 1, Formulae (61) and (84) to (89).
d_8	Outside diameter of collar [mm], Figure 12
d_9	Diameter of a central hole in a blank flange [mm], Figure 11
d_{B0}, d_{Be}, d_{Bs}	Diameter of bolt: nominal diameter, effective diameter, shank diameter [mm], Figure 3, Table A.1
d_{B2}, d_{B3}	Basic pitch diameter, basic minor diameter of thread [mm], see Figure 3
d_{B4}	Maximum possible outside contact diameter between bolt head or nut and flange or washer [mm], Formula (47)
d_{Gi},d_{Ge}, d_{Gt}	Diameter of gasket, interim, effective, theoretical [mm], Figure 4, Formula (56), Table 1
d_{K1}, d_{K2}	Extreme contact diameters (inside, outside) [mm], Formulae (46) and (47)
d_{G0}, d_{G1}, d_{G2}	Real, theoretical inside, theoretical outside contact diameters [mm], Figure 4
d_E, d_F, d_L d_S, $d_{X,}$ d_w	Average diameter of part or section designated by the subscript [mm], Figures 1 and 6 to 14
$d_{w1,}$ d_{w2}	Inside, Outside diameter of washer [mm], Figure 1, 2
e_0	Wall thickness of central plate of blank flange within diameter d_0 [mm], Figure 11

e_1	Minimum wall thickness at thin end of hub [mm], Figures 6, 7, 13, 14
e_2	Wall thickness at thick end of hub [mm], Figures 6, 7, 13, 14
e_D, e_E	Wall thickness of equivalent cylinder for load limit calculations, for flexibility calculations [mm], Formulae (17) and (18)
e_F, e_L	Effective axial thickness of flange, loose flange [mm], Formulae (10), (13) and (16)
e_{Fb}	Thickness of flange ring at diameter d_3 (bolt position) [mm], Formula (5)
e_{Ft}	Thickness of flange ring at diameter d_{Ge} (gasket force position), relevant for thermal expansion [mm], Formula (98)
$e_G(Q_{G0})$	Initial compressed gasket thickness of gasket under contact pressure Q_{G0} [mm], Formulae (106), (121) can be obtained from the tests according to EN 13555
$e_{G(A)}$	Compressed gasket thickness of gasket after all the situations (including plastic deformation) [mm], Formulae (106), (121) and Annex H
e_{Gt}	Initial theoretical uncompressed thickness of gasket [mm]
e_P, e_Q	Part of flange thickness with (e_P), without (e_Q) radial pressure loading [mm], Figures 6 to 14, such that $e_P+e_Q = e_F$
e_S	Thickness of connected shell [mm], Figures 6 to 10, 12 to 14
e_W	Washer thickness [mm], Figure 1, 2
e_X	Flange thickness at weak section [mm], Figure 11
f_B, f_E, f_F, f_L, f_S, f_W	Nominal design stress [MPa] of the part designated by the subscript, at design temperature [°C] or [K], as defined and used in pressure vessel codes (see Formulae (123), (127), (130) to (133), (140), (145), (146), (148), (150) and (151))
h_G, h_H, h_L	Lever arms [mm], Figure 1, Formulae (81) to (83) and (87) to (89)
h_P, h_Q, h_R, h_S, h_T	Lever arm corrections [mm], Formulae (77), (79) and (80), (31) and (37), (29), (30)
j_M, j_S	Sign number for moment, shear force (+1 or 1), Formulae (136) and (137)
k_Q, k_R, k_M, k_S	Correction factors, Formulae (32), (33), (138), (139)
l_B, l_s	Bolt axial dimensions [mm], Figure 2, Formulae (98) and (42)
l_e	$l_e = l_B - l_S$
l_H	Length of hub [mm], Figures 6, 7, 13, 14, Formulae (17), (18)
m	tightness factor for subsequent conditions (I>0) [-], (Annex G)
n_B	Number of bolts, Formulae (3), (6), (41), (42)
p_B	Pitch between bolts [mm], Formula (3)
p_t	Pitch of bolt thread [mm], Table A.1

r_0, r_1	Radii [mm], Figures 6, 12
r_2	Radius of curvature in gasket cross-section [mm], Figure 4
ΔU^{I}	Differential thermal axial expansions [mm], Formula (97)
Δe_{Gc}	Additional deflection of the gasket due to creep that can be defined from P_{QR} value following the method explained in Annex F (Formula F.3). Equal to 0 if no creep of the gasket is considered, Formulae (105), (106), (120) and (121)
Θ_F, Θ_L	Rotation of flange, loose flange, due to applied moment [rad], Annex C
Ψ	Load ratio of flange ring due to radial force, Formula (140)
Ψ_Z	Particular value of Ψ, Formula (130), Table 2
Φ_B, Φ_F, Φ_G, Φ_L, Φ_X,	Load ratio of part and/or cross-section designated by the subscript, to be calculated for all load conditions, Formulae (123), (129), (145), (151), (128), (149), (147)
α_B, α_F, α_G, α_L, α_W	Thermal expansion coefficient of the part designated by the subscript, averaged between T_0 and T_B, T_F, T_G, T_L, T_S, T_W [K^{-1}], Formula (97)
β, γ, δ, v, κ, λ, x	Intermediate variables, Formulae (19), (25) to (27), (62), (132), (133)
ε_{1+}, ε_{1-}	Scatter of initial bolt load of a single bolt, above nominal value, below nominal value, Annex B
ε_+, ε_-	Scatter for the global load of all the bolts above nominal value, below nominal value, Annex B
μ	Friction factor for bolting, see Annex B
μ_G	Friction factor between the gasket and the flange facing, Table (E.1) and Formula (104)
π	Numerical constant (π = 3,141593)
ρ	Diameter ratio as given in Formula (36)
φ_G	Angle of inclination of a sealing face [rad or deg], Figure 4, Table 1
φ_S	Angle of inclination of connected shell wall [rad or deg], Figures 8, 9

3.4 Terminology

3.4.1 Flanges

Integral flange:	Flange attached to the shell either by welding (e.g. neck weld, see Figure 6 to Figure 9, or slip on weld, see Figure 10 and Figure 13) or cast onto the envelope (integrally cast flanges, type 21)
Blank or blind flange:	Flat closure, see Figure 11
Loose flange:	Separate flange ring abutting a collar, see Figure 12
Hub:	Axial extension of flange ring, usually connecting flange ring to shell, see Figure 6 and Figure 7
Collar or stub:	Abutment for a loose flange, see Figure 12

3.4.2 Loading

External loads:	Forces and/or moments applied to the joint by attached equipment, e.g. weight and thermal expansion of pipes

3.4.3 Load conditions

Load condition:	State with set of applied simultaneous loads; designated by I.
Assembly condition:	Load condition due to initial tightening of bolts (bolting up), designated by I = 0
Subsequent condition:	Load condition subsequent to assembly condition, e.g. test condition, operating condition, conditions arising during start-up and shut-down; designated by I = 1, 2, 3 ...

3.4.4 Compliances

Compliance:	Inverse stiffness (axial), symbol Y, [mm/N]
Flexibility modulus:	Inverse stiffness modulus, excluding elastic constants of material:

- axial: symbol X, [1/mm]
- rotational: symbol Z, [1/mm^3]

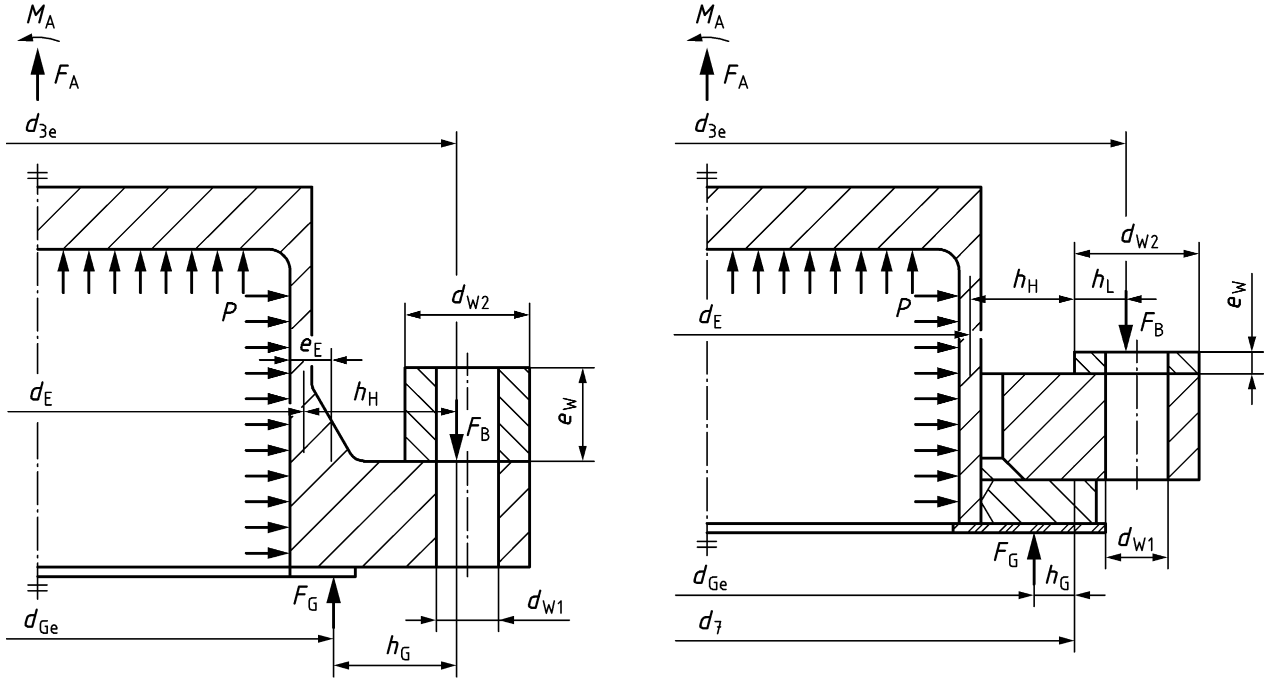


Figure 1 — Loads and lever arms

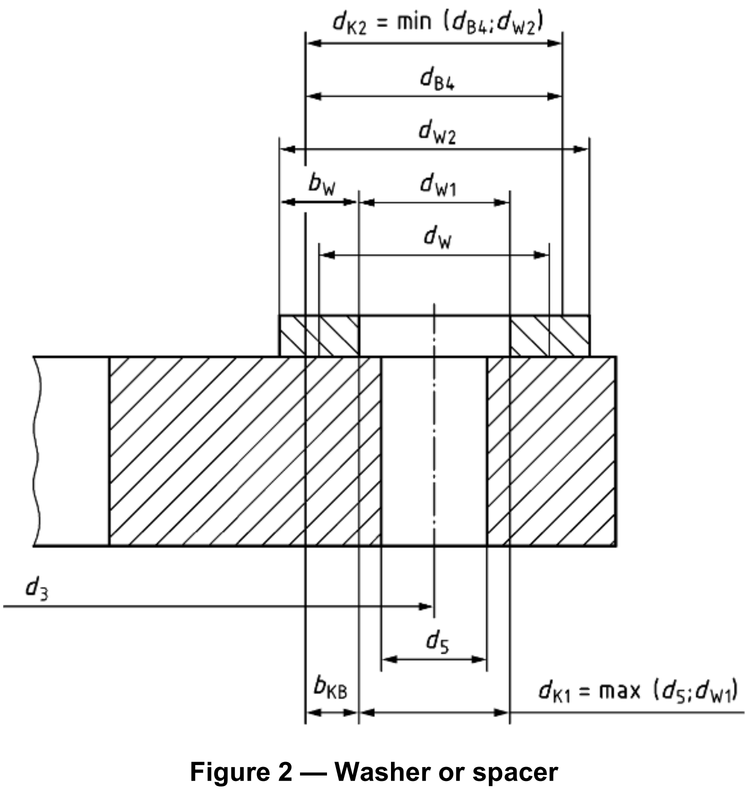
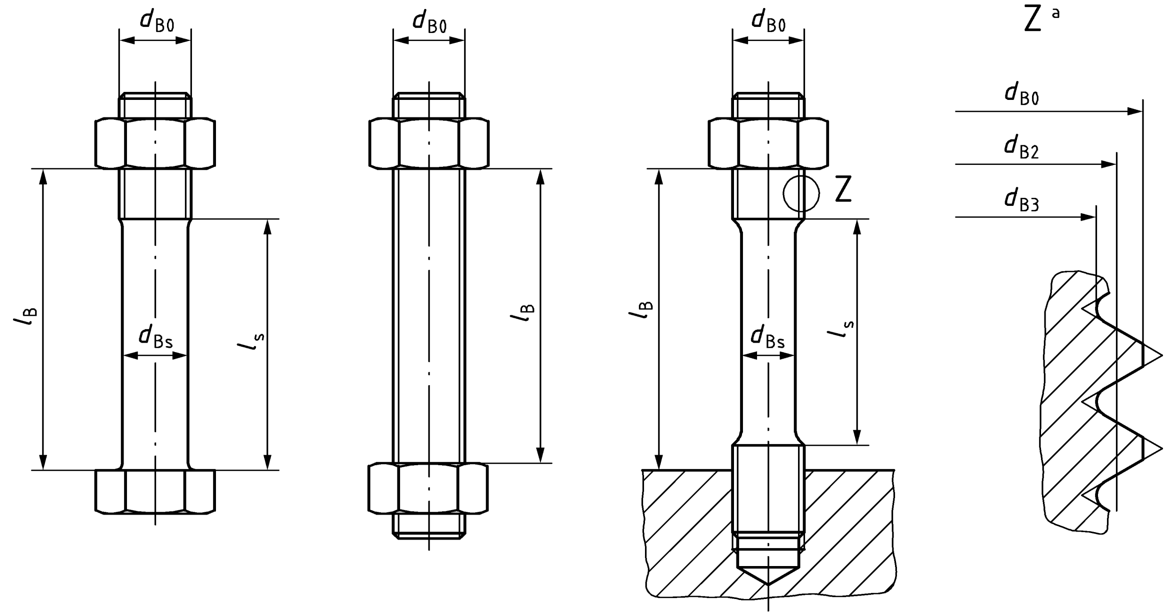


Figure 2 — Washer or spacer



$l_e = l_B - l_s$

Figure 3 — Bolts

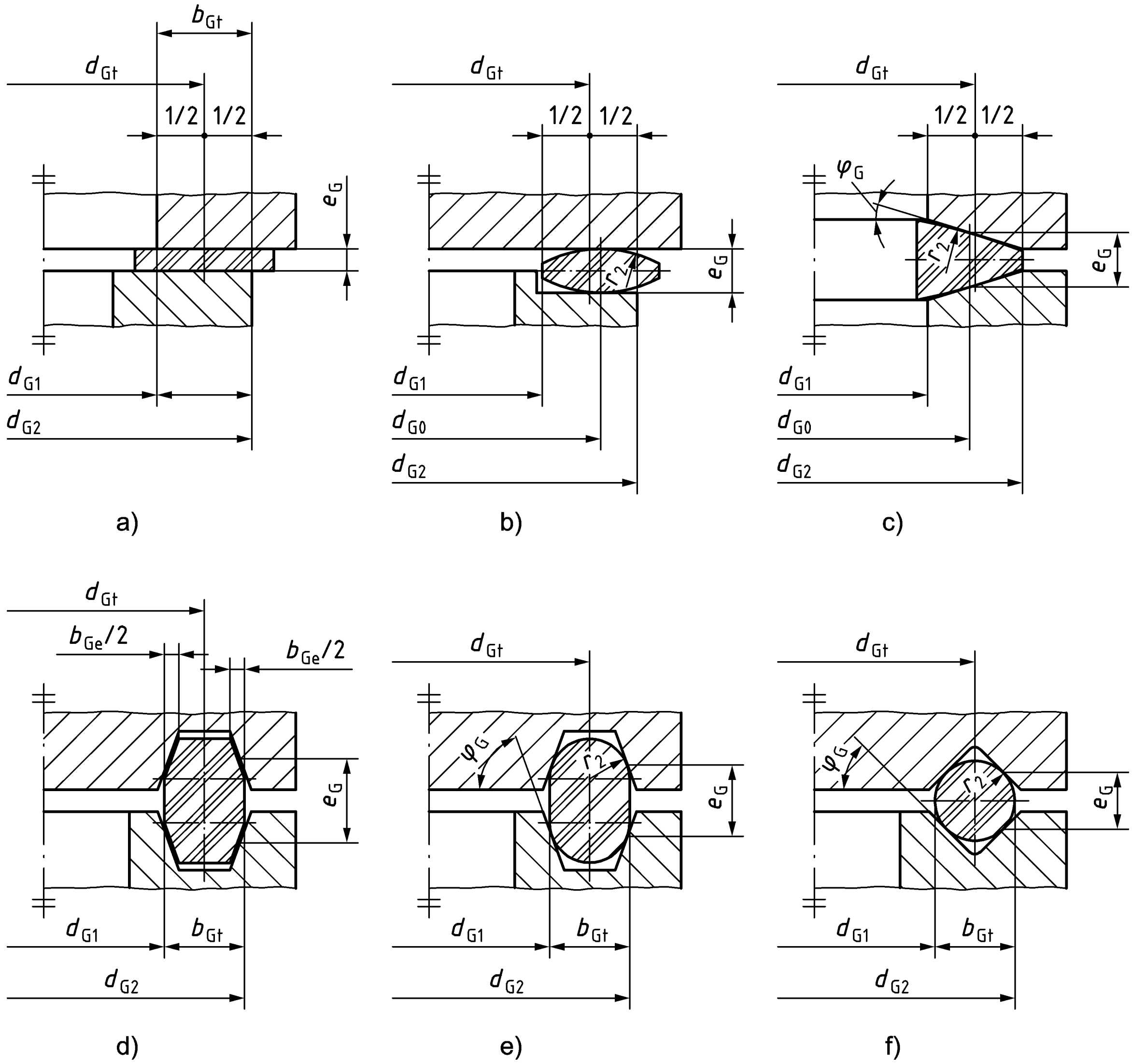
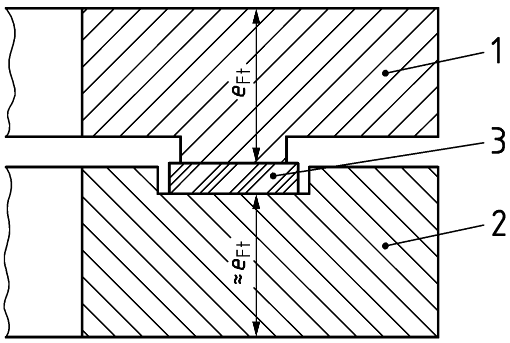


Figure 4 — Gaskets



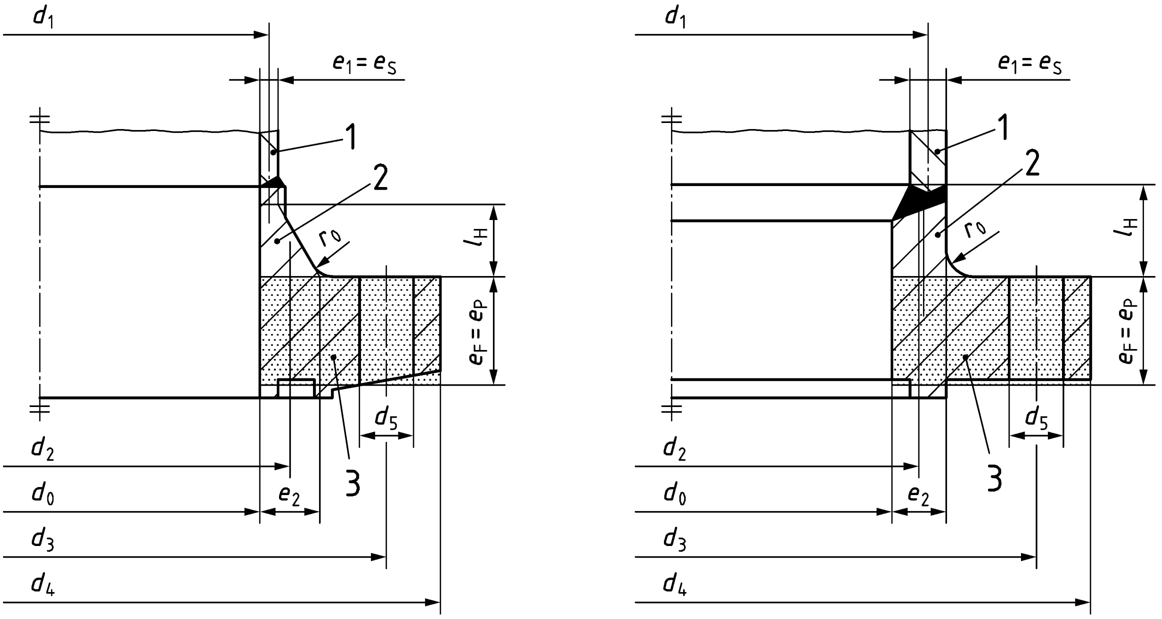
Key

1 male flange (tongue)

