DRAFT FOR DEVELOPMENT

DD 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

ICS 23.040.60

British Standards

National foreword

This Draft for Development is the UK implementation of CEN/TS 1591-3:2007.

This publication is not to be regarded as a British Standard.

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Comments arising from the use of this Draft for Development are requested so that UK experience can be reported to the European organization responsible for its conversion to a European standard. A review of this publication will be initiated not later than 3 years after its publication by the European organization so that a decision can be taken on its status. Notification of the start of the review period will be made in an announcement in the appropriate issue of *Update Standards*.

According to the replies received by the end of the review period, the responsible BSI Committee will decide whether to support the conversion into a European Standard, to extend the life of the Technical Specification or to withdraw it. Comments should be sent to the Secretary of the responsible BSI Technical Committee at British Standards House, 389 Chiswick High Road, London W4 4AL.

The UK participation in its preparation was entrusted to Technical Committee PSE/15, Flanges.

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

This publication does not purport to include all the necessary provisions of a contract. Users are responsible for its correct application.

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Flanges and their joints - Design rules for gasketed circular flange connections - Part 3: Calculation method for metal to metal contact type flanged joint

Brides et leurs assemblages - Règles de calcul des assemblages à brides circulaires avec joint - Partie 3 : Méthode de calcul pour les assemblages à brides de type contact métal-métal

Flansche und ihre Verbindungen - Regeln für die Auslegung von Flanschverbindungen mit runden Flanschen und Dichtung - Teil 3: Berechnungsmethode für Flanschverbindungen mit Dichtungen im Kraft-Nebenschluss

This Technical Specification (CEN/TS) was approved by CEN on 16 June 2007 for provisional application.

The period of validity of this CEN/TS is limited initially to three years. After two years the members of CEN will be requested to submit their comments, particularly on the question whether the CEN/TS can be converted into a European Standard.

CEN members are required to announce the existence of this CEN/TS in the same way as for an EN and to make the CEN/TS available promptly at national level in an appropriate form. It is permissible to keep conflicting national standards in force (in parallel to the CEN/TS) until the final decision about the possible conversion of the CEN/TS into an EN is reached.

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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

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Foreword

This document (CEN/TS 1591-3:2007) has been prepared by Technical Committee CEN/TC "Flanges and their joints", the secretariat of which is held by DIN.

EN 1591 *"Flanges and their joints — Design rules for gasketed flange connections"* consists of the following three parts:

— *Part 1: Calculation method*

— *Part 2: Gasket parameters*

— *Part 3: Calculation method for metal to metal contact type flanged joint (CEN/TS)*

— *Part 4: Qualification of personnel competency in the assembly of bolted joints fitted to equipment subject to the Pressure Equipment Directive*

According to the CEN/CENELEC Internal Regulations, the national standards organizations of the following countries are bound to announce this Technical Specification: 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 the United Kingdom.

Introduction

Bolted Flange connections with metal to metal contact are frequently used in industrial plants for severe working conditions (thermal transients, pressure fluctuations). The use of metal to metal contact allows to avoid the damage of the sealing component by limiting the gasket loading stress and to limit the load variations on the gasket.

This Technical Specification describes a calculation method which enables to determine the internal forces of the BFC in all the load conditions. It ensures structural integrity and control of leak-tightness in BFC with MMC (BFC types which are outside the scope of EN 1591-1).

The calculation method may be divided into three steps:

Determination of the bolt tightening to reach the MMC.

Determination of the bolt tightening to maintain the MMC and to satisfy the leak-tightness criteria in all the load conditions.

Checking of the admissibility of the load ratio.

1 Scope

The aim of this Technical Specification is to describe a calculation method dedicated to Bolted Flange Connections (BFC) with metal to metal contact (MMC). It is dedicated to BFC where MMC occurs in a region between the outside diameter of the gasket and the inside diameter of the bolt hole region. For MMC inside the gasket and for MMC outside the bolt hole region, the present method is not appropriate.

The calculation method proposed in this Technical Specification is mainly based on the method described in EN 1591-1, dedicated to floating type BFC. The behaviour of the complete flanges-bolts-gasket system is considered. In assembly condition as well as for all the subsequent load conditions, the BFC components are maintained together by internal forces. This leads to deformations and forces balances (see Annex F) which gives the basic relations between the forces variations in the BFC.

The calculation of BFC with MMC leads to the consideration of an additional force compared to the EN 1591-1 calculation method: the reaction force in the MMC area. It explains why two compliance equations are required in this Technical Specification (in the EN 1591-1 calculation method just one compliance equation is needed to determine the internal forces in all the load conditions).

Unlike EN 1591-1 where the internal forces variations are determined with the compliance relation between the assembly and the considered load condition, here, the internal forces variations are determined by using the compliance relations between two consecutive load conditions.

This method does not treat non-gasketed pipe joints.

1.1 Requirement for use of the Method

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

— special testing;

— proven practice.

— Use of standard flanges within permitted conditions

1.2 Geometry

The Method is applicable to the configurations having:

— flanges whose section is given or may be assimilated to those given in Figures 4 to 12 of EN 1591-1:2001;

— four or more identical bolts uniformly distributed;

— gasket designed for MMC application;

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

c) $\cos\varphi_S \geq 1/(1 + 0{,}01 \cdot d_S / e_S)$

NOTE 1 For explanations of symbols see Clause 3.

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

NOTE 3 The condition $e_F \geq p_B \cdot \sqrt[3]{(0{,}01...0{,}10)\, p_B / b_F}$ is for limitation of non-uniformity of gasket pressure due to spacing of bolts. The values 0,01 and 0,10 should be applied for soft (non-metallic) and hard (metallic) gaskets respectively. A more precise criterion is given in Annex A of EN 1591-1:2001.

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 Method:

— flanges of essentially non-axisymmetric geometry, e.g. split loose flanges, web reinforced flanges.

1.3 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 should 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 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.

1.4 Loads

This 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.

1.5 Mechanical model

The 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 1.2 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.

   This presupposition requires compliance with 1.2 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. 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 unloading modulus of elasticity of the gasket may increase with the gasket surface pressure. The Method uses a linear model: $E_G = E_0 + K_1 \cdot Q$. This is the unloading elasto-plastic secant modulus measured between 100 % and 33 % of the highest surface pressure reached on the gasket.

f) Relaxation of the gasket under compression is approximated (see 4.9 and Annex C).

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 (75).

i) Load changes between load conditions cause internal changes of bolt, gasket and MMC forces. These are calculated with account taken of elastic deformations of all components.

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 rough 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 tapped hole (see Equation (37)).

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 torsion moments and external shear loads, e.g. those due to pipework.

2 Normative references

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.

EN 1591-1:2001, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*

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

3 Notation

3.1 Use of figures

Figure 1 illustrates the two configurations of metal to metal contact.

Figure 2 shows the variables used in the calculation of the inside diameter of the MMC area.

3.2 Subscripts and special marks

3.2.1 Subscripts

A	Additional (F_A, M_A)
B	Bolt
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 (taken values 0, 1, 2 ...)
L	Loose flange
M	Metallic ring or metal to metal contact
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
W	Washer
Δ	Symbol for change or difference
av	average
c	calculated
e	effective
j	identifier of reference dots (Q_{Gj}, e_{Gj}) used to described the gasket behaviour in compression
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 connection, possibly different from the first

^{c} exponent marking deformation terms due to creep relaxation

^{i} Exponent marking terms related to the first compliance equation

^{ii} Exponent marking terms related to the second compliance equation

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 (36)

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 (41), (39)

C Coefficient to account for twisting moment in bolt load ratio, Equation (121)

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

E_B, E_F, E_G, E_L

E_M, E_W Modulus of elasticity of the part designated by the subscript, at the temperature of the part [MPa] (for E_G see prEN 1591-2)

F_A Additional external axial force [N], tensile force > 0, compressive force < 0, see Figure 1 of EN 1591-1:2001

F_B Bolt force (sum of all bolts) [N]

F_{BMMC} Bolt force (sum of bolts) required to reach MMC [N]

F_G Gasket force [N]

F_{GMMC} Gasket force required to reach the MMC [N]

F_M Metal to metal contact force [N]

F_Q Axial fluid-pressure force [N], Equation (74)

F_R Force resulting from F_A and M_A [N], Equation (75)

F_{RMMC} Force resulting from F_A and M_A corresponding to the tightening force F_{BMMC} [N]

$G(t)$ Relaxation function Equation (C.13)

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, Equation (121)
K_1	Rate of change of compressive modulus of elasticity of the gasket with compressive stress, prEN 1591-2
K_s	Systematic error due to the inaccuracy of the bolt tightening method
M_A	Additional external moment [N · mm], Figure 1 of EN 1591-1:2001
M_t	Bolt assembly torque [N · mm], Annex D of EN 1591-1:2001
$M_{t, B}$	twisting moment [N · mm] applied to bolt shanks as a result of application of the bolt assembly torque M_t, Equations (121) and (D.8) to (D.11) of EN 1591-1:2001
P	Pressure of the fluid [MPa], internal pressure > 0, external pressure < 0 (1 bar = 0,1 MPa)
Q	Mean effective gasket compressive stress [MPa], $Q = F_G/A_{Ge}$
$Q_{GMMCinf}$	Inferior boundary of the range of gasket compressive stress in which the MMC appears [MPa]
$Q_{GMMCsup}$	Superior boundary of the range of gasket compressive stress in which the MMC appears [MPa]
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 (93), (see prEN 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 (120), see prEN 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 prEN 1591-2 [MPa]
T_B, T_F, T_G, T_L,	
T_M, T_W	Temperature (average) of the part designated by the subscript [°C] or [K], Equation (77), (78) and (80), (81)
T_O	Temperature of connection at assembly [°C] or [K] (usually + 20 °C)
U	Axial displacement [mm]; ΔU according to Equations (76), (77), (78) and Equations (79), (80) and (81).
W_F, W_L, W_X	Resistance of the part and/or cross-section designated by the subscript [N · mm], Equations (123), (135), (137), (139)
X_B, X_G, X_M, X_W	Axial flexibility modulus of bolts, gasket, metallic compression limiter ring, washer [1/mm], Equations (37), (44), (52), (46)
X_{FB}	Axial flexibility modulus corresponding to local compression of flange at contact area with nut [1/mm], Equation (73)

X_{FG}	Axial flexibility modulus corresponding to local compression of flange at contact area with gasket [1/mm], Equation (55)
X_{FL}	Axial flexibility modulus corresponding to local compression of collar at contact area with loose flange [1/mm], Equation (63)
X_{FM}	Axial flexibility modulus corresponding to local compression of flange at metal to metal contact area [1/mm], Equation (59)
X_{LB}	Axial flexibility modulus corresponding to local compression of loose flange at contact area with nut [1/mm], Equation (71)
X_{LF}	Axial flexibility modulus corresponding to local compression of loose flange at contact area with collar [1/mm], Equation (67)
Y_G, Y_M, Y_Q, Y_R	Axial compliance of the bolted connection, related to F_G, F_M, F_Q, F_R [mm/N], Equations (83) to (86) and (89) to (92)
Z_F, Z_L	Rotational flexibility modulus of flange, loose flange [mm^{-3}], Equations (30), (34), (35)
$a(T_I,T_0)$	Shift function Equation (C.17)
b_0	Width of chamfer (or radius) of a loose flange [mm] see Figure 10 of EN 1591-1:2001, Equation (17) such that: $d_{7min} = d_6 + 2 \cdot b_0$
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 (38), (40), Table 1
b_{Mt}	Metal to metal contact area width [mm], Equation (48) and Figure 1
c_F, c_M, c_S	Correction factors, Equations (23), (127), (128)
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 of EN 1591-1:2001
d_1	Average diameter of hub, thin end [mm], Figures 4, 5, 11 and 12 of EN 1591-1:2001
d_2	Average diameter of hub, thick end [mm], Figures 4, 5, 11 and 12 of EN 1591-1:2001
d_3, d_{3e}	Bolt circle diameter, real, effective [mm], Figures 4 to 12 of EN 1591-1:2001
d_4	Outside diameter of flange [mm], Figures 4 to 12 of EN 1591-1:2001
d_5, d_{5t}, d_{5e}	Diameter of bolt hole, pierced, blind, effective [mm], Figures 4 to 12 of EN 1591-1:2001
d_6	Inside diameter of loose flange [mm], Figures 10, 12 of EN 1591-1:2001
d_7	Diameter of position of reaction between loose flange and stub or collar [mm], Figure 1 of EN 1591-1:2001, Equations (17), (43)
d_8	Outside diameter of collar [mm], Figure 10 of EN 1591-1:2001
d_9	Diameter of a central hole in a blank flange [mm], Figure 9 of EN 1591-1:2001
d_{B0}, d_{Be}, d_{Bs}	Diameter of bolt: nominal diameter, effective diameter, shank diameter [mm], Figure 2 of EN 1591-1:2001, Table B.1 of EN 1591-1:2001

d_{B2}, d_{B3}	Basic pitch diameter, basic minor diameter of thread [mm], see Figure 2 of EN 1591-1:2001
d_{Ge}, d_{Gt}	Diameter of gasket, effective, theoretical [mm], Figure 3 of EN 1591-1:2001, Table 1
d_{G1}, d_{G2}	Inside, outside diameter of theoretical contact area of gasket [mm], Figure 3 of EN 1591-1:2001
d_{M1}, d_{M2}	Inside, outside diameter of theoretical metal to metal contact area [mm]
d_{M1e}	Effective inside diameter of metal to metal contact area [mm], Equation (49), (50)
d_{Me}, d_{Mt}	Diameter of metal to metal contact area, effective, theoretical [mm], Equation (51), (47)
d_E, d_F, d_L	Average diameter of part or section designated by the subscript [mm], Equations (5) to (8), (10)
d_S, d_X	to (12), Figures 4 to 12 of EN 1591-1:2001
d_{W1}, d_{W2}	Inside, outside diameter of washers Equation (46) [mm]
e_0	Wall thickness of central plate of blank flange within diameter d_0 [mm], Figure 9 of EN 1591-1:2001
e_1	Minimum wall thickness at thin end of hub [mm], Figures 4, 5,11, 12 of EN 1591-1:2001
e_2	Wall thickness at thick end of hub [mm], Figures 4, 5, 11, 12 of EN 1591-1:2001
e_D, e_E	Wall thickness of equivalent cylinder for load limit calculations, for flexibility calculations [mm], Equations (9), (11), (12), (124)
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 (77), (78) and (80), (81)
e_{Fm}	Thickness of flange ring at diameter d_{Me} (metal to metal contact force position), relevant for thermal expansion and inside diameter of the metal to metal contact area [mm], Equation (77), (78) and (80), (81)
e_G	Thickness of gasket [mm], Figure 3 of EN 1591-1:2001
e_M	Thickness of metallic compression limiter ring [mm]
e_P, e_Q	Part of flange thickness with (e_P), without (e_Q) radial pressure loading [mm], Figures 4 to 12 of EN 1591-1:2001, such that $e_P + e_Q = e_F$
e_S	Thickness of connected shell [mm], Figures 4 to 8, 10 to 12 of EN 1591-1:2001
e_X	Flange thickness at weak section [mm], Figure 9 of EN 1591-1:2001
e_W	Washer thickness [mm]
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

h_G, h_H, h_L, h_M	Lever arms [mm], Equations (15), (16), (18), (19)
h_D	Difference of lever arms h_G and h_M [mm], Figure 1, Equation (14)
h_P, h_Q, h_R,	
h_S, h_T	Lever arm corrections [mm], Equations (13), (24) to (27), (32), (33)
j_M, j_S	Sign number for moment, shear force (+ 1 or – 1), Equation (129)
k_Q, k_R, k_M, k_S	Correction factors, Equation (28), (29), (130)
l_{5t}	Depth of the blind holes, Figure 5 of EN 1591-1:2001, Equation (3)
l_B, l_s	Bolt axial dimensions [mm], Figure 2 of EN 1591-1:2001, Equation (37)
l_e	$l_e = l_B - l_s$
l_H	Length of hub [mm], Figures 4, 5, 11, 12 of EN 1591-1:2001, Equation (9), (124)
n_B	Number of bolts, Equations (1), (4), (36), (37), (46)
p_B	Pitch between bolts [mm], Equation (1)
p_t	Pitch of bolt thread [mm], Table B.1 of EN 1591-1:2001
r_0, r_1	Radii [mm], Figures 4, 10 of EN 1591-1:2001
r_2	Radius of curvature in gasket cross-section [mm], Figure 3 of EN 1591-1:2001
z_{1B}, z_{1F}, z_{1L}	thickness concerned by the local compression at the inner diameter of the component designated by the subscript [mm], Equation (55) to (73)
z_{2B}, z_{2F}, z_{2L}	thickness concerned by the local compression at the outer diameter of the component designated by the subscript [mm], Equation (55) to (73)
ΔU	Differential axial expansions [mm], Equation (76) to (81)
Θ_F, Θ_L	Rotation of flange, loose flange, due to applied moment [rad], Equation (97) to (100), (104), (105)
Ψ	Load ratio of flange ring due to radial force, Equation (131)
Ψ_Z	Particular value of Ψ, Equation (123), 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 (120), (121), (122), (134), (136), (138)
Φ_{max}	Reduced maximum allowable load ratio, Equation (108)
α_B, α_F, α_G, α_L, α_W, α_M	Thermal expansion coefficient of the part designated by the subscript, averaged between T_0 and T_B, T_F, T_G, T_L, T_W, T_M [K^{-1}], Equation (77), (78) and (80), (81)
β, γ, δ, θ	Intermediate variables, Equations (9), (20), (21), (22), (43), (108), (124)
κ, λ, χ	

ε_0 — Strain (Annex C).

ε_{1+}, ε_{1-} — Scatter of initial bolt load of a single bolt, above nominal value, below nominal value, Annex D

ε_+, ε_- — Scatter for the global load of all the bolts above nominal value, below nominal value, Equations (109), to (111)

η — Tangent of the propagation angle of the local compression (see Annex B)

π — Numerical constant (π = 3,141593)

σ — Stress [MPa] (Annex C)

τ_R — Relaxation time [s] (Annex C)

ϕ_G — Angle of inclination of a sealing face [rad or deg], Figure 3 of EN 1591-1:2001, Table 2

ϕ_S — Angle of inclination of connected shell wall [rad or deg], Figures 6, 7 of EN 1591-1:2001

ξ_0, ξ_1, ω_0, ω_1 — Material parameters regarding the stress relaxation function of the gasket (see Annex C).

