PD CEN/TR 1591-5:2012

BSI Standards Publication

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

Part 5: Calculation method for full face gasketed joints

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National foreword

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

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

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ISBN 978 0 580 72952 2

ICS 23.040.60

Compliance with a British Standard cannot confer immunity from legal obligations.

This Published Document was published under the authority of the Standards Policy and Strategy Committee on 29 February 2012.

Amendments issued since publication

Date	Text affected

TECHNICAL REPORT

RAPPORT TECHNIQUE

TECHNISCHER BERICHT

CEN/TR 1591-5

February 2012

ICS 23.040.60

English Version

Flanges and their joints - Design rules for gasketed circular flange connections - Part 5: Calculation method for full face gasketed joints

Brides et leurs assemblages - Règles de calcul des assemblages à brides circulaires avec joint - Partie 5: Méthode de calcul pour assemblages avec joints pleine face

Flansche und ihre Verbindungen - Regeln für die Auslegung von Flanschverbindungen mit runden Flanschen und Dichtung - Teil 5: Berechnungsmethode für Verbindungen mit vollflächiger Dichtung

This Technical Report was approved by CEN on 12 December 2011. It has been drawn up by the Technical Committee CEN/TC 74.

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Ref. No. CEN/TR 1591-5:2012: E

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Foreword

This document (CEN/TR 1591-5:2012) has been prepared by Technical Committee CEN/TC 74 “Flanges and their joints”, the secretariat of which is held by DIN.

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

EN 1591 "*Flanges and their joints — Design rules for gasketed circular flange connections*" consists of the following 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;*

— *Part 5: Calculation method for full face gasketed joints* (CEN/TR).

1 Scope

This Technical Report gives guidance for the calculation of full face gasketed joints on the basis of the calculation method given in EN 1591-1.

2 Normative references

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

EN 1591-1:2001+A1:2009, *Flanges and their joints — Design rules for gasketed circular flange connections — Part 1: Calculation method*

3 Symbols and abbreviated terms

A_{Ge} effective gasket area (= $\pi * d_{Ge} * b_{Ge}$), [mm^2], see Equation (26)

b_{Ge} effective gasket width, (mm), see Figure 2

b_{Gi} interim value of effective gasket width, [mm]

b_{Gseal} effective sealing gasket width, [mm], Figure 9

b_{GQ} compressed gasket width, [mm], Figure 2

d_{F1} gasket force acting diameter for zone A, Equations (12), (14)

d_{F2}	gasket force acting diameter for zone B, Equations (7), (10)
d_{F3}	resultant gasket force acting diameter on outside area of the real gasket, [mm], Equation (16)
d_{F4}	resultant gasket force acting diameter on outside area of the equivalent gasket, [mm], Equation (19)
d_{G1}	inside diameter of gasket theoretical contact area, [mm], Figure 1
d_{G2}	outside diameter of gasket theoretical contact area, [mm], Figure 1
d_{G3}	outside diameter of bolt holes part for the equivalent gasket, [mm], Figure 1
d_{G4}	outside diameter of equivalent gasket, [mm], Figure 1
d_{Gi}	interim value of effective gasket diameter, [mm], Equation (25)
d_{Ge}	effective gasket diameter, [mm], Figure 2
d_3	real bolt circle diameter, [mm], Figure 1
d_{3e}	effective bolt circle diameter, [mm]
d_4	outside diameter of flange, [mm]
d_5	diameter of bolt hole, [mm], Figure 1
F_{Gmin}	minimum gasket force, [N], Equations (28), (29), (30), (31)
F_{Greal}	force on the real gasket, [N]
F_{Gequi}	force on the equivalent gasket, [N]
F_{GzoneA}	gasket force on the bolt holes zone A, [N], Equation (5), Figure 7
F_{GzoneB}	gasket force on zone B, [N], Equations (5), (6), Figure 7
F_{GzoneC}	gasket force on zone C, [N], Equations (5), (10), Figure 7
F_Q	axial fluid pressure force, [N], Equations (29), (31)
F_R	force resulting from external additional axial force and moment, [N], Equations (29), (31)
K_{seal}	ratio of effective to sealing average gasket stress, Equation (27), (30), (31)
n_B	number of bolts, Equation (11), (21), (23)
P	internal pressure, [MPa]
$Q(x)$	gasket stress evolution versus gasket radius, [MPa], Equation (2)
Q_A	minimum necessary compressive stress in gasket for assembly condition, [MPa], Equation (28), (30)
$Q_{AGe'}$	average stress on effective gasket area, [MPa], Equation (27)
$Q_{smin(L)I}$	mean effective required gasket compressive stress at load condition I, [MPa], Equations (29), (31)
$Q_{max,Y}$	yield stress characteristic of the gasket materials and construction, [MPa], Equation (22), (26)

Q_{seal} average stress on sealing gasket area, [MPa], Equation (27)

x radial position defined by $Q(x = 0) = 0$, [mm], Figures 4, 5, 6

x_{max} elastic behaviour gasket width, [mm], Figure 10

Subscripts

0 Initial bolt-up condition (assembly)

I Subsequent operating condition I

Superscripts

el elastic behaviour of the gasket is considered

pl plastic behaviour of the gasket is considered

4 Introduction

EN 1591-1:2001+A1:2009 and CEN/TS 1591-3 only deal with the Inside Bolt Circle (IBC) gaskets. For a Full Face (FF) gasket, the gasket reaction force application diameter is closer to the bolt circle diameter due to the gasket reaction on the outside part of the gasket.

The aim of this proposal is to give a method (based on EN 1591-1) enabling to include the full face gaskets within the scope of series EN 1591.

The method described in this document involves the following steps:

a) Geometrical definition of a homogeneous gasket (without any hole) equivalent in terms of reaction force to the real Full Face gasket with holes:

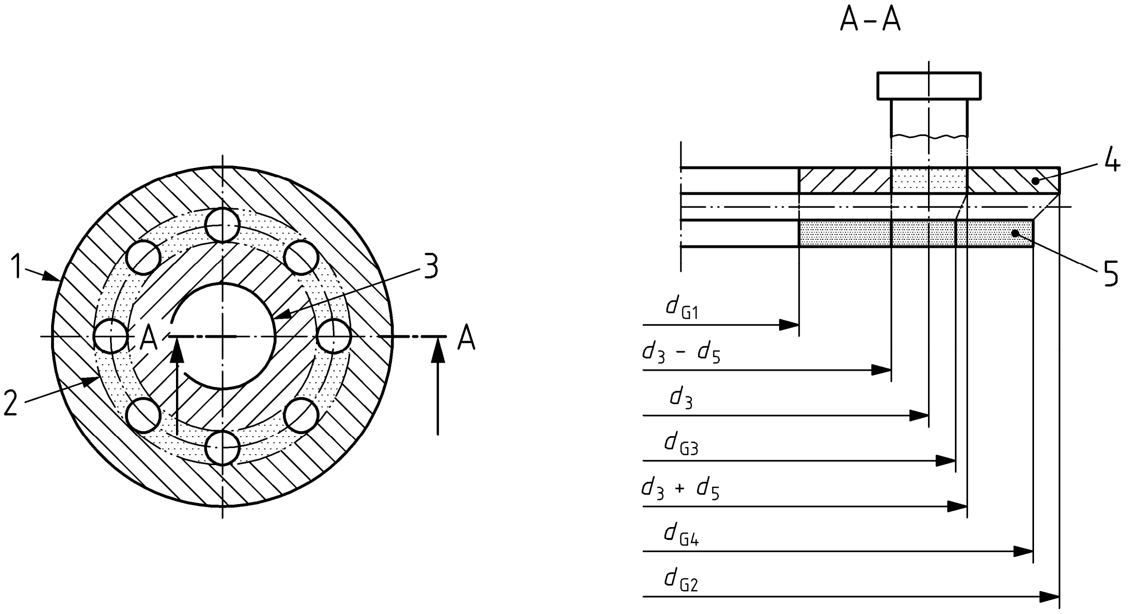
   1) Partitioning of the gasket into 3 parts (internal part, bolt holes part and external part);