2 female flange (groove)

3 gasket

Figure 5 — Details for tongue and groove facing



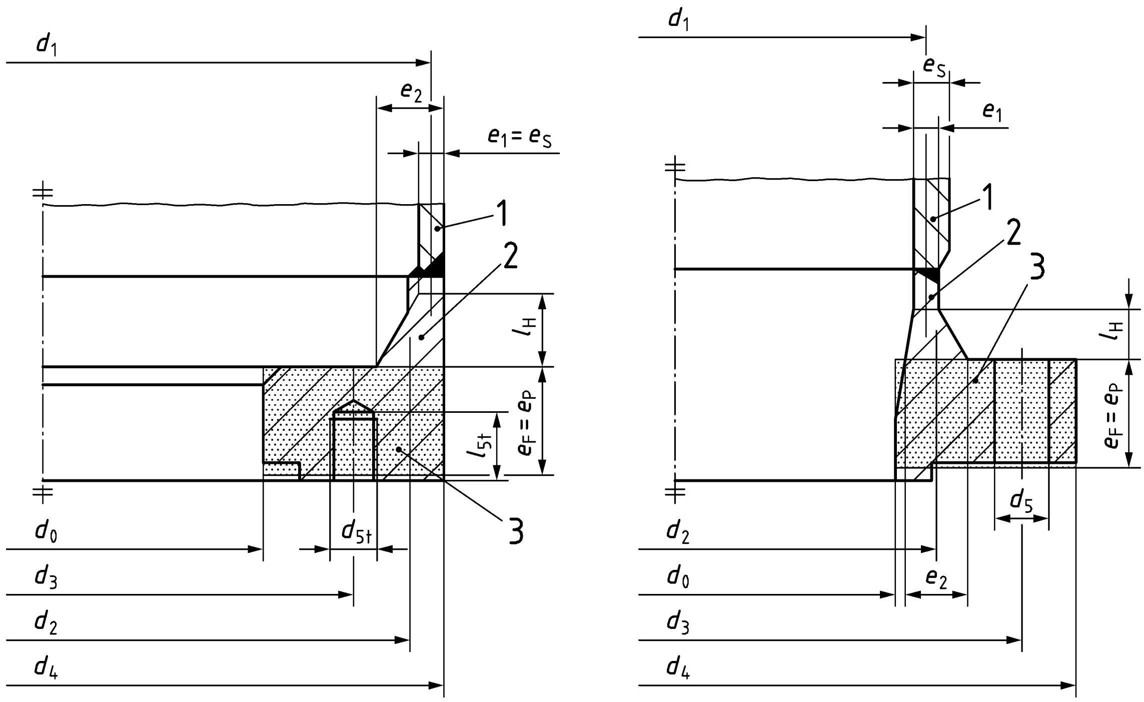
Key

1 shell

2 hub

3 ring

Figure 6 — Weld-neck flanges with cylindrical shells (example 1)



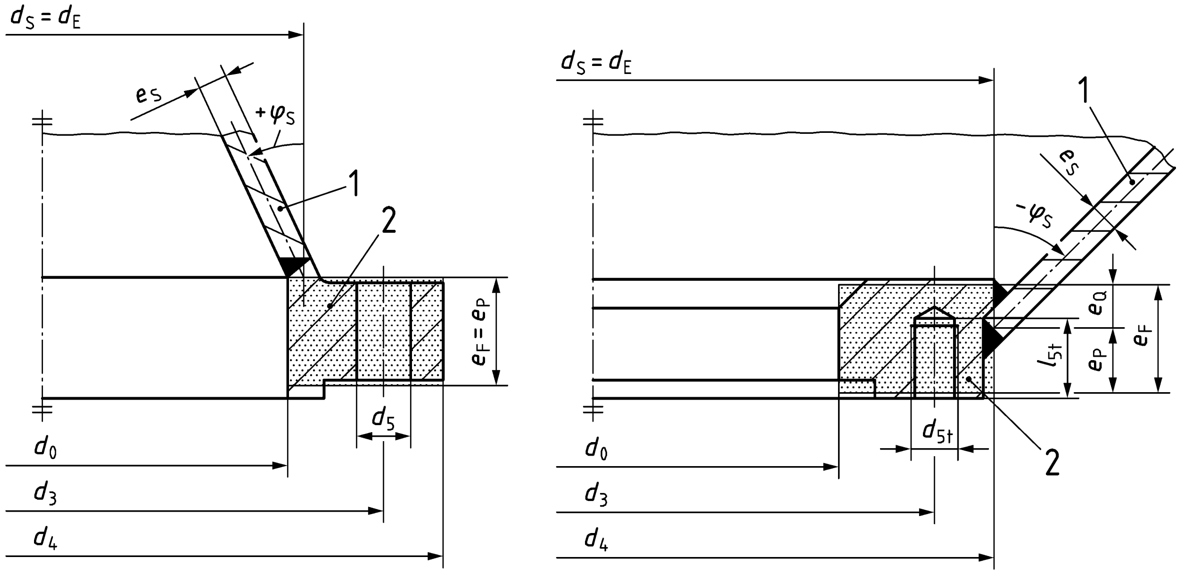
Key

1 shell

2 hub

3 ring

Figure 7 — Weld-neck flanges with cylindrical shells (example 2)

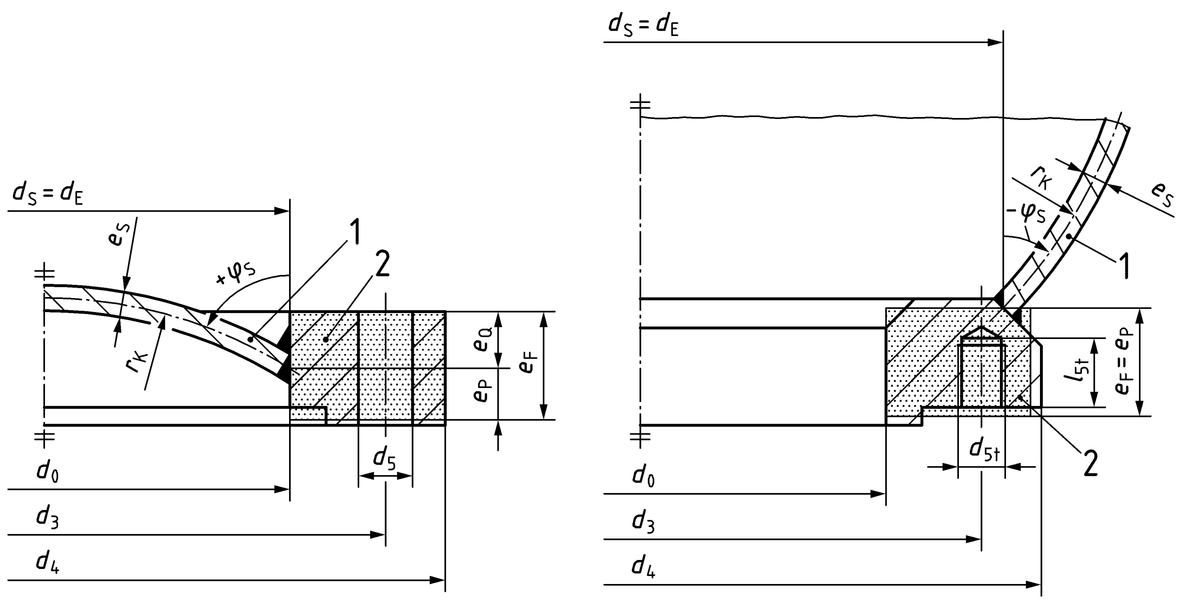


Key

1 shell

2 ring

Figure 8 — Flanges welded to conical shells

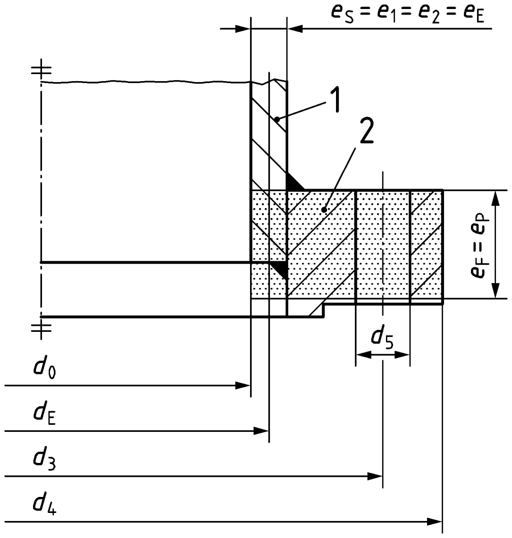


Key

1 shell

2 ring

Figure 9 — Flanges welded to spherical shells

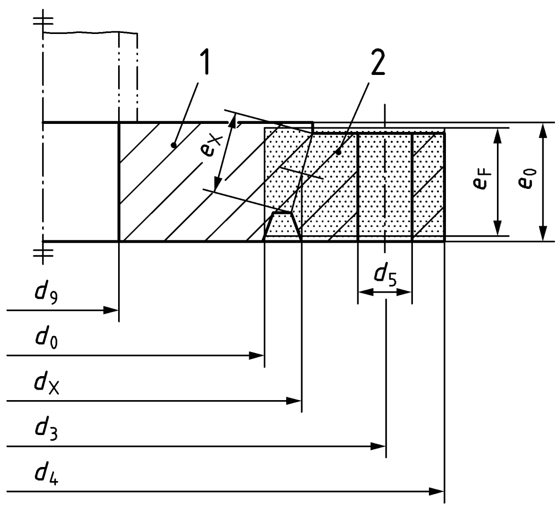


Key

1 shell

2 ring

Figure 10 — Weld-on plate flange

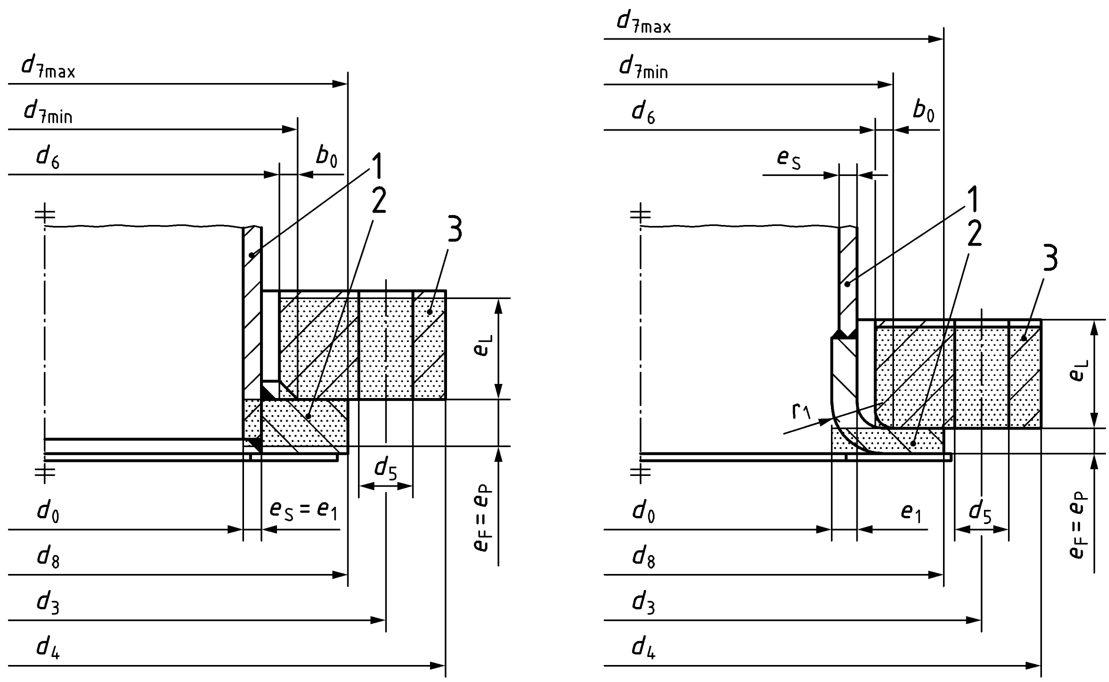


Key

1 plate

2 ring

Figure 11 — Blank flange



Key

1 shell

2 collar

3 loose flange

Figure 12 — Loose flanges with collar

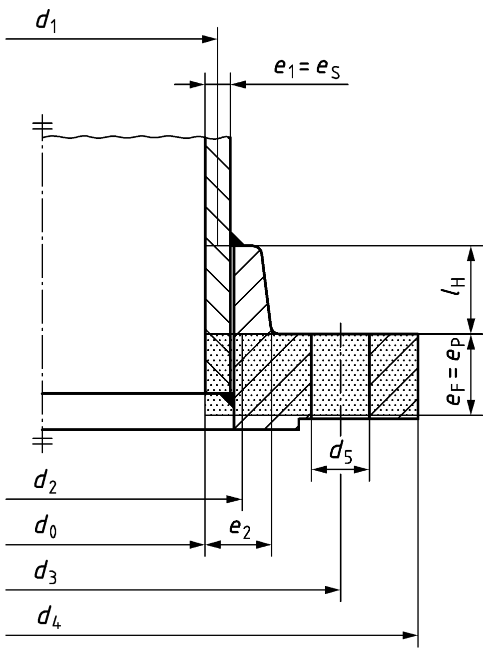


Figure 13 — Hubbed slip-on welded flange

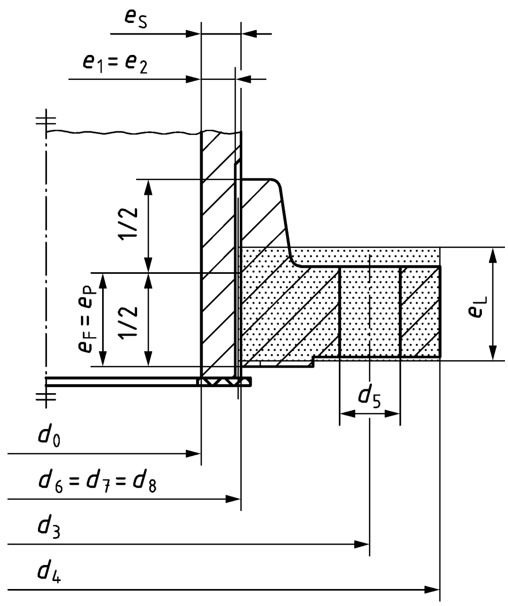


Figure 14 — Hubbed threaded flange

4 Requirements for use of the calculation method

4.1 General

Where permitted, the calculation method is an alternative to design validation by other means, e.g.:

— special testing;

— proven practice;

— use of standard flanges within permitted conditions.

The calculation method can also be used to assess the behaviour and admissibility of a bolted flange connection for a specified initial bolt force (see Clause 5).

4.2 Geometry

The calculation method is applicable to the configurations having:

a) flanges whose section is given or may be assimilated to those given in Figure 6 to Figure 14;

b) four or more identical bolts uniformly distributed;

c) gasket whose section and configuration after loading can be assimilated by one of those given in Figure 4 and Figure 5;

d) flange dimension which meet the following conditions:

1) $0{,}2 \leq b_F / e_F \leq 5{,}0; 0{,}2 \leq b_L / e_L \leq 5{,}0;$

2) $\cos\varphi \geq 1/(1 + 0{,}01 \dfrac{d_S}{e_S})$.

NOTE 1 For explanations of symbols see Clause 3.

NOTE 2 The condition $b_F/e_F \leq 5,0$ need not be met for collar in combination with loose flange.

Where corrosion allowance has been applied in the design it should be subtracted for the calculation on the area in contact with the fluid. For minus tolerances, reference should be made to other codes, for example EN 13445 and EN 13480.

The following configurations are outside the scope of the calculation method:

- flanges of essentially non-axisymmetric geometry, e.g. split loose flanges, web reinforced flanges;
- flange connections having direct or indirect metal to metal contact between flanges inside and/or outside the gasket, inside and/or outside the bolt circle.

4.3 Material

Values of nominal design stresses are not specified in this calculation method. They depend on other codes which are applied, for example these values are given in EN 13445 and EN 13480.

Nevertheless, since all significant design parameters are accounted for, the use of low safety factors is made possible by special use of nominal design stresses:

- for assembly conditions the nominal design stresses have the same values as for the hydraulic pressure tests (normally higher than for operating conditions);
- the nominal design stresses for the bolts are determined by the same rules as relevant for the flange and shell material e.g. same safety factor on yield stress.

4.4 Loads

This calculation method applies to the following load types:

- fluid pressure: internal or external;
- external loads: axial and lateral forces as torsion and bending moments;
- axial expansion of flanges, bolts and gasket, in particular due to thermal effects.

All conditions shall be taken into account (start-up, test, service, cleaning, maintenance, shut down, and other exceptional conditions) within the calculation as far as they have influence on the design.

Minimum required are calculations for the assembly conditions, the main operating and the initial test conditions. If the test shall not be repeated at any time, the calculations may be separated into two sets:

- A: Assembly + operating;
- B: Assembly + test.

The higher assembly bolt load shall be applied.

5 Checking the assembly for a specified initial tightening bolt force (or torque)

The details of calculation method as the calculation process are detailed in Annex D.

EN 1591-1 is based upon the principal that a selected leakage rate is to be achieved. Nevertheless, in the case where the aim of the calculation is to check the design for a given value of the tightening bolting force at assembly ($F_{B0,specified}$), the calculation shall be started using Formula (1) below instead of Formula (54) in 6.4.3.

$$F_{G0} = F_{B0,specified} \times (1 - \varepsilon_{-}) - F_{R0} \tag{1}$$

NOTE This formula involves the scatter of the bolting method in order to check the tightness criteria for the expected minimum value of tightening bolt force.

Then the calculation shall be performed in the usual way from Formula (55) to Formula (110). From the required initial bolt force calculated in Formula (110) two cases shall be considered:

— If the value F_{G0req} given by Formula (110) is higher than the initial value F_{G0} given by Formula (1), the value of $F_{B0,specified}$ is not sufficient to insure the tightness criteria. So the value of $F_{B0,\ specified}$ shall be increased to meet the tightness criteria. The calculation procedure from Formula (55) to Formula (110) shall be applied again.

— If the value F_{G0req} given by Formula (110) is lower than the initial value F_{G0} given by Formula (1), the value of $F_{B0,specified}$ is sufficient to insure the tightness criteria and therefore the calculation can be continued using the value of F_{G0} calculated by Formula (1) as the gasket force in assembly condition (I=0). In that case, the initial bolt force at assembly can be very much greater than the required one, and the Formula (119) shall be replaced by Formula (2), taking into account the lower bound of the applied initial bolt force at assembly phase.

$$F_{G0d} = \max\{F_{B0min} - F_{R0}; (2/3) \times (1 - 10/N_R) \times F_{B0max} - F_{R0}\} \tag{2}$$

6 Calculation parameters

6.1 General

The parameters defined in this clause are effective dimensions, areas and stiffness parameters.

6.2 Flange parameters

6.2.1 General

The formulae given in 6.2 shall be used for each of the two flanges and where applicable, the two collars of a joint.

Specific flange types are treated as follows:

— Integral flange: calculated as an equivalent ring with rectangular cross-section, dimensions $b_F \times e_F$ connected at diameter d_E to an equivalent shell of constant wall thickness e_E.

— Blank flange: calculated as an equivalent ring with rectangular cross-section, dimensions $b_F \times e_F$, connected at diameter $d_E = d_0$ to a plate of constant thickness e_0. It may have a central opening of diameter d_9. If a nozzle is connected at the opening the nozzle is not taken into account in the calculation.

— Loose flange: calculated as an equivalent ring with rectangular cross-section dimensions $b_L \times e_L$ without connection to a shell.

— Screwed flange: calculated as a loose flange with inside diameter equal to load transmission diameter, i.e. average thread diameter.

— Collar: The collar is treated in the same way as an integral flange.

