SLOVENSKI STANDARD

SIST EN 1591-1:2002

01-maj-2002

Prirobnice in prirobniļ ni spoji - Pravila za naļ rtovanje okroglih prirobniļ nih spojev s tesnili - 1. del: Naļ in izraļ una

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 - Regles de calcul des assemblages a brides circulaires avec joint - Partie 1: Méthode de calcul

Ta slovenski standard je istoveten z: EN 1591-1:2001

ICS:

23.040.60

SIST EN 1591-1:2002 **en**

EUROPEAN STANDARD

NORME EUROPÉENNE

EUROPÄISCHE NORM

EN 1591-1

April 2001

ICS 23.040.60

English version

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

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

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This European Standard was approved by CEN on 8 March 2001.

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EUROPEAN COMMITTEE FOR STANDARDIZATION
COMITÉ EUROPÉEN DE NORMALISATION
EUROPÄISCHES KOMITEE FÜR NORMUNG

Management Centre: rue de Stassart, 36 B-1050 Brussels

Ref. No. EN 1591-1:2001 E

Contents

Foreword

This European Standard was prepared by the 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of an identical text or by endorsement, at the latest by October 2001, and conflicting national standards shall be withdrawn at the latest by October 2001.

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

– ENV 1591-2 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 ENV 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.

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 ENV 1591-2, 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 ENV 1591-2. 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.

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to implement this European Standard: Austria, Belgium, Czech Republic, Denmark, Finland, France, Germany, Greece, Iceland, Ireland, Italy, Luxembourg, Netherlands, Norway, Portugal, Spain, Sweden, Switzerland and the 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. ENV 1591-2 gives values for gasket properties which can be used in the Calculation method.

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

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

c) $\cos\varphi_s \geq 1/(1 + 0{,}01\ d_s/e_s)$

NOTE 1 For explanations of symbols see clause 3.

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

NOTE 3 The condition $e_F \geq p_B \times \sqrt[3]{(0{,}01...0{,}10)\ p_B/b_F}$ 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 4 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) 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 ...). 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.

e) The modulus of elasticity of the gasket may increase with the compressive stress Q on the gasket. The Calculation method uses a linear model: $E_G = E_0 + K_1 \times Q$. This is the unloading elasto-plastic secant modulus measured between 100 % and 33 % of the highest stress (Q) in assembly conditions.

f) Creep of the gasket under compression is approximated by a creep factor g_c (see ENV 1591-2).

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 Q_{max} 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

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

prEN 1092-1:1997	*Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories, PN designated - Part 1: Steel flanges*
EN 1092-2	*Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories, PN designated - Part 2: Cast iron flanges*
prEN 1092-3:1994	*Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories - Part 3: Copper alloy and composite flanges, PN designated*
prEN 1092-4:1995	*Flanges and their joints - Circular flanges for pipes, valves, fittings and accessories, PN designated - Part 4: Aluminium alloy flanges*
ENV 1591-2	*Flanges and their joints - Design rules for gasketed circular flange connections - Part 2: Gasket parameters*

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

av – average

c – calculated

d – design

e – effective

max – maximum

min – minimum

nom – nominal

opt – optimal

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)

E_0 – Compressive modulus of elasticity of the gasket [MPa] at zero compressive stress Q = 0 [MPa] (see ENV 1591-2)

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] (for E_G see ENV 1591-2)

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 $\left(= \frac{\pi}{12} \times \min\ (d_{Be};\ d_{Bs})^3 \right)$, equation (71)

K_1 – Rate of change of compressive modulus of elasticity of the gasket with compressive stress, ENV 1591-2

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.

Q	– Mean effective gasket compressive stress [MPa], $Q = F_G/A_{Ge}$
Q_I	– Mean effective required gasket compressive stress at load condition I [MPa]
Q_{min}	– Minimum necessary compressive stress in gasket for assembly condition (on the effective gasket area) [MPa], equation (49), (see ENV 1591-2)
Q_{max}	– Maximum allowable compressive stress in the gasket (depends on the gasket materials, construction, dimensions and the roughness of the flange facings) [MPa], equation (72), see ENV 1591-2 (including safety margins, which are same for all load conditions)
$Q_{max,Y}$	– Yield stress characteristic of the gasket materials and construction, see Table 1, and ENV 1591-2 [MPa]
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
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 fo 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 – Creep factor for gasket, equation (46), see ENV 1591-2

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

r_2 – Radius of curvature in gasket cross-section [mm], Figure 3

ΔU – Differential axial expansions [mm], equation (45)

Θ_F, Θ_L – Rotation of flange, loose flange, due to applied moment [rad], annex E

Ψ – Load ratio of flange ring due to radial force, equation (82)

Ψ_Z – Particular value of Ψ, equation (74), 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, equation (71), (72), (73), (85), (87), (89), (91)

Φ_{max} – Reduced maximum allowable load ratio, equation (70)

α_B, α_F, α_G, α_L – 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, [K^{-1}]

β, γ, δ, ϑ, κ, λ, χ – Intermediate variables, equations (9), (17), (18), (19), (41), (70), (75), (77)

ε_{1+}, ε_{1-} – Scatter of initial bolt load of a single bolt, above nominal value, below nominal value, annex C