   2) determination of the outside diameter of an homogeneous gasket part equivalent (for gasket force) to the bolt hole part, having the same inside diameter, considering full elastic behaviour of the gasket (d_{G3}^{el});

   3) determination of the outside diameter of a homogeneous gasket part equivalent (for gasket force) to the external part, having an inside diameter of (d_{G3}^{el}) considering full elastic behaviour of the gasket (d_{G4}^{el});

   4) determination of the outside diameter of a homogeneous gasket equivalent (for gasket force) to the real gasket, having the same inside diameter, considering full plastic behaviour of the gasket (d_{G4}^{pl});

   5) determination of the outside diameter of the equivalent gasket (d_{G4}) as the maximum value of d_{G4}^{el} and (d_{G4}^{pl}) in order to maximize the gasket surface and the subsequent required bolt load.



Key

1 external area
2 bolt holes area
3 internal area
4 real gasket
5 equivalent gasket

Figure 1 — "FULL FACE" gasket areas division and definition of the equivalent gasket

b) Determination of the effective dimensions of the homogeneous equivalent gasket using the equations of EN 1591-1.

c) Modification of the tightness criteria verification, in order to take into account that only a part of the effective gasket width participates to the sealing (part inside the bolt hole circle).

5 Definition of an equivalent gasket

5.1 Gasket parts identification

The gasket can be divided into three parts as shown in Figure 1:

a) The <u>Internal part</u> (or sealing part) is the gasket part inside the bolt holes diameter. This part of the gasket is the part already treated in EN 1591-1 and is the part associated to the sealing behaviour of the bolted flange connection;

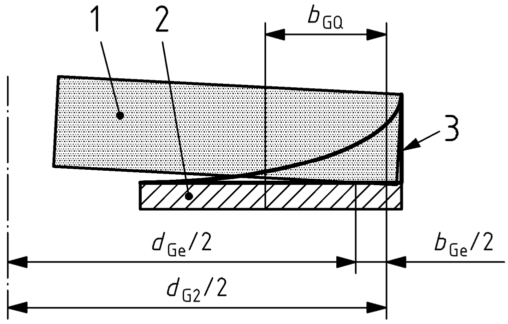
b) The <u>Bolt holes part</u> is the part of the gasket containing the holes enabling the bolt going through the gasket. The width of this part is equal to the bolt holes diameter;

c) The <u>External part</u> is the part outside the bolt holes diameter (d_5).

5.2 Gasket elastic deformation

5.2.1 General

For the compressed gasket width see Figure 2.



Key

1 flange

2 gasket

3 gasket stress distribution

Figure 2 — Compressed gasket width

5.2.2 Gasket modelling at unloading

The gasket elasticity modulus at unloading (E_G) depends on the initial gasket stress (Q = F_{G0}/A_{Ge}), see Figure 3. The test performed according to EN 13555 give tabulated values for E_G depending on the initial gasket stress F_{G0}/A_{Ge}. Thus the variation of E_G versus Q can be modelled by a linear relation for each interval [Q_n, Q_{n+1}].

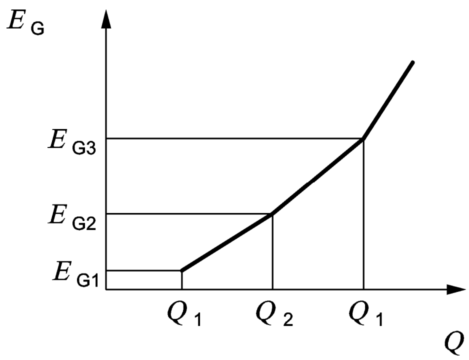


Figure 3 — Variation of E_G versus initial gasket stress Q

$$For\ Q_n \le Q \le Q_{n+1}$$

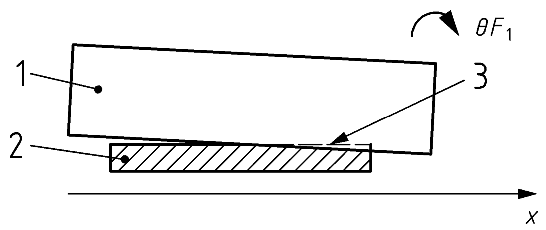
$$\begin{aligned} &E_G(Q) = A_n \cdot Q + B_n = dQ/d\varepsilon \\ &A_n = (E_{G,n+1} - E_{G,n}) / (Q_{n+1} - Q_n) \\ &B_n = E_n - \left[(E_{G,n+1} - E_{G,n}) / (Q_{n+1} - Q_n) \right] \cdot Q_n \end{aligned} \tag{1}$$

In the case of elastic deformation, considering the flange rotation (see Figure 4), leads to the following expression of Q versus radial distance x, using the same equations as in the CR 13642, with coefficients A_n and B_n for each initial gasket stress interval [Q_n, Q_{n+1}].

$For\ x_n \le x \le x_{n+1}$ with $x_0 = 0$

$$Q(x) = k \cdot (x - x_n) \cdot B_n \cdot \left[1 + \frac{1}{2} \cdot A_n \cdot k \cdot (x - x_n) \right] + \sum_{m=0}^{n-1} k \cdot (x_{m+1} - x_m) \cdot B_m \cdot \left[1 + \frac{1}{2} \cdot A_m \cdot k \cdot (x_{m+1} - x_m) \right] \tag{2}$$

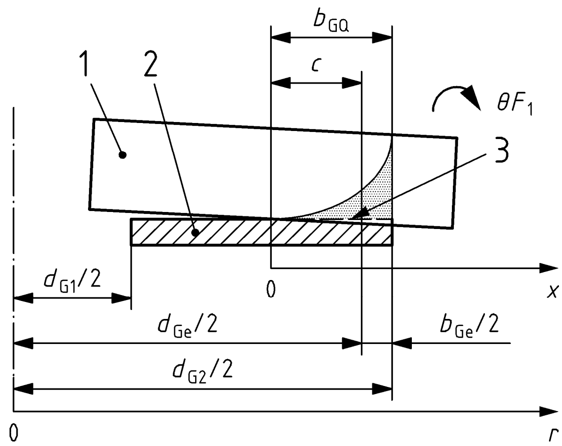
NOTE This expression of Q replaces the expression $Q(x) \approx E_0 \cdot k \cdot x \cdot (1 + ½ \cdot K_1 \cdot k \cdot x)$ given in document CR 13642:1999, Equation (4.4). (The value of n = 0, leads to CR 13642:1999, Equation (4.4)). It should be noted that the expression of EN 1591-1:2001+A1:2009, Table 1 is based upon the hypothesis of a gasket elastic modulus for unloading depending linearly of the initial gasket stress ($E_G = E_0 + K_1 \cdot Q$).