In Figure 6 to Figure 14 the equivalent ring is sketched by shaded area.

6.2.2 Flange ring

6.2.2.1 Bolt holes

Pitch between bolts:

$$p_B = \pi \times d_3 / n_B \qquad (3)$$

Effective diameter of bolt hole:

$$d_{5e} = d_5 \times \sqrt{\frac{d_5}{p_B}} \qquad (4)$$

Diameter of blind holes is assumed to be:

$$d_5 = d_{5t} \times l_{5t} / e_{Fb} \qquad (5)$$

Effective bolt circle diameter:

$$d_{3e} = d_3 \times (1 - \frac{2}{n_B^2}) \qquad (6)$$

NOTE 1 p_B and $\tilde{p}_B$ are equal as well as d_{3e} and $\tilde{d}_{3e}$.

NOTE 2 Formulae (3) to (6) do not apply to collars.

6.2.2.2 Effective dimensions of flange ring

The effective thickness e_F or e_L used below is the average thickness of the flange ring. It can be obtained by dividing the cross-section area of the ring A_F or A_L (including bolt holes) by the actual radial width of this section.

Since there is a large variety of shapes of flange cross-sections, formulae for the calculation of A_F or A_L are not given for specific flange types.

Integral flange and blank flange (see Figure 6 to Figure 11)

$$b_F = (d_4 - d_0) / 2 - d_{5e} \qquad (7)$$

$$b_L = d_L = e_L = 0 \qquad (8)$$

$$d_F = (d_4 + d_0) / 2 \qquad (9)$$

$$e_F = 2 \times A_F / (d_4 - d_0) \qquad (10)$$

Loose flange with collar (see Figure 12)

For collar:

$$b_F = (d_8 - d_0) / 2 \qquad (11)$$

$$d_F = (d_8 + d_0) / 2 \qquad (12)$$

$$e_F = 2 \times A_F / (d_8 - d_0) \tag{13}$$

For flange:

$$b_L = (d_4 - d_6)/2 - d_{5e} \tag{14}$$

$$d_L = (d_4 + d_6)/2 \tag{15}$$

$$e_L = 2 \times A_L / (d_4 - d_6) \tag{16}$$

6.2.3 Connected shell

6.2.3.1 Flange with tapered hub

A cylindrical shell (constant wall thickness e_S, average diameter d_S) integral with a tapered hub is treated as being an equivalent cylindrical shell of effective wall thickness e_E and effective average diameter d_E:

$$e_E = e_1 \times \left\{ 1 + \frac{(\beta - 1) \times l_H}{(\beta/3) \times \sqrt{d_1 \times e_1} + l_H} \right\} \tag{17}$$

$$e_D = e_1 \times \left\{ 1 + \frac{(\beta - 1) \times l_H}{\sqrt[4]{(\beta/3)^4 \times (d_1 \times e_1)^2 + l_H^4}} \right\} \tag{18}$$

$$\beta = \frac{e_2}{e_1} \tag{19}$$

$$d_E = \{\min(d_1 - e_1 + e_E; d_2 + e_2 - e_E) + \max(d_1 + e_1 - e_E; d_2 - e_2 + e_E)\}/2 \tag{20}$$

6.2.3.2 Flange without hub

For a shell (cylindrical or conical or spherical, constant wall thickness e_s, angle j_S and diameter d_S at junction with flange) directly connected to a flange ring, the effective dimensions are:

$$e_E = e_S \tag{21}$$

$$d_E = d_S \tag{22}$$

Formulae (21) and (22) are not applicable when a nozzle is connected to the central opening of a blank flange. This case is covered by 6.2.3.3.

6.2.3.3 Blank flange

For a blank flange, the effective dimensions to be used are:

$$e_E = 0 \tag{23}$$

$$d_E = d_0 \tag{24}$$

Formulae (23) and (24) apply whatever the blank flange configuration (without opening, with opening without nozzle, with opening with nozzle).

6.2.3.4 Collar

The formulae which are applicable are those of 6.2.3.1 or 6.2.3.2 depending on whether or not the collar has a hub.

6.2.4 Flexibility-related flange parameters

6.2.4.1 Integral flange and collar

$$\gamma = e_E \times d_F / (b_F \times d_E \times \cos\varphi_S) \tag{25}$$

$$\vartheta = 0{,}55 \times \cos\varphi_S \times \frac{\sqrt{d_E \times e_E}}{e_F} \tag{26}$$

$$\lambda = 1 - e_P / e_F = e_Q / e_F \tag{27}$$

NOTE e_P and e_Q are defined in Figures 4 to 12 (when $e_P = e_F$ then $e_Q = 0$).

$$c_F = (1 + \gamma \times \vartheta) / \left\{1 + \gamma \times \vartheta \times \left[4 \times \left(1 - 3 \times \lambda + 3 \times \lambda^2\right) + 6 \times (1 - 2 \times \lambda) \times \vartheta + 6 \times \vartheta^2\right] + 3 \times \gamma^2 \times \vartheta^4\right\} \tag{28}$$

$$h_S = 1{,}1 \times e_F \times \sqrt{e_E / d_E} \times (1 - 2 \times \lambda + \vartheta) / (1 + \gamma \times \vartheta) \tag{29}$$

$$h_T = e_F \times \left(1 - 2 \times \lambda - \gamma \times \vartheta^2\right) / (1 + \gamma \times \vartheta) \tag{30}$$

$$h_R = h_S \times k_R - h_T \times 0{,}5 \tan\varphi_S \tag{31}$$

$$k_Q = \begin{cases} +0{,}85 / \cos\varphi_S & \text{for conical or cylindrical shell} \\ +0{,}35 / \cos\varphi_S & \text{for spherical shell} \end{cases} \tag{32}$$

$$k_R = \begin{cases} -0{,}15 / \cos\varphi_S & \text{for conical or cylindrical shell} \\ -0{,}65 / \cos\varphi_S & \text{for spherical shell} \end{cases} \tag{33}$$

$$Z_F = 3 \times d_F \times c_F / \left(\pi \times b_F \times e_F^3\right) \tag{34}$$

$$Z_L = 0 \tag{35}$$

6.2.4.2 Blank flange

Diameter ratio:

$$\rho = d_9 / d_E \tag{36}$$

NOTE Reminder: for a blank flange, $d_E = d_0$ (according to Formula (24))

$$h_R = (d_E / 4) \times \left(1 - \rho^2\right) \times \left(0{,}7 + 3{,}3\,\rho^2\right) / \left[\left(0{,}7 + 1{,}3\,\rho^2\right) \times \left(1 + \rho^2\right)\right] \tag{37}$$

$$Z_F = 3 d_F / \left\{\pi \times \left[b_F \times e_F^3 + d_F \times e_0^3 \times \left(1 - \rho^2\right) / \left(1{,}4 + 2{,}6 \times \rho^2\right)\right]\right\} \tag{38}$$

$$Z_L = 0 \tag{39}$$

6.2.4.3 Loose flange with collar

For the collar use Formulae (25) to (35); for the loose flange use the following formula:

$$Z_L = 3 \times d_L / (\pi \times b_L \times e_L^3) \tag{40}$$

6.3 Bolt and washer parameters

6.3.1 General

The bolt dimensions are shown in Figure 2. Diameters of standard metric series bolts are given in Annex B.

6.3.2 Effective cross-section area of bolts

$$A_B = \{\min(d_{Be}; d_{Bs})\}^2 \times n_B \times \pi/4 \tag{41}$$

6.3.3 Flexibility modulus of bolts

$$X_B = \left(l_S / d_{Bs}^2 + l_e / d_{Be}^2 + 0{,}8 / d_{B0}\right) \times 4 / (n_B \times \pi) \tag{42}$$

The thickness of washers possibly present in the joint shall be included in lengths l_s and l_e.

6.3.4 Geometric parameters for washers and contact surfaces

NOTE Formulae presented for washers are also applicable to expansion sleeves.

6.3.4.1 Absence of washers

If there are no washers,

$$X_W = \tilde{X}_W = 0 \tag{43}$$

And ignore Formulae (44) to (50).

6.3.4.2 Presence of washers

$$b_W = (d_{W2} - d_{W1})/2 \tag{44}$$

$$d_W = (d_{W2} + d_{W1})/2 \tag{45}$$

$$d_{K1} = \max(d_5; d_{W1}) \tag{46}$$

$$d_{K2} = \min(d_{B4}; d_{W2}) \tag{47}$$

$$b_{KB} = (d_{K2} - d_{W1})/2 \tag{48}$$

NOTE 1 These formulae also apply for washer of flange number 2.

NOTE 2 In the usual case $d_{K1} = d_5$ and $d_{K2} = d_{B4}$.

6.3.5 Flexibility modulus of washers

$$X_W = \frac{e_W}{n_B \times \pi \times d_W \times b_W} \cdot \frac{2 \times b_W / (b_W + b_{KB}) + e_W / (b_W - b_{KB})}{1 + e_W / (b_W - b_{KB})} \tag{49}$$

$$\tilde{X}_W = \frac{\tilde{e}_W}{n_B \times \pi \times \tilde{d}_W \times \tilde{b}_W} \cdot \frac{2 \times \tilde{b}_W / (\tilde{b}_W + \tilde{b}_{KB}) + \tilde{e}_W / (\tilde{b}_W - \tilde{b}_{KB})}{1 + \tilde{e}_W / (\tilde{b}_W - \tilde{b}_{KB})} \tag{50}$$

NOTE X_W includes an estimated correction factor for different axial stresses in different sections.

6.4 Gasket parameters

6.4.1 General

The notation for dimensions of gaskets is given in Figure 4.

6.4.2 Theoretical dimensions

$$b_{Gt} = (d_{G2} - d_{G1})/2 \tag{51}$$

$$d_{Gt} = (d_{G2} + d_{G1})/2 \tag{52}$$

$$A_{Gt} = \pi \times d_{Gt} \times b_{Gt} \tag{53}$$

NOTE The theoretical gasket width b_{Gt} is the maximum which may result from a very high F_G.

6.4.3 Effective dimensions

The effective gasket width b_{Ge} depends on the force F_G applied to the gasket for many types of gasket. The value b_{Ge} is determined iteratively for the assembly condition with $F_G = F_{G0}$ and assumed to be unchanged for subsequent conditions.

NOTE 1 For a flat gasket, the effective gasket width is equal to twice the distance separating the outside diameter of the sealing face from the point of application of the gasket reaction (i.e. the resultant of compressive stress over the gasket width).

The value F_{G0} used for this determination represents the minimum force which shall be reached in assembly condition, to meet the leak-tightness criteria given in 7.4.

This minimum force is not known when starting the calculation. It will be obtained through the iterative calculation process beginning at this point and ending with 7.6, Formula (122).

To start calculation, any arbitrary value may be chosen for F_{G0}. Nevertheless, the use of a realistic value is recommended. In the case where the method is used with a specified initial bolt load, this initial value is given by the Formula (1) from Clause 5. In other cases the value from Formula (54) below is recommended.

$$F_{G0} \leq A_B * f_{B0} /3 - F_{R0} \tag{54}$$

Where F_{R0} is as given by 7.2.

Interim gasket width b_{Gi} shall be determined from the formulae in Table 1, starting with the first approximation given in this table.

Effective gasket width:

$$b_{Ge} = \min\{b_{Gi}; b_{Gt}\} \tag{55}$$

Effective gasket diameter:

The effective gasket diameter d_{Ge} is the diameter where the gasket force acts. It is determined from Table 1.

NOTE 2 For flat gaskets, d_{Ge} varies with b_{Ge}. In that case, b_{Ge} is twice the distance between the outside contact diameter of the gasket and the effective gasket diameter.

Effective gasket area:

$$A_{Ge} = \pi \times d_{Ge} \times b_{Ge} \tag{56}$$

NOTE 3 The method is not taking into account the effect on the gasket thickness if the gasket stress rises above the assembly level in a subsequent condition. The modification on the gasket thickness in such a case is considered to have negligible impact. If this phenomenon has to be taken into account, the alternative possible method given in Annex H can be used.

Initial gasket stress at assembly and associated thickness determination:

$$Q_{G0} = F_{G0} / A_{Ge} \tag{57}$$

$$E_{G0} = E_G(Q_{G0}) \tag{58}$$

Lever arm:

$$h_{G0} = (d_{3e} - d_{Ge})/2 \text{ for integral or blank flange} \tag{59}$$

$$h_{G0} = (d_{70} - d_{Ge})/2 \text{ for loose flange with collar} \tag{60}$$

$$d_{70} = \min\{\max\{d_{7\min}; (d_{Ge} + \chi \times d_{3e})/(1+\chi)\}; d_{7\max}\} \tag{61}$$

$$\chi = (Z_L \times E_{F0})/(Z_F \times E_{L0}) \tag{62}$$

NOTE 4 Formulae (61) and (62) only apply to loose flanges on a collar.

Formulae (55) to (62) are re-evaluated iteratively until the value b_{Ge} is constant within the required precision.

A precision of 5 % is enough. To obtain results almost independent of the operator, a precision of 0,1 % is however recommended.

6.4.4 Axial flexibility modulus of gasket

$$X_G = (e_G(Q_{G0})/A_{Gt}) \times (b_{Gt} + e_G(Q_{G0})/2) / (b_{Ge} + e_G(Q_{G0})/2) \tag{63}$$

The value of the compressed gasket thickness at assembly phase $e_G(Q_{G0})$ for the associated gasket stress Q_{G0} shall be determined from gasket compression curve obtained following test performed according to EN 13555.

Table 1 — Effective gasket geometry

Type	Gasket form	Formulae	
1	Flat gaskets, of low hardness, composite or pure metallic, materials Figure 3 a	First approximation: $b_{Gi} = b_{Gt}$ More accurate: $b_{Gi} = \sqrt{\dfrac{e_G (Q_{G0})/(\pi \times d_{Ge} \times E_{Gm})}{h_{G0} \times Z_F / E_{F0} + \tilde{h}_{G0} \times \tilde{Z}_F / \tilde{E}_{F0}} + \left[\dfrac{F_{G0}}{\pi \times d_{Ge} \times Q_{smax}}\right]^2}$ $E_{Gm} = E_{G0}$ for flat metallic ring gaskets with rectangular cross section. $E_{Gm} = 0{,}5 \times E_{G0}$ for non metallic flat gaskets In all cases: $d_{Ge} = d_{G2} - b_{Ge}$ Q_{smax} shall be taken at assembly phase temperature here NOTE An alternative (more precise and more complex), calculation method for b_{Gi} is given in Annex H.	(64) (65) (66) (67) (68)
2	Metal gaskets with curved surfaces, simple contact, Figures 3 b, 3 c	First approximation: $b_{Gi} = \sqrt{6 \times r_2 \times \cos \varphi_G \times b_{Gt} \times Q_{smax} / E_{G0}}$ More accurate: $b_{Gi} = \sqrt{\dfrac{6\, r_2 \times \cos \varphi_G \times F_{G0}}{\pi \times d_{Ge} \times E_{G0}} + \left[\dfrac{F_{G0}}{\pi \times d_{Ge} \times Q_{smax}}\right]^2}$ In all cases: $d_{Ge} = d_{G0}$	(69) (70) (71)
3	Metal octagonal section gaskets see Figure 3 d	In all cases: b_{Gi} = length b_{Ge} according to Figure 3d (Projection of contacting surfaces in axial direction.) $d_{Ge} = d_{Gt}$	(72) (73)
4	Metal oval or circular section gaskets, double contact see Figures 3 e, 3 f	First approximation: $b_{Gi} = \sqrt{12\, r_2 \times \cos \varphi_G \times b_{Gt} \times Q_{smax} / E_{G0}}$ More accurate: $b_{Gi} = \sqrt{\dfrac{12\, r_2 \times \cos \varphi_G \times F_{G0}}{\pi \times d_{Ge} \times E_{G0}} + \left[\dfrac{F_{G0}}{\pi \times d_{Ge} \times Q_{smax}}\right]^2}$ In all cases: $d_{Ge} = d_{Gt}$	(74) (75) (76)

6.4.5 Lever arms

6.4.5.1 All flanges

$$h_P = \left[\left(d_{Ge} - d_E \right)^2 \times \left(2d_{Ge} + d_E \right) / 6 + 2e_P^2 \times d_F \right] / d_{Ge}^2 \tag{77}$$

For blank flanges:

$$e_P = 0 \tag{78}$$

6.4.5.2 Integral flange and collar

$$h_Q = \left\{ h_S \times k_Q + h_T \times \left(2d_F \times e_P / d_E^2 - 0{,}5 \tan \varphi_S \right) \right\} \times \left(d_E / d_{Ge} \right)^2 \tag{79}$$

6.4.5.3 Blank flange

$$h_Q = \left(d_E / 8 \right) \times \left(1 - \rho^2 \right) \times \left(0{,}7 + 3{,}3\ \rho^2 \right) / \left(0{,}7 + 1{,}3\ \rho^2 \right) \times \left(d_E / d_{Ge} \right)^2 \tag{80}$$

6.4.5.4 Integral flange and blank flange

$$h_G = \left(d_{3e} - d_{Ge} \right) / 2 \tag{81}$$

$$h_H = \left(d_{3e} - d_E \right) / 2 \tag{82}$$

$$h_L = 0 \tag{83}$$

NOTE These formulae do not apply to collars.

6.4.5.5 Loose flange with collar

$$d_{7\min} \le d_7 \le d_{7\max} \tag{84}$$

$$d_{7\min} = d_6 + 2 \times b_0 \tag{85}$$

$$d_{7\max} = d_8 \tag{86}$$

$$h_G = \left(d_7 - d_{Ge} \right) / 2 \tag{87}$$

$$h_H = \left(d_7 - d_E \right) / 2 \tag{88}$$

$$h_L = \left(d_{3e} - d_7 \right) / 2 \tag{89}$$

As the value of d_7 is not known in advance, the following hypotheses can be made:

— For the flexibility calculations (i.e. up to the end of Clause 7), take for d_7 the value d_{70} given by Formula (61).

NOTE It follows that h_G, h_H and h_L can vary with each iteration necessary to calculate b_{Ge} and d_{Ge} (see 6.4.3).

— For the calculation of load ratios (Clause 8), the most favourable value between $d_{7\ \min}$ and $d_{7\ \max}$ can be used, as given in 8.6.

7 Forces

7.1 General

Different load conditions are indicated by the value of indicator "I". Case I = 0 is the assembly condition; higher values (I = 1,2...) are different test conditions, operating conditions and so on. The number of load conditions depends on the application. All potentially critical load conditions shall be calculated.

7.2 Applied loads

7.2.1 Assembly condition (I = 0)

Fluid pressure (internal or external) is zero: $P_0 = 0$

External bending moments and axial force combine to give a net force F_{R0} as in Formula (96) (load case I = 0), whereas lateral forces and torsion moment are equal to zero at assembly.

All temperatures are equal to the initial uniform value T_0.

7.2.2 Subsequent conditions (I = 1, 2 …)

7.2.2.1 Fluid pressure

$$A_Q = \frac{\pi \times d_{Ge}^2}{4} \tag{90}$$

NOTE d_{Ge} is the location of the forces acting on the gasket and not the location where the leak tightness is achieved. This is conservative, overestimating the load coming from the pressure of the fluid for large gasket width.

$$\left.\begin{array}{ll} \text{Internal fluid pressure} & P_I > 0 \\ \text{Unpressurised condition} & P_I = 0 \\ \text{External fluid pressure} & P_I < 0 \end{array}\right\} F_{QI} = A_Q \times P_I \tag{91}$$

7.2.2.2 Additional external loads

The connection can be submitted to 6 components of external loads: F_{XI} F_{YI}, F_{ZI}, M_{XI}, M_{YI}, M_{ZI}. The revolution axis of the assembly is named as the Z-axis, thus we have:

Axial force: $$F_{AI} = F_{ZI} \tag{92}$$

Resultant lateral force: $$F_{LI} = \sqrt{F_{XI}^2 + F_{YI}^2} \tag{93}$$

Resulting bending moment: $$M_{AI} = \sqrt{M_{XI}^2 + M_{YI}^2} \tag{94}$$

Torsion moment (due to friction): $$M_{TGI} = M_{ZI} \tag{95}$$

Additional external loads combine to give a net force F_{RI} as follows:

$$\left.\begin{array}{ll} \text{Axial tensile force} & F_{AI} > 0 \\ \text{Axial compression force} & F_{AI} < 0 \end{array}\right\} F_{RI} = F_{AI} \pm \left(4 / d_{3e}\right) \times M_{AI} \tag{96}$$

Select the sign in Formula (96) giving the more severe condition.