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 of EN 1591-1:2001 or slip on weld see Figures 8 and 11 of EN 1591-1:2001) or cast onto the envelope (integrally cast flanges, Type 21)

Blank flange: Flat closure, Figure 9 of EN 1591-1:2001

Loose flange: Separate flange ring abutting a collar, Figure 10 of EN 1591-1:2001

Hub: Axial extension of flange ring, usually connecting flange ring to shell, Figures 4, 5 of EN 1591-1:2001

Collar: Abutment for a loose flange, Figure 10 of EN 1591-1:2001

3.4.2 Loading

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

3.4.3 Loading 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. operating condition, test 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]

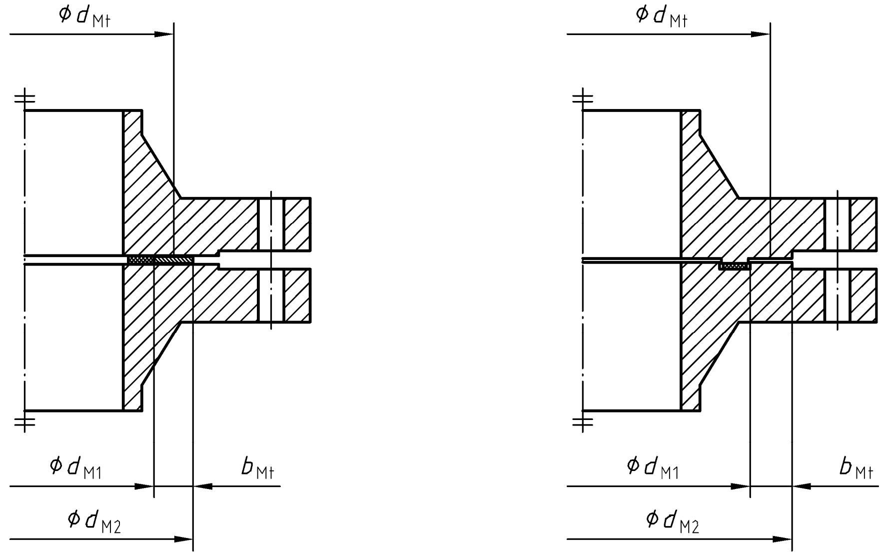


Figure 1 — 2 types of MMC BFC: gasket with limiter ring and gasket inserted in a groove, with theoretical dimensions

4 Calculation parameters

4.1 General

The parameters defined in this Clause are effective dimensions and areas as well as stiffness parameters. Most Parameters of 4.2 to 4.4 are extracted from EN 1591-1:2001.

4.2 Flange parameters

4.2.1 General

The formulae given in 4.2 shall be used for each of the two flanges and where applicable, the two collars of a connection.

Specific flange types are treated as follows:

Integral flange: calculated as an equivalent ring with rectangular cross-section, dimensions $b_F \cdot e_F$ 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 $b_F \cdot e_F$, 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 $b_L \cdot e_L$ 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 Figures 4 to 12 of EN 1591-1:2001, the equivalent ring is sketched by shaded area.

4.2.2 Flange ring

4.2.2.1 Bolt holes

Pitch between bolts:

$$p_{\mathrm{B}} = \pi \cdot d_3 / n_{\mathrm{B}} \tag{1}$$

Effective diameter of bolt hole:

$$d_{5\mathrm{e}} = d_5 \cdot \sqrt{d_5 / p_{\mathrm{B}}} \tag{2}$$

Diameter of blind holes is assumed to be:

$$d_5 = d_{5\mathrm{t}} \cdot l_{5\mathrm{t}} / e_{\mathrm{Fb}} \tag{3}$$

Effective bolt circle diameter:

$$d_{3\mathrm{e}} = d_3 \cdot \left(1 - 2 / n_{\mathrm{B}}{}^2\right) \tag{4}$$

NOTE 1 p_{B} and $\widetilde{p}_{\mathrm{B}}$ are equal as well as $d_{3\mathrm{e}}$ and $\widetilde{d}_{3\mathrm{e}}$.

NOTE 2 Equations (1) to (4) do not apply to collars.

4.2.2.2 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 flanges 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 Figures 4 to 9 of EN 1591-1:2001)

$$\begin{aligned} &b_{\mathrm{F}} = (d_4 - d_0)/2 - d_{5\mathrm{e}}; \qquad d_{\mathrm{F}} = (d_4 + d_0)/2 \\ &e_{\mathrm{F}} = 2 \cdot A_{\mathrm{F}} / (d_4 - d_0) \end{aligned} \tag{5}$$

$$b_{\mathrm{L}} = d_{\mathrm{L}} = e_{\mathrm{L}} = 0 \tag{6}$$

Loose flange with collar (see Figure 10 of EN 1591-1:2001)

For collar:

$$\begin{aligned} &b_{\mathrm{F}} = (d_8 - d_0)/2; \qquad d_{\mathrm{F}} = (d_8 + d_0)/2 \\ &e_{\mathrm{F}} = 2 \cdot A_{\mathrm{F}} / (d_8 - d_0) \end{aligned} \tag{7}$$

For flange:

$$b_{\mathrm{L}} = (d_4 - d_6)/2 - d_{5e}; \qquad d_{\mathrm{L}} = (d_4 + d_6)/2$$
$$e_{\mathrm{L}} = 2 \cdot A_{\mathrm{L}} / (d_4 - d_6) \tag{8}$$

4.2.3 Connected shell

4.2.3.1 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}:

$$e_{\mathrm{E}} = e_1 \cdot \left\{ 1 + \frac{(\beta - 1) \cdot l_{\mathrm{H}}}{(\beta/3) \cdot \sqrt{d_1 \cdot e_1} + l_{\mathrm{H}}} \right\} \quad \beta = \frac{e_2}{e_1} \tag{9}$$

$$d_{\mathrm{E}} = \{\min(d_1 - e_1 + e_{\mathrm{E}}; d_2 + e_2 - e_{\mathrm{E}}) + \max(d_1 + e_1 - e_{\mathrm{E}}; d_2 - e_2 + e_{\mathrm{E}})\}/2 \tag{10}$$

4.2.3.2 Flange without hub

For a shell (cylindrical or conical or spherical, constant wall thickness e_{s}, angle ϕ_{S} and diameter d_{S} at junction with flange) directly connected to a flange ring, the effective dimensions are:

$$e_{\mathrm{E}} = e_{\mathrm{S}}; \qquad d_{\mathrm{E}} = d_{\mathrm{S}} \tag{11}$$

The Equations (11) are not applicable when a nozzle is connected to the central opening of a blank flange. This case is covered by 4.2.3.3.

4.2.3.3 Blank flange

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

$$e_{\mathrm{E}} = 0; \qquad d_{\mathrm{E}} = d_0 \tag{12}$$

The Equations (12) apply whatever the blank flange configuration (without opening, with opening without nozzle, with opening with nozzle).

4.2.3.4 Collar

The equations which are applicable are those of 4.2.3.1 or 4.2.3.2 depending on whether or not the collar has a hub.

4.2.4 Lever arms

NOTE When the gasket is of flat type, the parameters h_{P} and h_{G} below can be calculated only when d_{Ge} has been determined, i.e. when the calculations given in 4.4.3 have been carried out. h_{M} below can be calculated only when d_{Me} has been determined at each load condition.

4.2.4.1 All flanges

$$h_{\mathrm{P}} = \left[(d_{\mathrm{Ge}} - d_{\mathrm{E}})^2 \cdot (2 \cdot d_{\mathrm{Ge}} + d_{\mathrm{E}})/6 + 2 \cdot e_{\mathrm{p}}^2 \cdot d_{\mathrm{F}}\right] / d_{\mathrm{Ge}}^2 \tag{13}$$

For blank flanges: $e_{\mathrm{p}} = 0$.

For a practical point of view the following difference of lever arms is defined:

$$h_D = h_M - h_G = \left(d_{Me} - d_{Ge}\right)/2 \tag{14}$$

4.2.4.2 Integral flange and blank flange

$$h_G = \left(d_{3e} - d_{Ge}\right)/2; \qquad h_H = \left(d_{3e} + d_E\right)/2$$
$$h_L = 0 \tag{15}$$

$$h_M = \left(d_{3e} - d_{Me}\right)/2 \tag{16}$$

NOTE These equations do not apply to collars.

4.2.4.3 Loose flange with collar

$$d_{7\,\mathrm{min}} \le d_7 \le d_{7\,\mathrm{max}}$$
$$d_{7\,\mathrm{min}} = d_6 + 2 \cdot b_0; \qquad d_{7\,\mathrm{max}} = d_8 \tag{17}$$

$$h_G = \left(d_7 - d_{Ge}\right)/2; \qquad h_H = \left(d_7 + d_E\right)/2$$
$$h_L = \left(d_{3e} - d_7\right)/2 \tag{18}$$

$$h_M = \left(d_7 - d_{Me}\right)/2 \tag{19}$$

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

— for the flexibility calculations, take for d_7 the value d_{70} given by Equation (43);

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 4.3.2).

— for the calculation of load ratios (Clause 6), the most favourable value between $d_{7\,\mathrm{min}}$ and $d_{7\,\mathrm{max}}$ can be used.

4.2.5 Flexibility-related flange parameters

NOTE When the gasket is of the flat type, the parameter h_Q below can be calculated only when d_{Ge} has been determined, i.e. when the calculations stated in 4.3.2 have been carried out.

4.2.5.1 Integral flange and collar

$$\gamma = e_E \cdot d_F / \left(b_F \cdot d_E \cdot \cos \varphi_S\right) \tag{20}$$

$$\theta = 0{,}55 \cos \varphi_S \cdot \sqrt{d_E \cdot e_E} / e_F \tag{21}$$

$$\lambda = 1 - e_P / e_F = e_Q / e_F \tag{22}$$

NOTE e_P and e_Q are defined in Figures 4 to 12 of EN 1591-1:2001 (when $e_P = e_F$, $e_Q = 0$).

$$c_F = \left(1 + \gamma \cdot \theta\right) / \left\{1 + \gamma \cdot \theta \cdot \left[4 \cdot \left(1 - 3\lambda + 3\lambda^2\right) + 6 \cdot \left(1 - 2\lambda\right) \cdot \theta + 6 \cdot \theta^2\right] + 3\gamma^2 \cdot \theta^4\right\} \tag{23}$$

$$h_S = 1{,}1_{eF} \cdot \sqrt{e_E / d_E} \cdot \left(1 - 2 \cdot \lambda + \theta\right) / \left(1 + \gamma \cdot \theta\right) \tag{24}$$

$$h_T = e_F \left(1 - 2\lambda - \gamma \cdot \theta^2\right) / \left(1 + \gamma \cdot \theta\right) \tag{25}$$

$$h_Q = \left\{ h_S \cdot k_Q + h_T \cdot \left(2d_F \cdot e_P / d_E{}^2 - 0{,}5 \cdot \tan \varphi_S \right) \right\} \cdot \left(d_E / d_{Ge} \right)^2 \tag{26}$$

$$h_R = h_S \cdot k_R - h_T \cdot 0{,}5 \tan \varphi_S \tag{27}$$

$$k_Q = \begin{Bmatrix} +0{,}85 / \cos \varphi_S & \text{for conical or cylindrical shell} \\ +0{,}35 / \cos \varphi_S & \text{for spherical shell} \end{Bmatrix} \tag{28}$$

$$k_R = \begin{Bmatrix} -0{,}15 / \cos \varphi_S & \text{for conical or cylindrical shell} \\ -0{,}65 / \cos \varphi_S & \text{for spherical shell} \end{Bmatrix} \tag{29}$$

$$\left. \begin{aligned} Z_F &= 3d_F \cdot c_F / \left(\pi \cdot b_F \cdot e_F{}^3 \right) \\ Z_L &= 0 \end{aligned} \right\} \tag{30}$$

4.2.5.2 Blank flange

Diameter ratio:

$$\rho = d_9 / d_E \tag{31}$$

NOTE reminder: for a blank flange, $d_E = d_0$ (according to Equation (12))

$$h_Q = \left(d_E / 8 \right) \cdot \left(1 - \rho^2 \right) \cdot \left(0{,}7 + 3{,}3\, \rho^2 \right) / \left(0{,}7 + 1{,}3\, \rho^2 \right) \cdot \left(d_E / d_{Ge} \right)^2 \tag{32}$$

$$h_R = \left(d_E / 4 \right) \cdot \left(1 - \rho^2 \right) \cdot \left(0{,}7 + 3{,}3\, \rho^2 \right) / \left[\left(0{,}7 + 1{,}3\, \rho^2 \right) \cdot \left(1 + \rho^2 \right) \right] \tag{33}$$

$$\left. \begin{aligned} Z_F &= 3d_F / \left\{ \pi \cdot \left[b_F \cdot e_F{}^3 + d_F \cdot e_0{}^3 \cdot \left(1 - \rho^2 \right) / \left(1{,}4 + 2{,}6 \rho^2 \right) \right] \right\} \\ Z_L &= 0 \end{aligned} \right\} \tag{34}$$

4.2.5.3 Loose Flange with collar

For the collar use Equations (20) to (30); for the loose flange use the following equation:

$$Z_L = 3 \cdot d_L / \left(\pi \cdot b_L \cdot e_L{}^3 \right) \tag{35}$$

4.3 Bolt parameters

4.3.1 General

The bolt dimensions are shown in Figure 2 of EN 1591-1:2001. Diameters of standard metric series bolts are given in Annex B of EN 1591-1:2001.

4.3.2 Effective cross-section area of bolts

$$A_B = \left\{ \min \left(d_{Be} ; d_{BS} \right) \right\}^2 \cdot n_B \cdot \pi / 4 \tag{36}$$

4.3.3 Flexibility modulus of bolts

$$X_B = \left(l_S / d_{BS}{}^2 + l_e / d_{Be}{}^2 + 0{,}8 / d_{B0} \right) \cdot 4 / \left(n_B \cdot \pi \right) \tag{37}$$

4.4 Gasket parameters

4.4.1 General

The notation for dimensions of gaskets is given in Figure 3 of EN 1591-1:2001.

prEN 1591-2 gives typical non-mandatory values for material properties. If data for the actual gasket is available, it should preferably be used.

4.4.2 Theoretical dimensions

$$b_{Gt} = (d_{G2} - d_{G1})/2; \qquad d_{Gt} = (d_{G2} + d_{G1})/2 \tag{38}$$

$$A_{Gt} = \pi \cdot d_{Gt} \cdot b_{Gt} \tag{39}$$

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

4.4.3 Effective dimensions

The effective gasket width b_{Ge} depends on the force F_G applied to the gasket for many types of gasket.

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 first calculation is performed with F_G value as described in 5.5.2.

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

Effective gasket width:

$$b_{Ge} = \min\{b_{Gi}; b_{Gt}\} \tag{40}$$

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:

$$A_{Ge} = \pi \cdot d_{Ge} \cdot b_{Ge} \tag{41}$$

Lever arm:

$$h_{G0} = \begin{cases} (d_{3e} - d_{Ge})/2 & \text{for integral flange or blank flange} \\ (d_{70} - d_{Ge})/2 & \text{for loose flange with collar} \end{cases} \tag{42}$$

$$\left.\begin{aligned} d_{70} &= \min\{\max(d_{7\,\min}; (d_{Ge} + \kappa \cdot d_{3e})/(1+\kappa)); d_{7\,\max}\} \\ \kappa &= (Z_L \cdot E_{F0})/(Z_F \cdot E_{L0}) \end{aligned}\right\} \tag{43}$$

NOTE 3 Equation (43) does only apply to loose flanges on a collar.

Equations (40) to (43) are re-evaluated iteratively until the value b_{Ge} is constant within the required precision.

NOTE 4 A precision of 5 % is enough. To obtain results almost independent of the operator, a precision of 0,1 % is however recommended.

4.4.4 Axial flexibility modulus of gasket

$$X_G = (e_G / A_{Gt}) \cdot (b_{Gt} + e_G / 2) / (b_{Ge} + e_G / 2) \tag{44}$$

Table 1 — Effective gasket geometry

Type	Gasket form	Formulae
1	Flat gaskets, of low hardness, composite or pure metallic, materials, see Figure 3a of EN 1591	First approximation: $b_{Gi} = b_{Gt}$ More accurate: $b_{Gi} = \sqrt{\dfrac{e_G / (\pi \cdot d_{Ge} \cdot E_{Gm})}{h_G \cdot Z_F / E_F + \tilde{h}_G \cdot \tilde{Z}_F / \tilde{E}_F} + \left[\dfrac{F_G}{\pi \cdot d_{Ge} \cdot Q_{max,y}}\right]^2}$ $E_{Gm} = E_0 + 0{,}5\, K_1 \cdot F_G / A_{Ge}$ $Z_F, \tilde{Z}_F$ according to Equation (30) or (34) In all cases: $d_{Ge} = d_{G2} - b_{Ge}$
2	Metal gaskets with curved surfaces, simple contact, see Figures 3b, 3c of EN 1591	First approximation: $b_{Gi} = \sqrt{6_{r2} \cdot \cos \varphi_G \cdot b_{Gt} \cdot Q_{max,y} / E_G}$ More accurate: $b_{Gi} = \sqrt{\dfrac{6_{r2} \cdot \cos \varphi_G \cdot F_G}{\pi \cdot d_{Ge} \cdot E_G} + \left[\dfrac{F_G}{\pi \cdot d_{Ge} \cdot Q_{max,y}}\right]^2}$ In all cases: $d_{Ge} = d_{G0}$
3	Metal octagonal section gaskets see Figure 3d of EN 1591	In all cases: b_{Gi} = length b_{Ge} according to Figure 3d of EN 1591 (Projection of contacting surfaces in axial direction.) $d_{Ge} = d_{Gt}$
4	Metal oval or circular section gaskets, double contact see Figures 3e, 3f of EN 1591	First approximation: $b_{Gi} = \sqrt{12_{r2} \cdot \cos \varphi_G \cdot b_{Gt} \cdot Q_{max,y} / E_G}$ More accurate: $b_{Gi} = \sqrt{\dfrac{12_{r2} \cdot \cos \varphi_G \cdot F_G}{\pi \cdot d_{Ge} \cdot E_G} + \left[\dfrac{F_G}{\pi \cdot d_{Ge} \cdot Q_{max,y}}\right]^2}$ In all cases: $d_{Ge} = d_{Gt}$

4.5 Washer parameters

It is common use to add washers between the nuts and the flanges in BFC. Here, we consider not only the axial thermal expansion of the washers (see 5.2.2.3 and 5.2.2.4) but also the axial flexibility of these components.

Washers are submitted to compression. The axial deformation of the washers due to compression is considered in the deformation compatibility equations (see Annex F) with the term $\Delta e_{\mathrm{W}}{}^{\mathrm{M}}$.

$$\Delta e_{\mathrm{W}}{}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{W}}}{E_{\mathrm{W}}} \cdot F_{\mathrm{B}}\right) \tag{45}$$

Where X_{W} represents the axial flexibility modulus of the washers.

For flat washers the axial flexibility modulus is:

$$X_{\mathrm{W}} = \frac{4 \cdot e_{\mathrm{W}}}{n_{\mathrm{B}} \cdot \pi \cdot (d_{\mathrm{W2}}{}^{2} - d_{\mathrm{W1}}{}^{2})} \tag{46}$$

For other kind of washers the axial flexibility modulus has to be determined following the manufacturer indications.

4.6 Calculation parameters for the metal to metal contact area

4.6.1 Metal to metal contact theoretical dimensions

Theoretical diameter of the metal to metal contact : d_{Mt}

$$d_{\mathrm{Mt}} = \frac{d_{\mathrm{M1}} + d_{\mathrm{M2}}}{2} \tag{47}$$

Theoretical width of the metal to metal contact : b_{Mt}

$$b_{\mathrm{Mt}} = \frac{d_{\mathrm{M2}} + d_{\mathrm{M1}}}{2} \tag{48}$$

(see Figure 1)

4.6.2 Metal to metal contact effective dimensions

4.6.2.1 Inside diameter of the metal to metal contact area

The inside diameter of the metal to metal contact area depends on the flanges geometries and on the rotation angles of the flanges:

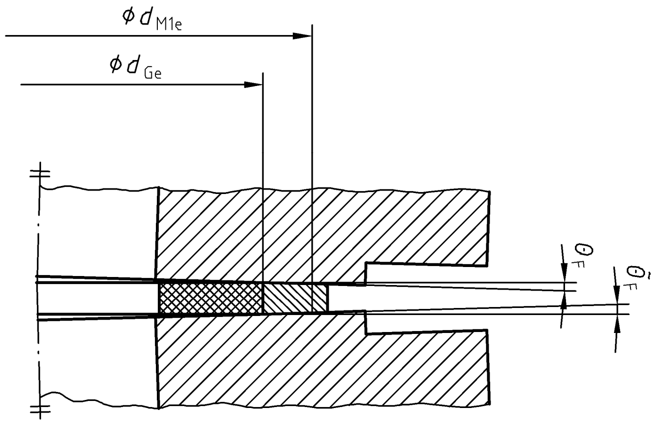
In the case of MMC with compression limiter ring:

$$d_{\mathrm{M1e}} = d_{\mathrm{Ge}} + 2 \times \frac{e_{\mathrm{G}} - \Delta e_{\mathrm{FG}} - \Delta \widetilde{e}_{\mathrm{FG}} - e_{\mathrm{M}}}{\Theta_{\mathrm{F}} + \widetilde{\Theta}_{\mathrm{F}}} \tag{49}$$

For MMC occurring with gasket inserted in a groove:

$$d_{\mathrm{M1e}} = d_{\mathrm{Ge}} + 2 \times \frac{\left(e_{\mathrm{G}} + e_{\mathrm{Ft}} + \tilde{e}_{\mathrm{Ft}}\right) - \left(\Delta e_{\mathrm{FG}} + \Delta\tilde{e}_{\mathrm{FG}} + e_{Fm} + \tilde{e}_{\mathrm{Fm}}\right)}{\Theta_{\mathrm{F}} + \tilde{\Theta}_{\mathrm{F}}} \tag{50}$$

e_{G} is the thickness of the gasket under compression.