Key
1 flange
2 gasket
3 effective contact area due to flange rotation

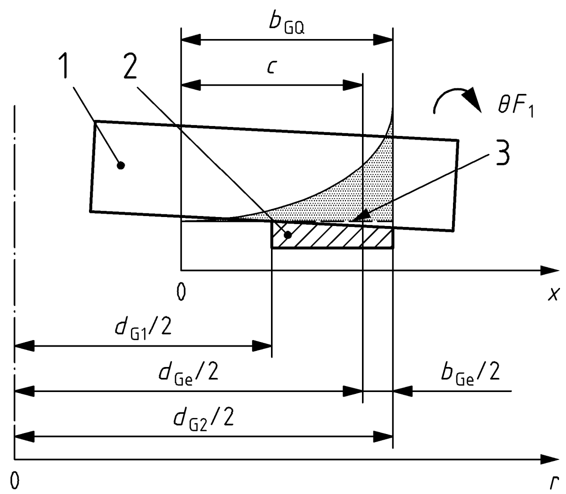
Figure 4 — Flange rotation

At this step there are two possible configurations, see Figure 5 and Figure 6:



Key
1 flange
2 gasket
3 gasket pressure contact profile on effective contact area

Figure 5 — Case with no contact on the inside part of the gasket



Key
1 bride (flange)
2 joint (gasket)
3 gasket pressure contact profile on effective contact area

Figure 6 — Case with contact on the inside part of the gasket

This leads to the following expressions:

$$r = d/2 = x + d_{G2}/2 - b_{GQ} \tag{3}$$

This last expression combined with the expression of $Q(x)$ (see Equation (2)) shown above leads to a polynomial of order 2 or 3 for the variable b_{GQ}.

Knowing d_{Ge} and F_G (which is the case in the determination of the effective gasket dimensions), this polynomial can be analytically solved, and the value of b_{GQ} found using Equation (4) below.

$$F_G = \pi \cdot d_{Ge} \cdot \int_{\max(d_{G1}/2 - d_{G2}/2 + b_{GQ};0)}^{b_{GQ}} Q(x)\,dx \tag{4}$$

5.2.3 Determination of the dimensions of the homogeneous part equivalent to the hole part (elastic case)

The gasket force on the hole part (zone A) is calculated by subtracting the force on the bolt holes surface (zone C) to the force on an homogeneous gasket part with dimensions equal to those of the bolt hole part (zone B), see Figure 7.

$$F_{G\,zoneA} = F_{G\,zoneB} - F_{G\,zoneC} \tag{5}$$

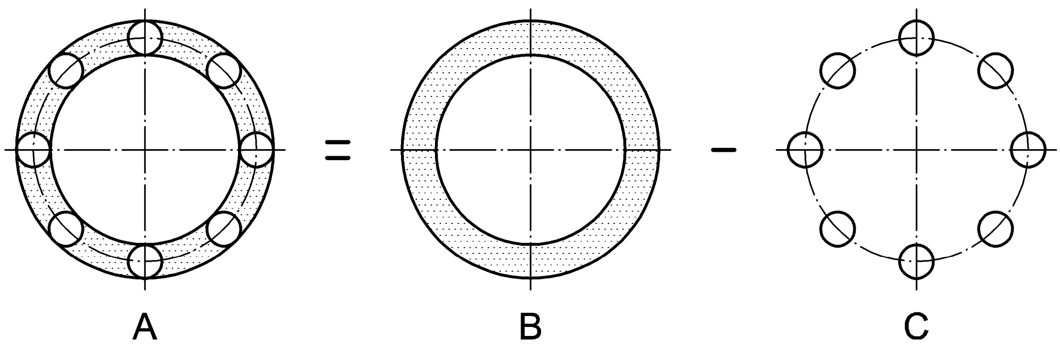


Figure 7 — Gasket force calculation on bolt hole part

The gasket force on zone B can be evaluated using Equation (6) assuming the reaction force is located at diameter d_{F2}.

$$(d_{F2}{}^{el} - d_{G2} + 2 \cdot b_{GQ}) \cdot \int_{x_{B1}{}^{el}}^{x_{B2}{}^{el}} Q(x)dx = \int_{x_{B1}{}^{el}}^{x_{B2}{}^{el}} x \cdot Q(x)dx \tag{6}$$

The value of reaction force diameter (d_{F2}) is determined using Equation (7)

$$F_{G\,zoneB} = \pi \cdot d_{F_2}{}^{el} \cdot \int_{x_{B1}{}^{el}}^{x_{B2}{}^{el}} Q(x)dx \tag{7}$$

With

$$x_{B1}{}^{el} = (d_3 - d_5 - d_{G2})/2 + b_{GQ} \tag{8}$$

$$x_{B2}{}^{el} = (d_3 + d_5 - d_{G2})/2 + b_{GQ} \tag{9}$$

$$\left(d_{F2}{}^{el} - d_{G2} + 2 \cdot b_{GQ}\right) \cdot \int_{x_{B1}{}^{el}}^{x_{B2}{}^{el}} Q(x)dx = \int_{x_{B1}{}^{el}}^{x_{B2}{}^{el}} xQ(x)dx \tag{10}$$

The gasket force on zone C can be estimated by the Equation (11), assuming that the gasket stress is uniform on the surfaces corresponding to the bolt holes.

$$F_{G\,zoneC} = \pi \cdot n_B \cdot \frac{d_5{}^2}{4} \cdot Q \cdot \left(\frac{d_3}{2} - \left(\frac{d_{G2}}{2} - b_{GQ}\right)\right) \tag{11}$$

Equation (12) expresses the fact that the gasket force on the part of the equivalent gasket corresponding to the bolt hole part on the real gasket, is equal to the gasket force on zone A. Using the Equations (12) to (14), the value of $d_{G3}{}^{el}$ can be determined.

$$F_{G\,zoneA} = \pi \cdot d_{F_1}{}^{el} \cdot \int_{x_{B1}{}^{el}}^{x_{B3}{}^{el}} Q(x)dx \tag{12}$$

$$x_{B3}{}^{el} = \left(d_{G3}{}^{el} - d_{G2}\right)/\,2 + b_{GQ} \tag{13}$$

$$\left(d_{F_1}{}^{el} - d_{G2} + 2 \cdot b_{GQ}\right) \cdot \int_{x_{B1}{}^{el}}^{x_{B3}{}^{el}} Q(x)dx = \int_{x_{B1}{}^{el}}^{x_{B3}{}^{el}} xQ(x)dx \tag{14}$$

Considering a gasket stress varying in the interval [Q_n, Q_{n+1}] described above, we get an order 4 polynomial equation of $x_{B3}{}^{el}$ (see Equation (15)) that can be analytically solved using algorithm providing the “zero” values of a function. Then, using Equation (13), the value of $d_{G3}{}^{el}$ can be assessed.