NOTE In the presence of external bending moment M_A, the most severe condition may be difficult to foresee because:

— on the side of the joint where the moment induces an additional tensile load (sign + in Formula (96)), load limits of flanges or bolts may govern, as well as minimum gasket compression;

— on the side of the joint where the moment induces an additional compression load (sign - in Formula (96)), load limit of gasket may be decisive.

Therefore, for good practice, it is suggested to consider systematically two load conditions (one for each sign in Formula (96)) whenever an external moment is applied, with different indices I being assigned to each case.

7.2.2.3 Thermal loads

Axial thermal expansion relative to the assembly condition (uniform temperature T_0) is treated by the formula below.

$$\begin{aligned}\Delta U^T{}_I &= l_B \times \alpha_{BI} \times (T_{BI} - T_0) - e_{Ft} \times \alpha_{FI} \times (T_{FI} - T_0) - e_L \times \alpha_{LI} \times (T_{LI} - T_0) - e_W \times \alpha_{WI} \times (T_{WI} - T_0) \\ &- e_{G(Q0)} \times \alpha_{GI} \times (T_{GI} - T_0) - \tilde{e}_{Ft} \times \tilde{\alpha}_{FI} \times (\tilde{T}_{Ft} - T_0) - \tilde{e}_L \times \tilde{\alpha}_{LI} \times (\tilde{T}_{LI} - T_0) - \tilde{e}_W \times \tilde{\alpha}_{WI} \times (\tilde{T}_{WI} - T_0)\end{aligned} \quad (97)$$

with:

$$e_{Ft} + \tilde{e}_{Ft} + e_L + \tilde{e}_L + e_{Gt} + e_W + \tilde{e}_W = l_B \quad (98)$$

7.3 Compliance of the joint

Lever arms are calculated from 6.4.5. For loose flanges, the assumption of Formulae (61) and (62) shall be used.

The following formulae apply for all load conditions (I = 0, 1, 2, …) with $Q_{G0} = F_{G0} / A_{Ge}$ for the determination of E_{Gi}.

$$Y_{BI} = Z_L \times h_L^2 / E_{LI} + \tilde{Z}_L \times \tilde{h}_L^2 / \tilde{E}_{LI} + X_B / E_{BI} + X_W / E_{WI} + \tilde{X}_W / \tilde{E}_{WI} \quad (99)$$

$$Y_{GI} = Z_F \times h_G^2 / E_{FI} + \tilde{Z}_F \times \tilde{h}_G^2 / \tilde{E}_{FI} + Y_{BI} + X_G / E_{GI} \quad (100)$$

$$Y_{QI} = Z_F \times h_G \times (h_H - h_P + h_Q) / E_{FI} + \tilde{Z}_F \times \tilde{h}_G \times (\tilde{h}_H - \tilde{h}_P + \tilde{h}_Q) / \tilde{E}_{FI} + Y_{BI} \quad (101)$$

$$Y_{RI} = Z_F \times h_G \times (h_H + h_R) / E_{FI} + \tilde{Z}_F \times \tilde{h}_G \times (\tilde{h}_H + \tilde{h}_R) / \tilde{E}_{FI} + Y_{BI} \quad (102)$$

NOTE In Formulae (99) to (102):

— only one term in which the parameters Z and E have the subscript F relates to each integral flange (or blank flange); for the same gasket side (side without ~, side with ~), any term in which Z and E have the subscript L is not applicable;

— two terms always relate to each loose flange;

— the first relates to the flange itself (term in which Z and E have the subscript L);

— the second relates to its collar (term in which Z and E have the subscript F).

If there is neither loose flange nor washers only one term exists (for the bolts) in Formula (99).

7.4 Minimum forces necessary for the gasket

7.4.1 Assembly condition (I = 0)

Minimum gasket force:

$$F_{\text{G0min}} = A_{\text{Ge}} \times Q_{\text{A}} \tag{103}$$

If no specific leak rate is requested, then use $\boldsymbol{Q}_{0,\text{min}}$ (from Annex G) instead of $\boldsymbol{Q}_{\text{A}}$.

7.4.2 Subsequent conditions (I = 1, 2, ….)

Force required ensuring:

- leak-tightness;
- no loss of contact at bolts or nuts due to external compression axial load on the joint or to negative fluid pressure;
- sufficient axial load on the gasket in order to counter its potential sliding due to external torsion moments and radial forces by friction at flange/gasket interface.

$$F_{\text{GImin}} = \max\left\{A_{\text{Ge}} \times Q_{\text{smin(L),I}}; -(F_{\text{QI}} + F_{RI}); \frac{F_{\text{LI}}}{\mu_{\text{G}}} + \frac{2 \times M_{\text{TGI}}}{\mu_{\text{G}} \times d_{\text{Gt}}} - \frac{2 \times M_{\text{AI}}}{d_{\text{Gt}}}\right\} \tag{104}$$

For gaskets according to Figure 4c) to Figure 4f), the third term of Formula (104) should be neglected.

If no specific leak rate is requested, then use m × | P_{I}| (with m from Annex G) instead of $Q_{\text{smin(L)I}}$.

When no specific data are available for the value of μ_{G}, the generic values from Table E.1 can be taken as an approximation.

NOTE 1 It is essential that the selection of $Q_{\text{S min(L)I}}$ depends on the initial gasket surface pressure Q_{A} which is applied in the assembly condition. Q_{A} and $Q_{\text{Smin(L)I}}$ are a pair of variables which are determined in a leakage test according to EN 13555 and which belong together. The lowest acceptable value of Q_{A} is equal to $Q_{\text{min(L)I}}$, in this case $Q_{\text{Smin(L)I}}$ = Q_{A}. The higher Q_{A} can be chosen, the lower $Q_{\text{Smin(L)I}}$ can get.

NOTE 2 A calculation can be performed with no specified leakage rate using the values of $Q_{0,\text{min}}$ and m in the table of Annex G. The expected leakage rate can be assessed from the average gasket surface pressure ($F_{\text{GI}}/A_{\text{Ge}}$) obtained in the first calculation for the considered situation and using the EN 13555 leakage diagram for the relevant gasket (type) and test conditions.

7.5 Internal forces in assembly condition (I = 0)

7.5.1 Required forces

To guarantee that the gasket force in subsequent conditions never falls below the value F_{GImin} given by Formula (104) the gasket force in the assembly condition shall be at least the following:

$$F_{\text{G}\Delta} = \max_{\text{all I}\neq 0}\left\{F_{\text{GI min}} \times Y_{\text{GI}} + F_{\text{QI}} \times Y_{\text{QI}} + \left(F_{\text{RI}} \times Y_{\text{RI}} - F_{\text{R0}} \times Y_{\text{R0}}\right) + \Delta U_{\text{I}} + \Delta e_{\text{Gc,I}}\right\} / Y_{\text{G0}} \tag{105}$$

Formula (105) does not take into account plastic deformation that can happen in the subsequent situations of assembly phase. Where this plastic deformation is considered significant, it is recommended to replace Formula (105) by Formula (106) below and that method detailed in Annex H or equivalent is used.

$$F_{G\Delta} = \max_{\text{all } I\neq 0}\left\{F_{GI\,min} \times Y_{GI} + F_{QI} \times Y_{QI} + \left(F_{RI} \times Y_{RI} - F_{R0} \times Y_{R0}\right) + \Delta U_I + \Delta e_{Gc,I} + \left[e_G(Q_{G0}) - e_{G(A)}\right]\right\}/Y_{G0} \quad (106)$$

The above alternative Formula (106) enables to take plastic deformation into account, by introducing the difference between the compressed gasket thickness after assembly ($e_G(Q_{G0})$) and the gasket compressed thickness after all situations have occurred ($e_{G(A)}$).

- When no plastic deformation happens in the subsequent situation after assembly we have $e_G(Q_{G0}) = e_{G(A)}$ and Formula (106) is equivalent to Formula (105).
- When plastic deformation happens in the subsequent situation after assembly we have $e_G(Q_{G0}) > e_{G(A)}$.

Taking into account what is also necessary for seating of the gasket the required gasket force and the corresponding bolt load are as follows:

$$F_{G0\,req} = \max\left\{F_{G0\,min}; F_{G\Delta}\right\} \quad (107)$$

$$F_{B0\,req} = F_{G0\,req} + F_{R0} \quad (108)$$

If the value F_{G0req} given by Formula (107) is higher than the value F_{G0} assumed up to this step, the calculation shall be repeated from Formula (55), using a higher value for F_{G0} until:

$$F_{G0\,req} \leq F_{G0} \quad (109)$$

On the contrary, if the value F_{G0req} given by Formula (107) is lower than the value F_{G0} assumed up to this step, this value is acceptable, because it gives a higher approximation of the true F_{G0req}.

The true value $F_{G0\,req}$ may be found through a number of iterations great enough so that:

$$F_{G0} \approx F_{G0req} \quad (110)$$

Within the required precision.

A precision of 5 % is enough, with F_{G0} greater than F_{G0req}. To obtain a result almost independent of the operator, a precision of 0,1 % is however recommended.

7.5.2 Accounting for bolt-load scatter at assembly

The actual force, F_{B0} is limited as follows:

$$F_{B0min} \leq F_{B0} \leq F_{B0max} \quad (111)$$

where

$$F_{B0min} = F_{B0\,av} \times \left(1 - \varepsilon_-\right) \quad (112)$$

$$F_{B0max} = F_{B0\,av} \times \left(1 + \varepsilon_+\right) \quad (113)$$

Annex B, gives theoretical indicative values of scatter of initial bolt load of a single bolt (ε_{1-}, ε_{1+}) in Table B.1, as possible Formulae enabling to assess the scatter for the global load of all the bolts bolt (ε_-, ε_+).

After assembly, the actual bolt force achieved shall be not less than the required minimum bolt force F_{B0req}:

$$F_{\text{B0min}} \geq F_{\text{B0 req}} \tag{114}$$

Consequently, the scatter of the bolt-tightening shall be taken account of in the following way:

a) nominal bolt assembly force, used to define the bolting-up parameters:

1) for bolt-tightening methods involving control of bolt-load:

$$F_{\text{B0nom}} \geq F_{\text{B0 req}} \Big/ \left(1 - \varepsilon_-\right) \tag{115}$$

2) for bolt-tightening methods involving no control of bolt-load:

The value to be selected for F_{B0nom} is the average bolt load F_{B0av} that can really be expected in practise for the method used, independently of F_{B0req}.

The following condition shall be met:

$$F_{\text{B0nom}} = F_{\text{B0av}} \geq F_{\text{B0req}} \Big/ \left(1 - \varepsilon_-\right) \text{ where } \varepsilon_{1-} = 0{,}5 \tag{116}$$

If not, the bolt-tightening method initially chosen is not valid and shall be changed.

NOTE For the common case of manual bolt-tightening, Annex B gives an estimate of $F_{\text{BO av}}$.

b) maximum forces to be used for load limit calculation (Clause 8) in assembly condition:

They shall be based on the nominal bolt assembly force selected according to a) above:

$$F_{\text{B0max}} = F_{\text{B0nom}} \times \left(1 + \varepsilon_+\right) \tag{117}$$

$$F_{\text{G0max}} = F_{\text{B0max}} - F_{\text{R0}} \tag{118}$$

The effective gasket width b_{Ge} shall not be recalculated for $F_{\text{G0 max}}$.

7.6 Internal forces in subsequent conditions (I = 1, 2, …)

To prevent leakage, the gasket force in all subsequent conditions shall be at least the minimum required F_{GImin} from Formula (104).

This corresponds to a gasket assembly force equal to $F_{\text{G}\Delta}$ from Formula (105) or (106).

If the admissibility of the forces in the connection has been proved for this value of the gasket forces in the assembly conditions, and in practise a bolt load F_{B0} (= F_{G0}) > $F_{\text{G}\Delta}$ + F_{R0} is applied, plastic deformations may occur in subsequent load conditions. In case of frequent re-assembly (which each of them may generate a bolt load $F_{\text{G}\Delta}$ + F_{R0}) it is important to avoid accumulation of the plastic deformations that may occur at start-up after each re-assembly. This is obtained by checking the load limits of the flange connection, in subsequent conditions, for an assembly gasket force F_{G0d} defined by the formula below.

$$F_{\text{G0d}} = \max\left\{F_{\text{G}\Delta}; (2/3) \times \left(1 - 10/N_{\text{R}}\right) \times F_{\text{B0 max}} - F_{\text{R0}}\right\} \tag{119}$$

As described in 5, when the calculation method is applied for a specified bolt load, Formula (2) shall be applied instead of Formula (119) to take into account the lower bound of applied bolt load that can be much greater than FG_{Δ}.

Subsequent gasket force and bolt load for load limit calculations then are

$$F_{\mathrm{GI}} = \left\{F_{\mathrm{G0d}} \times Y_{\mathrm{G0}} - \left[F_{\mathrm{QI}} \times Y_{\mathrm{QI}} + \left(F_{\mathrm{RI}} \times Y_{\mathrm{RI}} - F_{\mathrm{R0}} \times Y_{\mathrm{R0}}\right) + \Delta U_{\mathrm{I}}\right] - \Delta_{eGc,I}\right\} / Y_{\mathrm{GI}} \tag{120}$$

Formula (120) does not take into account plastic deformation that can happen in the subsequent situations of assembly phase. Where this plastic deformation is considered significant, it is recommended to replace Formula (120) by Formula (121) below and that method detailed in Annex H or equivalent is used.

$$F_{\mathrm{GI}} = \left\{F_{\mathrm{G0d}} \times Y_{\mathrm{G0}} - \left[F_{\mathrm{QI}} \times Y_{\mathrm{QI}} + \left(F_{\mathrm{RI}} \times Y_{\mathrm{RI}} - F_{\mathrm{R0}} \times Y_{\mathrm{R0}}\right) + \Delta U_{\mathrm{I}}\right] - \Delta_{\mathrm{eGc,I}} - \left[e_{\mathrm{G}}(Q_{\mathrm{G0}}) - e_{\mathrm{G(A)}}\right]\right\} / Y_{\mathrm{GI}} \tag{121}$$

The above alternative Formula (121) enables to take it into account, by introducing the difference between the compressed gasket thickness after assembly ($e_{\mathrm{G}}(Q_{\mathrm{G0}})$) and the gasket compressed thickness after all situations have occurred ($e_{\mathrm{G(A)}}$).

— When no plastic deformation happens in the subsequent situation after assembly we have $e_{\mathrm{G}}(Q_{\mathrm{G0}}) = e_{\mathrm{G(A)}}$ and Formula (121) is equivalent to Formula (120).

— When plastic deformation happens in the subsequent situation after assembly we have $e_{\mathrm{G}}(Q_{\mathrm{G0}}) > e_{\mathrm{G(A)}}$.

From values calculated from Formulae (120) or (121), the bolt force in subsequent conditions shall be calculated as follows:

$$F_{\mathrm{BI}} = F_{\mathrm{GI}} + \left(F_{\mathrm{QI}} + F_{\mathrm{RI}}\right) \tag{122}$$

Then in Clause 8 the admissibility is checked with the following approach:

— For assembly condition, F_{B0max} and F_{G0max} shall be used.

— For subsequent conditions, F_{BI} and F_{GI} shall be used.

8 Load limits

8.1 General

Loads on the joint system shall be within safe limits at all times. These limits are expressed in calculated load ratios.

Each load ratio Φ … shall be less than or equal to unity for all conditions (I = 0, 1, 2 ...).

The index I for the load condition is omitted in the following for simplification.

Nominal design stresses in assembly condition are the same as in test condition (see 4.3).

NOTE It is reminded that for bolting-up condition (I = 0), the forces to be considered are the maximum possible forces (see 7.5.2 b).

8.2 Bolts

Nominal design stress of bolts shall be determined by the same rules as used for nominal design stress of flanges and shells (see 4.3).

Bolt load ratio:

$$\Phi_{\mathrm{B}} = \frac{1}{f_{\mathrm{B}} \times c_{\mathrm{B}}} \sqrt{\left(\frac{F_{\mathrm{B}}}{A_{\mathrm{B}}}\right)^2 + 3 \times \left(c_{\mathrm{A}} \times \frac{M_{\mathrm{t,B}}}{I_{\mathrm{B}}}\right)^2} \leq 1{,}0 \tag{123}$$

Where c_A and c_B defined as follows:

c_A = 1 in assembly condition, for bolt material with minimum rupture elongation $A \geq 10$ % (124)

c_A = 4/3 in assembly condition, for bolt material with minimum rupture elongation $A < 10$ % (125)

c_A = 0 in all other loading conditions and in assembly conditions when no torque is applied on the bolt (tensioner) (126)

NOTE 1 The value c_A = 1 is based on a plastic limit criterion. Due to this criterion, some limited plastic strains may occur at periphery of the bolts in assembly condition.

Use of this criterion has been validated by industrial experience, for bolt material with sufficient ductility ($A \geq 10$ %).

The value c_A = 4/3 is based on an elastic limit criterion. Even with sufficiently ductile bolt material, it may be selected if a strict elastic behaviour of the bolts is wished in assembly condition.

Concerning c_B, it is recommended to apply nuts with specified proof load values not less than the minimum proof load values of the screws on which they are mounted (e_N x $f_N \geq 0{,}8$ x d_{B0} x f_B). If bolts are screwed in the flange, the engagement length of screws in threaded holes shall be sufficiently large ($l_{5t} \geq 0{,}8$ x d_{B0} x f_B/f_F). These two aspects are taken into account by introducing a correction factor $c_B \leq 1$, determined as follows:

$$c_B = \min\{1{,}0;\ e_N \times f_N/(0{,}8 \times d_{B0} \times f_B);\ l_{5t} \times f_F/(0{,}8 \times d_{B0} \times f_B)\} \tag{127}$$

NOTE 2 If $c_B < 1{,}0$ the design can be improved.

For good practice, a minimum load ratio should be observed and determined according to bolt material class (for example, Φ_{B0min} = 0,3 for commonly used bolt material).

8.3 Gasket

Gasket load ratio:

$$\Phi_G = \frac{F_G}{A_{Gt} \times Q_{smax}} \leq 1{,}0 \tag{128}$$

8.4 Integral flange and collar

Load ratio for flange, or collar

$\Phi_F = \lvert F_G \times h_G + F_Q \times (h_H - h_P) + F_R \times h_H \rvert / W_F \leq 1{,}0$	(129)
$W_F = (\pi/4) \times \{f_F \times 2 \times b_F \times e_F^2 \times (1 + 2 \times \Psi_{opt} \times \Psi_Z - \Psi_Z^2) + f_E \times d_E \times e_D^2 \times c_M \times j_M \times k_M\}$	(130)
$f_E = \min(f_F; f_S)$	(131)
$\delta_Q = P \times d_E / (f_E \times 2 \times e_D \times \cos\varphi_S)$	(132)
$\delta_R = F_R / (f_E \times \pi \times d_E \times e_D \times \cos\varphi_S)$	(133)

$$c_M = \begin{cases} \sqrt{1{,}33 \times \left[1 - 0{,}75 \times \left(0{,}5 \times \delta_Q + \delta_R\right)^2\right] \times \left[1 - \left(0{,}75 \times \delta_Q^2 + 1 \times \delta_R^2\right)\right]} \ \textit{for conical and cylindrical shell} \\ \sqrt{1{,}33 \times \left[1 - 0{,}75 \times \left(0{,}5 \times \delta_Q + \delta_R\right)^2\right] \times \left[1 - \left(0{,}25 \times \delta_Q^2 + 3 \times \delta_R^2\right)\right]} \ \textit{for spherical shell} \end{cases} \tag{134}$$

$$c_S = \begin{cases} \frac{\pi}{4} \times \sqrt{1 - 0{,}75 \times \left(0{,}5 \times \delta_Q + \delta_R\right)^2} + j_s \times \left(0{,}5 \times \delta_R - 0{,}75 \times \delta_Q\right) \textit{for conical and cylindrical shell} \\ \frac{\pi}{4} \times \sqrt{1 - 0{,}75 \times \left(0{,}5 \times \delta_Q + \delta_R\right)^2} + j_s \times \left(1{,}5 \times \delta_R - 0{,}25 \times \delta_Q\right) \textit{for spherical shell} \end{cases} \tag{135}$$

$$j_M = \frac{F_G \times h_G + F_Q \times \left(h_H - h_p\right) + F_R \times h_H}{\left|F_G \times h_G + F_Q \times \left(h_H - h_p\right) + F_R \times h_H\right|} \tag{136}$$

(j_M is equal to +1 or -1 depending on the sign of the expression $F_G \times h_G + F_Q \times \left(h_H - h_p\right) + F_R \times h_H$)

$$j_s = \pm 1 \tag{137}$$

$$-1 \le k_M \le +1 \tag{138}$$

$$0 \le k_S \le 1 \tag{139}$$

$$\Psi_{(j_S, k_M, k_S)} = \frac{f_E \times d_E \times e_D \times \cos\varphi_S}{f_F \times 2 \times b_F \times e_F} \times \left\{ \left(0{,}5 \times \delta_Q + \delta_R\right) \times \tan\varphi_S - \delta_Q \times 2 \times e_P / d_E + j_S \times k_S \times \sqrt{\frac{e_D \times c_M \times c_S \times (1 + j_S \times k_M)}{d_E \times \cos^3 \varphi_S}} \right\} \tag{140}$$

The values of j_S, k_M, *and* k_S to be used are defined in the calculation sequence described following Table 2.

$$\Psi_{opt} = j_M \times \left(2 \times e_P / e_F - 1\right) \text{ with } (-1 \le \Psi_{opt} \le +1) \tag{141}$$

$$\Psi_0 = \Psi_{(0.0.0)} \tag{142}$$

$$\Psi_{max} = \Psi_{(+1,+1,+1)} \tag{143}$$

$$\Psi_{min} = \Psi_{(-1,-1,+1)} \tag{144}$$

The value Ψ_Z in Formula (130) depends on j_M and Ψ_{opt} as given in Table 2.