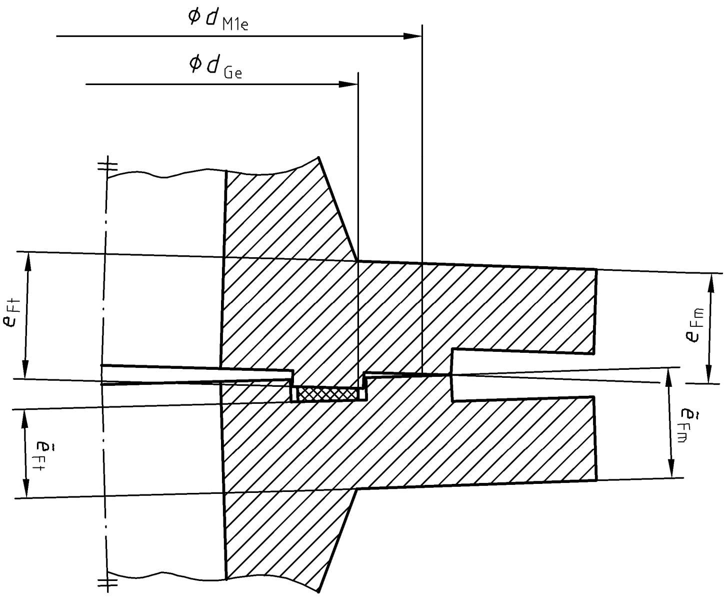


Figure 2 — Inside diameter of the MMC area

4.6.2.2 Effective diameter of the metal to metal contact area

The effective metal to metal contact diameter d_{Me} is the diameter where the metal to metal contact force acts.

$$d_{\mathrm{Me}} = \frac{3 d_{\mathrm{M2}}{}^{2} + 2 d_{\mathrm{M2}} \cdot d_{\mathrm{M1e}} + d_{\mathrm{M1e}}{}^{2}}{4 d_{\mathrm{M2}} + 2 d_{\mathrm{M1e}}} \tag{51}$$

4.6.3 Axial flexibility modulus of metallic compression limiter ring

$$X_{\mathrm{M}} = \frac{e_{\mathrm{M}}}{\pi \cdot b_{\mathrm{Me}} \cdot d_{\mathrm{Me}}} \tag{52}$$

4.7 Gasket compression behaviour

We define here the behaviour of the gasket in compression:

We consider the gasket thickness in compression as a function of the gasket compression stress is linear by pieces (see Annex A).

Such as for $Q_{\mathrm{Gj-1}} \leq Q_{\mathrm{G}} \leq Q_{\mathrm{Gj}}$, the thickness of the gasket in compression for a gasket compression stress Q_{G} is given by:

$$e_{\mathrm{G}}(Q_{\mathrm{G}}) = e_{\mathrm{Gj-1}} + (Q_{\mathrm{G}} - Q_{\mathrm{Gj-1}}) \cdot \frac{(e_{\mathrm{Gj}} - e_{\mathrm{Gj-1}})}{(Q_{\mathrm{Gj}} - Q_{\mathrm{Gj-1}})} \tag{53}$$

4.8 Local deformation parameters

4.8.1 General

Local compressions may occur at the contact areas between the different components of the connection. When significant, these local compressions have to be considered in the axial deformation balances.

Here below are given the expressions of local compressions at the different contact area in the connection.

(See also Annex B).

4.8.2 Local compression of flange at contact area with gasket

$$\Delta e_{\mathrm{FG}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{FG}}}{E_{\mathrm{F}}} \cdot F_{\mathrm{G}}\right) \tag{54}$$

with: X_{FG}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{FG}} = \begin{cases} \dfrac{\eta}{2 \cdot \pi \cdot d_{\mathrm{Ge}}} \cdot \ln\left(1 + \dfrac{2 \cdot e_{\mathrm{Ft}}}{\eta \cdot b_{\mathrm{Ge}}}\right) \\ \text{if } z_{1\mathrm{F}} \geq e_{\mathrm{Ft}} \text{ and } z_{2\mathrm{F}} \geq e_{\mathrm{Ft}} \\ \dfrac{\eta}{2 \cdot \pi \cdot d_{\mathrm{Ge}}} \cdot \ln\left(\dfrac{d_4 - d_{\mathrm{Ge}}}{b_{\mathrm{Ge}}}\right) + \dfrac{\eta}{\pi \cdot d_4} \cdot \ln\left(\dfrac{-\dfrac{2 \cdot e_{\mathrm{Ft}}}{\eta_2} + \dfrac{d_{\mathrm{Ge}} - b_{\mathrm{Ge}} - d_4}{\eta}}{-\dfrac{2 \cdot e_{\mathrm{Ft}}}{\eta_2} + \dfrac{d_{\mathrm{Ge}} - b_{\mathrm{Ge}} - d_4}{\eta}} \cdot \dfrac{d_{\mathrm{Ge}}}{d_{\mathrm{Ge}} - d_4}\right) \\ \text{if } z_{1\mathrm{F}} \geq e_{\mathrm{Ft}} \text{ and } z_{2\mathrm{F}} < e_{\mathrm{Ft}} \\ \dfrac{\eta}{2 \cdot \pi \cdot d_{\mathrm{Ge}}} \cdot \ln\left(\dfrac{d_{\mathrm{Ge}} - d_0}{b_{\mathrm{Ge}}}\right) + \dfrac{\eta}{\pi \cdot d_0} \cdot \ln\left(\dfrac{\dfrac{2 \cdot e_{\mathrm{Ft}}}{\eta_2} + \dfrac{d_{\mathrm{Ge}} - b_{\mathrm{Ge}} - d_0}{\eta}}{\dfrac{2 \cdot e_{\mathrm{Ft}}}{\eta_2} + \dfrac{d_{\mathrm{Ge}} - b_{\mathrm{Ge}} - d_0}{\eta}} \cdot \dfrac{d_{\mathrm{Ge}}}{d_{\mathrm{Ge}} - d_0}\right) \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Ft}} \text{ and } z_{2\mathrm{F}} \geq e_{\mathrm{Ft}} \\ \dfrac{\eta}{2 \cdot \pi \cdot d_{\mathrm{Ge}}} \cdot \ln\left(\dfrac{d_{\mathrm{Ge}} - d_0}{b_{\mathrm{Ge}}}\right) + \dfrac{\eta}{\pi \cdot d_0} \cdot \ln\left(\dfrac{d_4 - d_0}{d_4 + d_0} \cdot \dfrac{d_{\mathrm{Ge}}}{d_{\mathrm{Ge}} - d_0}\right) + \dfrac{4 \cdot (e_{\mathrm{Ft}} - z_{2\mathrm{F}})}{\pi \cdot (d_4^{\,2} - d_0^{\,2})} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Ft}},\ z_{2\mathrm{F}} < e_{\mathrm{Ft}} \text{ and } z_{1\mathrm{F}} \leq z_{2\mathrm{F}} \\ \dfrac{\eta}{2 \cdot \pi \cdot d_{\mathrm{Ge}}} \cdot \ln\left(\dfrac{d_4 - d_{\mathrm{Ge}}}{b_{\mathrm{Ge}}}\right) + \dfrac{\eta}{\pi \cdot d_4} \cdot \ln\left(\dfrac{d_0 - d_4}{d_0 + d_4} \cdot \dfrac{d_{\mathrm{Ge}}}{d_{\mathrm{Ge}} - d_4}\right) + \dfrac{4 \cdot (e_{\mathrm{Ft}} - z_{1\mathrm{F}})}{\pi \cdot (d_4^{\,2} - d_0^{\,2})} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Ft}},\ z_{2\mathrm{F}} < e_{\mathrm{Ft}} \text{ and } z_{2\mathrm{F}} < z_{1\mathrm{F}} \end{cases} \tag{55}$$

$$z_{1\mathrm{F}} = \frac{d_{\mathrm{Ge}} - b_{\mathrm{Ge}} - d_0}{2} \cdot \eta \tag{56}$$

$$z_{2\mathrm{F}} = \frac{d_4 - d_{\mathrm{Ge}} - b_{\mathrm{Ge}}}{2} \cdot \eta \tag{57}$$

4.8.3 Local compression of flange at metal to metal contact area

$$\Delta e_{\mathrm{FM}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{FM}}}{E_{\mathrm{F}}} \cdot F_{\mathrm{M}}\right) \tag{58}$$

with: X_{FM}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{FM}} = \begin{cases} \dfrac{\eta}{2\cdot\pi\cdot d_{\mathrm{Me}}}\cdot \ln\left(1+\dfrac{2\cdot e_{\mathrm{Fm}}}{\eta\cdot b_{\mathrm{Me}}}\right) \\ \text{if } z_{1\mathrm{F}} \ge e_{\mathrm{Fm}} \text{ and } z_{2\mathrm{F}} \ge e_{\mathrm{Fm}} \\ \dfrac{\eta}{2\cdot\pi\cdot d_{\mathrm{Me}}}\cdot \ln\left(\dfrac{d_4-d_{\mathrm{Me}}}{b_{\mathrm{Me}}}\right)+\dfrac{\eta}{\pi\cdot d_4}\cdot \ln\left(\dfrac{-\dfrac{2\cdot e_{\mathrm{Fm}}}{\eta_2}+\dfrac{d_{\mathrm{Me}}-b_{\mathrm{Me}}-d_4}{\eta}}{-\dfrac{2\cdot e_{\mathrm{Fm}}}{\eta_2}+\dfrac{d_{\mathrm{Me}}-b_{\mathrm{Me}}-d_4}{\eta}}\cdot\dfrac{d_{\mathrm{Me}}}{d_{\mathrm{Me}}-d_4}\right) \\ \text{if } z_{1\mathrm{F}} \ge e_{\mathrm{Fm}} \text{ and } z_{2\mathrm{F}} < e_{\mathrm{Fm}} \\ \dfrac{\eta}{2\cdot\pi\cdot d_{\mathrm{Me}}}\cdot \ln\left(\dfrac{d_{\mathrm{Me}}-d_0}{b_{\mathrm{Me}}}\right)+\dfrac{\eta}{\pi\cdot d_0}\cdot \ln\left(\dfrac{\dfrac{2\cdot e_{\mathrm{Fm}}}{\eta_2}+\dfrac{d_{\mathrm{Me}}-b_{\mathrm{Me}}-d_0}{\eta}}{\dfrac{2\cdot e_{\mathrm{Fm}}}{\eta_2}+\dfrac{d_{\mathrm{Me}}-b_{\mathrm{Me}}-d_0}{\eta}}\cdot\dfrac{d_{\mathrm{Me}}}{d_{\mathrm{Me}}-d_0}\right) \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Fm}} \text{ and } z_{2\mathrm{F}} \ge e_{\mathrm{Fm}} \\ \dfrac{\eta}{2\cdot\pi\cdot d_{\mathrm{Me}}}\cdot \ln\left(\dfrac{d_{\mathrm{Me}}-d_0}{b_{\mathrm{Me}}}\right)+\dfrac{\eta}{\pi\cdot d_0}\cdot \ln\left(\dfrac{d_4-d_0}{d_4+d_0}\cdot\dfrac{d_{\mathrm{Me}}}{d_{\mathrm{Me}}-d_0}\right)+\dfrac{4\cdot(e_{\mathrm{Fm}}-z_{2\mathrm{F}})}{\pi\cdot(d_4{}^2-d_0{}^2)} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Fm}},\ z_{2\mathrm{F}} < e_{\mathrm{Fm}} \text{ and } z_{1\mathrm{F}} \le z_{2\mathrm{F}} \\ \dfrac{\eta}{2\cdot\pi\cdot d_{\mathrm{Me}}}\cdot \ln\left(\dfrac{d_4-d_{\mathrm{Me}}}{b_{\mathrm{Me}}}\right)+\dfrac{\eta}{\pi\cdot d_4}\cdot \ln\left(\dfrac{d_0-d_4}{d_0+d_4}\cdot\dfrac{d_{\mathrm{Me}}}{d_{\mathrm{Me}}-d_4}\right)+\dfrac{4\cdot(e_{\mathrm{Fm}}-z_{1\mathrm{F}})}{\pi\cdot(d_4{}^2-d_0{}^2)} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{Fm}},\ z_{2\mathrm{F}} < e_{\mathrm{Fm}} \text{ and } z_{2\mathrm{F}} < z_{1\mathrm{F}} \end{cases} \tag{59}$$

$$z_{1\mathrm{F}} = \frac{d_{\mathrm{Me}}-b_{\mathrm{Me}}-d_0}{2}\cdot\eta \tag{60}$$

$$z_{2\mathrm{F}} = \frac{d_4-d_{\mathrm{Me}}-b_{\mathrm{Me}}}{2}\cdot\eta \tag{61}$$

4.8.4 Local compression of collar at contact area with loose flange

$$\Delta e_{\mathrm{FL}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{FL}}}{E_{\mathrm{F}}} \cdot F_{\mathrm{B}}\right) \quad (62)$$

with: X_{FL}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{FL}} = \begin{cases} \dfrac{\eta}{\pi \cdot (d_{7\,\min} + d_{7\,\max})} \cdot \ln\left(1 + \dfrac{4 \cdot e_{\mathrm{F}}}{\eta \cdot (d_{7\,\min} - d_{7\,\max})}\right) \\ \text{if } z_{1\mathrm{F}} \geq e_{\mathrm{F}} \text{ and } z_{2\mathrm{F}} \geq e_{\mathrm{F}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\min} + d_{7\,\max})} \cdot \ln\left(\dfrac{2 \cdot d_8 - (d_{7\,\min} + d_{7\,\max})}{d_{7\,\min} - d_{7\,\max}}\right) + \dfrac{\eta}{\pi \cdot d_8} \cdot \ln\left(\dfrac{-\dfrac{2 \cdot e_{\mathrm{F}}}{\eta_2} + \dfrac{d_{7\,\min} - d_8}{\eta}}{-\dfrac{2 \cdot e_{\mathrm{F}}}{\eta_2} + \dfrac{d_{7\,\min} + d_8}{\eta}} \cdot \dfrac{(d_{7\,\min} + d_{7\,\max})}{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_8}\right) \\ \text{if } z_{1\mathrm{F}} \geq e_{\mathrm{F}} \text{ and } z_{2\mathrm{F}} < e_{\mathrm{F}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\min} + d_{7\,\max})} \cdot \ln\left(\dfrac{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_0}{d_{7\,\min} - d_{7\,\max}}\right) + \dfrac{\eta}{\pi \cdot d_0} \cdot \ln\left(\dfrac{\dfrac{2 \cdot e_{\mathrm{F}}}{\eta_2} + \dfrac{d_{7\,\max} - d_0}{\eta}}{\dfrac{2 \cdot e_{\mathrm{F}}}{\eta_2} + \dfrac{d_{7\,\max} + d_0}{\eta}} \cdot \dfrac{(d_{7\,\min} + d_{7\,\max})}{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_0}\right) \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{F}} \text{ and } z_{2\mathrm{F}} \geq e_{\mathrm{F}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\min} + d_{7\,\max})} \cdot \ln\left(\dfrac{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_0}{d_{7\,\min} - d_{7\,\max}}\right) + \dfrac{\eta}{\pi \cdot d_0} \cdot \ln\left(\dfrac{d_8 - d_0}{d_8 + d_0} \cdot \dfrac{(d_{7\,\min} + d_{7\,\max})}{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_0}\right) + \dfrac{4 \cdot (e_{\mathrm{F}} - z_{2\mathrm{F}})}{\pi \cdot (d_8{}^2 - d_0{}^2)} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{F}}, z_{2\mathrm{F}} < e_{\mathrm{F}} \text{ and } z_{1\mathrm{F}} \leq z_{2\mathrm{F}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\min} + d_{7\,\max})} \cdot \ln\left(\dfrac{2 \cdot d_8 - (d_{7\,\min} + d_{7\,\max})}{d_{7\,\min} - d_{7\,\max}}\right) + \dfrac{\eta}{\pi \cdot d_8} \cdot \ln\left(\dfrac{d_0 - d_8}{d_0 + d_8} \cdot \dfrac{(d_{7\,\min} + d_{7\,\max})}{(d_{7\,\min} + d_{7\,\max}) - 2 \cdot d_8}\right) + \dfrac{4 \cdot (e_{\mathrm{F}} - z_{1\mathrm{F}})}{\pi \cdot (d_8{}^2 - d_0{}^2)} \\ \text{if } z_{1\mathrm{F}} < e_{\mathrm{F}}, z_{2\mathrm{F}} < e_{\mathrm{F}} \text{ and } z_{2\mathrm{F}} < z_{1\mathrm{F}} \end{cases} \quad (63)$$

$$z_{1\mathrm{F}} = \frac{d_{7\,\min} - d_0}{2} \cdot \eta \quad (64)$$

$$z_{2\mathrm{F}} = \frac{d_8 - d_{7\,\max}}{2} \cdot \eta \quad (65)$$

4.8.5 Local compression of loose flange at contact area with collar

$$\Delta e_{\mathrm{LF}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{LF}}}{E_{\mathrm{L}}} \cdot F_{\mathrm{B}}\right) \tag{66}$$