$$
\begin{aligned}
0 = -&\left[1/6 \cdot \pi \cdot B_n \cdot A_n \cdot k^2 \cdot \left(x_{B1}{}^{el}\right)^3 + 1/2 \cdot \pi \cdot B_n \cdot k \cdot \left(x_{B1}{}^{el}\right)^2\right] \cdot \left[d_{G2} - 2 \cdot b_{GQ}\right] \\
&- 2 \cdot \left[1/8 \cdot \pi \cdot B_n \cdot A_n \cdot k^2 \cdot \left(x_{B1}{}^{el}\right)^4 + 1/3 \cdot \pi \cdot B_n \cdot k \cdot \left(x_{B1}{}^{el}\right)^2\right] \\
&+ F_{G\,zoneC} - F_{G\,zoneB} + 1/2 \cdot \pi \cdot B_n \cdot k \cdot \left[d_{G2} - 2 \cdot b_{GQ}\right] \cdot \left[x_{B3}{}^{el}\right]^2 \\
&+ \left[2/3 \cdot \pi \cdot B_n \cdot k + \left(1/6 \cdot \pi \cdot B_n \cdot A_n \cdot k^2\right) \cdot \left(d_{G2} - 2 \cdot b_{GQ}\right)\right] \cdot \left[x_{B3}{}^{el}\right]^3 \\
&+ \left[\left(1/4 \cdot \pi \cdot B_n \cdot A_n \cdot k^2\right)\right] \cdot \left[x_{B3}{}^{el}\right]^4
\end{aligned}
\qquad (15)
$$

The external diameter of the equivalent gasket in the elastic case ($d_{G4}{}^{el}$) is determined using the same approach as for the bolt hole part. Equation (16) is stating the equality of gasket force for the real and the equivalent gaskets on their external parts. As for $x_{B3}{}^{el}$, we get an order 4 polynomial equation of $x_{B4}{}^{el}$ (see Equation (20)) that can be analytically solved using algorithm providing the “zero” values of a function. Then, using Equation (17), the value of $d_{G4}{}^{el}$ can be assessed

$$\pi \cdot d_{F4}{}^{el} \cdot \int_{x_{B3}{}^{el}}^{x_{B4}{}^{el}} Q(x)dx = \pi \cdot d_{F3}{}^{el} \cdot \int_{x_{B2}{}^{el}}^{b_{GQ}} Q(x)dx \qquad (16)$$

$$x_{B4}{}^{el} = \left(d_{G4}{}^{el} - d_{G2}\right) / 2 + b_{GQ} \qquad (17)$$

$$\left(d_{F3}{}^{el} - d_{G2} + 2 \cdot b_{GQ}\right) \cdot \int_{x_{B3}{}^{el}}^{x_{B4}{}^{el}} Q(x)dx = \int_{x_{B3}{}^{el}}^{x_{B4}{}^{el}} xQ(x)dx \qquad (18)$$

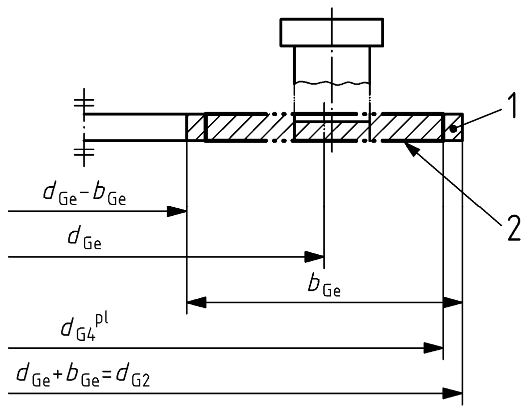
$$\left(d_{F4}{}^{el} - d_{G2} + 2 \cdot b_{GQ}\right) \cdot \int_{x_{B2}{}^{el}}^{b_{GQ}} Q(x)dx = \int_{x_{B2}{}^{el}}^{b_{GQ}} xQ(x)dx \qquad (19)$$

The equation to solve is then:

$$
\begin{aligned}
0 = & -\left[1/6\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot\left(x_{B3}^{el}\right)^3 + 1/2\cdot\pi\cdot B_n\cdot k\cdot\left(x_{B1}^{el}\right)^2\right]\cdot\left[d_{G2}-2\cdot b_{GQ}\right] \\
& -2\cdot\left[1/8\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot\left(x_{B3}^{el}\right)^4 + 1/3\cdot\pi\cdot B_n\cdot k\cdot\left(x_{B3}^{el}\right)^3\right] \\
& -\left(1/6\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot b_{GQ}{}^3 + 1/2\cdot\pi\cdot B_n\cdot k\cdot b_{GQ}{}^2\right)\cdot\left[d_{G2}-2\cdot b_{GQ}\right] \\
& +\left(1/6\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot\left(x_{B2}^{el}\right)^3 + 1/2\cdot\pi\cdot B_n\cdot k\cdot\left(x_{B2}^{el}\right)^2\right)\cdot\left[d_{G2}-2\cdot b_{GQ}\right] \\
& -2\cdot\left(1/8\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot b_{GQ}{}^4 + 1/3\cdot\pi\cdot B_n\cdot k\cdot b_{GQ}{}^3\right) \\
& +2\cdot\left(1/8\cdot\pi\cdot B_n\cdot A_n\cdot k^2\cdot\left(x_{B2}^{el}\right)^4 + 1/3\cdot\pi\cdot B_n\cdot k\cdot\left(x_{B2}^{el}\right)^3\right) \\
& +\left[d_{G2}-2\cdot b_{GQ}\right]\cdot(1/2\cdot\pi\cdot B_n\cdot k)\cdot\left(x_{B4}^{el}\right)^2 \\
& +\left[2/3\cdot\pi\cdot B_n\cdot k + \left(1/6\cdot\pi\cdot B_n\cdot A_n\cdot k^2\right)\cdot\left(d_{G2}-2\cdot b_{GQ}\right)\right]\cdot\left(x_{B4}^{el}\right)^3 \\
& +\left[\left(1/4\cdot\pi\cdot B_n\cdot A_n\cdot k^2\right)\right]\cdot\left(x_{B4}^{el}\right)^4
\end{aligned}
\tag{20}
$$

5.2.4 Determination of the dimensions for the homogeneous gasket equivalent to the real gasket (plastic case)

In the plastic case, the gasket stress is assumed to be uniform (value $Q_{\max,Y}$) along its radius. The gasket force on the real ($F_{\mathrm{Greal}}{}^{\mathrm{pl}}$) and the equivalent gasket ($F_{\mathrm{Gequi}}{}^{\mathrm{pl}}$) are given by the Equations (21) and (22). The equivalent gasket is defined to have equal values for these two forces. Thus finally the value of the external diameter for the equivalent gasket ($d_{\mathrm{G4}}{}^{\mathrm{el}}$) is given by Equation (23), see also Figure 8.