Table 2 — Determination of Ψ_Z

j_M	Range of Ψ_{opt}	k_M	Ψ_Z (j_S, k_M, k_S)
$j_M = +1$	$\Psi_{max} \leq \Psi_{opt}$	$k_M = +1$	$\Psi_Z = \Psi_{max}$
	$\Psi_0 \leq \Psi_{opt} < \Psi_{max}$	$k_M = +1$	$\Psi_Z = \Psi_{opt}$
	$\Psi_{opt} < \Psi_0$	$k_M < +1$	$\Psi_Z = \Psi_{(-1, k_M, +1)}$
$j_M = -1$	$\Psi_{opt} \leq \Psi_{min}$	$k_M = -1$	$\Psi_Z = \Psi_{min}$
	$\Psi_{min} < \Psi_{opt} \leq \Psi_0$	$k_M = -1$	$\Psi_Z = \Psi_{opt}$
	$\Psi_0 < \Psi_{opt}$	$k_M > -1$	$\Psi_Z = \Psi_{(+1, k_M, +1)}$

The sequence of calculation shall be as follows:

a) Calculate e_D from Formula (18), β having previously been calculated by Formula (19).

b) Calculate fE, δQ, δR, cM from Formulae (131) to (134).

(If the value in the root giving c_M is negative the hub is overloaded).

c) Calculate $c_s(j_S = +1)$, $c_s(j_S = -1)$, j_M, Ψ_{opt}, Ψ_0, Ψ_{max}, Ψ_{min}, from Formulae (135) to (144).

(If $\Psi_{max} < -1$ or $\Psi_{min} > +1$ the ring is overloaded).

d) Determine k_M et Ψ_Z according to Table 2. When that table gives $k_M < +1$ or $k_M > -1$ or k_M without any more precision, the value of k_M shall be determined so that W_F is maximum in Formula (130) as calculated at step e) which follows. The value of Ψ_Z associated with k_M is given in Formula (140);

e) Calculate W_F et Φ_F from Formulae (130), (129).

8.5 Blank flange

Load ratio for blank flange:

$$\Phi_F = \max\left\{\left|F_B \times h_G + F_Q \times \left(1-\rho^3\right) \times d_{Ge}/6 + F_R \times (1-\rho) \times d_{Ge}/2\right|; \left|F_B \times h_G + F_Q \times \left(1-\rho^3\right) \times d_{Ge}/6\right|; \left|F_R \times (1-\rho) \times d_{Ge}/2\right|\right\} / W_F \leq 1{,}0 \quad (145)$$

$$W_F = (\pi/4) \times f_F \times \left\{2 \times b_F \times e_F^2 + d_0 \times (1-\rho) \times e_0^2\right\} \quad (146)$$

If there is a possible critical section where $e_X < e_F$ (see Figure 11), then calculate additionally the following load ratio:

$$\Phi_X = F_B \times (d_3 - d_X)/(2 \times W_X) \leq 1{,}0 \quad (147)$$

$$W_{\text{X}} = (\pi/4) \times f_{\text{F}} \times \left\{ (d_4 - 2\, d_{5\text{e}} - d_{\text{X}}) \times e_{\text{F}}^2 + d_{\text{X}} \times e_{\text{X}}^2 \right\} \tag{148}$$

8.6 Loose flange with collar

Load ratio for loose flange:

$$\Phi_{\text{L}} = F_{\text{B}} \times h_{\text{L}} / W_{\text{L}} \le 1{,}0 \tag{149}$$

$$W_{\text{L}} = (\pi/2) \times f_{\text{L}} \times b_{\text{L}} \times e_{\text{L}}^2 \tag{150}$$

Load ratio for collar can be evaluated arbitrarily from 8.4 or from Formula (151). The more favourable result (i.e. the smaller of both Φ_{F} values) is valid.

Formula (151) only applies to connections using a flat gasket with $(d_{\text{G2}} - d_7) > 0$.

$$\Phi_F = \frac{\left| F_{\text{Q}} + F_{\text{R}} \right| \times h_{\text{H}}}{(\pi/4) \times d_{\text{E}} \times \left[f_{\text{E}} \times \min\left\{ e_E^2 ; e_F^2 \right\} + \min\left\{ f_{\text{F}} \times e_F^2 ; Q_{\max} \times (d_{\text{G2}} - d_7)^2 / 4 \right\} \right]} \le 1{,}0 \tag{151}$$

The lever arms h_{G}, h_{H}, h_{L} may be determined by variation of the diameter d_7 in such a way that Formulae (149) to (151) and (129) to (144) all give the most favourable result, i.e. max (Φ_{L}; Φ_{F}) is minimum.

In the case of $F_{\text{Q}}+F_{\text{R}} > 0$ the most favourable result is generally obtained near $d_{7\ \text{min}}$ according to Formulae (84) to (86). In the assembly condition (with $F_{\text{Q}} = 0$ and $F_{\text{R}} = 0$), in contrast the optimum is near $d_{7\ \text{max}}$.

Diameter d_7 may be different depending on the load condition. For the assembly condition ($I = 0$) the load limit calculations may be performed with a value d_7 differing from the value d_{70} given in Formula (61).

Annex A
(informative)

Dimensions of standard metric bolts

Table A.1 — Metric bolt meters

Bolt size [a]	d_{B0}	d_{Be} [b]	d_{Bs}	
			c	d
M6 x 1	6	5,06	-	5,3
M8 x 1,25	8	6,83	-	7,1
M10 x 1,5	10	8,59	-	9,0
M12 x 1,75	12	10,36	8,5	10,8
M14 x 2 [e]	14	12,12	10,0	12,7 [f]
M16 x 2	16	14,12	12,0	14,7
M18 x 2,5 [e]	18	15,65	-	16,3 [f]
M20 x 2,5	20	17,65	15,0	18,3
M22 x 2,5 [e]	22	19,65	-	20,3 [f]
M24 x 3	24	21,19	18,0	22,0
M27 x 3	27	24,19	20,5	25,0 [f]
M30 x 3,5	30	26,72	23,0	27,7
M33 x 3,5 [e]	33	29,72	25,5	30,7 [f]
M36 x 4	36	32,25	27,5	33,4
M39 x 4	39	35,25	30,5	36,4 [f]
M42 x 4,5	42	37,78	32,5	39,0
M45 x 4,5	45	40,78	35,5	42,0 [f]
M48 x 5	48	43,31	37,5	44,7
M52 x 5	52	47,31	41,0	48,7 [f]
M56 x 5,5	56	50,84	44,0	52,4
M60 x 5,5 [e]	60	54,84	-	56,4
M64 x 6	64	58,37	51,0	60,1
M68x 6 [e]	68	62,37	-	64,1
M72 x 6	72	66,37	58,5	68,1
M76 x 6 [e]	76	70,37	-	72,1
M80 x 6	80	74,37	66,0	76,1
M90 x 6	90	84,37	75,0	86,1
M100 x 6	100	94,37	84,0	96,1

[a] For M6 to M64, the pitch p_t is that of the normal series (according to ISO 261); up to and including M64 the nominal dimensions conform to EN ISO 4014 and EN ISO 4016.

[b] The value of d_{Be} corresponds to the following definition:

$d_{Be} = (d_{B2} + d_{B3})/2$ (see Figure 2); $d_{Be} = d_{B0} - 0,9382 \times p_t$

[c] Diameter of neck for necked-down bolts (dimensions not standardized by EN or ISO)

[d] Body diameter for rolled thread (approximately equal to the basic pitch diameter d_{B2} according to ISO 724).

[e] Non-preferred sizes.

[f] Dimensions not standardized by EN or ISO.

Annex B
(informative)

Tightening

B.1 Scatter of initial bolt load of a single bolt — Indicative values ε_{1-} and ε_{1+} for a single bolt

Table B.1 — Scatter of initial bolt load of a single bolt — Indicative values ε_{1-} and ε_{1+} for a single bolt

Bolting up (tightening) method; Measuring method	Factors affecting scatter	Scatter value [a, b, c, d]	
		ε_{1-}	ε_{1+}
Wrench: operator feel or uncontrolled	Friction, Stiffness, Qualification of operator	0,3 + 0,5 x μ	0,3 + 0,5 x μ
Impact wrench	Friction, Stiffness, Calibration	0,2 + 0,5 x μ	0,2 + 0,5 x μ
Torque wrench = Wrench with measuring of torque (only)	Friction, Calibration, Lubrification	0,1 + 0,5 x μ	0,1 + 0,5 x μ
Hydraulic tensioner; Measuring of hydraulic pressure	Stiffness, Bolt length, Calibration	0,2	0,4
Wrench or hydraulic tensioner; Measuring of bolt elongation	Stiffness, Bolt length, Calibration	0,15	0,15
Wrench, Measuring of turn of nut (nearly to bolt yield)	Stiffness, Friction, Calibration	0,10	0,10
Wrench, Measuring of torque and turn of nut (nearly to bolt yield)	Calibration	0,07	0,07

[a] Very experienced operators can achieve scatter less than given values (e.g. ε = 0,2 instead of ε = 0,3 with torque wrench); for inexperienced operators scatter can be greater than shown.

[b] Tabulated scatter values are for a single bolt, the scatter of the total bolt load will be less, for statistical reasons, see B.2.

[c] With hydraulic tensioner, ε_{1+} et ε_{1-} are not equal, due to the fact that an additional load is supplied to the bolt while turning the unit to contact, prior to load transfer to the nut.

[d] μ is the friction coefficient which can be assumed between bolt and nut.

B.2 Scatter for the global load of all the bolts

All bolt-tightening methods involve some degree of inaccuracy. The resulting scatter values for a set of n_B bolts are $\boldsymbol{\varepsilon_+}$ and ε_-, respectively above and below the target value. Table B.1 gives indicative values ε_{1+} and ε_{1-} for single bolts.

When the accuracy of the tightening of one bolt is not influenced by the other bolts, the scatter values ε_+ and ε_- for the total bolt load are reasonably expressed in terms of n_B, ε_{1+} and ε_{1-} as described below.

$$\varepsilon_+ = \varepsilon_{1+}\left(1 + 3/\sqrt{n_B}\right)/4 \quad \text{(B.1)}$$

$$\varepsilon_{-} = \varepsilon_{1-}\left(1+3/\sqrt{n_B}\right)/4 \qquad (B.2)$$

B.3 Manual uncontrolled tightening

By manual use of standard ring wrenches (without additional lever arm, without hammer impacts and without measuring of force or torque), the achieved average initial bolt force is limited by the wrench length (about 20 x d_{B0}), the power of the operator (maximum value about 1000 N) and the friction at the bolts (μ_B > 0,1).

For d_{B0}<24 mm, an initial bolt stress greater than 600 MPa may be achieved and the bolt may be destroyed if the operator has no feeling.

For d_{B0}>36 mm, the achieved initial bolt stress is less than 200 MPa, what is not sufficient in most cases.

For manual uncontrolled tightening by sufficient experienced operators the following estimate for the average total bolt force may be made:

$$F_{B0\,av} = \min\left(A_B \times f_{B0}; n_B \times 200000\right) \qquad (B.3)$$

Where A_B is expressed in [mm^2], f_{BO} in [MPa] and F_{BOav} in [N].

However, such uncontrolled tightening is not recommended.

B.4 Assembly using torque wrench

The nominal torque applied to tighten bolt is:

$$M_{t,nom} = k_B \times F_{B0nom}/n_B \qquad (B.4)$$

Hence the nominal bolt assembly force is:

$$F_{B0nom} = n_B \times M_{t,nom}/k_B \qquad (B.5)$$

The general formula for k_B is:

$$k_B = p_t/(2\times\pi) + \mu_t \times d_t/(2\times\cos\alpha) + \mu_n \times d_n/2 \qquad (B.6)$$

where

d_n: mean contact diameter under nut or bolt head;

d_t: mean contact diameter on thread;

μ_n: friction coefficient under nut or bolt head;

μ_t: friction coefficient on thread;

p_t: thread pitch;

α: half thread-angle.

In Formula (B.6), the first term is due to inclination of the thread helix angle, the second is due to friction between threads, and the third is due to friction under the nut (or bolt head).

For threads of ISO triangular profile, the expression of k_B becomes:

$$k_B = 0{,}159 \times p_t + 0{,}577 \times \mu_t \times d_{B2} + 0{,}5 \times \mu_n \times d_n \qquad (B.7)$$

where

d_{B2} is the mean thread diameter (see Figure 2).

The values given below for μ_t, μ_n, are very rough estimated values, the highest being for austenitic steels. Precise friction coefficients shall be gathered from the lubricant manufacturer, knowing the precise used lubricant reference.

— 0,10 to 0,15 for smooth, lubricated surfaces

— 0,15 to 0,25 for average, "normal" conditions (B.8)

— 0,20 to 0,35 for rough, dry surfaces

NOTE Use of simple torque wrench without torque multiplier device is limited to about $M_{t,nom} \approx$ 1 000 Nm.

Nominal twisting moment on bolt shanks

This moment is approximately equal to the part of assembly torque due to the friction coefficient on threads. From Formulae (B.4) and (B.7), it writes:

$$M_{t,B\,nom} = (0{,}159 \times p_t + 0{,}577 \times \mu_t \times d_{B2}) \times F_{B0nom} / n_B \qquad (B.9)$$

B.5 Assembly using bolt tensioner

When assembling and tightening a joint using Hydraulic Bolt Tensioning the factors affecting the achieved scatter and achieved residual bolt load are different to torquing.

The most significant factor in bolt tensioning is Tool Load Loss, this should be determined as follows.

To achieve a Nominal Bolt force $F_{B0\ nom}$ in all cases of Hydraulic tensioning a Tool load loss factor should be added to determine the Applied Bolt Force required to produce the required Nominal Bolt force $F_{B0\ nom}$. This additional Applied Force should be considered when selecting Bolt material and tightening method to ensure that the applied load in a bolt will not take the bolt into plastic deformation.

Tool Load loss occurs due to the transfer loss when the pressure is released from the Hydraulic tensioner by the load transfers from the threads of the tensioner to the threads of the nut. This causes thread deflection in the nut and consequent load loss. (An additional but smaller load loss occurs as the nut settles onto the flange nut spot face as load transfers to the nut.)

Tool load loss is dependent upon the diameter of bolt, the thread pitch and the load and increases in significance as the L/D ratio (Effective length/ root diameter of the bolt) reduces.

When the L/D ratio is below 4 generally torque is more accurate.

Alternative methods of calculating tool load loss factor are described in standards such as Norsok L005.

Tensioning offers the advantage of multiple tools tightening bolts simultaneously with a direct axial pull, this gives even gasket compression. However on most standard flanges it is impossible to fit tensioning tools on every bolt on one side of the joint (one flange) due to clearance issues. Common practice has developed to apply 50 % tensioning to applications other than highly critical joints such as nuclear or high cost such as sub sea.

In typical applications, tensioner is fitted to every other bolt and 50 % of the bolts are then tightened. The tensioners are then moved to the other 50 % of bolts which are then tightened.

As the second set of bolts tighten the gasket compresses further resulting in load loss in the first set of bolts. Therefore for speed i.e. to complete tightening in two passes it is common practice to apply a Flange Load Loss Factor to the first set of bolts and tighten them to a higher level to allow for the loss as the gasket compresses.

This factor and procedure is often referred to as Flange Load loss factor and shall be planned and considered when tensioning bolts to ensure that neither the flange gasket nor bolt is taken beyond desired stress /Load limits during assembly /tightening or at any phase.

Where Flange load loss factor creates an issue then the 100 % figure can be used with repeat passes until no nut movement occurs. This should be done system by system to ensure no foul play.

Annex C
(informative)

Flange rotations

C.1 General

The flange rotations which can be expected in practice are dependent among other parameters, on the true initial bolt force applied at bolting-up. Also, some (small) plastic deformation may occur, both at bolting-up and in subsequent conditions. Therefore:

— only lower and upper limits to the rotations can be evaluated, assuming successively minimum and maximum possible values of initial bolt load;

— only the elastic part of the rotations can be calculated.

C.2 Use of flange rotation

If the gasket manufacturer specifies a maximum acceptable value of flange rotation for the gasket, then the calculated values shall be checked to ensure that they are less than the maximum acceptable value.

Measured values of $\Theta_F + \tilde{\Theta}_F$, respectively $\Theta_L + \tilde{\Theta}_L$, can be used to control the bolt load during assembly.

C.3 Calculation of flange rotations

The elastic rotation of each flange or collar may be calculated from the following Formula (C.1) and for loose flanges from Formula (C.2):

$$\Theta_F = \frac{Z_F}{E_F} \times \left\{ F_G \times h_G + F_Q \times \left(h_H - h_P + h_Q\right) + F_R \times \left(h_H + h_R\right) \right\} \tag{C.1}$$

$$\Theta_L = (Z_L / E_L) \times F_B \times h_L \tag{C.2}$$

The preceding formulae are applicable to all loading conditions (I = 0, 1, 2 …) provided use of appropriate values of E_F, E_L, F_Q, F_R, F_G et F_B for each condition:

— E_{FI}, E_{LI}: same values as elsewhere;

— F_{QI}, F_{RI}: values according to Formulae (91) and (96);

— F_{GI}, F_{BI}: use minimum possible values of gasket and bolt loads to calculate minimum rotations, respectively maximum possible values to calculate maximum rotations.