with: X_{LF}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{LF}} = \begin{cases} \dfrac{\eta}{\pi \cdot (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})} \cdot \ln\left(1 + \dfrac{4 \cdot e_{\mathrm{L}}}{\eta \cdot (d_{7\,\mathrm{min}} - d_{7\,\mathrm{max}})}\right) \\ \text{if } z_{1\mathrm{L}} \geq e_{\mathrm{L}} \text{ and } z_{2\mathrm{L}} \geq e_{\mathrm{L}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})} \cdot \ln\left(\dfrac{2 \cdot d_4 - (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{d_{7\,\mathrm{min}} - d_{7\,\mathrm{max}}}\right) + \dfrac{\eta}{\pi \cdot d_4} \cdot \ln\left(\dfrac{-\dfrac{2 \cdot e_{\mathrm{L}}}{\eta_2} + \dfrac{d_{7\,\mathrm{min}} - d_4}{\eta}}{-\dfrac{2 \cdot e_{\mathrm{L}}}{\eta_2} + \dfrac{d_{7\,\mathrm{min}} + d_4}{\eta}} \cdot \dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_4}\right) \\ \text{if } z_{1\mathrm{L}} \geq e_{\mathrm{L}} \text{ and } z_{2\mathrm{L}} < e_{\mathrm{L}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})} \cdot \ln\left(\dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_6}{d_{7\,\mathrm{min}} - d_{7\,\mathrm{max}}}\right) + \dfrac{\eta}{\pi \cdot d_6} \cdot \ln\left(\dfrac{\dfrac{2 \cdot e_{\mathrm{L}}}{\eta_2} + \dfrac{d_{7\,\mathrm{max}} - d_6}{\eta}}{\dfrac{2 \cdot e_{\mathrm{L}}}{\eta_2} + \dfrac{d_{7\,\mathrm{max}} + d_6}{\eta}} \cdot \dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_6}\right) \\ \text{if } z_{1\mathrm{L}} < e_{\mathrm{L}} \text{ and } z_{2\mathrm{L}} \geq e_{\mathrm{L}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})} \cdot \ln\left(\dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_6}{d_{7\,\mathrm{min}} - d_{7\,\mathrm{max}}}\right) + \dfrac{\eta}{\pi \cdot d_6} \cdot \ln\left(\dfrac{d_4 - d_6}{d_4 + d_6} \cdot \dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_6}\right) + \dfrac{4 \cdot (e_{\mathrm{L}} - z_{2\mathrm{L}})}{\pi \cdot (d_4^{\,2} - d_6^{\,2})} \\ \text{if } z_{1\mathrm{L}} < e_{\mathrm{L}},\, z_{2\mathrm{L}} < e_{\mathrm{L}} \text{ and } z_{1\mathrm{L}} \leq z_{2\mathrm{L}} \\ \dfrac{\eta}{\pi \cdot (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})} \cdot \ln\left(\dfrac{2 \cdot d_4 - (d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{d_{7\,\mathrm{min}} - d_{7\,\mathrm{max}}}\right) + \dfrac{\eta}{\pi \cdot d_4} \cdot \ln\left(\dfrac{d_6 - d_4}{d_6 + d_4} \cdot \dfrac{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}})}{(d_{7\,\mathrm{min}} + d_{7\,\mathrm{max}}) - 2 \cdot d_4}\right) + \dfrac{4 \cdot (e_{\mathrm{L}} - z_{1\mathrm{L}})}{\pi \cdot (d_4^{\,2} - d_6^{\,2})} \\ \text{if } z_{1\mathrm{L}} < e_{\mathrm{L}},\, z_{2\mathrm{L}} < e_{\mathrm{L}} \text{ and } z_{2\mathrm{L}} \leq z_{1\mathrm{L}} \end{cases} \tag{67}$$

$$z_{1\mathrm{L}} = \frac{d_{7\,\mathrm{min}} - d_6}{2} \cdot \eta \tag{68}$$

$$z_{2\mathrm{L}} = \frac{d_4 - d_{7\,\mathrm{max}}}{2} \cdot \eta \tag{69}$$

4.8.6 Local compression of loose flange at contact area with nut

$$\Delta e_{\mathrm{LB}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{LB}}}{E_{\mathrm{L}}}\cdot F_{\mathrm{B}}\right) \tag{70}$$

with: X_{LB}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{LB}} = \begin{cases} \dfrac{\eta}{n_B \cdot \pi \cdot d_5}\cdot \ln\left(\dfrac{\dfrac{2\cdot e_{\mathrm{L}}}{\eta_2}+\dfrac{d_{\mathrm{FBext}}-d_5}{\eta}}{\dfrac{2\cdot e_{\mathrm{L}}}{\eta_2}+\dfrac{d_{\mathrm{FBext}}+d_5}{\eta}}\cdot\dfrac{d_{\mathrm{FBext}}+d_5}{d_{\mathrm{FBext}}-d_5}\right) & z_{2\mathrm{L}} \geq e_{\mathrm{L}} \\ \dfrac{\eta}{n_{\mathrm{B}} \cdot \pi \cdot d_5}\cdot \ln\left(\dfrac{d_4-d_3-2\cdot d_{\mathrm{FBext}}}{d_4-d_3+d_5}\cdot\dfrac{d_{\mathrm{FBext}}+d_5}{d_{\mathrm{FBext}}-d_5}\right)+\dfrac{4\cdot\left(e_{\mathrm{L}}-\left(\dfrac{d_4-d_3-d_{\mathrm{FBext}}}{2}\right)\eta\right)}{n_B\cdot\pi\cdot((d_4-d_3)^2-d_5{}^2)} & z_{2\mathrm{L}} < e_{\mathrm{L}} \end{cases} \tag{71}$$

4.8.7 Local compression of flange at contact area with nut

$$\Delta e_{\mathrm{FB}}^{\mathrm{M}} = -\Delta\left(\frac{X_{\mathrm{FB}}}{E_{\mathrm{F}}}\cdot F_{\mathrm{B}}\right) \tag{72}$$

with: X_{FB}: the flexibility parameter of flange in the local compression area:

$$X_{\mathrm{FB}} = \begin{cases} \dfrac{\eta}{n_B \cdot \pi \cdot d_5}\cdot \ln\left(\dfrac{\dfrac{2\cdot e_{\mathrm{Fb}}}{\eta_2}+\dfrac{d_{\mathrm{FBext}}-d_5}{\eta}}{\dfrac{2\cdot e_{\mathrm{Fb}}}{\eta_2}+\dfrac{d_{\mathrm{FBext}}+d_5}{\eta}}\cdot\dfrac{d_{\mathrm{FBext}}+d_5}{d_{\mathrm{FBext}}-d_5}\right) & z_{2\mathrm{F}} \geq e_{\mathrm{Fb}} \\ \dfrac{\eta}{n_{\mathrm{B}} \cdot \pi \cdot d_5}\cdot \ln\left(\dfrac{d_4-d_3-2\cdot d_{\mathrm{FBext}}}{d_4-d_3+d_5}\cdot\dfrac{d_{\mathrm{FBext}}+d_5}{d_{\mathrm{FBext}}-d_5}\right)+\dfrac{4\cdot\left(e_{\mathrm{Fb}}-\left(\dfrac{d_4-d_3-d_{\mathrm{FBext}}}{2}\right)\eta\right)}{n_B\cdot\pi\cdot((d_4-d_3)^2-d_5{}^2)} & z_{2\mathrm{F}} < e_{\mathrm{Fb}} \end{cases} \tag{73}$$

4.9 Relaxation phenomenon of the gasket

4.9.1 General

Relaxation phenomenon of the gasket consists in the decrease of gasket compression after the increase of the load applied on the gasket (mechanical and (or) thermal).

This is an irreversible phenomenon. It means that a decrease of the load applied on the gasket (mechanical and (or) thermal) does not lead to an increase of the gasket compression.

We assume that the relaxation phenomenon occurs after the loading of the gasket.

4.9.2 Consideration of the relaxation phenomenon

Relaxation of the gasket will be considered after every load condition. It means that after determining the internal reactions corresponding to a loading situation, an additional calculation will be performed in order to determine the internal reactions after the relaxation phenomenon (only in the case where the load on the gasket is increased (mechanical or thermal)).

4.9.3 Relaxation behaviour of the gasket

Different models described in [1] may be used to reproduce the relaxation behaviour of the gasket material. Among them, a realistic model in Annex C provides a stress relaxation response for gasket that may be consider as viscoelastic material.

Other models or test data may be suitable to consider the relaxation behaviour of the gasket exposed to compression and (or) elevated temperature.

It means that the gasket surface pressure is determined for a given gasket compression after a given period of time.

This leads to the determination of the remaining gasket surface pressure after relaxation.

A new internal reaction value on the gasket after relaxation is obtained. Once determined, the new internal reaction on the gasket is introduced in the compliance equations in order to update the other forces and deformations in the connection.

5 Internal forces (in the connection)

5.1 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 applications. All potentially critical load conditions shall be calculated.

5.2 Applied loads

5.2.1 Assembly condition (I = 0)

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

External loads F_{A0} and M_{A0} combine to give a net force F_{R0} as in Equation (75) (load case I = 0).

All temperatures are equal to the initial uniform value T_0.

5.2.2 Subsequent conditions (I = 1,2 …)

5.2.2.1 Fluid pressure

$$\left.\begin{array}{lll} \text{Internal fluid pressure} & & P_I > 0 \\ \text{Unpressureized condition} & & P_I = 0 \\ \text{External fluid pressure} & & P_I < 0 \end{array}\right\} F_{QI} = (\pi / 4) \times d_{Ge}{}^2 \cdot P_I \qquad (74)$$

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.

5.2.2.2 Additional external loads

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

$$\left.\begin{array}{lll} \text{Axial tensile force} & & F_{AI} > 0 \\ \text{Axial compression force} & & F_{AI} < 0 \end{array}\right\} F_{RI} = F_{AI} \pm (4 / d_{3e}) \cdot M_{AI} \qquad (75)$$

Select the sign in Equation (75) giving the more severe condition.

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

— on the side of the connection where the moment induces an additional tensile load (sign + in Equation (75)), load limits of flanges or bolts may govern, as well as minimum gasket compression;

— on the side of the connection where the moment induces an additional compression load (sign - in Equation (75)), 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 Equation (75)) whenever an external moment is applied, with different indices I being assigned to each case.

5.2.2.3 Thermal loads in the gasket area

Axial thermal expansion in the gasket area between load condition I and load condition I + 1 to be considered in the first compliance equation is treated by the equation here below.

$$\Delta U_{\mathrm{I}\to\mathrm{I}+1}{}^{\mathrm{i}} = \Delta U_{0\to\mathrm{I}+1}{}^{\mathrm{i}} - \Delta U_{0\to\mathrm{I}}{}^{\mathrm{i}} \tag{76}$$

where

$$\begin{aligned}\Delta U_{0\to\mathrm{I}+1}{}^{\mathrm{i}} = {} & l_{\mathrm{B}} \cdot \alpha_{\mathrm{B}} \cdot \left(T_{\mathrm{BI}+1} - T_{\mathrm{B0}}\right) - e_{\mathrm{w}} \cdot \alpha_{w} \cdot \left(T_{\mathrm{WI}+1} - T_{\mathrm{W0}}\right) - e_{\mathrm{Lt}} \cdot \alpha_{\mathrm{L}} \cdot \left(T_{\mathrm{LI}+1} - T_{\mathrm{L0}}\right) - e_{\mathrm{Ft}} \cdot \alpha_{\mathrm{F}} \cdot \left(T_{\mathrm{FI}+1} - T_{\mathrm{F0}}\right) \\ & - e_{\mathrm{G}} \cdot \alpha_{\mathrm{G}} \cdot \left(T_{\mathrm{GI}+1} - T_{\mathrm{G0}}\right) - \tilde{e}_{\mathrm{Ft}} \cdot \tilde{\alpha}_{\mathrm{F}} \cdot \left(\tilde{T}_{\mathrm{F}+1} - \tilde{T}_{\mathrm{F0}}\right) - \tilde{e}_{\mathrm{Lt}} \cdot \tilde{\alpha}_{\mathrm{L}} \cdot \left(\tilde{T}_{\mathrm{LI}+1} - \tilde{T}_{\mathrm{L0}}\right) \\ & - \tilde{e}_{\mathrm{W}} \cdot \tilde{\alpha}_{\mathrm{W}} \cdot \left(\tilde{T}_{\mathrm{WI}+1} - \tilde{T}_{\mathrm{W0}}\right)\end{aligned} \tag{77}$$

and

$$\begin{aligned}\Delta U_{0\to\mathrm{I}}{}^{\mathrm{i}} = {} & l_{\mathrm{B}} \cdot \alpha_{\mathrm{B}} \cdot \left(T_{\mathrm{BI}} - T_{\mathrm{B0}}\right) - e_{\mathrm{w}} \cdot \alpha_{w} \cdot \left(T_{\mathrm{WI}} - T_{\mathrm{W0}}\right) - e_{\mathrm{Lt}} \cdot \alpha_{\mathrm{L}} \cdot \left(T_{\mathrm{LI}} - T_{\mathrm{L0}}\right) - e_{\mathrm{Ft}} \cdot \alpha_{\mathrm{F}} \cdot \left(T_{\mathrm{FI}} - T_{\mathrm{F0}}\right) \\ & - e_{\mathrm{G}} \cdot \alpha_{\mathrm{G}} \cdot \left(T_{\mathrm{GI}} - T_{\mathrm{G0}}\right) - \tilde{e}_{\mathrm{Ft}} \cdot \tilde{\alpha}_{\mathrm{F}} \cdot \left(\tilde{T}_{\mathrm{FI}} - \tilde{T}_{\mathrm{F0}}\right) - \tilde{e}_{\mathrm{Lt}} \cdot \tilde{\alpha}_{\mathrm{L}} \cdot \left(\tilde{T}_{\mathrm{LI}} - \tilde{T}_{\mathrm{L0}}\right) \\ & - \tilde{e}_{\mathrm{W}} \cdot \tilde{\alpha}_{\mathrm{W}} \cdot \left(\tilde{T}_{\mathrm{WI}} - \tilde{T}_{\mathrm{W0}}\right)\end{aligned} \tag{78}$$

where:

$$\tilde{e}_{\mathrm{Ft}} + e_{\mathrm{Ft}} + e_{\mathrm{Lt}} + \tilde{e}_{\mathrm{Lt}} + e_{\mathrm{G}} + e_{\mathrm{W}} + \tilde{e}_{\mathrm{W}} = l_{\mathrm{B}}$$

5.2.2.4 Thermal loads in the metal to metal contact area

Axial thermal expansion in the metal to metal contact area between load condition I and load condition I + 1 to be considered in the second compliance equation is treated by the equation here below.

$$\Delta U_{\mathrm{I}\to\mathrm{I}+1}{}^{\mathrm{ii}} = \Delta U_{0\to\mathrm{I}+1}{}^{\mathrm{ii}} - \Delta U_{0\to\mathrm{I}}{}^{\mathrm{ii}} \tag{79}$$

where

$$\begin{aligned}\Delta U_{0\to\mathrm{I}+1}{}^{\mathrm{ii}} = {} & e_{\mathrm{G}} \cdot \alpha_{\mathrm{G}} \left(T_{\mathrm{GI}+1} - T_{\mathrm{G0}}\right) + e_{\mathrm{Ft}} \cdot \alpha_{\mathrm{F}} \cdot \left(T_{\mathrm{FI}+1} - T_{\mathrm{F0}}\right) - \tilde{e}_{\mathrm{Ft}} \cdot \tilde{\alpha}_{\mathrm{F}} \cdot \left(\tilde{T}_{\mathrm{FI}+1} - \tilde{T}_{\mathrm{F0}}\right) \\ & - e_{\mathrm{M}} \cdot \alpha_{\mathrm{M}} \cdot \left(T_{\mathrm{MI}+1} - T_{\mathrm{M0}}\right) - e_{\mathrm{Fm}} \cdot \alpha_{\mathrm{F}} \cdot \left(T_{\mathrm{FI}+1} - T_{\mathrm{F0}}\right) - \tilde{e}_{\mathrm{Fm}} \cdot \tilde{\alpha}_{\mathrm{F}} \cdot \left(\tilde{T}_{\mathrm{FI}+1} - \tilde{T}_{\mathrm{F0}}\right)\end{aligned} \tag{80}$$

and

$$\Delta U_{0\to\mathrm{I}}{}^{\mathrm{ii}} = e_{\mathrm{G}} \cdot \alpha_{G} \cdot (T_{\mathrm{GI}} - T_{\mathrm{G0}}) - e_{\mathrm{Ft}} \cdot \alpha_{\mathrm{F}} \cdot (T_{\mathrm{FI}} - T_{\mathrm{F0}}) + \widetilde{e}_{\mathrm{Ft}} \cdot \widetilde{\alpha}_{\mathrm{F}} \cdot (\widetilde{T}_{\mathrm{FI}} - \widetilde{T}_{\mathrm{F0}}) - e_{\mathrm{M}} \cdot \alpha_{\mathrm{M}} \cdot (T_{\mathrm{MI}} - T_{\mathrm{M0}}) \\ - e_{\mathrm{Fm}} \cdot \alpha_{\mathrm{F}} \cdot (T_{\mathrm{FI}} - T_{\mathrm{F0}}) - \widetilde{e}_{\mathrm{Fm}} \cdot \widetilde{\alpha}_{\mathrm{F}} \cdot (\widetilde{T}_{\mathrm{FI}} - \widetilde{T}_{\mathrm{F0}}) \quad (81)$$

5.3 Compliance of the connection

5.3.1 First compliance equation

The first set of compliance terms corresponding to the first compliance equation (see Annex F for more details) is as follow: compliance equation is determined from the forces balance and the first deformation compatibility equation:

$$\Delta\left(Y_{\mathrm{G}}^{\mathrm{i}} \cdot F_{\mathrm{G}}\right) + \Delta\left(Y_{\mathrm{M}}^{\mathrm{i}} \cdot F_{\mathrm{M}}\right) + \Delta\left(Y_{\mathrm{Q}}^{\mathrm{i}} \cdot F_{\mathrm{Q}}\right) + \Delta\left(Y_{\mathrm{R}}^{\mathrm{i}} \cdot F_{\mathrm{R}}\right) + \Delta U^{\mathrm{i}} = 0 \quad (82)$$

with

$$Y_{\mathrm{G}}^{\mathrm{i}} = W + \frac{Z_{\mathrm{F}} \cdot h_{\mathrm{G}}{}^{2} + X_{\mathrm{FG}}}{E_{\mathrm{F}}} + \frac{X_{\mathrm{G}}}{E_{\mathrm{G}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{G}}{}^{2} + \widetilde{X}_{\mathrm{FG}}}{\widetilde{E}_{\mathrm{F}}} \quad (83)$$

$$Y_{\mathrm{M}}^{\mathrm{i}} = W + \frac{Z_{\mathrm{F}} \cdot h_{\mathrm{G}} \cdot h_{\mathrm{M}}}{E_{\mathrm{F}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{G}} + \widetilde{h}_{\mathrm{M}}}{\widetilde{E}_{\mathrm{F}}} \quad (84)$$

$$Y_{\mathrm{Q}}^{\mathrm{i}} = W + \frac{Z_{\mathrm{F}} \cdot h_{\mathrm{G}} \cdot \left(h_{\mathrm{H}} + h_{\mathrm{Q}} - h_{\mathrm{P}}\right)}{E_{\mathrm{F}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{G}} \cdot \left(\widetilde{h}_{\mathrm{H}} + \widetilde{h}_{\mathrm{Q}} - \widetilde{h}_{\mathrm{P}}\right)}{\widetilde{E}_{\mathrm{F}}} \quad (85)$$

$$Y_{\mathrm{R}}^{\mathrm{i}} = W + \frac{Z_{\mathrm{F}} \cdot h_{\mathrm{G}} \cdot \left(h_{\mathrm{H}} + h_{\mathrm{R}}\right)}{E_{\mathrm{F}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{G}} \cdot \left(\widetilde{h}_{\mathrm{H}} + \widetilde{h}_{\mathrm{R}}\right)}{\widetilde{E}_{\mathrm{F}}} \quad (86)$$

$$W = \frac{X_{\mathrm{B}}}{E_{\mathrm{B}}} + \frac{X_{\mathrm{W}}}{E_{\mathrm{W}}} + \frac{\widetilde{X}_{\mathrm{W}}}{\widetilde{E}_{\mathrm{W}}} + \frac{X_{\mathrm{LB}} + Z_{\mathrm{L}} \cdot h_{\mathrm{L}}{}^{2} + X_{\mathrm{LF}}}{E_{\mathrm{L}}} + \frac{\widetilde{X}_{\mathrm{LB}} + \widetilde{Z}_{\mathrm{L}} \cdot \widetilde{h}_{\mathrm{L}}{}^{2} + \widetilde{X}_{\mathrm{LF}}}{\widetilde{E}_{\mathrm{L}}} + \frac{X_{\mathrm{FL}}}{E_{\mathrm{F}}} + \frac{\widetilde{X}_{\mathrm{FL}}}{\widetilde{E}_{\mathrm{F}}} \quad (87)$$