Key
1 load on the real gasket
2 load on the equivalent gasket

Figure 8 — Equivalent gasket external diameter in plastic case

$$F_{G\,real}{}^{pl} = Q_{\max,Y} \cdot \left(\pi \cdot d_{Ge} \cdot b_{Ge} - \pi \cdot n_B \cdot d_5{}^2 / 4\right) \tag{21}$$

$$F_{G\,equi}{}^{pl} = \pi \cdot Q_{\max,Y} \cdot d_{Ge} \cdot \left(d_{G4}{}^{pl} - d_{Ge}\right) \tag{22}$$

$$d_{G4}{}^{pl} = d_{G2} - \left(n_B \cdot d_5{}^2\right) / \left(4 \cdot d_{Ge}\right) \tag{23}$$

5.2.5 General case

The external diameter of the equivalent gasket for the general case d_{G4} is taken as the maximum value of $d_{G4}{}^{el}$ and $d_{G4}{}^{pl}$, in order to maximize the gasket area and thus the initial required bolt load (safe approximation).

$$d_{G4} = \max(d_{G4}{}^{el}; d_{G4}{}^{pl}) \tag{24}$$

6 Effective gasket geometry

In this phase, the effective geometry of the equivalent homogeneous gasket is determined using the iterative process of EN 1591-1. Due to the proximity of the gasket reaction force diameter (d_{Ge}) to the effective bolt circle diameter (d_{3e}), many convergence problems occur using the iterative method.

This problem can be solved using another approach. All the equations concerning the iterative process to determine the value of b_{Ge} can be gathered in one unique equation expressing the value of b_{Gi} (interim value of b_{Ge}) or d_{Gi} (interim value of d_{Ge}) at iteration $n + 1$ in function of b_{Gi} or d_{Gi} at iteration n.

$$d_{Gi}(n+1) = f\left(d_{Gi}(n)\right) \tag{25}$$

This leads to the following relation for a stress value of (F_{G0}/A_{Ge}) varying in interval [Q_k, Q_{k+1}]:

$$d_{GI(n+1)} = \max\left[d_{Gt}, d_{G4} - \sqrt{\frac{e_G / (\pi \cdot d_{GI(n)} \cdot (B_k + 0{,}5\ A_k \cdot F_{G0}/A_{Ge}))}{(d_{3e} - d_{GI(n)})/2 \cdot Z_F/E_{F0} + (\tilde{d}_{3e} - d_{GI(n)})/2 \cdot \tilde{Z}_F/\tilde{E}_{F0}} + \left[\frac{F_{G0}}{\pi \cdot d_{GI(n)} \cdot Q_{\max,y}}\right]^2} \right] \tag{26}$$

Near the convergence value, $d_{Gi}(n + 1)$ and $d_{Gi}(n)$ are nearly equal, so the solution of the equation above is the solution of the equation $f(x) = x$. Several algorithms can be used to solve that kind of equation, where the “zero” value of a function is researched.

NOTE This approach could be used for the cases of IBC gaskets. A validation on several assemblies with IBC gasket types has been performed. Both new direct method and existing iterative method gave the same values for b_{Ge}.

7 Modification of tightness criteria equations

For the FULL FACE gasket, a part of the effective gasket width can overlap the bolt holes area. But, only one part of the gasket will participate to the sealing behaviour (see Figure 9). For the elastic behaviour, considering the stress profile, the maximum stress level will be reached on the external part of the gasket. The average gasket stress value, Q_{AGe}, on the gasket effective width is higher than the average gasket stress participating to the sealing behaviour, Q_{seal} (see Figure 9). For the plastic behaviour, the gasket stress is uniformly equal to $Q_{max,Y}$ along the gasket.

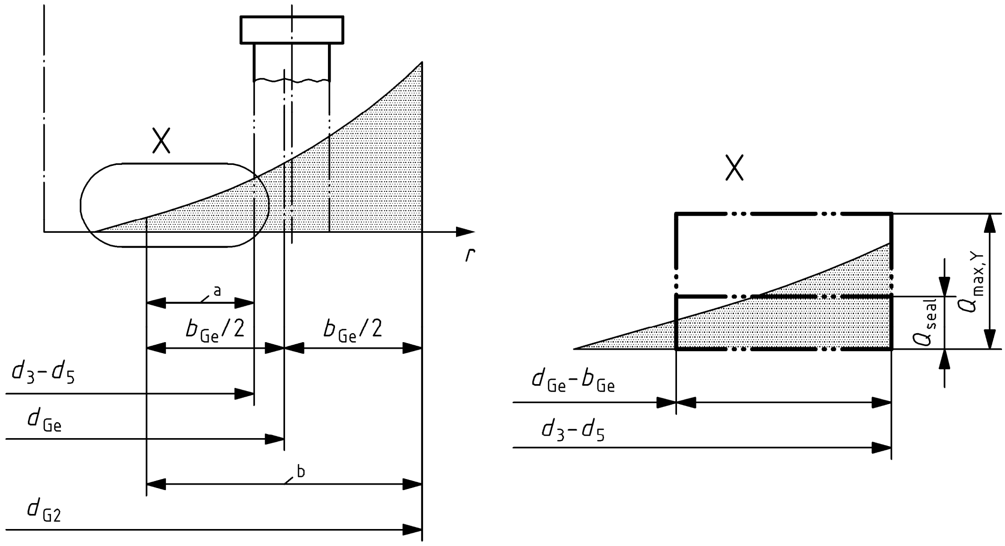
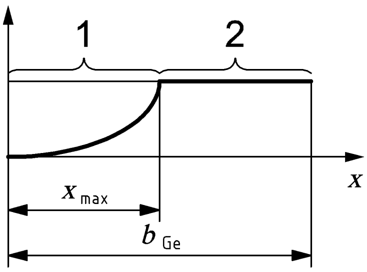


Figure 9 — Effective sealing and mechanical parts

In order to asses the average gasket stress value in the area participating to the sealing behaviour (Q_{seal}), a gasket stress profile built with a combination of pure elastic and pure plastic behaviour is considered (see Figure10).



Key
1 elastic
2 plastic

Figure 10 — Elastic/plastic gasket behaviour combination

For the three configurations ($b_{Ge} < x_{max}$, $b_{Gseal} < x_{max} < b_{Ge}$, $x_{max} < b_{Gseal} < b_{Ge}$), the value of Q_{seal} can be determined knowing the value of Q_{AGe}. This enables to calculate the ratio, $K_{seal} \geq 1$ (Equation 27), comparing the average gasket stress on the effective average gasket part to the gasket stress on the sealing part.

$$K_{seal} = Q_{AGe} / Q_{Seal} \tag{27}$$

In EN 1591-1:2001+A1:2009, the tightness criteria are checked with Equation (27) and Equation (28) (there defined as EN 1591-1:2001+A1:2009, Equation (49) and Equation (50)), defining the required gasket force at assembly and in operating conditions, in order to insure that the minimal average gasket contact pressures (Q_A for assembly and $Q_{smin(L)I}$ for operating condition) are reached.