These values are given by the following formulae:

— For bolting up condition (I = 0):

$$F_{\text{B0min}} = F_{\text{B0nom}} \times (1 - \varepsilon_-) \tag{C.3}$$

$$F_{\text{B0max}} = F_{\text{B0nom}} \times (1 + \varepsilon_+) \tag{C.4}$$

$$F_{\text{G0min}} = F_{\text{B0min}} - F_{\text{R0}} \tag{C.5}$$

$$F_{\text{G0max}} = F_{\text{B0max}} - F_{\text{R0}} \tag{C.6}$$

— For subsequent conditions (I ≠ 0):

Minimum and maximum values of F_{GI}, F_{BI} are obtained from the following formulae:

$$F_{\text{GImin}} = \{F_{\text{G0min}} \times Y_{\text{G0}} - [F_{\text{QI}} \times Y_{\text{QI}} + (F_{\text{RI}} \times Y_{\text{RI}} - F_{\text{R0}} \times Y_{\text{R0}}) + \Delta U_{\text{I}}] - \Delta_{\text{eGc,I}}\}/Y_{\text{GI}} \tag{C.7}$$

$$F_{\text{GImax}} = \{F_{\text{G0max}} \times Y_{\text{G0}} - [F_{\text{QI}} \times Y_{\text{QI}} + (F_{\text{RI}} \times Y_{\text{RI}} - F_{\text{R0}} \times Y_{\text{R0}}) + \Delta U_{\text{I}}] - \Delta_{\text{eGc,I}}\}/Y_{\text{GI}} \tag{C.8}$$

$$F_{\text{BImin}} = F_{\text{GImin}} + (F_{\text{QI}} + F_{\text{RI}}) \tag{C.9}$$

$$F_{\text{BImax}} = F_{\text{GImax}} + (F_{\text{QI}} + F_{\text{RI}}) \tag{C.10}$$

Annex D
(informative)

Use of the calculation method

D.1 Calculation method principle

Calculation for sealing performance is based on elastic analysis of the load/deformation relations between all parts of the flange connection, corrected by a possible plastic behaviour of the gasket material. Calculation for mechanical resistance is based on (plastic) limit analysis of the flange-shell combination. Both internal and external loads are considered. Load conditions covered include initial assembly, hydrostatic test, and all significant subsequent operating conditions. The calculation steps are broadly as follows:

a) First, the required minimum initial bolt load (to be reached at bolting-up) is determined, so that in any subsequent specified load condition, the residual force on the gasket will never be less than the minimum mean value required for the gasket (value is gasket data from EN 1591-2 for instance). The determination of this load is iterative, because it depends on the effective gasket width, which itself depends on the initial bolt load.

b) Then, the internal forces that result from the selected value of initial bolt load are derived for all load conditions, and the admissibility of combined external and internal forces is checked as follows:

   1) bolting-up condition: the check is performed against the maximum possible bolt force that may result from the bolting-up procedure;

   2) test and operating conditions: checks are performed against the minimum necessary forces (except when using the special procedure involving a specified bolt load described in 1), to ensure that the connection will be able to develop these minimum forces without risk of yielding, except in highly localised areas. Higher actual initial bolting results in (limited) plastic deformation in subsequent conditions (test, operation). But the checks so defined assure that these deformations will not reduce the bolt force to a value less than the minimum required.

If necessary, the flange rotations may be estimated in all load conditions, using Annex C, and the values obtained, compared with the relevant gasket limits which could apply.

EN 1591-1 is based upon the principle that a selected leakage rate is to be achieved. In that case, the gasket sealing coefficients have to be taken from results from tests performed according to EN 13555 or directly from EN 1591-2:2008. The gasket leakage behaviour is measured according to EN 13555, using Helium. Available, incomplete, models for leakage rate conversion and their limitations are given in Annex I.

But, where there is no requirement on limitation of leakage, the calculation can be performed using gasket coefficients not related to leakage rate (see table of Annex G), the other coefficients are to be obtained in accordance with EN 13555 or directly from EN 1591-2:2008. In that case the attempted leakage rate can be estimated from the calculated gasket contact pressure and the leakage diagram obtain for that specific gasket type during EN 13555 tests.

The load calculated by the procedures outlined in this standard represent the minimum bolt load that should be applied to the gasket to achieve the required tightness class. Increasing bolt load within acceptable load ratios of the flanges / bolt / gasket, reduces leak rates and produces a conservative design. The designer may choose a bolt load between the load to achieve the tightness class and the load limited by the load ratios.

D.2 Mechanical model

The calculation method is based on the following mechanical model:

a) Geometry of both flanges and gasket is axisymmetric. Small deviations such as those due to a finite number of bolts are permitted. Application to split loose flanges or oval flanges is not permitted.

b) The flange ring cross-section (radial cut) remains undeformed. Only circumferential stresses and strains in the ring are treated; radial and axial stresses and strains are neglected. This presupposition requires compliance with condition 4.2 a).

c) The flange ring is connected to a cylindrical shell. A tapered hub is treated as being an equivalent cylindrical shell of calculated wall thickness, which is different for elastic and plastic behaviour, but always between the actual minimum and maximum thickness. Conical and spherical shells are treated as being equivalent cylindrical shells with the same wall thickness; differences from cylindrical shell are explicitly taken into account in the calculation formula. At the connection of the flange ring and shell, the continuity of radial displacement and rotation is accounted for in the calculation.

d) The gasket is modelled by elastic behaviour with a plastic correction. For gaskets in incompressible materials which permit large deformations (for example: flat gaskets with rubber as the major component), the results given by the calculation method can be excessively conservative (i.e. required bolting load too high, allowable pressure of the fluid too low, required flange thickness too large, etc.) because it does not take account of such properties.

e) The gasket contacts the flange faces over a (calculated) annular area. The effective gasket width (radial) b_{Ge} may be less than the true width of gasket. This effective width b_{Ge} is calculated for the assembly condition (I = 0) and is assumed to be unchanged for all subsequent load conditions (I = 1, 2 ...) even in the case where the gasket stress is higher in one of the subsequent conditions (involved E_G values modifications are not considered to be of importance for the calculation result). There is no chronological order specified for the subsequent conditions numbering. The calculation of b_{Ge} includes the elastic rotation of both flanges as well as the elastic and plastic deformations of the gasket (approximately) in assembly condition.

f) The modulus of elasticity of the gasket at unloading (E_G) may increase with the compressive stress Q_G on the gasket. This modulus of elasticity is the unloading elasto-plastic secant modulus measured between 100 % and 33 % for several gasket stress levels. The calculation method uses the highest stress (Q_G) in assembly condition.

g) Creep of the gasket under compression can be approximated by a creep factor P_{QR} (see Annex F).

h) Thermal and mechanical axial deformations of flanges, bolts and gasket are taken into account.

i) Loading of the flange joint is axisymmetric. Any non-axisymmetric bending moment is replaced by an equivalent axial force, which is axisymmetric according to Formula (96).

j) Load changes between load conditions cause internal changes of bolt and gasket forces. These are calculated with account taken of elastic deformations of all components. To ensure leaktightness, the required initial assembly force is calculated (see 7.5) to ensure that the required forces on the gasket are achieved under all conditions (see 7.4 and 7.6).

k) Load limit proofs are based on limit loads for each component. This approach prevents excessive deformations. The limits used for gaskets, which depend on Q_{smax} are only approximations.

The model does not take account of the following:

l) Bolt bending stiffness and bending strength. This is a simplification. However the tensile stiffness of the bolts includes (approximately) the deformation within the threaded part in contact with the nut or threaded hole (see Formula (42)).

m) Creep of flanges and bolts (assuming that the materials have been selected in order to avoid excessive creep).

n) Different radial deformations at the gasket (this simplification has no effect for identical flanges).

o) Fatigue proofs (usually not taken into account by codes like this).

D.3 Required checks

Checks for admissibility of loads imply safety factors which are those applied to material yield stress or strength in the determination of the nominal design stresses used in the calculation method.

The assembly bolt load shall be sufficiently large to ensure leak tightness for all subsequent load conditions. Additionally, it is recommended to specify the tightening procedure with the required parameters (e.g. torque, tension...).

— The load ratios for bolts, gasket and both flanges are to be checked for all load conditions (including assembly and load conditions).

— If no leakage rate is specified, the expected leakage rate can be determined from the calculated gasket load in all situations, using gasket leakage data from test according to EN 13555.

D.4 Calculation sequence

The calculation sequence is detailed below in accordance with the clause numbers defined in this document.

6 Calculation Parameters

6.2 First flange and second flange parameters

6.2.2 b_F, d_F, e_F (or b_L, d_L, e_L)[1)] and $\tilde{b}_F, \tilde{d}_F, \tilde{e}_F$ (or $\tilde{b}_L, \tilde{d}_L, \tilde{e}_L$)[1)]
d_{3e} ($\tilde{d}_{3e} = d_{3e}$)

6.2.3 e_E, e_D, d_E and $\tilde{e}_E, \tilde{e}_D, \tilde{d}_E$

6.2.4 h_R and $\tilde{h}_R$

Z_F, Z_L and $\tilde{Z}_F, \tilde{Z}_L$

6.3 Bolts and washers parameters

6.3.2 A_B

6.3.3 X_B

6.3.4 $b_W, d_W, d_{K1}, d_{K2}, b_{KB}$

6.3.5 X_W and $\tilde{X}_W$

6.4 Gasket parameters

NOTE An alternative calculation method taking into account plastic deformation that can happen in the subsequent situations of assembly phase is also proposed in Annex H. The application of this annex can replace the calculation sequence proposed from 6.4 to Clause 7. Then the calculation from Clause 8 can be applied.

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1) Only in the case of loose flanges.

6.4.2 b_{Gt}, d_{Gt}, A_{Gt}

6.4.3 F_{G0} from Formula (58) or from Formula (1) of 5. if an initial bolt force ($F_{B0,specified}$) is specified
b_{Ge}, d_{Ge}, A_{Ge} (first approximation)
E_{G0} , h_{G0} et $\tilde{h}_{G0}$

6.4.4 X_G

Table1 b_{Ge}, d_{Ge}, A_{Ge} (more precise)

6.4.5 h_P, h_Q, h_G, h_H (and h_L) 1) and $\tilde{h}_P$, $\tilde{h}_Q$, $\tilde{h}_G$ (= $\tilde{h}_{G0}$), $\tilde{h}_H$ (and $\tilde{h}_L$)1)

7 Forces

7.2 Applied loads

7.2.1 T_0, F_{R0} (I = 0)

7.2.2 A_Q, F_{QI}, F_{AI}, F_{LI}, M_{AI}, M_{TGI}, F_{RI}, ΔU^T_I (I > 0)

7.3 Compliance of the joint
Y_{GI}, Y_{QI}, Y_{RI}, Y_{BI},

7.4 Minimum gasket forces
F_{G0min}, F_{GImin}

7.5 Internal forces in assembly condition

7.5.1 $F_{G\Delta}$ (from Formula (109) or from Formula (110) associated to Annex H)
$F_{G0\ req}$, $F_{B0\ req}$ (if $F_{G0\ req} > F_{G0}$ then calculation shall be reinitiate from 6.4.3)

7.5.2 $F_{B0\ nom}$, $F_{B0\ max}$, $F_{G0\ max}$

7.6 Internal forces in subsequent conditions

F_{G0d} (from Formula (119) or from Formula (2) if an initial bolt load is specified as described in 5)

F_{GI} (from Formula (120) or from Formula (121) associated to Annex H), F_{BI}

8 Load limits

8.2 Bolts
Φ_B

8.3 Gasket
Φ_G

8.4/ 8.5/ 8.6 First flange and second flange[2]
Φ_F (or Φ_L), possibly Φ_X, and $\tilde{\Phi}_F$ (and $\tilde{\Phi}_L$), possibly $\tilde{\Phi}_X$

2) For simplicity, possible optimisation of d7 (if the flange is loose) is not shown (see 8.6).

Annex E
(informative)

Gasket/flange face friction coefficients examples

Table E.1 — Gasket/flange face friction coefficients examples

Generic gasket type	Value for μ_G
PTFE based gaskets	0,05
Graphite based gaskets	0,1
Fibre based gaskets / rubber gaskets	0,25
Flat metallic gaskets	0,15

These friction factors are probably very conservative. An experimental determination of the friction factor should be preferred.

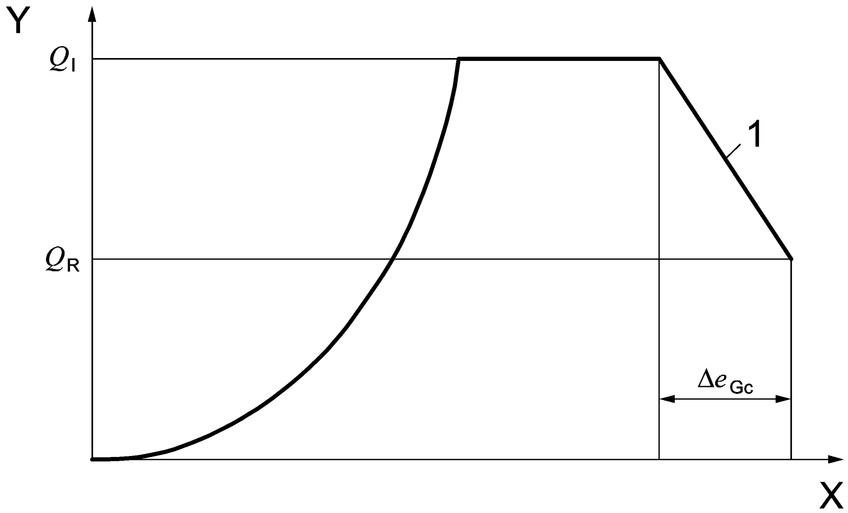
NOTE These lower limits tabulated values may lead to very conservative calculations because they do not take into account the specific roughness of the flange surfaces.

Annex F
(normative)

Determination of $\Delta_{eGc,I}$ based on a given P_{QR}

F.1 Determination of the deflection occurring during a P_{QR} test

NOTE In EN 13555:2004 and EN 1591-1:2001+A1:2009, the gasket creep was taken into account by the factor P_{QR}. This version of the standard is no longer using the factor P_{QR} to take the gaskets creep into account. This phenomenon is now directly handled by the treatment of gasket deflection including the viscous contribution.



Key

X gasket deflection

Y gasket surface pressure

1 creep-relaxation phase under a defined stiffness

Figure F.1 — P_{QR} test

The former P_{QR} is measured for a specific stiffness of the assembly (K), a specific initial gasket stress (Q_I) and a temperature (T). According to EN 13555:2004, the measured creep factor $P_{QR\,[K,\,QI,\,T]}$ for such a specific set of conditions is defined by the Formula below as the ratio of the residual stress to the initial stress:

$$P_{QR\,[K,\,QI,\,T]} = Q_R/Q_I \tag{F.1}$$

Thus for a specific set of conditions [K, QI, T], the viscous contribution of creep in the gasket deflection during the test, can be evaluated by the Formula below.

$$\Delta e_{Gc[K,QI,T],test} = \frac{A_{Gt[test]} \times Q_I \times \left(1 - P_{QR[K,Q_I,T]}\right)}{K} \tag{F.2}$$

where

— $A_{Gt[test]}$ is the gasket surface area during the test following EN 13555:2004 defined by:

$AGt = \frac{\pi}{4} \times \left(d_{Gext}{}^2 - d_{G\mathrm{int}}{}^2\right)$ where d_{Gext} and d_{Gint} are respectively the external and the internal diameters of the gasket used for the test (typically a PN40 DN40 gasket size) [mm^2];

— K is the stiffness of the test rig used for the test following EN 13555:2004) [N/mm];

— T is the temperature (during the test) used for the test following EN 13555:2004 (that shall be chosen as close as possible to the temperature of the calculated assembly) [°C];

— Q_I is the initial gasket contact pressure [MPa];

— $P_{\mathrm{QR}\,[\mathrm{K,\,QI,\,T}]}$ is the value for the creep factor derived from the test following EN 13555:2004 performed for the specific set of conditions (stiffness, initial contact pressure and temperature).

F.2 Determination of the deflection to be taken into account in the calculation

In the calculation, the stiffness of the assembly regarding the gasket is defined by $1/Y_G$. Thus, the value of K for the P_{QR} test has to be as close as possible as the value of $1/Y_G$ of the calculated assembly. Then the, viscous contribution of creep in the gasket deflection to be taken in the calculation is defined by the following formula:

$$\Delta e_{\mathrm{Gc}} = K \times Y_{\mathrm{G}} \times \Delta e_{\mathrm{Gc[K,QI,T],test}} \quad \text{(F.3)}$$

Annex G
(informative)

Sealing gasket parameter when no leakage rate is specified

Table G.1 — Sealing gasket parameter when no leakage rate is specified

Gasket type and material	Q_{0min} MPa	**m**
Non-metallic flat gaskets (soft) and flat gaskets with metal insertion		
Rubber	0,5	0,9
PTFE	10	1,3
Expanded PTFE (ePTFE)	12	1,3
Expanded graphite without metal insertion	10	1,3
Expanded graphite with perforated metal insertion	15	1,3
Expanded graphite with adhesive flat metal insertion	10	1,3
Expanded graphite with metallic sheet laminated in thin layers with standing high stresses	15	1,3
Non asbestos fibre with binder (thickness < 1mm)	40	1,6
Non asbestos fibre with binder (thickness >= 1mm)	35	1,6
Grooved steel gaskets with soft layers on both sides		
PTFE layers on soft steel	10	1,3
PTFE layers on stainless steel	10	1,3
Graphite layers on soft steel	15	1,3
Graphite layers on low alloy heat resistant steel	15	1,3
Graphite layers on stainless steel	15	1,3
Silver layers on heat resistant stainless steel	125	1,8
Spiral wound gaskets with soft filler		
Spiral wound gaskets with PTFE filler, outer support-ring only	20	1,6
Spiral wound gaskets with PTFE filler, inner and outer support-rings	20	1,6
Spiral wound gaskets with graphite filler, outer support-ring only	20	1,6
Spiral wound gaskets with graphite filler, inner and outer support-rings	50	1,6
Solid metal gaskets		
Aluminium (Al) (soft)	50	2,0
Copper (cu) or brass (soft))	100	2,0
Iron (Fe) (soft)	175	2,0
Steel (soft)	200	2,0
Steel, low alloy, heat resistant	225	2,0
Stainless steel	250	2,0
Stainless steel, heat resistant	300	2,0
Covered metal-jacketed gaskets		
Soft iron or steel jacket with graphite filler and covering	20	1,3
Low alloy steel (4 % to 6 % chrome) or stainless steel jacket with graphite filler and covering	20	1,3
Stainless steel jacket with expanded PTFE filler and covering	10	1,3
Nickel alloy jacket with expanded PTFE filler and covering	10	1,3
Metal-jacketed gaskets		
Aluminium (soft) jacket with graphite filler	50	1,6
Copper or brass (soft) jacket with graphite filler	60	1,8
Soft iron or steel jacket with graphite filler	80	2,0
Low alloy steel (4 % to 6 % chrome) or stainless steel jacket with graphite filler	100	2,2

Annex H
(informative)

Alternative calculation procedure taking into account the plastic deformation of the gasket in subsequent load conditions procedures (after assembly)

H.1 Introduction

In the main part of this calculation method possible plastic deformations of the gasket should be taken into account only for assembly $\left(e_{G(0)} \le e_{G(N)} = e_{Gt}\right)$; for subsequent load conditions they are neglected for simplicity.

However, if in a subsequent load condition the gasket stress rises above the assembly level (e.g. due to external fluid pressure) and/or if the resistance of the gasket material lowers (e.g. due to elevated temperature), then additional plastic deformations of the gasket occur.

From this follows $\left\lfloor e_{G(0)} - e_{G(A)} \right\rfloor > 0$ to be applied in Formulae (106) and (121).

All dimensions of a gasket may be changed by loading, but they may also remain unchanged. The gasket thickness may change from $e_{G(N)} = e_{Gt}\left(\underline{n}ew\right) to\ e_{G(0)} = e_{G(Q0)}\ to\ e_{G(I)} = e_{Ge(I)}$. The last change gives the deciding smallest gasket thickness $e_{G(A)}$ (after all had applied).

The effective width b_{Ge} may also change by changed loading; it may be connected with a change of d_{Ge} and A_{Ge}. Finally the greatest width $b_{Ge} = b_{Ge(A)}$ should be taken to get conservative results.

The theoretical width b_{Gt} may also change by loading (it may be increased if e_{Ge} is decreased); however this effect is not large and it is neglected for simplicity.

H.2 Calculation procedure

H.2.1 General description

The calculation of plastic deformations of the gasket should be based on the corresponding forces $F_{G(I)}$.

From a known actual gasket force after assemblage $F_{G(0)}$ the corresponding actual force in all subsequent load conditions (I = 1,2, ... I_{max}) may be calculated as follows:

$$F_{G(I)} = \left\{F_{G(0)} * Y_{G(0)} - .\Delta U_{G(I)} - \left\lfloor e_{G(0)} - e_{G(A)} \right\rfloor\right\} / Y_{G(I)} \tag{H.1}$$

Herein: $\Delta U_{G(I)} = \left[F_{QI} \times Y_{QI} + \left(F_{RI} \times Y_{RI} - F_{R0} \times Y_{R0}\right) + \Delta U_I\right]$

The iterative calculation starts with $e_{G(A)} = e_{G(0)}$.

A required precision for a minimum change of the gasket thickness should be assumed:

$$\Delta e_{G(A)} = e_{G(A),old} - e_{G(A),new} \qquad \text{e.g.: } \Delta e_{G(A),min} = 0{,}02\ \text{mm} \tag{H.2}$$

For all load conditions (I = 0, 1, 2, ... I_{max}) the effective dimensions of the gasket $\left(b_{Ge(I)}, d_{Ge(I)}, A_{Ge(I)} \text{ and } e_{Ge(I)}\right)$ should be determined according to one of the following subclauses H.3 to H.5.