ΔU^{i}: following 5.2.2.3

5.3.2 Second compliance equation

The second compliance equation is determined from the forces balance and the second deformation compatibility equation:

$$\Delta\left(Y_{\mathrm{G}}^{\mathrm{ii}} \cdot F_{\mathrm{G}}\right) + \Delta\left(Y_{\mathrm{M}}^{\mathrm{ii}} \cdot F_{\mathrm{M}}\right) + \Delta\left(Y_{\mathrm{Q}}^{\mathrm{ii}} \cdot F_{\mathrm{Q}}\right) + \Delta\left(Y_{\mathrm{R}}^{\mathrm{ii}} \cdot F_{\mathrm{R}}\right) + \Delta U^{\mathrm{ii}} = 0 \quad (88)$$

with:

$$Y_{\mathrm{G}}^{\mathrm{ii}} = -\left(\frac{Z_{\mathrm{F}} \cdot h_{\mathrm{G}} \cdot h_{\mathrm{D}} + X_{\mathrm{FG}}}{E_{\mathrm{F}}} + \frac{X_{\mathrm{G}}}{E_{\mathrm{G}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{G}} \cdot \widetilde{h}_{\mathrm{D}} + \widetilde{X}_{\mathrm{FG}}}{\widetilde{E}_{\mathrm{F}}}\right) \quad (89)$$

$$Y_{\mathrm{M}}^{\mathrm{ii}} = \left(\frac{X_{\mathrm{FM}} - Z_{\mathrm{F}} \cdot h_{\mathrm{M}} \cdot h_{\mathrm{D}}}{E_{\mathrm{F}}} + \frac{X_{\mathrm{M}}}{E_{\mathrm{M}}} + \frac{\widetilde{X}_{\mathrm{FM}} - \widetilde{Z}_{\mathrm{F}} \cdot \widetilde{h}_{\mathrm{M}} \cdot \widetilde{h}_{\mathrm{D}}}{\widetilde{E}_{\mathrm{F}}}\right) \quad (90)$$

$$Y_{\mathrm{Q}}^{\mathrm{ii}} = -\left(\frac{Z_{\mathrm{F}} \cdot \left(h_{\mathrm{H}} + h_{\mathrm{Q}} - h_{\mathrm{P}}\right) \cdot h_{\mathrm{D}}}{E_{\mathrm{F}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \left(\widetilde{h}_{\mathrm{H}} + \widetilde{h}_{\mathrm{Q}} - \widetilde{h}_{\mathrm{P}}\right) \cdot \widetilde{h}_{\mathrm{D}}}{\widetilde{E}_{\mathrm{F}}} \right) \tag{91}$$

$$Y_{\mathrm{R}}^{\mathrm{ii}} = -\left(\frac{Z_{\mathrm{F}} \cdot \left(h_{\mathrm{H}} + h_{\mathrm{R}}\right) \cdot h_{\mathrm{D}}}{E_{\mathrm{F}}} + \frac{\widetilde{Z}_{\mathrm{F}} \cdot \left(\widetilde{h}_{\mathrm{H}} + \widetilde{h}_{\mathrm{R}}\right) \cdot \widetilde{h}_{\mathrm{D}}}{\widetilde{E}_{\mathrm{F}}} \right) \tag{92}$$

ΔU^{ii}: following 5.2.2.4

5.4 Determination of the minimum forces necessary for the gasket

5.4.1 Assembly condition (I = 0)

Minimum gasket force:

$$F_{\mathrm{G0\,min}} = A_{\mathrm{Ge}} \times Q_{\mathrm{min}} \tag{93}$$

The F_{G0} value obtained in 5.6.3 shall be higher than $F_{\mathrm{G0\,min}}$. If not, the additional tightening above F_{BMMC} shall be adapted to ensure that F_{G0} is higher than $F_{\mathrm{G0\,min}}$ after additional tightening above F_{BMMC}.

5.4.2 subsequent conditions (I = 1, 2...)

At every load condition I, the condition here below shall be verified:

$$A_{\mathrm{Ge}} \times Q_{\mathrm{I}} \le F_{\mathrm{GI}} \tag{94}$$

where QI is the required gasket surface pressure in order to ensure the required leak rate at the load condition I for the internal pressure PI and the temperature TI and for the maximum surface pressure previously applied on the gasket.

If the condition is not satisfied, the additional tightening above F_{BMMC} shall be adapted to fulfill this condition.

5.5 Determination of the appearance of the MMC in assembly condition (I = 0)

5.5.1 General

The principle is to determine:

First, the range [$Q_{\mathrm{Gj-1}}$; Q_{Gj}] of gasket surface pressure values in which the MMC is achieved, then the force to be applied on the gasket F_{GMMC} to reach the MMC by “successive approach”.

The required tightening force F_{BMMC} to reach MMC is then deduced.

5.5.2 Determination of the gasket surface pressure range in which MMC appears in assembly condition (I = 0)

A gasket surface pressure Q_{Gj} from one of the n couples defined in 4.7 and Annex A is considered and it is determined whether the MMC is achieved at this gasket surface pressure or not.

If the MMC is achieved it means that MMC appears in a range [$Q_{\mathrm{Gk-1}}$; Q_{Gk}] where k ≤ j.

If the MMC is not achieved, the calculation process is repeated with a higher gasket surface pressure, until the range of appearance is found (see Figure 3 here below).

First, the force F_{G} to be applied on the gasket to obtain the gasket surface pressure Q_{Gj} is determined.

A first approximation by considering the theoretical dimensions of the gasket is done:

$$F_{\mathrm{G}} = Q_{\mathrm{Gj}} \cdot A_{\mathrm{Gt}} \tag{95}$$

From this gasket force, the effective dimensions of the gasket (b_{Ge}, d_{Ge}, A_{Ge}) are determined following equations described in 4.4.3, with $F_{\mathrm{G0}} = F_{\mathrm{G}}$.

Once A_{Ge} has been determined, a new value of F_{G} is obtained:

$$F_{\mathrm{G}} = Q_{\mathrm{Gj}} \cdot A_{\mathrm{Ge}} \tag{96}$$

We proceed by iteration until the value F_{G} is constant within the required precision.

Once F_{G} and h_{G} have been determined, the rotation angles of the flanges are determined:

$$\Theta_{\mathrm{F}} = \frac{Z_{\mathrm{F}}}{E_{\mathrm{F}}} \cdot \left(F_{\mathrm{G}} \cdot h_{\mathrm{G}} + F_{\mathrm{R}} \cdot \left(h_{\mathrm{H}} + h_{\mathrm{R}}\right)\right) \tag{97}$$

$$\widetilde{\Theta}_{\mathrm{F}} = \frac{\widetilde{Z}_{\mathrm{F}}}{\widetilde{E}_{\mathrm{F}}} \cdot \left(F_{\mathrm{G}} \cdot h_{\mathrm{G}} + F_{\mathrm{R}} \cdot \left(\widetilde{h}_{\mathrm{H}} + \widetilde{h}_{\mathrm{R}}\right)\right) \tag{98}$$

$$\Theta_{\mathrm{L}} = \frac{Z_{\mathrm{L}}}{E_{\mathrm{L}}} \cdot \left(F_{\mathrm{B}} \cdot h_{\mathrm{L}}\right) \tag{99}$$

$$\widetilde{\Theta}_{\mathrm{L}} = \frac{\widetilde{Z}_{\mathrm{L}}}{\widetilde{E}_{\mathrm{L}}} \cdot \left(F_{\mathrm{B}} \cdot \widetilde{h}_{\mathrm{L}}\right) \tag{100}$$

The potential internal diameter of MMC is obtained with the expression given in 4.6.2.1

At this stage, d_{M1e} shall be compared with d_{M2}.

It means while $d_{\mathrm{M1e}} > d_{\mathrm{M2}}$, the whole calculation described in 5.5.2 shall be repeated with a higher level of gasket surface pressure.

Then the range [$Q_{\mathrm{Gj-1}}$; Q_{Gj}], in which MMC appears, is determined. This range is defined as [Q_{GMMCinf}; Q_{GMMCsup}].

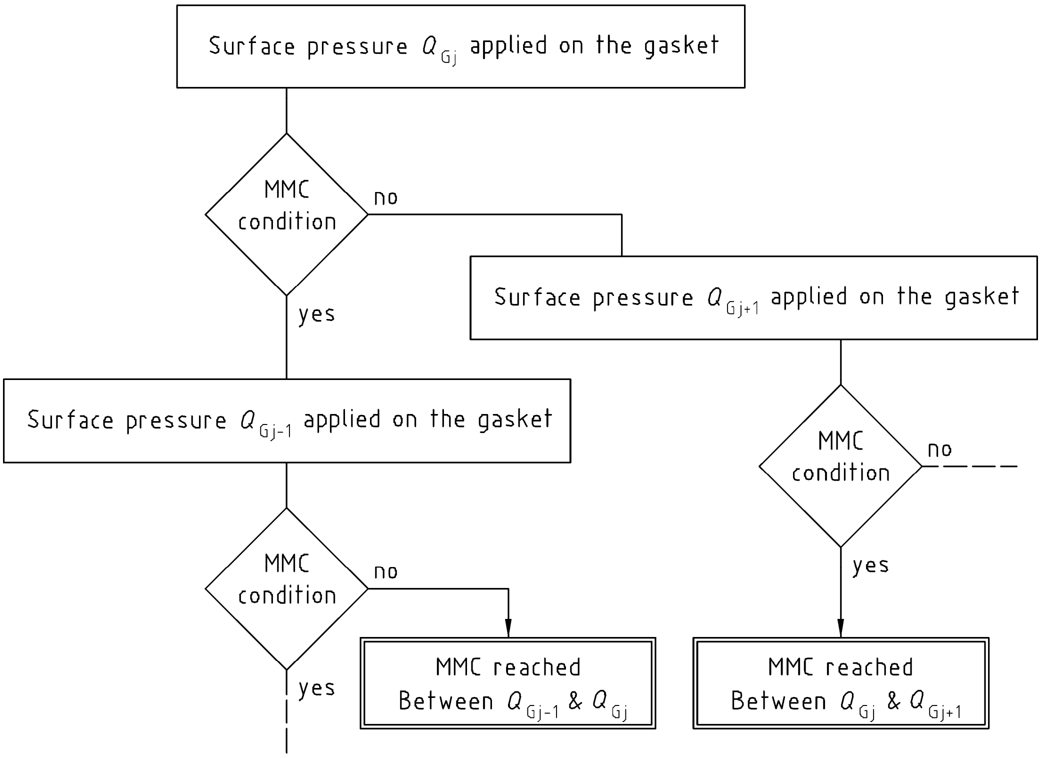


Figure 3 — Determination of the gasket surface pressure range in which MMC appears

5.5.3 Determination of the force to be applied on the gasket to achieve the MMC

Here, we proceed by convergence (like dichotomy) in order to determine F_{GMMC}, such as the corresponding gasket surface pressure Q_{G} belongs to [Q_{GMMCinf} ; Q_{GMMCsup}], and for which $d_{\text{M1e}} = d_{\text{M2}}$.

In the range [Q_{GMMCinf} ; Q_{GMMCsup}], the gasket behaviour in compression has been defined linear (see 4.7).

The calculation is started with F_{GMMC} such as:

$$F_{\text{GMMC}} = \frac{Q_{\text{GMMC}} + Q_{\text{GMMC inf}}}{2} \cdot A_{\text{Ge}} \tag{101}$$

The gasket thickness corresponding to this load is determined following the equation of 4.7.

Then, the rotation angle of the flanges are determined following Equations (97) to (100)

We obtain the potential internal diameter of MMC by the expression given in 4.6.2.1:

d_{M1e} shall be compared with d_{M2}.

The force to be applied on the gasket to achieve the MMC : F_{GMMC} ($d_{\text{M1e}} = d_{\text{M2}}$) is determined by convergence and the corresponding bolts tightening force is F_{BMMC}.

NOTE In 5.5.2 and 5.5.3 the gasket compression curve in the right joint configuration is assumed to be known. However, when this is not the case other gasket data may be used to determine F_{GMMC}, such as the gasket stress for which MMC appears under a compression press. In this case, it should be checked that the flange rotation angles are small enough.

5.6 Determination of the required bolt tightening in assembly condition (I = 0) to maintain the MMC in operation and to satisfy the leak-tightness criteria

5.6.1 General

Different initial bolt tightening values of F_{B0} between F_{BMMC} and $F_{B0\,max}$ may be considered. $F_{B0\,max}$ is the maximum allowable bolt tightening in assembly condition.

The first bolt tightening value F_{B0} to be considered is $F_{B0\,max}$. From this initial bolt tightening, the internal forces are calculated in all subsequent load conditions. It shall be checked that the MMC is maintained and the leak-tightness criteria verified at all the subsequent load conditions.

If the conditions about MMC and leak-tightness are verified, then a smaller bolt tightening may be selected within the range [F_{BMMC}; $F_{B0\,max}$] and the calculation repeated. The required minimum bolt force calculated to maintain the MMC and to ensure the required leak-rate in all the load conditions $F_{B0\,req}$ can be determined.

If there is no appropriate bolt tightening, the design of the bolted flange connection shall be changed.

5.6.2 Determination of the maximum allowable bolt tightening in assembly condition (I=0)

The maximum allowable bolt tightening in assembly condition is the bolt tightening for which the maximum bolt load ratio is reached (see Equation (121) in 6.4)

5.6.3 Determination of the reaction forces F_{G0} and F_{M0} after the additional tightening above F_{BMMC}

5.6.3.1 General

$$F_{G0} = \frac{1}{Y_{G0}^{\,ii} - Y_{M0}^{\,ii}} \cdot \begin{pmatrix} Y_{GMMC}^{\,ii} \cdot F_{GMMC} + Y_{M0}^{\,ii} \cdot \left(F_{BMMC} - F_{B0} + F_{R0} - F_{RMMC} - F_{GMMC}\right) \\ - Y_{R0}^{\,ii} \cdot F_{R0} + Y_{RMMC}^{\,ii} \cdot F_{RMMC} \end{pmatrix} \tag{102}$$

$$F_{M0} = \frac{1}{Y_{M0}^{\,ii} - Y_{G0}^{\,ii}} \cdot \begin{pmatrix} Y_{GMMC}^{\,ii} \cdot F_{GMMC} + Y_{G0}^{\,ii} \cdot \left(F_{BMMC} - F_{B0} + F_{R0} - F_{RMMC} - F_{GMMC}\right) \\ - Y_{R0}^{\,ii} \cdot F_{R0} + Y_{RMMC}^{\,ii} \cdot F_{RMMC} \end{pmatrix} \tag{103}$$

NOTE In most cases, $F_{R0} = F_{RMMC}$.

Several values of F_{B0} shall be considered. The first value is $F_{B0} = F_{B0\,max}$ (see above).

5.6.3.2 Initial calculation

A first calculation is performed by considering the results obtained in 5.5.

The effective metal to metal contact diameter d_{Me} is obtained with Equation (51) in which the d_{M1e} value corresponds to the one obtained in 5.5.3

The lever arm corresponding to the metal to metal contact reaction h_M: following Equation (16 or 19)

d_{Ge} is determined in 5.5.2

h_G and X_G are deduced from this value of d_{Ge}.

X_{FG} and X_{FM} are obtained as defined in 4.8.2 and 4.8.3.

The first compliance terms values $Y_{M0}^{\,ii}$, $Y_{G0}^{\,ii}$, $Y_{R0}^{\,ii}$ are then determined following the equations of 5.3.2

A first calculation of F_{G0} and F_{M0} is then performed following Equations (102) and (103) above.

5.6.3.3 Updating of calculation parameters

From the initial calculation performed in 5.6.3.2, new values of the compliance terms $Y_{M0}{}^{ii}$, $Y_{G0}{}^{ii}$, $Y_{R0}{}^{ii}$ shall then be determined.

From the new value of F_G obtained in 5.6.3.2: we deduce the new values of e_G, d_{Ge}, b_{Ge}, A_{Ge}, h_G, X_G (following 4.4.3 and 4.4.4).

From the new value of F_G and F_M obtained in 5.6.3.2: we deduce new flange rotation angles.

$$\Theta_F = \frac{Z_F}{E_F} \cdot \left(F_G \cdot h_G + F_M \cdot h_M + F_R \cdot \left(h_H + h_R\right)\right) \tag{104}$$

$$\widetilde{\Theta}_F = \frac{\widetilde{Z}_F}{\widetilde{E}_F} \cdot \left(F_G \cdot h_G + F_M \cdot h_M + F_R \cdot \left(\widetilde{h}_H + \widetilde{h}_R\right)\right) \tag{105}$$

For rotation angles of loose flange see Equations (99) and (100).

From the new flange rotation angles: we deduce the new effective MMC area dimensions d_{M1e}, d_{Me} and h_M.

Then we have to update the values of the compliance terms: $Y_{M0}{}^{ii}$, $Y_{G0}{}^{ii}$, $Y_{R0}{}^{ii}$.

We repeat the F_{G0} and F_{M0} calculations following Equation (102) and (103).

Here, we proceed by iteration. We repeat 5.6.3.3 calculations until the values F_{G0} and F_{M0} are constant within the required precision.

It leads to the final values of F_{G0} and F_{M0} obtained after applying an additional tightening above F_{BMMC}, as well as the flange rotation angles and the inside diameter of the MMC area.

5.6.4 Determination of the forces FG, FM and FB at the subsequent load conditions

5.6.4.1 General

Principle: we assume here that we know the forces at the load condition I and we determine the forces values at the load condition I + 1 from the forces balance and the 2 compliance equations given in 5.3. The compliance terms depend on the elasticity moduli E_B, E_F, E_G, E_L, E_M, E_W, and also on the effective dimensions of the gasket and the MMC area.

5.6.4.2 Initial calculation

The values of the compliance terms are determined following 5.3.

At the state I:

For the first compliance equation: $Y_{MI}{}^{i}$, $Y_{GI}{}^{i}$, $Y_{RI}{}^{i}$, $Y_{QI}{}^{i}$

For the second compliance equation: $Y_{MI}{}^{ii}$, $Y_{GI}{}^{ii}$, $Y_{RI}{}^{ii}$, $Y_{QI}{}^{ii}$

At the state I + 1:

In a first approximation, the compliance terms are calculated by considering the elasticity moduli E_B, E_F, E_M values at the state I + 1 and the effective dimensions and E_G corresponding to the state I.