— Assembly condition (I = 0)

$$FG_{0\,\min} = A_{Ge} \cdot Q_A \tag{28}$$

— Operating conditions (I = 1, 2, ...)

$$FG_{I\,\min} = \max\left\{A_{Ge} \cdot Q_{s\,\min(L)I}\,; -\left(F_{QI} + F_{RI}\right)\right\} \tag{29}$$

In the case of "FULL FACE" gaskets, Q_{seal} shall be greater than Q_A at assembly condition and greater than $Q_{smin(L)I}$ at operating conditions to insure the required gasket contact pressures. Thus the equation checking the tightness criteria become in the case of "FULL FACE" gaskets:

— Assembly condition (I = 0)

$$FG0_{\min} = A_{Ge} \cdot K_{seal} \cdot Q_A \tag{30}$$

— Operating conditions (I = 1, 2, ...)

$$FGI_{\min} = \max\left\{A_{Ge} \cdot K_{seal} \cdot Q_{s\min(L)I}; -\left(F_{QI} + F_{RI}\right)\right\} \tag{31}$$

8 Application of the proposed method on several standard assemblies

8.1 General

A software has been developed to apply the calculation method described above. The first calculations have been performed on standard assemblies.

8.2 Definition of assemblies

8.2.1 Dimensions (EN 1092-1)

a) PN6 DN200/DN600

b) PN10 DN200/DN600/DN1000/DN1600/DN2000

c) PN16 DN100/DN1000/DN2000

d) PN25 DN200/DN1000/DN1600/DN2000

e) PN40 DN100/DN600

8.2.2 Material

a) Flanges: E = 210 000 MPa, nominal design stress = 210 MPa.

b) Bolts: E = 210 000 MPa, nominal design stress = 427 MPa.

c) Gasket:

1) $E_G = A_i \cdot Q + B_i$ with $A_i = 0$ and $B_i = 100$ (uniform values for all the gasket stress intervals).

2) Q_A = 0,5 MPa et $Q_{smin(L)}$ = 0,9 · PN.

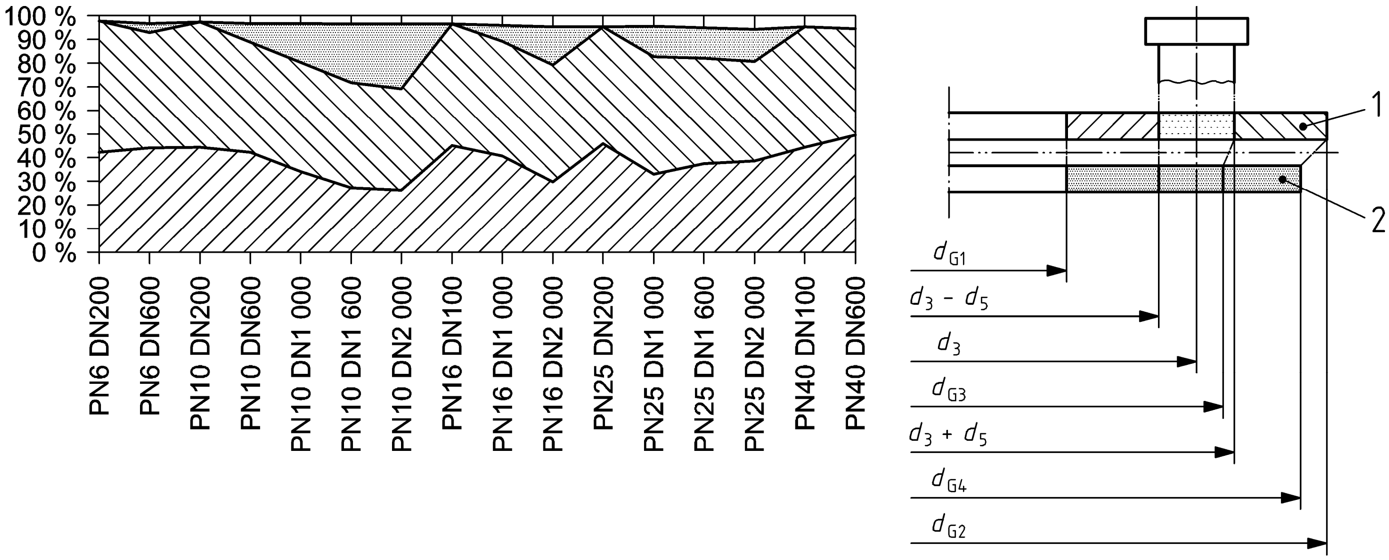
3) Gasket creep factor: P_{QR} = 1.

4) $Q_{max,Y} = Q_{smax}$ = 28 MPa for all the situations.

8.3 Results

The theoretical width of the equivalent gasket (homogeneous) is always higher than 94 % of the theoretical width of the real gasket (with holes). The gasket width participating to the sealing behaviour varies between 25 % and 50 % of the theoretical real gasket width (see Figure 11). It shall be noted that the gasket stress is considered uniform along the gasket circumference (variation between bolts ignored).

The theoretical width of the equivalent gasket (homogeneous) is always higher than 93 % of the theoretical width of the real gasket (with holes).



Key

relative theoretical width = $(d_{G4} - d_{G1})/(d_{G2} - d_{G1})$

relative mechanical effective width = $2 \cdot b_{Ge}/(d_{G2} - d_{G1})$

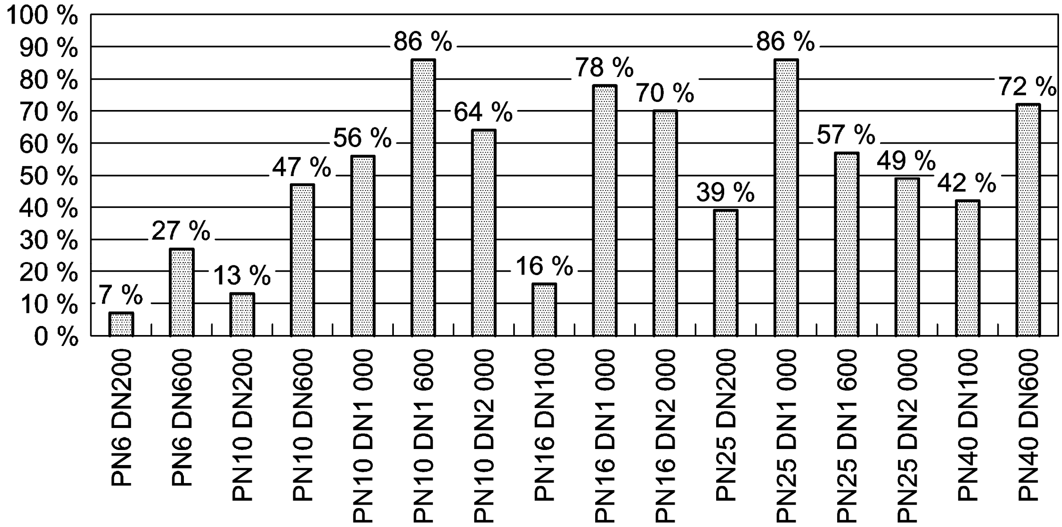
relative sealing width = $2 \cdot b_{Gseal}/(d_{G2} - d_{G1})$

1 real gasket

2 equivalent gasket

Figure 11 — Relative equivalent gasket dimensions

The required bolt load at assembly phase is lower (10 % to 80 %) than the value required by a TAYLOR FORGE based calculation method max {WA; WP}, see Figure 12.

