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Flanges and their joints - Design rules for gasketed circular flange connections - Part 1: Calculation method

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

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

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Flanges and their joints - Design rules for gasketed circular flange connections - Part 1: Calculation method

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This European Standard was approved by CEN on 8 March 2001 and includes Amendment 1 approved by CEN on 7 February 2009.

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Contents

Page

Foreword

This document (EN 1591-1:2001+A1:2009) 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of an identical text or by endorsement, at the latest by September 2009, and conflicting national standards shall be withdrawn at the latest by September 2009.

This document includes Amendment 1, approved by CEN on 2009-02-07.

This document supersedes EN 1591-1:2001.

The start and finish of text introduced or altered by amendment is indicated in the text by tags [A1⟩ ⟨A1].

This European Standard has been prepared under a mandate given to CEN by the European Commission and the European Free Trade Association. This European Standard is considered as a supporting standard to other application and product standards which in themselves support an essential safety requirement of a New Approach Directive and will appear as a normative reference in them.

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

EN 1591 consists of two parts:

– EN 1591-1 Flanges and their joints – Design rules for gasketed circular flange connections – Part 1: Calculation method

– [A1⟩ EN 1591-2 ⟨A1] Flanges and their joints – Design rules for gasketed circular flange connections – Part 2: Gasket parameters

The Calculation method satisfies both leaktightness 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 forces and bending moments;

– effect of temperature difference between bolts and flange ring

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:

1) 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 A1⟩ EN 1591-2 ⟨A1, 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.

2) 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:

   — bolting-up condition: the check is performed against the maximum possible bolt force that may result from the bolting-up procedure;
   — test and operating conditions: checks are performed against the minimum necessary forces, to ensure that the connection will be able to develop these minimum forces without risk of yielding, except in highly localized 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 E, and the values obtained, compared with the relevant gasket limits which could apply.

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.

NOTE Where flanges are used to comply with other codes the Calculation method does not specify values for nominal stresses.

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.

The minimum force required on the gasket for leak tightness considerations may be established by two different ways:

1) Use of tabulated gasket factors, for example those given in A1⟩ EN 1591-2 ⟨A1, which are based on industrial experience and correspond to mainly gas and steam leak rates.

2) Derivation from measured leak rate versus gasket stress data, if available for the gasket, for example as in A1⟩ EN 1591-2 ⟨A1. This permits design to be based on any specified maximum leak rate.

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.

In the present stage of development, the Calculation method is not applicable to joints with narrow metal-to-metal contact (with the exception of joints with spacer seated flanges (see annex G)), or to joints whose rigidity varies appreciably across gasket width.

A chart illustrating the calculation process is given in annex F.

!A1) 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.

The objective for the publication of this new edition of EN 1591-1:2001 is to keep the standard in line with EN 1591-2:2008. The calculation methodology and interpretation of gasket data is the subject of on going work in Joint Working Group CEN/TC54/TC69/TC74/TC267/TC269/JWG. This publication is therefore transitory and will be updated in due course.

EN 1591-1 is based upon the principle that a selected leakage rate is to be achieved. But, where there is no requirement on limitation of leakage, the following two modifications are suggested:

– In Equation (49) the gasket surface pressure Q_A may be replaced by $Q_{0,min}$ taken from EN 13445-3:2002, Annex G;

– In Equation (50) the gasket surface pressure $Q_{smin(L)I}$ may be replaced by $Q_{I,min} = m_I \times |P_I|$, with m_I taken from EN 13445-3:2002, Annex G. (A1!

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

1 Scope

1.1 General

This European Standard defines a Calculation method for bolted, gasketed, circular flange joints. Its purpose is to ensure structural integrity and control of leaktightness. A1 The following equations use gasket parameters based on definitions and test methods specified in EN 13555. A1

1.2 Requirement for use of the Calculation method

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.

1.3 Validity

1.3.1 Geometry

The Calculation method is applicable to the configurations having:

— flanges whose section is given or may be assimilated to those given in Figures 4 to 12;

— four or more identical bolts uniformly distributed;

— gasket whose section and configuration after loading can be assimilated by one of those given in Figure 3;

— flange dimension which meet the following conditions:

a) $0,2 \le b_F/e_F \le 5,0$; $0,2 \le b_L/e_L \le 5,0$

b) $e_F \le \max \{e_2; d_{B0}; p_B \times \sqrt[3]{(0,01 ... 0,10) \times p_B / b_F}\}$

c) $\cos\varphi \ge 1/(1 + 0,01 \dfrac{d_s}{e_s})$

NOTE For explanations of symbols see clause 3.