For the first compliance equation: $Y_{MI+1}{}^{i}$, $Y_{GI+1}{}^{i}$, $Y_{RI+1}{}^{i}$, $Y_{QI+1}{}^{i}$

For the second compliance equation: $Y_{MI+1}{}^{ii}$, $Y_{GI+1}{}^{ii}$, $Y_{RI+1}{}^{ii}$, $Y_{QI+1}{}^{ii}$

The values F_{GI+1} and F_{MI+1} are deduced from these calculated compliance values, the values of F_{GI} and F_{MI} and the compliance equations of 5.3.1 and 5.3.2.

$$F_{MI+1} = \left(\frac{1}{Y_{MI+1}{}^{ii} - \dfrac{Y_{MI+1}{}^{i} \cdot Y_{GI+1}{}^{ii}}{Y_{GI+1}{}^{i}}} \right) \cdot \left(\begin{array}{l} Y_{GI}{}^{ii} \cdot F_{GI} + Y_{MI}{}^{ii} \cdot F_{MI} + Y_{QI}{}^{ii} \cdot F_{QI} + Y_{RI}{}^{ii} \cdot F_{RI} \\ - Y_{QI+1}{}^{ii} \cdot F_{QI+1} - Y_{RI+1}{}^{ii} \cdot F_{RI+1} - \Delta U^{ii} \\ - \dfrac{Y_{GI+1}{}^{ii}}{Y_{GI+1}{}^{i}} \cdot \left(\begin{array}{l} Y_{GI}{}^{i} \cdot F_{GI} + Y_{MI}{}^{i} \cdot F_{MI} + Y_{QI}{}^{i} \cdot F_{QI} + Y_{RI}{}^{i} \cdot F_{RI} \\ - Y_{QI+1}{}^{i} \cdot F_{QI+1} - Y_{RI+1}{}^{i} \cdot F_{RI+1} - \Delta U^{i} \end{array} \right) \end{array} \right) \qquad (106)$$

$$F_{GI+1} = \frac{1}{Y_{GI+1}{}^{i}} \cdot \left(\begin{array}{l} Y_{GI}{}^{i} \cdot F_{GI} + Y_{MI}{}^{i} \cdot F_{MI} + Y_{QI}{}^{i} \cdot F_{QI} + Y_{RI}{}^{i} \cdot F_{RI} \\ - Y_{MI+1}{}^{i} \cdot F_{MI+1} - Y_{QI+1}{}^{i} \cdot F_{QI+1} - Y_{RI+1}{}^{i} \cdot F_{RI+1} - \Delta U^{i} \end{array} \right) \qquad (107)$$

5.6.4.3 Updating of the calculation parameters

From the first approximation of F_{MI+1} and F_{GI+1} obtained in 5.6.4.2, the calculation parameters such as the effective dimensions of the gasket and the MMC area, the rotation angles of the flanges, the elasticity modulus of the gasket are updated.

— The effective dimensions of the gasket: b_{Ge}, d_{Ge}, A_{Ge} (convergence may be used depending on the gasket type). The effective dimensions of the gasket (b_{Ge}, d_{Ge}, A_{Ge}) are determined following equations described in 4.4.3

— E_G may be updated depending on the value of F_{GI+1}.

— The axial flexibility modulus of the gasket $X_{G.}$

— The lever arms: h_G, h_P, h_Q.

— The rotation angles of the flanges and the effective dimensions of the MMC.

Then the compliance terms values at the step I + 1 are updated.

From these new values of the compliance terms at the step I + 1, new values of F_{GI+1} and F_{MI+1} are calculated.

Here again, we proceed by iteration. We repeat 5.6.4.3 calculations until the values F_{GI+1} and F_{MI+1} are constant within the required precision.

It leads to the values of F_{G+1} and F_{M+1}, as well as the flange rotation angles and the inside diameter of the MMC area at the step I + 1.

At this stage of the calculation, the comparison between the inside and the outside diameters of MMC area leads to the determination of the MMC maintain at the step I + 1.

If $d_{M1e} \leq d_{M2}$, then the MMC is maintain. It means that the calculation process described in 5.6.4 is repeated to check the maintain of MMC at the next load condition.

If $d_{M1e} > d_{M2}$, then the MMC is lost at step I + 1. The calculation shall be performed again from 5.6.3 with a higher initial bolt tightening within the range [F_{BMMC}; F_{B0max}].

6 Checking of the admissibility of the load ratio

6.1 General

At this stage, we have determined the required bolt tightening to maintain the MMC at all the load conditions and to satisfy leak-tightness criteria at all the load conditions. We also dispose of all the forces values at all the load conditions.

In this clause, the load ratio on the gasket, the bolts and the flanges are calculated to verify the admissibility of the loading at every load condition.

Loads on the connection 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.

For wide flanges a more stringent requirement applies to integral flanges having $\chi = d_4 / d_0 > 2{,}0$ and loose flanges having $\chi = d_4 / d_6 > 2{,}0$ instead of $\Phi \le 1{,}0$ it shall be:

$$\Phi \le \Phi_{\max} = \mathrm{Min}\left\{1{,}0;\ 0{,}6 + 1 \Big/ \sqrt{5{,}25 + (\chi - 1)^2}\right\} \tag{108}$$

6.2 Accounting for bolt load scatter at assembly

All bolt-tightening methods involve some degree of inaccuracy. The resulting scatter value for the set of n_B bolts ε_+ and ε_- which result from this, respectively above and below the target value, are defined by Equations (109) to (111). Annex D 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 global load of all the bolts is reasonably expressed in terms of n_B, ε_{1+}, and ε_{1-} as described below.

When the systematic error due to the inaccuracy of the bolt tightening method Ks is known, the following equation defines values ε_+, and ε_- for the global load of all the bolts:

$$\varepsilon_+ = K_S + (\varepsilon_{1+} - K_S) / \sqrt{n_b} \tag{109a}$$

$$\varepsilon_- = K_S + (\varepsilon_{1-} - K_S) / \sqrt{n_b} \tag{109b}$$

When the systematic error due to the inaccuracy of the bolt tightening method Ks is not known, a reasonable approximation of Ks is given by the following equation:

$$K_S = 0{,}25 \cdot \varepsilon_{1+} \tag{110a}$$

or

$$K_S = 0{,}25 \cdot \varepsilon_{1-} \tag{110b}$$

In this case, the following equations follow:

$$\varepsilon_+ = \varepsilon_{1+} \left(1 + 3/\sqrt{n_b}\right) / 4 \tag{111a}$$

$$\varepsilon_- = \varepsilon_{1-} \left(1 + 3/\sqrt{n_b}\right) / 4 \tag{111b}$$

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

$$F_{B0\ min} \leq F_{B0} \leq F_{B0\ max} \tag{112}$$

where:

$$F_{B0\ min} = F_{B0\ av} \cdot (1 - \varepsilon_-) \tag{113}$$

$$F_{B0\ max} = F_{B0\ av} \cdot (1 + \varepsilon_+) \tag{114}$$

After assembly, the actual bolt force achieved shall be not less than $F_{B0\ req}$ the required minimum bolt force calculated to maintain the MMC and to ensure the required leak-rate in all the load conditions.

$$F_{B0\ min} \geq F_{B0\ req} \tag{115}$$

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:

— For bolt-tightening methods involving control of bolt-load :

$$F_{B0\ nom} \geq F_{B0\ req} / (1 - \varepsilon_-) \tag{116}$$

— For bolt-tightening methods involving no control of bolt-load:

the value to be selected for $F_{B0\ nom}$ is the average bolt load $F_{B0\ av}$ that can really be expected in practise for the method used, independently of $F_{B0\ req}$.

The following condition shall be met:

$$F_{B0\ nom} = F_{B0\ av} \geq F_{B0\ req} / (1 - \varepsilon_-) \text{ where } \varepsilon_- \text{ is as defined in Annex D} \tag{117}$$

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

b) Maximum forces to be used for load limit calculation:

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

$$F_{B0\ max} = F_{B0\ nom} \cdot (1 + \varepsilon_+) \tag{118}$$

$$F_{G0\ max} = F_{B0\ max} - F_{R0} \tag{119}$$

6.3 Gasket load ratio

This subclause is identical to the subclause of EN 1591-1:2001 dedicated to the gasket load ratio.

Gasket load ratio:

$$\Phi_G = F_G / (A_{Gt} \cdot Q_{max}) \leq 1 \tag{120}$$

6.4 Bolts load ratio

This subclause is identical to the subclause of EN 1591-1:2001 dedicated to the bolts load ratio.

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

Bolt load ratio:

$$\Phi_{B} = \frac{1}{f_{B}} \sqrt{\left(\frac{F_{B}}{A_{B}}\right)^{2} + 3\left(C\frac{M_{t,B}}{I_{B}}\right)^{2}} \leq 1 \tag{121}$$

where:

$$I_{B} = \left(= \frac{\pi}{12} \cdot \min\left(d_{Be}; d_{Bs}\right)^{3} \right)$$

C = 1 in assembly condition, for bolt material with minimum rupture elongation A ≥ 10 %

C = 4/3 in assembly condition, for bolt material with minimum rupture elongation A < 10 %

C = 0 in all other loading conditions

NOTE 1 In the assembly condition, the value to be considered for the twisting moment $M_{t,B}$ acting on bolt shanks is the maximum possible value, defined as:

$$M_{t,B\,max} = M_{t,B\,nom} \cdot \left(1 + \varepsilon_{+}\right)$$

$M_{t,B\,nom}$ can be determined according to Annex D (informative) of EN 1591-1:2001, for the bolting-up methods involving application of the torque to the nut.

With hydraulic tensioners, $M_{t,B}$ = 0.

NOTE 2 The value C = 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 ≥ 10 %).

The value C = 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.

NOTE 3 It is recommended to observe a minimum load ratio $\Phi_{B0\,min}$ = 0,3 in assembly condition, because smaller initial bolt load is not good practice.

6.5 Flanges load ratio

6.5.1 Integral flange and collar

Load ratio for flange, or collar (for collar Φ_{max} = 1,0):

$$\Phi_{F} = \left|F_{G} \cdot h_{G} + F_{M} \cdot h_{M} + F_{Q} \cdot \left(h_{H} - h_{P}\right) + F_{R} \cdot h_{H}\right| / W_{F} \leq \Phi_{max} \tag{122}$$

$$W_{F} = \left(\pi / 4\right) \cdot \left\{f_{F} \cdot 2 \cdot b_{F} \cdot e_{F}^{2} \cdot \left(1 + 2 \cdot \Psi_{opt} \cdot \Psi_{Z} - \Psi_{Z}^{2}\right) + f_{E} \cdot d_{E} \cdot e_{D}^{2} \times \cdot c_{M} \cdot j_{M} \cdot k_{M}\right\} \tag{123}$$

$$e_{D} = e_{1} \cdot \left\{1 + \frac{(\beta - 1) \cdot l_{H}}{\sqrt[4]{(\beta / 3)^{4} \cdot (d_{1} \cdot e_{1})^{2} + l_{H}^{4}}}\right\} \tag{124}$$

$$f_E = \min(f_F; f_S) \tag{125}$$

$$\delta_Q = P \cdot d_E / (f_E \cdot 2 \cdot e_D \cdot \cos\varphi_S); \qquad \delta_R = F_R / (f_E \cdot \pi \cdot d_E \cdot e_D \cdot \cos\varphi_S) \tag{126}$$

$$c_M = \begin{Bmatrix} \sqrt{1{,}33 \cdot [1 - 0{,}75 \cdot (0{,}5 \cdot \delta_Q + \delta_R)^2] \cdot [1 - (0{,}75 \cdot \delta_Q^2 + 1 \cdot \delta_R^2)]} \text{ for conical and cylindrical shell} \\ \sqrt{1{,}33 \cdot [1 - 0{,}75 \cdot (0{,}5 \cdot \delta_Q + \delta_R)^2] \cdot [1 - (0{,}25 \cdot \delta_Q^2 + 3 \cdot \delta_R^2)]} \text{ for spherical shell} \end{Bmatrix} \tag{127}$$

$$c_S = \begin{Bmatrix} \frac{\pi}{4} \cdot [\sqrt{1 - 0{,}75 \cdot (0{,}5 \cdot \delta_Q + \delta_R)^2} + j_S \cdot (0{,}5 \cdot \delta_R - 0{,}75 \cdot \delta_Q \text{ for conical and cylindrical shell} \\ \frac{\pi}{4} \cdot [\sqrt{1 - 0{,}75 \cdot (0{,}5 \cdot \delta_Q + \delta_R)^2} + j_S \cdot (1{,}5 \cdot \delta_R - 0{,}25 \cdot \delta_Q \text{ for spherical shell} \end{Bmatrix} \tag{128}$$

$$j_M = \text{sign}\{F_G \cdot h_G + F_M \cdot h_M + F_Q \cdot (h_H - h_P) + F_R \cdot h_H\}; \tag{129}$$

$$j_S = \pm 1 - 1 \le k_M \le +1; \quad 0 \le k_S \le 1 \tag{130}$$

$$\Psi_{(j_s, k_M, k_S)} = \frac{f_E \cdot d_E \cdot e_D \cdot \cos\varphi_S}{f_F \cdot 2 \cdot b_F \cdot e_F} \cdot \left\{ (0{,}5 \cdot \delta_Q + \delta_R) \cdot \tan\varphi_S - \delta_Q \cdot 2 \cdot e_p / d_E + j_S \cdot k_S \cdot \sqrt{\frac{e_D \cdot c_M \cdot c_S \cdot (1 + j_S \cdot k_M}{d_E \cdot \cos^3 \varphi_S}} \right\} \tag{131}$$

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

$$\Psi_{opt} = j_M \cdot (2 \cdot e_P / e_F - 1); \; (-1 \le \Psi_{opt} \le +1) \tag{132}$$

$$\begin{aligned} \Psi_{max} &= \Psi_{(+1, +1, +1)} \\ \Psi_0 &= \Psi_{(0, 0, 0)} \\ \Psi_{min} &= \Psi_{(-1, -1, +1)} \end{aligned} \tag{133}$$

The value Ψ_Z in Equation (123) 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	$\Psi_{Z\,(j_S, k_M, k_S)}$
$j_M = +1$	$\Psi_{max} \le \Psi_{opt}$	$k_M = +1$	$\Psi_Z = \Psi_{max}$
	$\Psi_0 \le \Psi_{opt} \le \Psi_{max}$	$k_M = +1$	$\Psi_Z = \Psi_{opt}$
	$\Psi_{opt} \le \Psi_0$	$k_M < +1$	$\Psi_Z = \Psi_{(-1, k_M, +1)}$
$j_M = -1$	$\Psi_{opt} \le \Psi_{min}$	$k_M = -1$	$\Psi_Z = \Psi_{min}$
	$\Psi_{min} \le \Psi_{opt} \le \Psi_0$	$k_M = -1$	$\Psi_Z = \Psi_{opt}$
	$\Psi_0 \le \Psi_{opt}$	$k_M > -1$	$\Psi_Z = \Psi_{(+1, k_M, +1)}$

The sequence of calculation shall be as follows:

a) Calculate e_D from Equation (124), β having previously been calculated by Equation (9).

b) Calculate f_E, δ_Q, δ_R, c_M from Equations (125) to (127).

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

c) Calculate $c_{S\,(j\,=\,+\,1)}$, $c_{S\,(js\,=\,-\,1)}$, j_M, Ψ_{opt}, Ψ_0, Ψ_{max}, Ψ_{min} from Equations (128), (129), (131), (132), (133).

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

d) Determine k_M and Ψ_Z according to Table 2. When that table gives $k_M < +1$ or $k_M > -1$ or k_M without no more precision, the value of k_M shall be determined so that W_F is maximum in Equation (123) as calculated at step e) which follows. The value of Ψ_Z associated with k_M is given by Equation (131).

e) Calculate W_F, Φ_F from Equations (123), (122).

6.5.2 Loose flange

$$\Phi_L = F_B \cdot h_L / W_L \le \Phi_{max} \tag{134}$$

$$W_L = (\pi / 2) \cdot f_L \cdot b_L \cdot e_L^2 \tag{135}$$

6.5.3 Blank flange

Load ratio for blank flange:

$$\Phi_F = \max \left\{ \begin{matrix} \left| \left(F_G + F_Q + F_R\right) \cdot h_G + F_M \cdot h_M + F_Q \cdot \left(1 - \rho^3\right) \cdot d_{Ge} / 6 + F_R \cdot (1 - \rho) \cdot d_{Ge} / 2 \right|; \\ \left| \left(F_G + F_Q + F_R\right) \cdot h_G + F_M \cdot h_M + F_Q \cdot \left(1 - \rho^3\right) \cdot d_{Ge} / 6 \right|; \left| F_R \cdot (1 - \rho) \cdot d_{Ge} / 2 \right| \end{matrix} \right\} / W_F \le 1 \tag{136}$$

$$W_F = (\pi / 4) \cdot f_F \cdot \left\{ 2 \cdot b_F \cdot e_F^2 + d_0 \cdot (1 - \rho) \cdot e_0^2 \right\} \tag{137}$$

If there is a possible critical section where $e_X < e_F$ (see Figure 9 of EN 1591-1:2001), then calculate additionally the following load ratio:

$$\Phi_X = F_B \cdot (d_3 - d_X) / (2 \cdot W_X) \le 1 \tag{138}$$

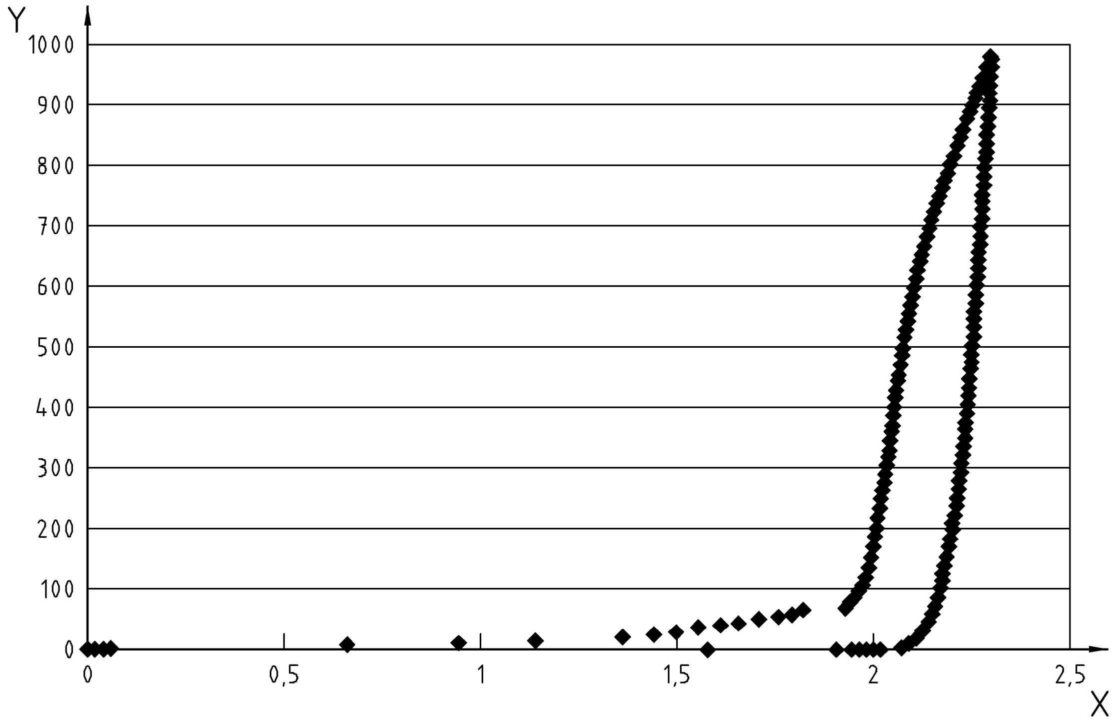
$$W_X = (\pi / 4) \cdot f_F \cdot \left\{ (d_4 - 2 \cdot d_{5e} - d_X) \cdot e_F^2 + d_X \cdot e_X^2 \right\} \tag{139}$$

Annex A
(informative)

Example of Gasket compression curve

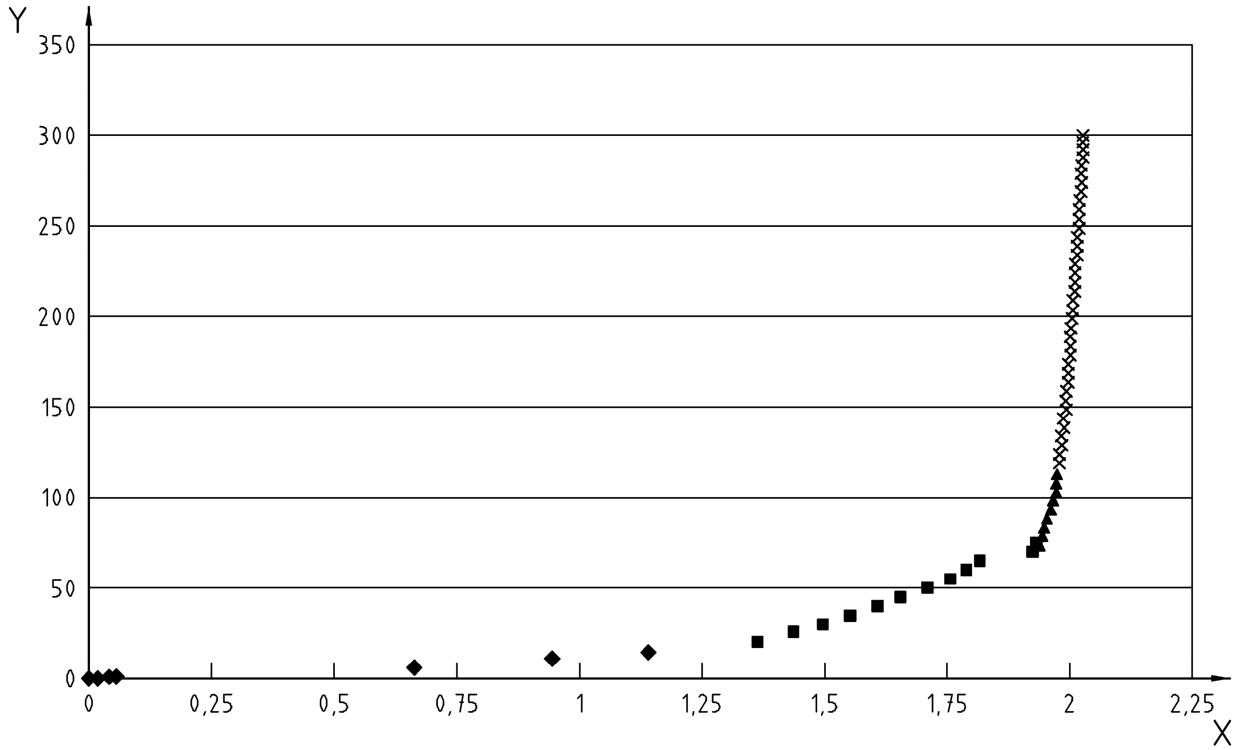
An example of linear approximation of the gasket compression curve is given here below.