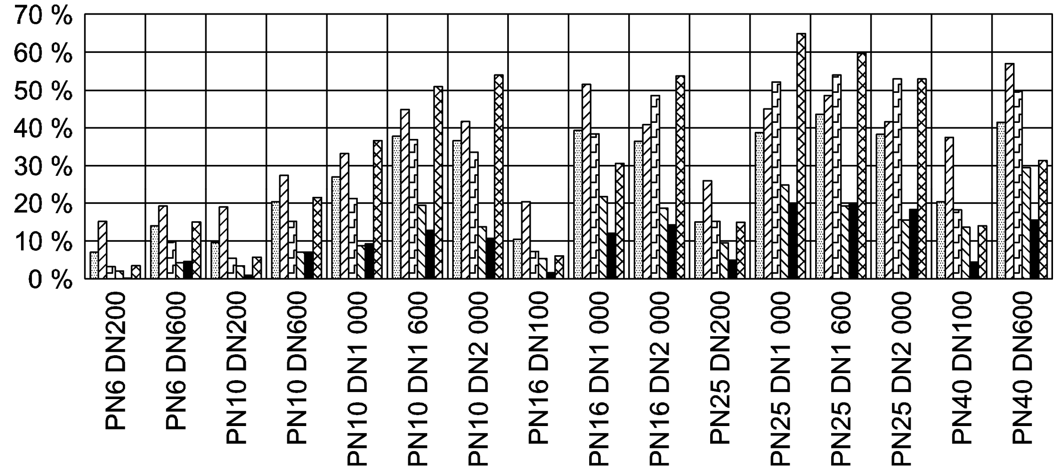


Key

Ratio of F_{B0req} obtained using the new method based on EN 1591-1 / Maximum value of WA and WP obtained using the existing TAYLOR FORGE based method.

Figure 12 — Required bolt up load ratio (EN 1591-1 based proposed method/ TAYLOR-FORGE based method)

In EN 1591-1, the load ratios are calculated for all the elements of the bolted joint (flanges/bolt/gasket). All the load ratios shall be lower than 100 % to insure the bolted flange connection mechanical integrity. All the calculated load ratios are lower than 70 %, for all the elements of the bolted flange joint and for all the studied conditions (bolt up + operating condition with P = PN). This shows that the method is in accordance with the existing flange standard (see Figure 13).



Key

bolt load ratio (bolt up) bolt load ratio (service) gasket load ratio (bolt up)

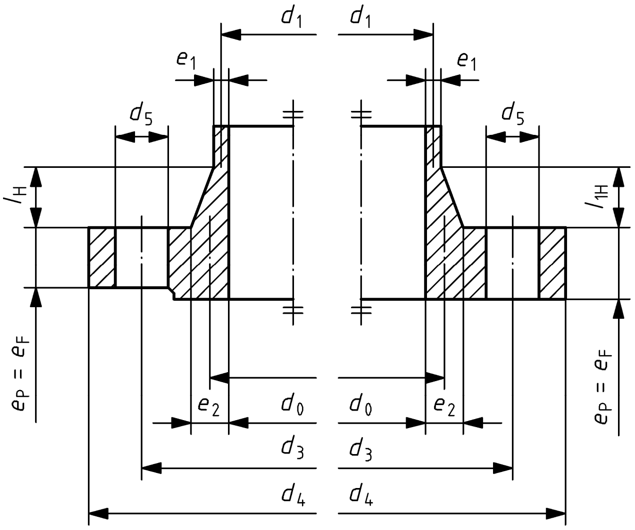
gasket load ratio (service) flange load ratio (bolt up) flange load ratio (service)

Figure 13 — Load ratios for all the elements

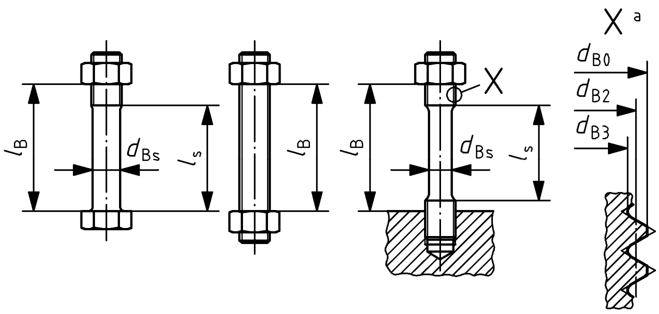
9 Application of the proposed method on a non-standard assembly

9.1 Dimensions

The considered gasketed joints involves two integral flanges (internal diameter 2 800 mm), with a collar, and a nominal pressure of 25 bar.



FLANGE		
d_0	2 800	mm
d_1	2 817,5	mm
d_2	2 852	mm
d_4	3 391	mm
d_3	3 219	mm
d_5	74	mm
d_6	0	mm
d_8	0	mm
d_9	0	mm
d_S	0	mm
e_0	0	mm
e_1	17,5	mm
e_2	52	mm
e_L	0	mm
e_P	164	mm
e_Q	0	mm
e_S	0	mm
l_H	34,5	mm
phi_S	0	°

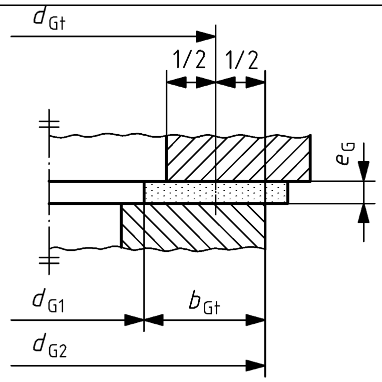


[a] X: ***detail of threaded part***

$l_s = 0$; $dB_s = dB_0$

$dB_e = 1/2 * (dB_2 + dB_3)$

BOLT		
n_B	**64**	
d_{B0}	**72**	**mm**
d_{Be}	**66,37**	**mm**



GASKET		
d_{G1}	2 894	mm
d_{G2}	3 380	mm
e_G	6	mm

9.2 Material data

a) Flanges: E = 210 000 MPa, nominal design stress = 210 MPa (bolt up), 170 MPa (service)

b) Bolts: E = 210 000 MPa, nominal design stress = 427 MPa (bolt up), 300 MPa (service)

c) Gasket:

d) $E_G = B + A * Q$ with 5 studied cases:

Case	1	2	3	4	5
B	13	100	200	200	200
A	0	0	0	10	15

1) Q_A = 0,5 MPa et $Q_{smin(L)}$ = 0,9 * PN

2) Gasket creep factor: P_{QR} = 1

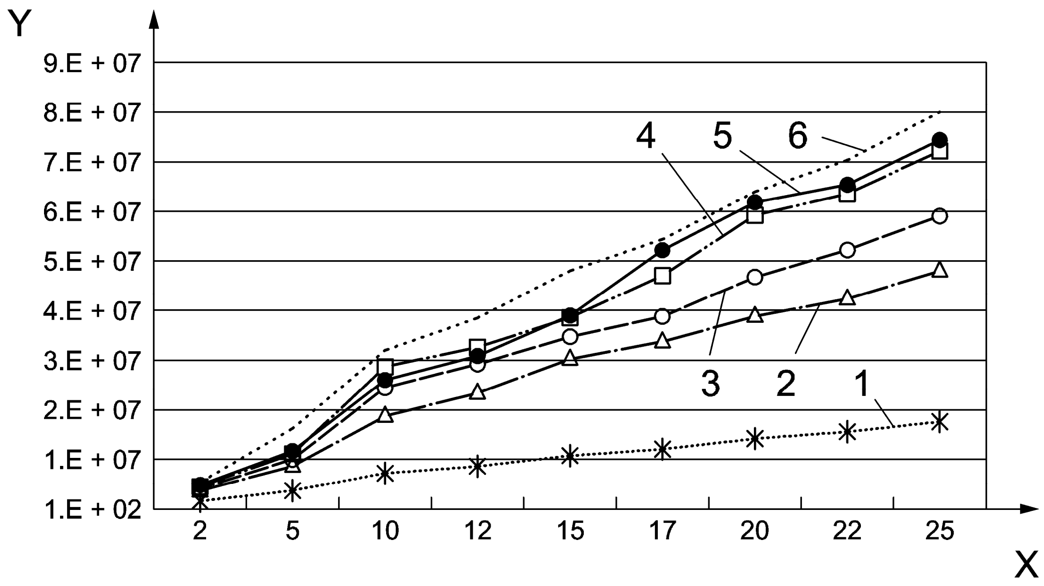
3) $Q_{max,Y} = Q_{smax}$ = 28 MPa for all the situations

e) Situations:

Bolt up and service (internal pressure variation between 2 bar and 25 bar at room temperature without additional external force) are studied.

9.3 Results

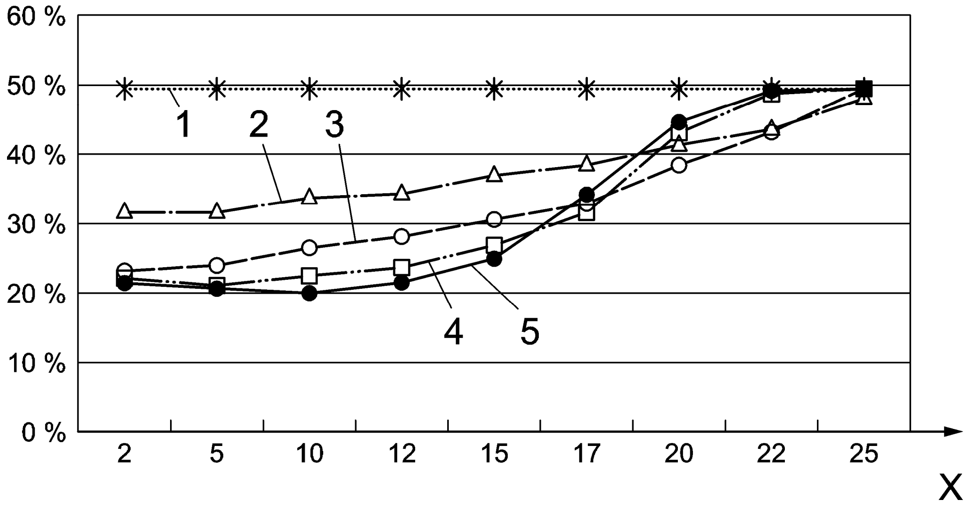
For all the studied gasket mechanical behaviours and internal pressures, the required bolt load with the new method is lower than the required bolt load using the existing method as shown in Figure 14. The initial required bolt load is especially much lower with the new method for the soft gaskets. In fact, the gasket mechanical behaviour is not taken into account in the existing method (based on the TAYLOR FORGE).



Key
X internal pressure (bar)
Y initial required bolt load (N)
1 A = 0; B = 13
2 A = 0; B = 100
3 A = 0; B = 200
4 A = 10; B = 200
5 A = 15; B = 200
6 CODAP-PD5500: max $\{WA; WP\}$

Figure 14 — Initial required bolt load comparison

The width participating to the sealing behaviour varies between 20 % and 50 % of the width of the real gasket, see Figure 15. At bolt up phase, the gasket width participating to the sealing behaviour with the new method is almost always greater than the effective gasket width calculated with existing method. Especially for soft gasket or when "plastic" term predominates in the equation determining the effective gasket width in EN 1591-1:2001+A1:2009, Table 1, see Figure 16.

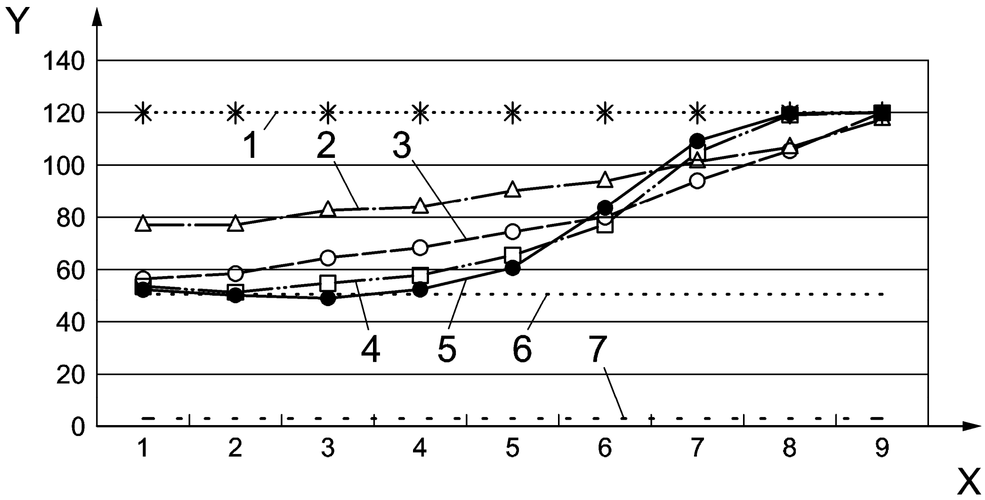


Key

X pressure (bar)
1 A = 0; B = 13
2 A = 0; B = 100
3 A = 0; B = 200
4 A = 10; B = 200
5 A = 15; B = 200

Figure 15 — Relative effective sealing width

For the service situation, the existing method gives a constant value of 2,5 mm for the efficient gasket width whatever the bolted connection dimension is. In series EN 1591, the effective gasket width value is considered equal between bolt up and subsequent situations. Thus it leads to a huge difference (2,5 mm to 120 mm) between the gasket widths given by both methods.



Key
X pressure (bar)
Y sealing gasket width (mm)
1 A = 0; B = 13
2 A = 0; B = 100
3 A = 0; B = 200
4 A =10; B = 200
5 A = 15; B = 200
6 assembly Taylor Forge (b')
7 operating Taylor Forge (b'')

Figure 16 — Gasket sealing width comparison

10 Conclusion

The method detailed in this document is based on the determination of a homogeneous equivalent gasket. Then the existing equations of EN 1591-1 are applied to this equivalent homogeneous gasket, with a slight mathematical modification in the way of solving the equations. At last, the equations checking the tightness criteria are modified in order to assure that the required average stress is applied on the gasket part participating to the sealing behaviour i.e. inside the bolt circle.

The method introduced above enables to take the full face gaskets into account without deeply modifying the equations of EN 1591-1. It enables to get a better estimation of the gasket width participating to the sealing behaviour of the assembly with a lower required bolt load than the existing method.

Providing additional validations using finite element modelling and experiments, this new method could easily be introduced in the series EN 1591 (and even directly in EN 1591-1).

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