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

NOTE The condition $e_F \ge p_B \times \sqrt[3]{(0,01 ... 0,10)\, p_B / b_F}$ is for limitation of non-uniformity of gasket pressure due to spacing of bolts. The values 0,01 and 0,10 are to be applied for soft (non-metallic) and hard (metallic) gaskets respectively. A more precise criterion is given in annex A.

NOTE Attention may need to be given to the effects of tolerances and corrosion on dimensions; reference should be made to other codes under which the calculation is made, for example values are given in 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, except the special case of spacer-seated flanges, which is covered in annex G.

1.3.2 Materials

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.

Design stresses for bolts are to be determined as for flanges and shells. The model of the gaskets 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.

1.3.3 Loads

This Calculation method applies to the following load types:

– fluid pressure: internal or external;

– external loads: axial forces and bending moments;

– axial expansion of flanges, bolts and gasket, in particular due to thermal effects.

1.3.4 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 1.3.1 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 explicity taken into account in the calculation formula.

This presupposition requires compliance with 1.3.1 c).

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 contacts the flange faces over a (calculated) annular area. The effective gasket width (radial) bGe may be less than the true width of gasket. This effective width bGe is calculated for the assembly condition (I = 0) and is assumed to be unchanged for all subsequent load conditions (I = 1,2 ...). The calculation of bGe includes the elastic rotation of both flanges as well as the elastic and plastic deformations of the gasket (approximately) in assembly condition.

e) A1 The modulus of elasticity of the gasket may increase with the compressive stress Q on the gasket. The 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) in assembly condition. A1

f) A1 Creep of the gasket under compression is approximated by a creep factor P_{QR}. A1

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

h) Loading of the flange joint is axisymmetric. Any non-axisymmetric bending moment is replaced by an equivalent axial force, which is axisymmetric according to equation (44).

i) 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 5.4) to ensure that the required forces on the gasket are achieved under all conditions (see 5.3 and 5.5).

j) load limit proofs are based on limit loads for each component. This approach prevents excessive deformations. The limits used for gaskets, which depend on Qmax are only approximations.

The model does not take account of the following:

k) Bolt bending stiffness and bending strength. This is a conservative 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 equation (34)).

l) Creep of flanges and bolts.

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

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

o) external torsional moments and external shear loads, e.g. those due to pipework.

2 Normative references

A1 The following referenced documents are indispensable for the application of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies. A1

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

A1 EN 1092-2:1997 A1, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 2: Cast iron flanges*

A1 EN 1092-3:2003, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 3: Copper alloy flanges* A1

A1 EN 1092-4:2002 A1, *Flanges and their joints — Circular flanges for pipes, valves, fittings and accessories, PN designated — Part 4: Aluminium alloy flanges*

A1 *deleted text* A1

A1 EN 13555:2004, *Flanges and their joints — Gasket parameters and test procedures relevant to the design rules for gasketed circular flange connections* A1

3 Notation

3.1 Use of figures

Figures 1 to 12 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.

For standard flange types, according to EN 1092, the relevant figures are the following:

Type 01	Figure 8
Type 02	Figure 10
Type 04	Figure 10
Type 05	Figure 9
Type 07	Figure 10
Type 11	Figure 4
Type 12	Figure 11
Type 13	Figure 12
Type 21	Figure 4 to 7

3.2 Subscripts and special marks

3.2.1 Subscripts

A – Additional (F_A, M_A)

B – Bolt

C – Creep of gasket (g_C)