ε_{+}, ε_{-} – Scatter for the global load of all the bolts above nominal value, below nominal value, equations (60), (61)

π – Numerical constant (π = 3,141593)

φ_G – Angle of inclination of a sealing face [rad or deg], Figure 3, Table 2

φ_S – Angle of inclination of connected shell wall [rad or deg], Figures 6, 7

3.4 Terminology

3.4.1 Flanges

Integral flange: Flange attached to the shell either by welding (e.g. neck weld, see Figures 4 to 7 or slip on weld see Figures 8 and 11) or cast onto the envelope (integrally cast flanges, type 21)

Blank flange: Flat closure, Figure 9

Loose flange: Separate flange ring abutting a collar, Figure 10

Hub: Axial extension of flange ring, usually connecting flange ring to shell, Figures 4, 5

Collar: Abuttment for a loose flange, Figure 10

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]

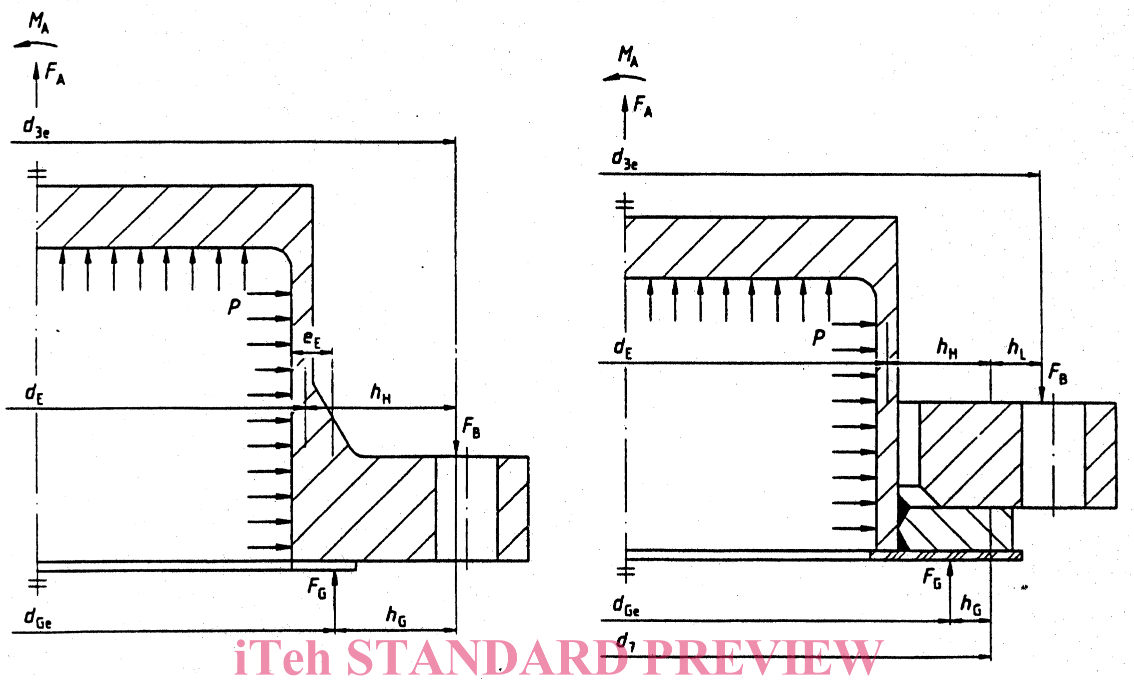
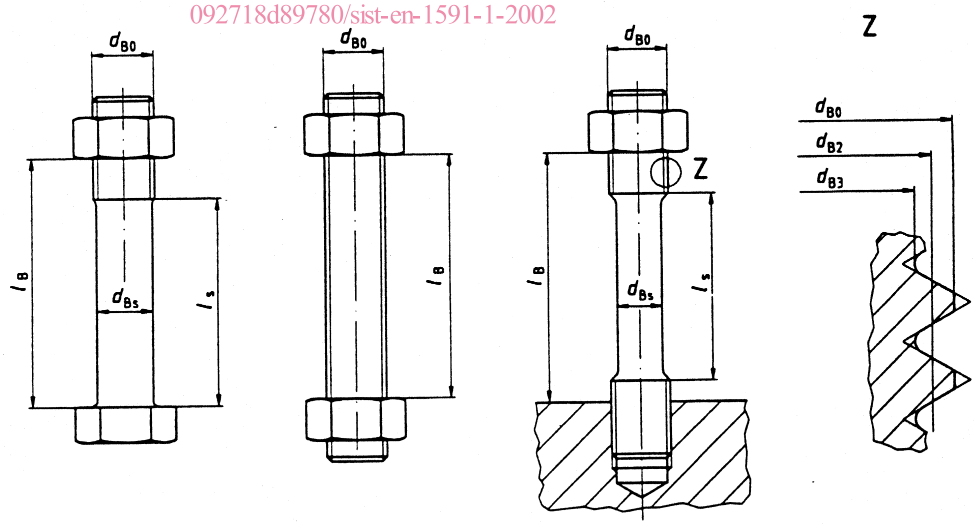


Figure 1 — Loads and lever arms



$l_e = l_B - l_s$

Figure 2 — Bolts