H.2.2 No additional plastic deformation

If no load condition is indicated $e_{Ge} < e_{G(A)}$ then additional plastic deformation not occurs. The actual effective dimensions $b_{Ge}, d_{Ge}, A_{Ge} \text{ and } e_{G(A)}$ remain unchanged. The calculation continues within 7.4 (either with Formula (110) or with Formula (125)).

H.2.3 Additional plastic deformation

If any load condition is indicated $e_{Ge} < e_{G(A)} = e_{G(A),old}$ then some additional plastic deformation occurs. A new value $e_{G(A)} = e_{G(A),new}$ should be assumed. The following rules are proposed:

(a) $e_{G(A),new} \approx 0{,}5 * \{\min[e_{Ge(I)}] + e_{G(A),old}\}$ however:

(b) $e_{G(A),new} \geq 0{,}8 * e_{G(A),old}$ however:

(c) $e_{G(A),new} \leq e_{G(A),old} - \Delta e_{G(A),min}$

For changes $\Delta e_{G(A)} \geq e_{G(A),min}$ the effective dimensions $b_{Ge(I)}, d_{Ge(I)}, A_{Ge(I)} \text{ and } e_{Ge(I)}$ should be calculated again. The calculation returns to 6.4.

For changes $\Delta e_{G(A)} < \Delta e_{G(A),min}$ the smallest $e_{G(I)} = e_{G(A)}$ and the greatest $b_{Ge(I)}$, with corresponding $d_{Ge(I)}$, and $A_{Ge(I)}$ should be applied for all further calculations.

The calculation continues within 7.4 (either with Formula (110) or with Formula (125)).

H.3 Flat gaskets

H.3.1 Flat gaskets with small or median deformations

H.3.1.1 Basic formulae

Due to the rotation of flanges $\left(\Theta_{F(1)} + \Theta_{F(2)} = \Theta_G\right)$ the relative axial deformation of the gasket (Ψ) is variable over the width (in radial direction).

If at a radius r_0 the deformation is Ψ_0, then at a radius $r_0 + \Delta r$ the deformation is:

$$\Psi = \Psi_0 + \Theta_G * \Delta r / e_{Gt} \tag{H.3}$$

$$\Theta_G = H_G * F_G \tag{H.4}$$

$$H_G = \left(h_G * Z_F / E_F\right)_{(1)} + \left(h_G * Z_F / E_F\right)_{(2)} \tag{H.5}$$

From the axial deformation $\Psi_{(r)}$ results an axial compressive stress $Q_{(r)}$ (or vice versa). Equilibrium of axial force $\left(\int Q_{(r)} * dr\right)$ and a moment $\left(\int Q_{(r)} * (r - r_o) * dr\right)$ gives the effective gasket width $\left(b_{Ge}\right)$ and the corresponding diameter $\left(\mathrm{d}_{Ge}\right)$ where the gasket force acts, and where also the actual effective gasket thickness $\left(e_{Ge}\right)$ should be found:

$$e_{Ge} = e_{Gt} * [1 - \Psi_o] - \Theta_{G(I)} * (b_{Gt} - b_{Ge} / 2) \tag{H.6}$$

H.3.1.2 Calculations for a general material law

The average (mean) gasket compressive stress Q_{av} as a function of the average deformation $\Psi = 1 - e_G / e_{Gt}$ may be found by measurements according to EN 13555 (gasket between rigid plates).

This function may be applied also to determine local variable compressive stress $Q_{(\Psi)}$ depending on local variable deformation (gasket between rotating flanges).

It is usefull to have the function by a formula (not only by extended tables) e.g. as follows:

$$Q_{(\Psi)} = \min\{[(C_1 + (C_2 + C_3 * \Psi) * \Psi) * \Psi] / [1 + (B_1 + B_2 * \Psi) * \Psi]; Q_Y\} \tag{H.7}$$

Other formulae are possible. Always should be expected $dQ / d\Psi = D_G > 0$ *for all* Ψ $(0 < \Psi < \Psi_{max})$.

The required integrals may be calculated by a numerical procedure for 6 points (5 sectors) over the theoretical gasket width:

$\Psi_0 = \Psi_{(0)}$ should be assumed.

$$\Psi_{(J)} = \max\{0; \Psi_0 + J * \Delta_\Psi\} \text{ for } J = 1,2,3,4,5. \tag{H.8}$$

$$\Delta_\Psi = \Theta_G * b_{Gt} / (5 * e_{Gt}) \tag{H.9}$$

$$\Psi = \{\Psi_{(0)}, \Psi_{(1)}, \Psi_{(2)}, \Psi_{(3)}, \Psi_{(4)}, \Psi_{(5)}\} = List\ \Psi \tag{H.10}$$

$$Q = \{Q_{(0)}, Q_{(1)}, Q_{(2)}, Q_{(3)}, Q_{(4)}, Q_{(5)}\} = List\ Q \tag{H.11}$$

The average of $Q_{(\Psi)}$ over the theoretical gasket width and the effective gasket width should be as follows:

$$Q_{av} = 0{,}1 * Q_{(0)} + 0{,}2 * Q_{(1)} + 0{,}2 * Q_{(2)} + 0{,}2 * Q_{(3)} + 0{,}2 * Q_{(4)} + 0{,}2 * Q_{(5)} \tag{H.12}$$

$$b_{Ge} = b_{Gt} * [0{,}2 * Q_{(0)} + 0{,}32 * Q_{(1)} + 0{,}24 * Q_{(2)} + 0{,}16 * Q_{(3)} + 0{,}08 * Q_{(4)} + 0{,}0 * Q_{(5)}] / Q_{av} \tag{H.13}$$

If $A_{Gt} * Q_{av} \neq F_G$ then the assumed value Ψ_0 should be changed and the calculation should be repeated. (For continuation with H.2.3 a rough approximation for Ψ_0 is sufficient).

If $b_{Ge} > 0{,}8 * b_{Gt}$ then proceed to H.3.2.

Otherwise the effective gasket thickness e_{Ge} follows from Formula (H.6).

H.3.1.3 Calculations for a simplified material law

The general material law according to Formula (H.7) should be simplified by $B_1 = B_2 = 0$ and $C_2 = C_3 = 0$. Then it corresponds to a linear-elastic-ideal-plastic material $(C_1 = D_G)$:

$$Q_{(\Psi)} = \min\{D_G * \Psi; Q_Y\} \tag{H.14}$$

The required integrals may be calculated analytically. For small loading (elastic behaviour) and large theoretical gasket width, the contact width is $b_{Gc} = b_{G0}$ (independent on the gasket force):

$$b_{G0} = \{2 * e_{Gt} / [\pi * d_G * D_G * H_G]\}^{(1/2)} \tag{H.15}$$

For this case (elastic with $b_{G0} = b_{Gt}$) results $b_{Ge} = (2/3) * b_{G0}$. For three other cases the results are different. They all should approximately collect as follows:

$$b_{Ge} = b_{Gt} * \max\left\{\left[1 + 5 * (b_{Gt} / b_{G0})^4\right]^{(-1/4)};\ F_G / (A_{Gt} * Q_Y)\right\} \tag{H.16}$$

If $b_{Ge} > 0{,}8 * b_{Gt}$ then proceed to H.3.2.

Otherwise the effective gasket thickness e_{Ge} follows from Formula (H.6) with:

$$\Psi_0 = (b_{Go} - b_{Gt}) * \Theta_G / e_{Gt} \tag{H.17}$$

It may be $\Psi_0 < 0$.

H.3.2 Flat gaskets with greater deformations

If $b_{Ge} \approx b_{Gc} \approx b_{Gt}$ then the axial gasket compressive stress becomes nearly uniform over the width.

If in such situation is $F_{G(I)} > A_{Gt} * Q_{Y(I)}$, then greater plastic deformations should be possible. They may be estimated as follows:

Determine a factor for support by radial friction (or shear respectively) at the gasket surfaces:

$$c_G = 1 + 0{,}5 * k_G * [1 + k_G] / [1 + \mu_G * k_G] \tag{H.18}$$

If both radial boundaries of the gasket are open (free):

$$k_G = 1 * \mu_G * b_{Gt} / e_{Ge} \tag{H.19}$$

If only one boundary of the gasket is open, the other should be closed:

$$k_G = 2 * \mu_G * b_{Gt} / e_{Ge} \tag{H.20}$$

If both boundaries of the gasket are closed, then may be assumed:

$$c_G = 5 \tag{H.21}$$

A boundary may be treated to be closed only if the radial gap between the two flanges is smaller than the actual gasket thickness.

For all load conditions the corresponding load ratio should be calculated (preliminary load limit check):

$$\Phi_{G(I)} = F_{G(I)} / (A_{Gt} * Q_{Y(I)} * c_G) \tag{H.22}$$

If $\Phi_{G(I)} < 1$ then no additional plastic deformation occurs.

If $\Phi_{G(I)} > 1$ then some additional plastic deformation occurs.

The effective thickness of the gasket e_{Ge} decreases to a yet unknown value. This value should be found by repeated calculations with different assumed e_{Ge} until $\Phi_{G(I)} \approx 1$. However this should not be necessary, because a decrease of $e_{G(A)} = e_{Ge}$ leads to a decrease of $F_{G(I)}$. Therefore at this point a very rough estimate should be sufficient:

$$e_{Ge,new} = e_{Ge,old} / \Phi_{G(I)} \qquad \text{(H.23)}$$

H.4 Metal gaskets with curved surfaces (Figures 3b, c, e, f)

The effective gasket width essential should depend on the gasket force.

$$b_{Gi} = \sqrt{\frac{6 * k_C * r_2 * \cos\varphi_G * F_{G(I)}}{\pi * d_{Ge} * E_{G(I)}} + \left[\frac{F_{G(I)}}{\pi * d_{Ge} * Q_{Y(I)}}\right]^2} \qquad \text{(H.24)}$$

where

E_G = Modulus of elasticity of the gasket for unloading/reloading at the considered temperature

Q_y = yield stress of the gasket metal

Gasket with simple contact :

$$K_C = 1 \qquad \text{(H.25)}$$

$$d_{Ge} = d_{G0} \qquad \text{(H.26)}$$

Gasket with double contact:

$$K_C = 2 \qquad \text{(H.27)}$$

$$d_{Ge} = d_{Gt} \qquad \text{(H.28)}$$

Follow: b_{Ge}, A_{Ge} from Formulae (61), (62).

$$e_{G(I)} = e_{G(N)} - \left(b_{Ge} / k_C\right)^2 / \left(4 * r_2 * \left(\cos\varphi_G\right)^3\right) \qquad \text{(H.29)}$$

H.5 Metal gaskets with octagonal section (Figure 3d)

Effective dimensions determined for assembly in 6.4 (Table 1) should remain unchanged during ail subsequent load cases. No additional deformations should occur.

Annex I
(informative)

Available, incomplete models for conversion of the leakage rates in different conditions (based on certain flow models)

I.1 Introduction and warning

WARNING — At present, due to the incomplete understanding of involved mechanisms of leakage, this annex cannot be used to determine the on-site leakage rate. It shall be emphasised that the correlations given in the present document are based on a very simplified model where all the leaks are supposed to be concentrated in straight cylindrical capillaries. This is not realistic for the moment to use them without any high additional background. This annex is giving a sum-up of the general correlation trends that are available.

The calculation method developed in the EN 1591 standards suite, enables to design bolted flange connections for a given Tightness Class. The evaluation of the expected Tightness Class can also be performed by a post-calculation analysis in the case where no particular Tightness Class had been taken into account for the calculation.

The link between the Tightness Class rate and the gasket solicitations (contact pressure, internal fluid pressure and temperature) is performed by Lab-testing according to EN 13555. This standard is specifying Helium as the test medium for the sealing test. For practical reasons, the gasket is compressed with non-deformable platens, and has specified dimensions. Moreover, a limited set of conditions (as internal fluid pressure or temperature) are investigated for performing the tests.

When performing a calculation according to EN 1591-1, the input parameter should be selected regarding the life conditions (internal pressure, temperature, external loads,…) for the bolted flange connection. As seen above, the leakage rates are assessed in Lab conditions that may be different from the real conditions on site. This raised the issue of assessment of the expected leakage rate in the real condition depending on the leakage rate measured in Lab.

After a brief introduction of the theory and formulae involved in the flow phenomena, highlighting the difficulties and restrictions of the correlation calculations, a practical method based on current available knowledge is described in this document.

I.2 Flow theory fundamentals

I.2.1 Transport modes

The transport of matter in the context of leakage through gasketed joints (bolted flange connections with gaskets) takes place by diffusion or permeation and by viscous flow. The driving force in diffusion is solely due to differences of concentration which cause matter to be transported in the direction of the lower concentration. Viscous flow on the other hand is triggered by pressure differences. In gasket problems both phenomena are inextricably superimposed. At low pressure differences between the sealed volume and the surroundings diffusion or molecular flow predominates whilst at higher pressure differences the viscous flow fraction prevails. A detailed discussion of these material transport phenomena in sealing problems may be found in [7] and [8].

The derivation of flow theory relationships is effected on the basis of a simplified modelling of gasket body through which seepage is occurring in the sealing face regions. To characterise the flow along the dividing surface between the gasket and flange sealing face reference [7] for example, resorted to a triangular groove as a model from [9] for flange face turning grooves. By means of this the boundary leakage of gasketed joints with compact gaskets e.g. metal gaskets can be modelled. For the description of flow through the body of fibre gaskets, especially those with long fibres, models with capillaries as leakage channels are suitable. To what extent these models are applicable to other jointing materials with different pore or leakage channel structures (Graphite, PTFE, Mica . . .) can only be checked by experiment.

I.2.2 Case of gases

I.2.2.1 Parallel capillary model

The current gas transport laws determined on the basis of the "parallel capillary model" (introduced in [16] & [17]) are summarised in Table 1. The quantity which characterises flow is the mean free path of the gas atoms divided by the capillary diameter which is designated the Knudsen Number. At very small Knudsen Numbers laminar continuum flow (Hagen-Poiseuille) occurs whilst at very large values the flow is molecular and between the two mixed flow incorporating both components occurs.

Table I.1 — Flow type depending on Knudsen number

Knudsen Number K_n	Type of Flow	Calculation Formula
$K_n \ll 1$	Laminar	$\dot{m} = \frac{M}{RT}\frac{nr^4\pi}{16\eta L}\left(p_i^2 - p_o^2\right)$
$1 \le K_n \le 100$	mixed	$\dot{m} = \left(\frac{nr^4\pi}{8\eta}\bar{p}\frac{M}{RT} + \frac{4}{3}\psi\sqrt{\frac{2\pi M}{RT}}nr^3\right)\frac{\Delta p}{L}$
$K_n \gg 100$	molecular	$\dot{m} = \frac{4nr^3}{3L}\sqrt{\frac{2\pi M}{RT}}\Delta p$

$K_n = l/d$	Mean free path length / capillary diameter	R	Universal gas constant
$\dot{m}$	Mass flow (Leak rate in gaskets)	T	Absolute temperature
ψ	Adzumi constant	M	Molar mass
η	Dynamic viscosity	n	Number of capillaries
$p_i - p_o$	External – , internal pressure	r	Radius of capillaries
$\bar{p}$	Mean pressure	L	Length of capillaries
Δp	Pressure difference		

As stated in EN 1779 [1] it is generally assumed that for Helium leakage rates below or equal to 10^{-7} Pa·m^3/s molecular flow takes place. The conditions for visco-laminar flow are given if the leakage rate is above 10^{-5} Pa·m^3/s. As seen through the formulae above, the dependence of the leakage rate on pressure, temperature and type of fluid will be different for both flow types.

I.2.2.2 Dusty gas model

Another flow model is the dusty gas model [10] which according to [11] is better justified experimentally than the capillary model. In the dusty gas model the porous medium is represented by firmly anchored hard spheres (dust) which are distributed throughout the space. The interaction with these obstacles causes the gas transport resistance. The dusty gas model - originally derived for pure diffusion - can be extended to a viscous flow component at high pressure gradients or differences. [11] gives information on the application of this model for fibre based gaskets.

I.2.3 Case of liquids: Parallel capillary model

As defined in [24], for liquid, the flow is considered laminar (not turbulent) through the capillary due to its diameter size. Thus, the liquid leakage rate will follow the Poiseuille's law.

$$L = n \times \rho \times \frac{\pi}{8} \times \frac{r^4}{\eta \times L}\left(p_\mathrm{i} - p_\mathrm{o}\right) \qquad \text{(I. 1)}$$

(using the parameter definitions of I.2.2.1 and with ρ: density [g.m^{-3}]

I.3 Factors of influence on the leakage rate of gaskets and gasketed joints

I.3.1 List of identified factors

According to both models described above ("parallel capillary model" and "dusty gas model") the important quantities influencing the leakage rate are:

- the absolute pressure and the pressure difference;
- the temperature;
- the molar mass and viscosity of the gas;
- the gasket geometry (width, thickness);
- the microstructure of the gasket (geometry, number of capillaries, specific permeability, structural constant).

I.3.2 Limits and restriction of the proposed models

Although self-evident it has to be emphasised that the effect of the gasket stress in the bolted flange connection - probably the critical parameter - is not included directly in either model.

In view of the approximate description of actual gaskets and gasketed joints given by the above models, it cannot be expected that the derived mathematical relationships will allow quantitative statements concerning leakage rate to be made. Under specified conditions however qualitative statements on the effect of the aforementioned factors of influence on the leakage of gaskets can be produced.

A prerequisite for the validity of all mathematical relationships is of course, that no interaction (e.g. oxidation, swelling) between the medium being contained and the gasket occurs and furthermore that no change in the gasket material over time, caused in particular by higher temperatures, takes place (e.g. hardening of rubber-binded fibre gaskets). In such cases there is no correlation between the initial condition and the various stages of ageing.

Before discussing the individual dependencies it shall be stressed at the outset that even a merely approximately realistic theoretical understanding of the gasket structure is not possible with either model and will therefore not be pursued further. Along with this also stands the fact already mentioned in the relationship, that the effect of gasket stress in the form of a change of the gasket structure has not been included. With regard to this, experimental investigations are an indispensable requirement.

I.3.3 Dependence on pressure

I.3.3.1 General

NOTE This subclause is based on [6].

According to Table I.1, for a given leak, whose geometrical dimensions are not varied by the pressure applied, the following relations have to be used in order to determine the effect of pressure variations on the leakage flow.

I.3.3.2 Definition

$p_{\mathrm{LOW,(ref)}}$ and $p_{\mathrm{LOW,(act)}}$	[Pa]	different low pressures for “reference” and “actual” conditions
$p_{\mathrm{HIGH,(ref)}}$ and $p_{\mathrm{HIGH,(act)}}$	[Pa]	different high pressures for “reference” and “actual” conditions
$L_{\mathrm{P,(ref)}}$ and $L_{\mathrm{P,(act)}}$	[mg/s]	leakage rates related to the pressure differences for “reference” and “actual” conditions

I.3.3.3 Molecular flow

$$L_{\mathrm{P,(act)}} = L_{\mathrm{P,(ref)}} \frac{\Delta p_{\mathrm{(act)}}}{\Delta p_{\mathrm{(ref)}}} \tag{I. 2}$$

with pressure differences

$$\Delta p_{\mathrm{(act)}} = p_{\mathrm{HIGH,(act)}} - p_{\mathrm{LOW,(act)}} \quad \text{with} \quad p_{\mathrm{HIGH,(act)}} \geq p_{\mathrm{LOW,(act)}}$$

$$\Delta p_{\mathrm{(ref)}} = p_{\mathrm{HIGH,(ref)}} - p_{\mathrm{LOW,(ref)}} \quad \text{with} \quad p_{\mathrm{HIGH,(ref)}} \geq p_{\mathrm{LOW,(ref)}}$$

I.3.3.4 Visco-laminar flow

$$L_{\mathrm{P,(act)}} = L_{\mathrm{P,(ref)}} \frac{\Delta p_{\mathrm{(act)}} * \overline{P}_{\mathrm{(act)}}}{\Delta p_{\mathrm{(ref)}} * \overline{P}_{\mathrm{(ref)}}} \tag{I. 3}$$

with pressure mean values

$$\overline{P}_{\mathrm{(ref)}} = \frac{(p_{\mathrm{HIGH,(ref)}} + p_{\mathrm{LOW,(ref)}})}{2} \tag{I. 4}$$

$$\overline{P}_{\mathrm{(act)}} = \frac{(p_{\mathrm{HIGH,(act)}} + p_{\mathrm{LOW,(act)}})}{2} \tag{I. 5}$$

I.3.3.5 Practical use

Because the flow regime is rarely clearly identified (especially if the medium is not Helium), a practical conservative approach is proposed here to convert the leakage rate values depending on the pressure value based on VDI 2200: 2007 [12].