Figure A.1 represents the raw data of a compression test performed on a covered metal jacketed gasket (graphite covering, stainless steel jacket and graphite filler) fitted with an outer metallic ring.



Key
X gasket deflection (mm)
Y gasket compression stress (MPa)

Figure A.1 — Example of compression curve obtained by test: Covered metal jacketed gasket with outer ring



Key

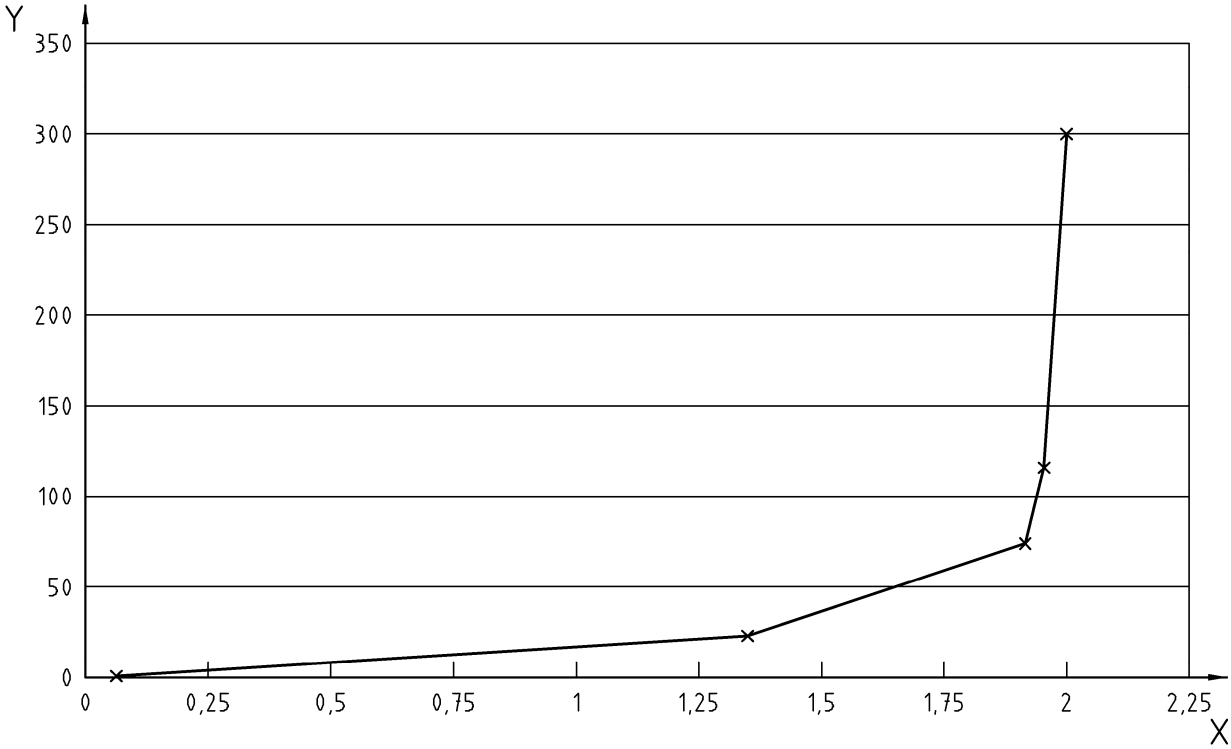
X gasket deflection (mm)

Y gasket compression stress (MPa)

Figure A.2 — Compression curve up to 300 MPa decomposed in four parts

In Figure A.2, the compression curve represented between 1 MPa and 300 MPa is decomposed in four parts.

Figure A.3 shows the approximation by linear parts of the gasket compression curve between 1 MPa and 300 MPa.



Key

X gasket deflection (mm)

Y gasket compression stress (MPa)

Figure A.3 — Approximation by linear parts of the compression curve

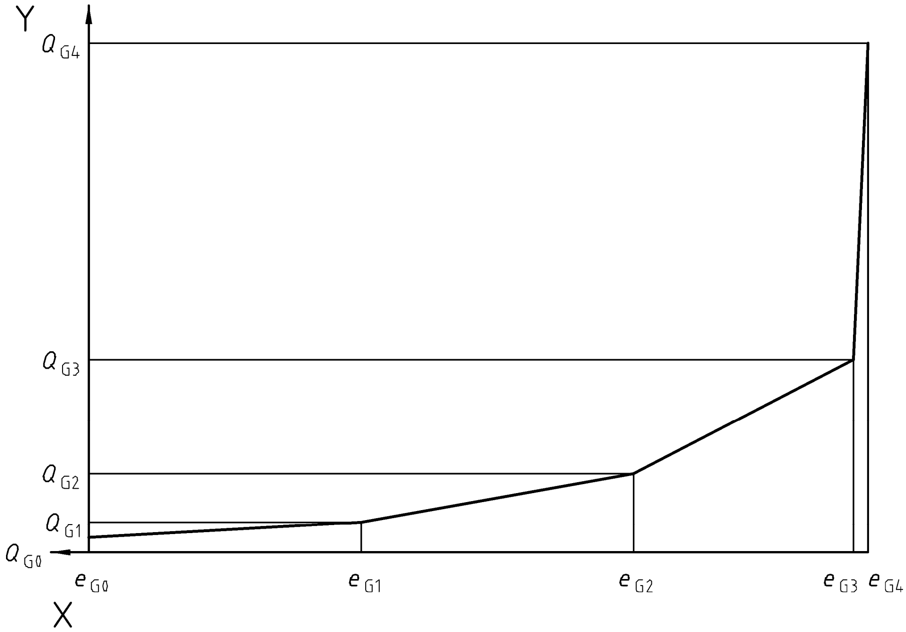
The behaviour of the gasket in compression is defined by couples (Q_{Gj}, e_{Gj}) with $0 \leq j \leq n$. As an example, the compression curve in Figure A.3 is approximated by four linear parts thanks to five couples (Q_{Gj}, e_{Gj}).

Such as for $Q_{Gj-1} \leq Q_G \leq Q_{Gj}$, the thickness of the gasket in compression for a gasket compression stress Q_G is given by:

$$e_G\left(Q_G\right) = e_{Gj-1} + \left(Q_G - Q_{Gj-1}\right) \cdot \frac{(e_{Gj} - e_{Gj-1})}{(Q_{Gj} - Q_{Gj-1})}$$

The definition of couples (Q_{Gj}, e_{Gj}) depends on the gasket type, dimensions and MMC configuration.

The compression curve obtained by test (EN 13555) to determine Qcrit (with the right platens configuration) may be used to obtain the couples (Q_{Gi}, e_{Gi}) required to define the behaviour of the gasket in compression.



Key
X gasket deflection (mm)
Y gasket compression stress (MPa)

Figure A.4 — Example of linear approximation of the gasket behaviour in compression showing couples (Q_{Gj}, e_{Gj})

Annex B
(informative)

Local compression deformation

B.1 Axial flexibility modulus in the case of local compression area

Local compression phenomena may be not negligible in some calculation cases. Local compression phenomena may occur on:

The flange (or collar) at the interface with the gasket.

The flange (or collar) at the interface with the metal to metal contact area.

The collar at the interface with the loose flange.

The loose flange at the interface with the collar.

The flange (or loose flange) at the interface with the nut.

Local compression phenomena are considered in the calculation with a model of a rectangular ring cross section submitted to compression on both the upper and lower faces (see Figures B.1 to B.3 below). The effect of the compression is assumed to propagate within the ring thickness with a 45° angle (η = 1).

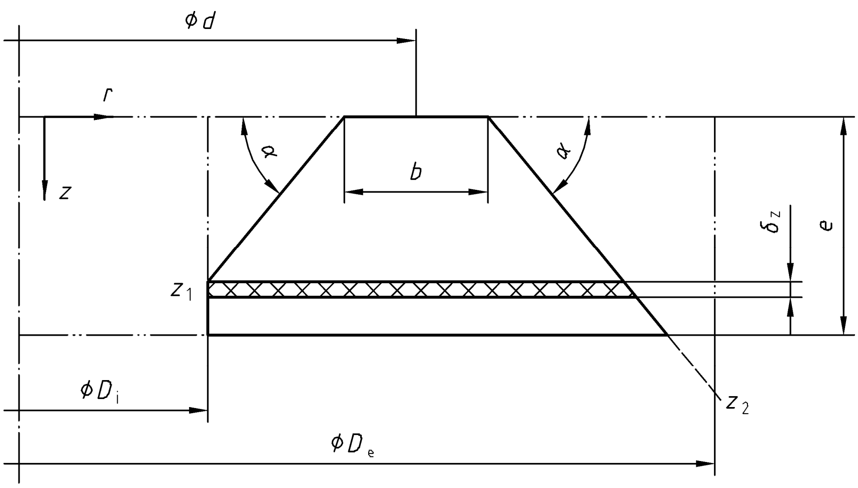


Figure B.1 — Schematic representation of a cross section where local compression occurs

$$\Delta e = \int_0^e \Delta\varepsilon(z) \quad dz = \int_0^e \frac{F}{S(z) \cdot E} \quad dz \Rightarrow X = \frac{\Delta e \cdot E}{F} = \int_0^e \frac{dz}{S(z)} \qquad \text{(B.1)}$$

According to the geometrical configuration, there are five different expressions of $S(z)$, and so, five expressions of the flexibility parameter X.

Let write $\eta = \tan\alpha$

1) $z_1 \geq e \quad z_2 \geq e$

$$X = \frac{\eta}{2\pi d} \cdot \ln\left(1 + \frac{2e}{b \cdot \eta}\right) \tag{B.2}$$

2) $z_1 \geq e \quad z_2 < e$

$$X = \frac{\eta}{2\pi d} \cdot \ln\left(\frac{D_\mathrm{e} - d}{b}\right) + \frac{\eta}{\pi\, De} \cdot \ln\left(\frac{\dfrac{-2e}{\eta^2} + \dfrac{d - b - D_\mathrm{e}}{\eta}}{\dfrac{-2e}{\eta^2} + \dfrac{d - b + D_\mathrm{e}}{\eta}} \cdot \frac{d}{d - D_\mathrm{e}}\right) \tag{B.3}$$

3) $z_1 < e \quad z_2 \geq e$

$$X = \frac{\eta}{2\pi d} \cdot \ln\left(\frac{d - D_\mathrm{i}}{b}\right) + \frac{\eta}{\pi D_\mathrm{i}} \cdot \ln\left(\frac{\dfrac{2e}{\eta^2} + \dfrac{d + b - D_\mathrm{i}}{\eta}}{\dfrac{2e}{\eta^2} + \dfrac{d + b + D_\mathrm{i}}{\eta}} \cdot \frac{d}{d - D_\mathrm{i}}\right) \tag{B.4}$$

4) $z_1 < e \quad z_2 < e \quad z_1 \leq z_2$

$$X = \frac{\eta}{2\pi d} \cdot \ln\left(\frac{d - D_\mathrm{i}}{b}\right) + \frac{\eta}{\pi D_\mathrm{i}} \cdot \ln\left(\frac{D_\mathrm{e} - D_\mathrm{i}}{D_\mathrm{e} + D_\mathrm{i}} \cdot \frac{d}{d - D_\mathrm{i}}\right) + \frac{4 \cdot (e - z_2)}{\pi\,(D_\mathrm{e}^{\,2} - D_\mathrm{i}^{\,2})} \tag{B.5}$$

5) $z_1 < e \quad z_2 < e \quad z_2 \leq z_1$

$$X = \frac{\eta}{2\pi d} \cdot \ln\left(\frac{D_\mathrm{e} - d}{b}\right) + \frac{\eta}{\pi D_\mathrm{e}} \cdot \ln\left(\frac{D_\mathrm{i} - D_\mathrm{e}}{D_\mathrm{i} + D_\mathrm{e}} \cdot \frac{d}{d - D_\mathrm{e}}\right) + \frac{4 \cdot (e - z_1)}{\pi\,(D_\mathrm{e}^{\,2} - D_\mathrm{i}^{\,2})} \tag{B.6}$$

The expressions of X in the different local compression area are obtained by adapting the geometrical variables to the considered contact area.

Annex C
(informative)

Relaxation of the gasket: Three-parameter solid model

C.1 Constitutive equation for one-dimensional response of viscoelastic materials

We use here the mechanical analogy to give the constitutive equation for one-dimensional response of a viscoelastic material.

Mechanical analogy:

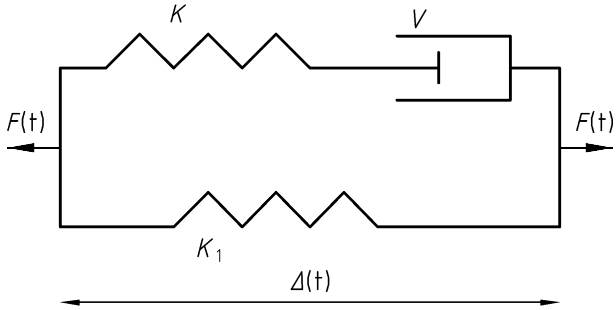


Figure C.1 — Schematic representation of the mechanical analogy

It consists in a Maxwell model (linear spring and linear viscous damper in series) and a linear spring associated in parallels.

In Annex C, quantities associated with the Maxwell element are denoted by a subscript M and with the spring by a subscript S.

Force analysis:

$$F(t) = F_{\mathrm{M}}(t) + F_{\mathrm{S}}(t) \tag{C.1}$$

Geometry:

$$\Delta(t) = \Delta_{\mathrm{M}}(t) = \Delta_{\mathrm{S}}(t) \tag{C.2}$$

Force-elongation relations:

$$F_{\mathrm{S}}(t) = K_1 \cdot \Delta_{\mathrm{S}}(t) \tag{C.3}$$

$$\dot{\Delta}_{\mathrm{M}} = \frac{\dot{F}_{\mathrm{M}}}{K} + \frac{F_{\mathrm{M}}}{V} \tag{C.4}$$

It leads to:

$$\frac{1}{K} \cdot \dot{F} + \frac{1}{V} \cdot F = \left(1 + \frac{K_1}{K}\right) \cdot \dot{\Delta} + \frac{K_1}{V} \cdot \Delta \tag{C.5}$$

In terms of stress and strain, it leads to the following constitutive equation:

$$\omega_0 \cdot \sigma + \omega_1 \cdot \dot{\sigma} = \xi_0 \cdot \varepsilon + \xi_1 \cdot \dot{\varepsilon} \tag{C.6}$$

where:

$$\omega_0 = \frac{1}{\mu}; \quad \omega_1 = \frac{1}{E}; \quad \xi_0 = \frac{E_1}{\mu}; \quad \xi_1 = \left(1 + \frac{E_1}{E}\right) \tag{C.7}$$

E and E_1 in N.mm^{-2}

μ in N · mm^{-2} · s

By application of the Laplace transform to the constitutive equation, we obtain:

$$(\omega_0 + \omega_1 \cdot a) \cdot \overline{\sigma} = (\xi_0 + \xi_1 \cdot a) \cdot \overline{\varepsilon} \tag{C.8}$$

Stress relaxation response:

Let $\varepsilon(t) = \varepsilon_0 \,, t \geq 0$ then:

$$\overline{\varepsilon}(a) = \varepsilon_0 \cdot \frac{1}{a} \tag{C.9}$$

The Laplace transform of the stress history is then given by:

$$\overline{\sigma}(a) = \frac{\varepsilon_0}{a} \cdot \frac{\xi_0 + \xi_1 \cdot a}{\omega_0 + \omega_1 \cdot a} \tag{C.10}$$

which is also written:

$$\overline{\sigma}(a) = \varepsilon_0 \cdot \left[\frac{\xi_0}{\omega_0} \cdot \frac{1}{a} + \left(\frac{\xi_1}{\omega_1} - \frac{\xi_0}{\omega_0} \right) \cdot \frac{1}{a + \frac{\omega_0}{\omega_1}} \right] \tag{C.11}$$

It leads to the expression of the stress history corresponding to the step strain history:

$$\sigma(t) = \varepsilon_0 \left[\frac{\xi_0}{\omega_0} + \left(\frac{\xi_1}{\omega_1} - \frac{\xi_0}{\omega_0} \right) \cdot e^{-\left(\frac{\omega_0}{\omega_1}\right) \cdot t} \right] \tag{C.12}$$

It may also be written:

$$G(t) = \frac{\sigma(t)}{\varepsilon_0} = G_\infty + (G_0 - G_\infty) \cdot e^{-\frac{t}{\tau_R}} \tag{C.13}$$

with:

$$G_\infty = \frac{\xi_0}{\omega_0}; \quad G_0 = \frac{\xi_1}{\omega_1}; \quad \tau_R = \frac{\omega_1}{\omega_0} \tag{C.14}$$

$G(t)$ is called relaxation function.

C.2 Consideration of the Temperature effect

Here, the theoretical model known as “time-temperature superposition” is used. It describes viscoelastic behaviour with respect to time and temperature.