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

M – Moment

P – Pressure

Q – Net axial force due to pressure

R – Net axial force due to external force

S – Shell, shear

T – Shell, modified

X – Weak cross-section

Δ – Symbol for change or difference

[A1⟩ actual the actual dimensions considered in the calculation ⟨A1]

av – average

c – calculated

d – design

e – effective

max – maximum

min – minimum

nom – nominal

opt – optimal

[A1⟩ ref dimensions of reference in EN 13555:2004, 7.4 ⟨A1]

req – required

s – non-threaded part of bolt

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], equation (33)
A_F, A_L	Gross radial cross-section area (including bolt holes) of flange ring, loose flange [mm^2], equations (5), (7), (8)
A_{Ge}, A_{Gt}	Gasket area, effective, theoretical [mm^2], equations (39), (36)
C	Coefficient to account for twisting moment in bolt load ratio, equation (71)

[A1⟩ *deleted text* ⟨A1]

E_B, E_F, E_G, E_L	Modulus of elasticity of the part designated by the subscript, at the temperature of the part [MPa] [A1⟩ *deleted text* ⟨A1]
F_A	Additional external axial force [N], tensile force > 0, compressive force < 0, see Figure 1
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, equation (51)

F_Q Axial fluid-pressure force [N], equation (43)

F_R Force resulting from F_A and M_A [N], equation (44)

I Load condition identifier, for assembly condition I = 0, for subsequent conditions I = 1, 2, 3, ...

I_B Plastic torsion modulus [mm^3] of bolt shanks $\left[= \frac{\pi}{12} \times \min\left(d_{Be}; d_{Bs}\right)^3 \right]$ equation (71)

Ⓐ1 *deleted text* Ⓐ1

K_s Systematic error due to the inaccuracy of the bolt tightening method

M_A Additional external moment [N × mm], Figure 1

M_t Bolt assembly torque [N × mm], annex D

$M_{t,B}$ Twisting moment [N × mm] applied to bolt shanks as a result of application of the bolt assembly torque M_t, equations (71) and (D.8) to (D.11)

N_R Number of re-assemblies and re-tightenings during service life of joint, equation (67)

P Pressure of the fluid [MPa], internal pressure > 0, external pressure < 0 (1 bar = 0,1 MPa)

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

Ⓐ1 P_{QR} Creep factor which is the ratio of the residual and the original gasket surface pressure at load conditions [-], Equation (51), (68) Ⓐ1

Q Mean effective gasket compressive stress [MPa], $Q = F_G/A_{Ge}$

Ⓐ1 Q_A – Gasket surface pressure at assembly prior to the unloading which is necessary for the validity of $Q_{S\,min\,(L)I}$ in service conditions [MPa], Equation (49)

$Q_{S\,min\,(L)}$ – Minimum level of gasket surface pressure required for tightness class L after off-loading at load conditions [MPa], Equation (50)

$Q_{min\,(L)}$ – Minimum level of gasket surface pressure required for tightness class L on assembly (on the effective gasket area) [MPa], lowest acceptable value for Q_A

$Q_{S\,max}$ – Maximum gasket surface pressure that can be safely imposed upon the gasket at the service temperature without damage [MPa], Equation (72a), (72b)

Q_{max} – Maximum gasket surface pressure that can be safely imposed upon the gasket at the service temperature without damage (for actual geometry of the gasket used in bolted flange connection) [MPa] ,Equation (72b), (72c)

$Q_{max,Y}$ – Maximum gasket surface pressure that can be safely imposed upon the gasket at the service temperature without damage (independent from the geometry of the gasket) [MPa], Equation (72a) Ⓐ1

T_B, T_F, T_G, T_L Temperature (average) of the part designated by the subscript [°C] or [K], equation (45)

T_O — Temperature of joint at assembly [°C] or [K] (usually + 20 °C)

U — Axial displacement [mm]; ΔU according to equation (45)

W_F, W_L, W_X — Resistance of the part and/or cross-section designated by the subscript [N × mm], equations (74), (86), (88), (90)

X_B, X_G — Axial flexibility modulus of bolts, gasket [1/ mm], equations (34), (42)

Y_G, Y_Q, Y_R — Axial compliance of the bolted joint, related to F_G, F_Q, F_R [mm/N], equations (46), (47), (48)

Z_F, Z_L — Rotational flexibility modulus of flange, loose flange [mm ^{3}], equations (27), (31), (32)

b_0 — Width of chamfer (or radius) of a loose flange [mm] see Figure 10, equation (15) such that:

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

b_F, b_L — Effective width of flange, loose flange [mm], equations (5) to (8)

b_{Gi}, b_{Ge}, b_{Gt} — Gasket width (radial), interim, effective, theoretical [mm], equations (35), (38), Table 1

A1⟩ c_1 — Numerical constant for a gasket type, e.g. c_1 = 1/20 for fibre based sheet gasket materials, for gaskets for which no value is available, c_1 = 0 can be used, Equations (72a), (72b) ⟨A1

c_F, c_M, c_S — Correction factors, equations (20), (78), (79)

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 4 to 12

d_1 — Average diameter of hub, thin end [mm], Figures 4, 5, 11 and 12

d_2 — Average diameter of hub, thick end [mm], Figures 4, 5, 11 and 12

d_3, d_{3e} — Bolt circle diameter, real, effective [mm], Figures 4 to 12

d_4 — Outside diameter of flange [mm], Figures 4 to 12

d_5, d_{5t}, d_{5e} — Diameter of bolt hole, pierced, blind, effective [mm], Figures 4 to 12

d_6 — Inside diameter of loose flange [mm], Figures 10, 12

d_7 — Diameter of position of reaction between loose flange and stub or collar [mm], Figure 1, equations (15), (41)

d_8 — Outside diameter of collar [mm], Figure 10

d_9 — Diameter of a central hole in a blank flange [mm], Figure 9

d_{B0}, d_{Be}, d_{Bs} — Diameter of bolt: nominal diameter, effective diameter, shank diameter [mm], Figure 2, Table B.1

d_{B2}, d_{B3} — Basic pitch diameter, basic minor diameter of thread [mm], see Figure 2

d_{Ge}, d_{Gt} — Diameter of gasket, effective, theoretical [mm], Figure 3, Table 1

d_{G1}, d_{G2}	Inside, outside diameter of theoretical contact area of gasket [mm], Figure 3
d_E, d_F, d_L d_S, d_X	Average diameter of part or section designated by the subscript [mm], equations (5) to (8), (10) to (12), Figures 4 to 12
e_0	Wall thickness of central plate of blank flange within diameter d_0 [mm], Figure 9
e_1	Minimum wall thickness at thin end of hub [mm], Figures 4, 5, 11, 12
e_2	Wall thickness at thick end of hub [mm], Figures 4, 5, 11, 12
e_D, e_E	Wall thickness of equivalent cylinder for load limit calculations, for flexibility calculations [mm], equations (9), (11), (12), (75)
e_F, e_L	Effective axial thickness of flange, loose flange [mm], equations (5) to (8)
e_{Fb}	Thickness of flange ring at diameter d_3 (bolt position) [mm] equation (3)
e_{Ft}	Thickness of flange ring at diameter d_{Ge} (gasket force position), relevant for thermal expansion [mm], equation (45)
e_G	Thickness f_o gasket [mm], Figure 3
e_P, e_Q	Part of flange thickness with (e_P), without (e_Q) radial pressure loading [mm], Figures 4 to 12, such that e_P+e_Q = e_F
e_S	Thickness of connected shell [mm], Figures 4 to 8, 10 to 12
e_X	Flange thickness at weak section [mm], Figure 9
f_B, f_E, f_F, f_L, f_S	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
g_C	[A1⟩ Creep factor for gasket according to EN 1591-1:2001, which is replaced by creep factor P_{QR}. If the creep factor g_C is still used, alternative rules for calculation are specified in Annex H ⟨A1]
h_G, h_H, h_L	Lever arms [mm], Figure 1, equations (14), (16)
h_P, h_Q, h_R, h_S, h_T	Lever arm corrections [mm], equations (13), (21) to (24), (29), (30)
j_M, j_S	Sign number for moment, shear force (+1 or 1), equation (80)
k_Q, k_R, k_M, k_S	Correction factors, equation (25), (26), (81)
l_B, l_s	Bolt axial dimensions [mm], Figure 2, equation (34)
l_e	$l_e = l_B - l_S$
l_H	Length of hub [mm], Figures 4, 5, 11, 12, equation (9), (75)
n_B	Number of bolts, equations (1), (4), (33), (34)
p_B	Pitch between bolts [mm], equation (1)
p_t	Pitch of bolt thread [mm], Table B.1
r_0, r_1	Radii [mm], Figures 4, 10