If $\Delta p_{(act)} > \Delta p_{(ref)}$, then $L_{P,(act)} = L_{P,(ref)} \dfrac{\Delta p_{(act)} * \overline{P}_{(act)}}{\Delta p_{(ref)} * \overline{P}_{(ref)}}$ (I. 6)

If $\Delta p_{(act)} < \Delta p_{(ref)}$, then $L_{P,(act)} = L_{P,(ref)} \dfrac{\Delta p_{(act)}}{\Delta p_{(ref)}}$ (I. 7)

I.3.4 Dependence on temperature

I.3.4.1 General

NOTE This subclause is based on [1].

For a given leak, whose geometrical dimensions are not varied by temperature changes, the following relations have to be used in order to determine the effect of temperature variations on the leakage flow.

I.3.4.2 Definition

$L_{T,(ref)}$ and $L_{T,(act)}$	[mg/s]	leakage rates related to $\boldsymbol{T}_{(ref)}$ and $\boldsymbol{T}_{(act)}$ for “reference” and “actual” conditions
$T_{(ref)}$ and $T_{(act)}$	[K]	different absolute temperatures for “reference” and “actual” conditions
$\eta_{T,(ref)}$ and $\eta_{T,(act)}$	[Pa·s]	dynamic viscosities related to $\boldsymbol{T}_{(ref)}$ and $\boldsymbol{T}_{(act)}$ for “reference” and “actual” conditions

I.3.4.3 Molecular flow

$$L_{T,(act)} = L_{T,(ref)} \sqrt{\frac{T_{(ref)}}{T_{(act)}}} \qquad \text{(I. 8)}$$

I.3.4.4 Visco-laminar flow

$$L_{T,(act)} = L_{T,(ref)} \frac{\eta_{T,(ref)}}{\eta_{T,(act)}} \qquad \text{(I. 9)}$$

I.3.4.5 Practical use

Because the differentiation between the type of flow is most of the time a problem, (especially if the medium is not Helium), a practical conservative approach is proposed here to convert the leakage rate values depending on the temperature.

$$L_{T,(act)} = L_{T,(ref)} * MAX\left[\sqrt{\frac{T_{(ref)}}{T_{(act)}}};\frac{\eta_{T,(ref)}}{\eta_{T,(act)}}\right] \qquad \text{(I. 10)}$$

I.3.5 Dependence on the type of fluid

I.3.5.1 General

NOTE This subclause is based on [1].

For a given leak the relation of the leakage rate of two different gaseous fluids is determined by the following. It shall be noted that, these formulae do not take into account the change in the dimension of a leak path either the modifications in the gasket material due the interaction with the internal medium.

I.3.5.2 Definition

$L_{\text{Media,(ref)}}$ and $L_{\text{Media,(act)}}$	[mg/s]	leakage rates related to “reference” and “actual” gaseous medium
$M_{\text{Media,(ref)}}$ and $M_{\text{Media,(act)}}$	[kg / Mol]	Mol masses of “reference” and “actual” gaseous medium
$\eta_{\text{Media,(ref)}}$ and $\eta_{\text{Media,(act)}}$	[Pa·s]	Dynamic viscosities of “reference” and “actual” gaseous medium

I.3.5.3 Molecular flow

$$L_{\text{Media,(act)}} = L_{\text{Media,(ref)}} \cdot \sqrt{\frac{M_{\text{Media,(act)}}}{M_{\text{Media,(ref)}}}} \tag{I. 11}$$

I.3.5.4 Visco-laminar flow

$$L_{\text{Media,(act)}} = L_{\text{Media,(ref)}} \cdot \frac{\eta_{\text{Media,(ref)}}}{\eta_{\text{Media,(act)}}} \tag{I. 12}$$

I.3.5.5 Practical use

Because the differentiation between the type of flow is most of the time a problem, (especially if the medium is not Helium), a practical conservative approach is proposed here to convert the leakage rate values depending on the gaseous medium.

$$L_{\text{Media,(act)}} = L_{\text{Media,(ref)}} \cdot MAX\left[\frac{\eta_{\text{Media,(ref)}}}{\eta_{\text{Media,(act)}}}; \sqrt{\frac{M_{\text{Media,(act)}}}{M_{\text{Media,(ref)}}}}\right] \tag{I. 13}$$

I.3.6 Influence of the gasket thickness

NOTE This subclause is based on [15].

A comparison of the leakage rates from gaskets of the same type but of different thicknesses is generally of little use. Differences in manufacture lead to different microstructures which do not allow unambiguous dependence of the leakage rate on gasket thickness to be expected. In addition a shift in the ratios of the contributions of boundary leakage along the interface and through the body of the gasket occurs. Furthermore the consolidation under compression and with it the change in microstructure, is thickness dependent.

As is to be expected experimental results with thicker gaskets for the most part show higher leakage rates under otherwise identical conditions. Occasionally an inversion of the relationship for the same gasket type has also been reported e.g. for CAF material [8], [11].

In view of this there appears to be no reliable correlation between leakage rate and gasket thickness and a reliable transfer of leakage rates to other gasket thicknesses seems to be ruled out.

I.3.7 Influence of gasket width

NOTE This subclause is based on [15].

A pre-requirement for an inversely proportional relationship between leakage rate and gasket width for uniform homogenous compressive stress is a sufficiently wide gasket, in comparison with the leakage channel length.

For fibre based gaskets the characteristic leakage channel length (in the model the capillary length) can be treated as equivalent to the fibre length. Therefore the width of the gasket shall clearly be greater than the fibre length. If the width of the gasket is reduced beyond this then a disproportionately high increase of the leakage is to be seen as was shown in [8] for CAF material.

A deviation from the inverse proportional relationship can also occur with thick gaskets. In this case because of the smaller supporting effect the same compressive stress leads to a higher mean deformation of narrow gaskets.

In view of this, there appears to be no reliable correlation between leakage rate and gasket width directly usable for all the gasket types.

I.3.8 Influence of gasket stress

NOTE This subclause is based on [15].

An inverse exponential relationship between leakage rate and pressure on the gasket face both for loading and unloading after pre-deformation in CAF materials was first demonstrated by Bierl [8] and subsequently confirmed approximately for CAF in [13] and also for other types of gaskets in [14].

But, due to the huge variety of gasket types, this relation cannot be generalised for all the gasket types. The Influence of the gasket stress is taken into account by the Lab-test performed according to EN 13555. The tabulated results in EN 1591-2, coming from the test performed according to EN 13555 are giving the leakage rate values depending on the actual gasket stress as on the gasket stress at assembly.

The determination of the expected leakage rate depending on the gasket stress has to be performed using either:

— the leakage diagram obtained from EN 13555;

— interpolation between data points in EN 1591-2 or other database (guidelines on the way to perform the interpolation are planned to be given in another document);

— a modelling of the leakage vs. gasket stress diagram obtained for that particular gasket type (if possible).

In view of this, there appears to be no general modelling for correlation between leakage rate and gasket stress currently available and directly usable for all the gasket types. The best that can be done is the creation of a specific model (including specific parameter values) for each specific gasket type or reference.

I.3.9 Influence of other factors

As stated in [1], other factors as the direction of flow (internal pressure higher or lower than the external pressure) or the presence of humidity, oil, grease and other contamination can have an impact on the effective leakage rate. But, these factors will not be taken into account in the further conversion calculations due to their difficulty to be modelled.

I.3.10 Conclusion on the factors of influence

As seen above, a prerequisite for the validity of all mathematical relationships developed above is of course, that no interaction (e.g. oxidation, swelling,…) between the medium being contained and the gasket occurs and furthermore that no change in the gasket material over time, caused in particular by higher temperatures or pressures, takes place. In such cases, there is no correlation between the initial condition and the various stages of ageing.

Due to the variety in the gasket types (material, structure, …), the leak paths numbers, locations and shape and the number of parameters involve in the effective leakage rate, only rough qualitative statements on the effect of the aforementioned factors of influence on the leakage of gaskets can be produced. From the relationships derived on the basis of the dusty gas and capillary models developed here, it cannot be expected that the derived mathematical relationships will allow quantitative statements concerning leakage rate to be made. Moreover, several experiments performed in Lab had shown the limits of the direct application of this formula for pressure and medium dependence for example. With regard to this experimental investigations are an indispensable requirement to get the effective leakage rate in real conditions.

However, this sub-clause gives a list of the typical factors influencing the leakage rate values. When possible, formulae are given to evaluate “roughly” the evolution of the leakage rate depending on the variation of the considered parameter. Because only “rough” estimation could be extracted from the proposed conversion models, the influence factors introduced above are neglected when there is no associated straightforward and “universal” (for all the gasket types) model is available. There is no point to introduce a too high complexity in these conversion calculations knowing that the results will only give a “rough” estimation on the converted leakage rate value.

I.4 Practical application for EN 1591-1 calculations

I.4.1 General

The conversion principles and formulae described above can be used to convert the leakage rate values from a first set of conditions to another set of conditions. Considering the calculation of bolted flange connection, the conversion can be used to estimate “on site” (called “actual” condition) expected leakage rates from ‘in Lab” (called “reference” condition) measured leakage rate during test according to EN 13555 (see Table I.2).

This enables to get a “rough” trend for the “on site” expected leakage rate for the considered bolted flange connection and/or the considered group of bolted flange connections. The conversion rules can also be used in the opposite way i.e. to determine the “reference” leakage rate that shall be used given the “on site’ leakage criteria (see Table I.3).

It shall be reemphasised here that the conversion tables presented here are based on the formulae developed above, and therefore have the same limits and level of imprecision. They are not valid for the estimation of “on-site” leakage rates.

I.4.2 Determination of a trend for the leakage rate for the flange connection in “actual” from “reference” conditions

Table I.2 — Determination of a trend for the leakage rate for the flange connection in “actual” from “reference” conditions

	Step	Coefficient definition	result	unit
From “reference” to “actual” Leakage rate	Determination of the “reference” leakage rate from EN 13555 test results for each situation		$L_{[dG, P, T, medium](ref)}$	mg/s/m
	Correction for real flange connection dimension	$K_D = \frac{d_{Gmean,(act)}}{1000}$	$L_{[dG](act),[P, T, medium](ref)}$	mg/s
	Correction for fluid pressure (only the pressure effect in the leak path is taken into account, the effect of pressure on the flange connection mechanical behaviour is not taken into account here)	If $\Delta p_{(act)} > \Delta p_{(ref)}$, then $K_P = \frac{\Delta p_{(act)} * \overline{P}_{(act)}}{\Delta p_{(ref)} * \overline{P}_{(ref)}}$ If $\Delta p_{(act)} < \Delta p_{(ref)}$, then $K_P = \frac{\Delta p_{(act)}}{\Delta p_{(ref)}}$	$L_{[dG, P](act),[T, medium](ref)}$	mg/s
	Correction for temperature (only the temperature effect on the fluid is taken into account, the temperature effect on the gasket material behaviour is not taken into account)		$L_{[dG, P,T](act),[medium](ref)}$	mg/s
	Correction for the medium and determination of the “actual” leakage rate.	$K_T = MAX\left[\frac{\eta_{Medium,(ref)}}{\eta_{Medium,(act)}}; \sqrt{\frac{T_{Medium,(ref)}}{T_{Medium,(act)}}}\right]$ $K_M = MAX\left[\frac{\eta_{Medium,(ref)}}{\eta_{Medium,(act)}}; \sqrt{\frac{M_{Medium,(act)}}{M_{Medium,(ref)}}}\right]$	$L_{[dG, P,T, medium](act)}$	mg/s

$L_{[dG, P,T, medium](act)} = K_D{*}K_P{*}K_T{*}K_M{*}L_{[dG, P, T, medium](ref)}$

I.4.3 Determination of a trend for the leakage rate for the flange connection in “reference” from “actual” conditions

Table I.3 — Determination of a trend for the leakage rate for the flange connection in “reference” from “actual” conditions

	Step	Coefficient definition	result	unit
From “actual” to “reference” Leakage rate	Leakage rate criteria for real conditions		$L_{[\text{dG, P,T, medium}](\text{act})}$	mg/s
	Correction for the medium and determination of the “actual” leakage rate.	$K'_{\text{M}} = MAX\left[\frac{\eta_{\text{Medium,(act)}}}{\eta_{\text{Medium,(ref)}}};\sqrt{\frac{M_{\text{Medium,(ref)}}}{M_{\text{Medium,(act)}}}}\right]$	$L_{[\text{dG, P,T}](\text{act}),[\text{medium}](\text{ref})}$	mg/s
	Correction for temperature (only the temperature effect on the fluid is taken into account, the temperature effect on the gasket material behaviour is not taken into account)	$K'_{\text{T}} = MAX\left[\frac{\eta_{\text{Medium,(act)}}}{\eta_{\text{Medium,(ref)}}};\sqrt{\frac{T_{\text{Medium,(act)}}}{T_{\text{Medium,(ref)}}}}\right]$	$L_{[\text{dG, P}](\text{act}),[\text{T, medium}](\text{ref})}$	mg/s
	Correction for fluid pressure (only the pressure effect in the leak path is taken into account, the effect of pressure on the flange connection mechanical behaviour is not taken into account here)	If $\Delta p_{(\text{act})} < \Delta p_{(\text{ref})}$, then $K'_{P} = \frac{\Delta p_{(\text{ref})} * \overline{P}_{(\text{ref})}}{\Delta p_{(\text{act})} * \overline{P}_{(\text{act})}}$ If $\Delta p_{(\text{act})} > \Delta p_{(\text{ref})}$, then $K'_{\text{P}} = \frac{\Delta p_{(\text{ref})}}{\Delta p_{(\text{act})}}$	$L_{[\text{dG}](\text{act}),[\text{P, T, medium}](\text{ref})}$	mg/s
	Correction for real flange connection dimension	$K'_{\text{D}} = \frac{d_{\text{Gmean,(act)}}}{1000}$	$L_{[\text{dG, P, T, medium}](\text{ref})}$	mg/s/m

$L_{[\text{dG, P,T, medium}](\text{ref})} = K'_{\text{M}} * K'_{\text{T}} * K'_{\text{P}} * K'_{\text{D}} * L_{[\text{dG, P, T, medium}](\text{act})}$

with:

— K_D, K_P, K_T and K_M = correction coefficient for bolted flange dimension, internal pressure, temperature and medium change for conversion from “reference” to “actual” conditions;

— K'_D, K'_P, K'_T and K'_M = correction coefficient for bolted flange dimension, internal pressure, temperature and medium change for conditions from “actual” to “reference” conditions;

— $L_{[X,\ Y](act),[Z,W](ref)}$ = Leakage rate with X and Y parameters corresponding to “actual” conditions and with Z and W parameters corresponding to “reference” conditions [mg/s or mg/s/m];

— $d_{Gmean\ (act)}$: Mean diameter of the gasket on the considered bolted flange connection = (internal gasket diameter + external gasket diameter)/2 [mm].

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 to provide a means of conforming to Essential Requirements of the New Approach Directive 97/23/EC of the European Parliament and of the Council of 29 May 1997 on the approximation of the laws of the Member States concerning pressure equipment.

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 on Pressure Equipment

Clause(s)/ subclause(s) of this EN 1591-1	Essential Requirements (ERs) of Directive 97/23/EC on Pressure Equipment Annex I	Qualifying remarks/Notes
5, 6, 7	2	Design:
	2.1	To be designed to ensure safety throughout intended life - to incorporate appropriate safety coefficients.
	2.2	To be designed for adequate strength
	2.2.1	To be designed for loadings appropriate to its intended use.
	2.2.2	To be designed for appropriate strength based on a calculation method.
	2.2.3 (a)	Requirements to be met by applying one of the following methods - design by formula.
	2.2.3 (b)	Design calculations to establish the resistance of equipment, in particular -account to be taken of combinations of temperature & pressure; -maximum stresses & peak stresses to be within limits.

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

Bibliography

[1] EN 1779:1999, *Non-destructive testing — Leak testing — Criteria for method and technique selection*

[2] EN ISO 4014:2011, *Hexagon head bolts — Product grades A and B (ISO 4014:2011)*

[3] EN ISO 4016:2011, *Hexagon head bolts — Product grade C (ISO 4016:2011)*

[4] ISO 261:1998, *ISO general purpose metric screw threads — General plan*

[5] ISO 724:1993, *ISO general-purpose metric screw threads — Basic dimensions*

[6] CR 13642:1999, *Flanges and their joints — Design rules for gasketed circular flange connections — Background information*

[7] Micheely, A :1997: Untersuchungen an Rohrleitungsflanschen bei Betriebsbedingungen unter besonderer Berücksichtigung des Leckverhaltens (Investigations on piping flanges under operating conditions with particular regard to leakage behaviour). Dissertation Universität Dortmund 1977

[8] Bierl, A. 1978: Untersuchungen der Leckraten von Gummi-Asbest-Dichtungen in Flanschverbindungen (Investigations on the leakage rates of rubber-asbestos gaskets in flanged joints). Dissertation Ruhr-Universität Bochum 1978

[9] Carlson, L.W; Irvine, T.F.: 1961: Fully developed pressure drop in triangular ducts.- Trans. ASME Ser. C, J. Heat Transfer 83 (1961) pp 441/444

[10] Jackson, R.: 1977 - Transport in Porous Catalysts. Elsevier Scientific Publishing Comp. Amsterdam - Oxford - New York 1977

[11] Kämpkes, W : 1982 .- Einflüsse der Dichtungsgeometrie auf die Gasleckage an Rohrleitungsflanschverbindungen mit IT-Flachdichtungen (Influence of gasket geometry on gas leakage in pipeline flanged joints with flat CAF gaskets) - Dissertation Universität Dortmund 1982

[12] VDI 2200:2007. Tight flange connections – Selection, calculation, design and assembly of bolted flange connections

[13] Tückmantel, H. J.:1988: Die Berechnung statischer Dichtverbindungen unter Berücksichtigung der maximal zulässigen Leckmenge auf der Basis einer neuen Dichtungstheorie (The design of static sealed joints in view of the maximum permissible leakage based on a new gasket theory) - Konstruktion 40 (1988) S. 116/120

[14] Gusko, A.; Kockelmann, H. 1993: Dichtheitsuntersuchungen an Flanschverbindungen in Anlehnung an DIN E 28091 (Tightness investigations on flanged joints following DIN E 28091). VIII. Int. Dichtungskolloquium, TH Köthen, 21/22. April 1993

[15] H. Kockelmann:1996: Leckageraten von Dichtungen für Flanschverbindungen - Einflußgrößen, Anforderungen, messtechnische Erfassung und leckageratenbezogene Dichtungskennwerte (leakage rate of gaskets in bolted flange connections based on flow theory) Chemie Ingenieur Technik 68 (1996) 219/227

[16] J. Bear: 1972: Dynamics of Fluids in Porous Media, Amsterdam, 1972

[17] J. Bear, Y. Bachmat 2007: Introduction to Modeling of Transport Phenomena in Porous Media, Springer-Verlag GmbH, February 2007

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

[19] EN 1092-2:1997, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 2: Cast iron flanges*

[20] EN 1092-3:2003, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 3: Copper alloy flanges*

[21] EN 1092-4:2002, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 4: Aluminium alloy flanges*

[22] EN 1591-2:2008, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 2: Gasket parameters*

[23] CEN/TS 1591-3:2007, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 3: Calculation method for metal to metal contact type flanged joint*

[24] ISO 12807:1996, *Safe transport of radioactive materials – Leakage testing on packages*

[25] EN 1759-1:2004, *Flanges and their joint — Circular flanges for pipes, valves, fittings and accessories Class designated — Part 1: Steel flanges, NPS ½ to 24*

[26] EN 1759-3:2003, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories Class designated — Part 3: Copper alloy flanges*

[27] EN 1759-4:2003, *Flanges and their joint — Circular flanges for pipes, valves, fittings and accessories Class designated — Part 4: Aluminium alloy flanges*

[28] EN 13445-3, *Unfired pressure vessels — Part 3: Design*

[29] EN 13480-3, *Metallic industrial piping — Part 3: Design and calculation*

[30] NORSOK L-005: 2006 Edition 2

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