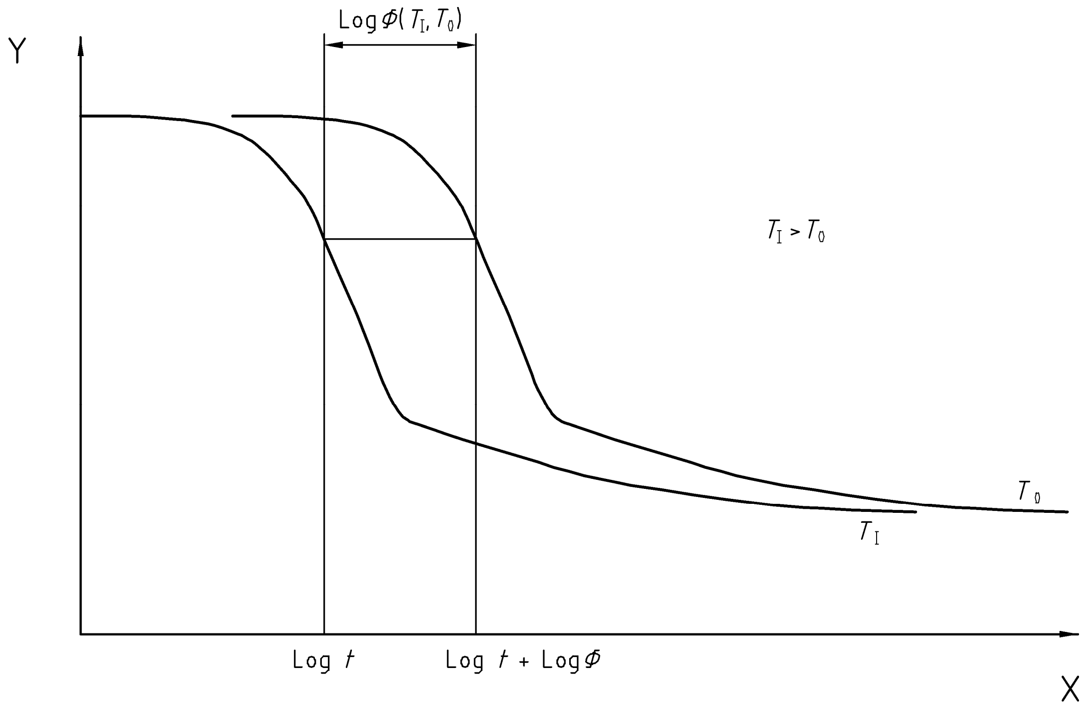
We consider here the expression of the relaxation function as a function of log t:

$$G(t, T) = E(\log t, T) \tag{C.15}$$

$E(\log t, T_0$represents the relaxation function at temperature $T = T_0$

$E(\log t, T_I)$ represents the relaxation function at temperature $T = T_I$

Assuming that $E(\log t, T_0)$ is known, $E(\log t, T_I)$ is obtained by shifting the $E(\log t, T_0)$ curve to the left by an amount of log $\Phi(T_I, T_0)$ (see figure here below).



Key
X log (time)
Y relaxation function

Figure C.2 — Curves of $E(\log t, T)$ for two different temperatures T_0 and T_I

$$G(t, T_I) = G(\phi(T_I, T_0) \cdot t, T_0) \tag{C.16}$$

Which is also commonly expressed by:

$$G(t, T_I) = G\left(\frac{t}{a(T_I, T_0)}, T_0\right) \tag{C.17}$$

$a(T_I, T_0)$ is called “Shift function”.

The expression of the relaxation function for temperature T_I is:

$$G(t, T_\mathrm{I}) = \frac{\sigma(t)}{\varepsilon_0} = G_\infty + (G_0 - G_\infty) \cdot e^{-\frac{t}{\tau_\mathrm{R}(T_\mathrm{I})}} \qquad \text{(C.18)}$$

with:

$$\tau_\mathrm{R}(T_\mathrm{I}) = a(T_\mathrm{I}, T_0) \cdot \tau_\mathrm{R}(T_0) \qquad \text{(C.19)}$$

a (T_I, T_0) is determined by using the time-temperature superposition principle. Several stress relaxation tests performed at different levels of temperatures are required to obtained the “shift function” (see reference [1]).

Annex D
(informative)

Scatter of bolting-up methods

Table D.1 — Indicative values ε_{1-} and ε_{1+} (6.2) 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	$0,3 + 0,5\ \mu$	$0,3 + 0,5\ \mu$
Impact wrench	Friction, Stiffness, Calibration	$0,2 + 0,5\ \mu$	$0,2 + 0,5\ \mu$
Torque wrench = Wrench with measuring of torque (only)	Friction, Calibration, Lubrication	$0,1 + 0,5\ \mu$	$0,1 + 0,5\ \mu$
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, 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 6.2.

[c] With hydraulic tensioner, ε_{1+} and 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.

Annex E
(informative)

Calculation sequences

Step 1: Determination of lever arms (except those depending on dGe and dMe) and flexibility of the flanges.

(See 4.2)

b_F, d_F, e_F (and b_L, d_L, e_L) and $\tilde{b}_F$, $\tilde{d}_F$, $\tilde{e}_F$ (and $\tilde{b}_L$, $\tilde{d}_L$, $\tilde{e}_L$)

d_{3e} ($\tilde{d}_{3e} = d_{3e}$)

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

h_H, h_L, h_R, h_Q and $\tilde{h}_H$, $\tilde{h}_L$, $\tilde{h}_R$, $\tilde{h}_Q$

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

Step 2: Determination of the flexibility of the bolts

(See 4.3)

A_B, X_B

Step 3: Determination of the theoretical dimensions of the gasket

(See 4.4.2)

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

Step 4: Determination of the appearance of MMC in assembly conditions

(See 5.5)

Step 4.1: determination of the range [Q_{Gj-1}; Q_{Gj}] **where the MMC appears**

— determination of the force F_{Gj} to apply a given gasket surface pressure Q_{Gj} (see 5.5.2).

F_G, b_{Ge}, d_{Ge}, A_{Ge}

— determination of the gasket thickness, gasket flexibility and lever arms depending on d_{Ge} (see 5.5.2).

e_G, h_G, $\tilde{h}_G$, X_G

— determination of the local compression (if required) and the rotation angles (see 5.5.2).

Δe_{FG}, $\Delta\tilde{e}_{FG}$, θ_F, $\tilde{\theta}_F$, θ_L, $\tilde{\theta}_L$

— determination of d_{M1e} (4.6.2.1)

— check of the MMC condition:

$d_{M1e} \leq d_{M2}$: Yes the MMC is reached and [Q_{Gj-1}; Q_{Gj}] is the range where the MMC appears

$d_{M1e} > d_{M2}$: no, the MMC is not reached when applying a gasket surface pressure equal to Q_{Gj}. The step 4.1 must be repeated with a higher gasket surface pressure.

Step 4.2: determination of the force F_{GMMC} for which the MMC appears

(See 5.5.3)

Step 5: determination of the forces F_{G0} & F_{M0} after applying a given initial tightening above F_{BMMC}:

(See 5.6.2 & 5.6.3):

d_{Me}, h_M

e_W, X_W

$\theta_F, \tilde{\theta}_F, \theta_L, \tilde{\theta}_L$

$\Delta e_{FG}, \Delta\tilde{e}_{FG}$

$X_{FG}, X_{FM}, X_{FL}, X_{LF}, X_{LB}, \tilde{X}_{FG}, \tilde{X}_{FM}, \tilde{X}_{FL}, \tilde{X}_{LF}, \tilde{X}_{LB}$

$Y_{G0}^{ii}, Y_{M0}^{ii}, Y_{R0}^{ii}$

F_{G0}, F_{M0} first approximation

Calculation:

$b_{Ge}, d_{Ge}, A_{Ge}, e_G, h_G, \tilde{h}_G, X_G$

e_W, X_W

$\theta_F, \tilde{\theta}_F, \theta_L, \tilde{\theta}_L$

$\Delta e_{FG}, \Delta\tilde{e}_{FG}$

d_{M1e}, d_{Me}, h_M

$X_{FG}, X_{FM}, X_{FL}, X_{LF}, X_{LB}, \tilde{X}_{FG}, \tilde{X}_{FM}, \tilde{X}_{FL}, \tilde{X}_{LF}, \tilde{X}_{LB}$

$Y_{G0}^{ii}, Y_{M0}^{ii}, Y_{R0}^{ii}$

F_{G0}, F_{M0}

Iteration is repeated until the required accuracy is reached on F_{G0} and F_{M0}.

Step 6: Determination of internal forces at the load condition I + 1, knowing the internal forces at the load condition I.

$Y_{\text{MI}+1}{}^{\text{i}}, Y_{\text{GI}+1}{}^{\text{i}}, Y_{\text{RI}+1}{}^{\text{i}}, Y_{\text{QI}+1}{}^{\text{i}}, Y_{\text{MI}+1}{}^{\text{ii}}, Y_{\text{GI}+1}{}^{\text{ii}}, Y_{\text{RI}+1}{}^{\text{ii}}, Y_{\text{QI}+1}{}^{\text{ii}}$

$F_{\text{GI}+1}, F_{\text{MI}+1}$ (first approximation)

$b_{\text{Ge}}, d_{\text{Ge}}, A_{\text{Ge}}, e_{\text{G}}, h_{\text{G}}, \widetilde{h}_{\text{G}}, X_{\text{G}}, E_{\text{G}}$

$e_{\text{W}}, X_{\text{W}}$

$\theta_{\text{F}}, \widetilde{\theta}_{\text{F}}, \theta_{\text{L}}, \widetilde{\theta}_{\text{L}}$

$\Delta e_{\text{FG}}, \Delta\widetilde{e}_{\text{FG}}$

$d_{\text{M1e}}, d_{\text{Me}}, h_{\text{M}}$

$X_{\text{FG}}, X_{\text{FM}}, X_{\text{FL}}, X_{\text{LF}}, X_{\text{LB}}, \widetilde{X}_{\text{FG}}, \widetilde{X}_{\text{FM}}, \widetilde{X}_{\text{FL}}, \widetilde{X}_{\text{LF}}, \widetilde{X}_{\text{LB}}$

$Y_{\text{MI}+1}{}^{\text{i}}, Y_{\text{GI}+1}{}^{\text{i}}, Y_{\text{RI}+1}{}^{\text{i}}, Y_{\text{QI}+1}{}^{\text{i}}, Y_{\text{MI}+1}{}^{\text{ii}}, Y_{\text{GI}+1}{}^{\text{ii}}, Y_{\text{RI}+1}{}^{\text{ii}}, Y_{\text{QI}+1}{}^{\text{ii}}$

$F_{\text{GI}+1}, F_{\text{MI}+1}, F_{\text{BI}+1}$

Iteration is repeated until the required accuracy is reached on $F_{\text{GI}+1}$, $F_{\text{MI}+1}$ and $F_{\text{BI}+1}$.

This step is repeated for all the load conditions.

After the calculation of $F_{\text{GI}+1}$, $F_{\text{MI}+1}$ and $F_{\text{BI}+1}$ at the load condition I + 1, in case of increase of thermal or mechanical load on the gasket, the step 7 must be achieved.

The MMC is checked to be maintained after each calculation condition.

The leak-tightness criteria are checked after each calculation condition.

At the end of this step,

If the MMC is maintained and the leak-tightness criteria respected at every calculation conditions, then the initial tightening can be decreased (but the tightening must still remain within the range [F_{BMMC}; $F_{\text{G0 max}}$]) and the step 5 must be repeated

If the MMC is lost in one of the calculation conditions, the calculation can be stopped, the MMC cannot be maintained for the defined bolted joint and the design of this joint must be revised.

If the leak-tightness criteria are not respected in one of the calculation conditions, the calculation can be stopped, the leak-tightness criteria cannot be respected for the defined bolted joint and the design of this joint must be revised.

Step 7: Determination of the internal forces after relaxation of the gasket

$F_{\mathrm{GI}+1}$ after relaxation

$b_{\mathrm{Ge}}, d_{\mathrm{Ge}}, A_{\mathrm{Ge}}, e_{\mathrm{G}}, h_{\mathrm{G}}, \widetilde{h}_{\mathrm{G}}, X_{\mathrm{G}}, E_{\mathrm{G}}$

$e_{\mathrm{W}}, X_{\mathrm{W}}$

$\theta_{\mathrm{F}}, \widetilde{\theta}_{\mathrm{F}}, \theta_{\mathrm{L}}, \widetilde{\theta}_{\mathrm{L}}$

$\Delta e_{\mathrm{FG}}, \Delta\widetilde{e}_{\mathrm{FG}}$

$d_{\mathrm{M1e}}, d_{\mathrm{Me}}, h_{\mathrm{M}}$

$X_{\mathrm{FG}}, X_{\mathrm{FM}}, X_{\mathrm{FL}}, X_{\mathrm{LF}}, X_{\mathrm{LB}}, \widetilde{X}_{\mathrm{FG}}, \widetilde{X}_{\mathrm{FM}}, \widetilde{X}_{\mathrm{FL}}, \widetilde{X}_{\mathrm{LF}}, \widetilde{X}_{\mathrm{LB}}$

$Y_{\mathrm{MI}+1}{}^{\mathrm{i}}, Y_{\mathrm{GI}+1}{}^{\mathrm{i}}, Y_{\mathrm{RI}+1}{}^{\mathrm{i}}, Y_{\mathrm{QI}+1}{}^{\mathrm{i}}, Y_{\mathrm{MI}+1}{}^{\mathrm{ii}}, Y_{\mathrm{GI}+1}{}^{\mathrm{ii}}, Y_{\mathrm{RI}+1}{}^{\mathrm{ii}}, Y_{\mathrm{QI}+1}{}^{\mathrm{ii}}$

$F_{\mathrm{MI}+1}, F_{\mathrm{BI}+1}$

Iteration is repeated until the required accuracy is reached on $F_{\mathrm{MI}+1}$

Step 8: Determination of the load ratio

(See Clause 6)

$F_{\mathrm{B0\,nom}}, F_{\mathrm{B0\,max}}$

from these values, we repeat the whole calculations from step 5 in order to determine the internal forces to be considered in the load ratio calculation at all the load conditions.

At all the load conditions:

$\Phi_{\mathrm{B}}, \Phi_{\mathrm{G}}, \Phi_{\mathrm{F}}, \Phi_{\mathrm{L}}, \widetilde{\Phi}_{\mathrm{F}}, \widetilde{\Phi}_{\mathrm{L}}, (\Phi_{\mathrm{X}}, \widetilde{\Phi}_{\mathrm{X}})$

Annex F
(informative)

Determination of the compliance equations

The consideration of the MMC leads to the determination of the reaction force on the MMC area F_{M}. It means that an additional compliance equation is required.

To determine F_{B}, F_{G} and F_{M}, we dispose of the 3 following equations:

— Forces balance:

$$F_{\mathrm{B}} = F_{\mathrm{G}} + F_{\mathrm{M}} + F_{\mathrm{Q}} + F_{\mathrm{R}} \quad \text{(F.1)}$$

The first deformation compatibility equation:

$$\begin{aligned} \Delta l_{\mathrm{B}}^{\mathrm{M}} + \Delta l_{\mathrm{B}}^{\mathrm{T}} = {} & \Delta e_{\mathrm{W}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{W}}^{\mathrm{M}} + \Delta e_{\mathrm{W}}^{\mathrm{T}} + \Delta \tilde{e}_{\mathrm{W}}^{\mathrm{T}} + \Delta e_{\mathrm{LB}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{LB}}^{\mathrm{M}} - \Delta \theta_{\mathrm{L}}^{\mathrm{M}} \cdot h_{\mathrm{L}} - \Delta \tilde{\theta}_{\mathrm{L}}^{\mathrm{M}} \cdot \tilde{h}_{\mathrm{L}} + \Delta e_{\mathrm{L}}^{\mathrm{T}} + \Delta \tilde{e}_{\mathrm{L}}^{\mathrm{T}} \\ & + \Delta e_{\mathrm{LF}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{LF}}^{\mathrm{M}} + \Delta e_{\mathrm{FL}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{FL}}^{\mathrm{M}} - \Delta \theta_{\mathrm{F}}^{\mathrm{M}} \cdot h_{\mathrm{G}} - \Delta \tilde{\theta}_{\mathrm{F}}^{\mathrm{M}} \cdot \tilde{h}_{\mathrm{G}} + \Delta e_{\mathrm{Ft}}^{\mathrm{T}} + \Delta \tilde{e}_{\mathrm{Ft}}^{\mathrm{T}} + \Delta e_{\mathrm{FG}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{FG}}^{\mathrm{M}} \\ & + \Delta e_{\mathrm{G}}^{\mathrm{M}} + \Delta e_{\mathrm{G}}^{\mathrm{T}} \end{aligned} \quad \text{(F.2)}$$

Second deformation compatibility equation:

$$\begin{aligned} & \Delta \theta_{\mathrm{F}}^{\mathrm{M}} \cdot \frac{d_{\mathrm{Ge}}}{2} + \Delta \tilde{\theta}_{\mathrm{F}}^{\mathrm{M}} \cdot \frac{\tilde{d}_{\mathrm{Ge}}}{2} + \Delta e_{\mathrm{Ft}}^{\mathrm{T}} + \Delta \tilde{e}_{\mathrm{Ft}}^{\mathrm{T}} + \Delta e_{\mathrm{FG}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{FG}}^{\mathrm{M}} + \Delta e_{\mathrm{G}}^{\mathrm{M}} + \Delta e_{\mathrm{G}}^{\mathrm{T}} = \\ & \quad \Delta \theta_{\mathrm{F}}^{\mathrm{M}} \cdot \frac{d_{\mathrm{Me}}}{2} + \Delta \tilde{\theta}_{\mathrm{F}}^{\mathrm{M}} \cdot \frac{d_{\mathrm{Me}}}{2} + \Delta e_{\mathrm{Fm}}^{\mathrm{T}} + \Delta \tilde{e}_{\mathrm{Fm}}^{\mathrm{T}} + \Delta e_{\mathrm{FM}}^{\mathrm{M}} + \Delta \tilde{e}_{\mathrm{FM}}^{\mathrm{M}} + \Delta e_{\mathrm{M}}^{\mathrm{M}} + \Delta e_{\mathrm{M}}^{\mathrm{T}} \end{aligned} \quad \text{(F.3)}$$

All the deformation terms are written as a function of the corresponding forces in the 2 deformation compatibility equations here above in order to obtain the 2 compliance equation given here below.

First compliance equation

The first compliance equation is determined from the forces balance and the first deformation compatibility equation:

$$\Delta\left(Y_{\mathrm{G}}^{\mathrm{i}} \cdot F_{\mathrm{G}}\right) + \Delta\left(Y_{\mathrm{M}}^{\mathrm{i}} \cdot F_{\mathrm{M}}\right) + \Delta\left(Y_{\mathrm{Q}}^{\mathrm{i}} \cdot F_{\mathrm{Q}}\right) + \Delta\left(Y_{\mathrm{R}}^{\mathrm{i}} \cdot F_{\mathrm{R}}\right) + \Delta U^{\mathrm{i}} = 0 \quad \text{(F.4)}$$

Second compliance equation

The second compliance equation is determined from the forces balance and the second deformation compatibility equation:

$$\Delta\left(Y_{\mathrm{G}}^{\mathrm{ii}} \cdot F_{\mathrm{G}}\right) + \Delta\left(Y_{\mathrm{M}}^{\mathrm{ii}} \cdot F_{\mathrm{M}}\right) + \Delta\left(Y_{\mathrm{Q}}^{\mathrm{ii}} \cdot F_{\mathrm{Q}}\right) + \Delta\left(Y_{\mathrm{R}}^{\mathrm{ii}} \cdot F_{\mathrm{R}}\right) + \Delta U^{\mathrm{ii}} = 0 \quad \text{(F.5)}$$

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