DEVELOPMENT OF A NEW METHOD FOR "FULL FACE" GASKETED BOLTED FLANGE CONNECTIONS BASED ON EUROPEAN STANDARD EN1591

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ABSTRACT
The new analytical calculation method for gasketed circular flange connections NF EN1591-1 [3], has been issued in Europe in 2001. It is an alternative to the existing TAYLOR FORGE gasketed flange connections calculation rules. This standard, based on tightness criteria, gives a more detailed analytical modelling of the behaviour of the connection.

However, the “full face” gaskets, located on each side of the bolt circle, are out of the scope of NF EN1591-1[3], whereas a method partly based on the TAYLOR FORGE rules is proposed in the NF EN 13445-3 [6] and CODAP® [10]. A proposal for the calculation of “full face” gasketed joints, based on [3] is presented here.

NOMENCLATURE

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<tr>
<th>Symbol</th>
<th>Description</th>
<th>Equation/Reference</th>
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<tr>
<td>A_{Ge}</td>
<td>Effective gasket area</td>
<td>(\pi d_{Ge} b_{Ge})</td>
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<tr>
<td>b_{Ge}</td>
<td>Effective gasket width</td>
<td>(Figure3 &amp; Figure6)</td>
</tr>
<tr>
<td>b_{Gi}</td>
<td>Interim value of effective gasket width</td>
<td></td>
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<tr>
<td>b_{Gseal}</td>
<td>Effective sealing gasket width</td>
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<tr>
<td>b_{GQ}</td>
<td>Compressed gasket width</td>
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<td>d_{F1}</td>
<td>Gasket force acting diameter for zoneA</td>
<td></td>
</tr>
<tr>
<td>d_{F2}</td>
<td>Gasket force acting diameter for zoneB</td>
<td></td>
</tr>
<tr>
<td>d_{F3}</td>
<td>Resultant gasket force acting diameter on outside area of the real gasket</td>
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<td>d_{F4}</td>
<td>Resultant gasket force acting diameter on outside area of the equivalent gasket</td>
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<td>d_{G1}</td>
<td>Inside diameter of gasket contact area</td>
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<tr>
<td>d_{G2}</td>
<td>Outside diameter of gasket contact area</td>
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<td>d_{G3}</td>
<td>Outside diameter of bolt holes part for the equivalent gasket</td>
<td>(Figure2)</td>
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<tr>
<td>d_{G4}</td>
<td>Outside diameter of equivalent gasket</td>
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<td>d_{Gi}</td>
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<td>d_{d}</td>
<td>Real bolt circle diameter</td>
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<tr>
<td>d_{3}</td>
<td>Outside diameter of flange</td>
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<tr>
<td>d_{5}</td>
<td>Diameter of bolt hole</td>
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<td>F_{Gmin}</td>
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<td>F_{Gequi}</td>
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<td>F_{GzoneC}</td>
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<tr>
<td>F_{Q}</td>
<td>Axial fluid pressure force</td>
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<td>F_{R}</td>
<td>Force resulting from external additional axial force and moment</td>
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<td>Internal pressure</td>
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<td>Q_{min}</td>
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Subscript

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<td>I</td>
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Superscript

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<td>pl</td>
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INTRODUCTION

The new European analytical calculation method for gasketed circular flange connections [3], is based on elastic interaction between bolt, flanges and gasket. This method is introduced as an alternative to the TAYLOR-FORGE for gasketed joints calculation in several European standards and codes on pressure vessel (NF EN13445-3 [6], CODAP® [10], piping (NF EN 13480-3 [7]), valves (NF EN 12516-2 [8]) and boilers (NF EN 12953-3 [5]), especially when sealing achievement is of major importance.

The full face gaskets located on each side of the bolt circle, are out of the scope of [3], whereas a method partly based on the TAYLOR FORGE rules is proposed in [6] and [10] for example. A significant work on the subject of full face flange design has been published by Blach et al. [1], where an analytical method for full face gasket was presented. During PVP2005, Bouzid et al [9], proposed another important paper detailing an analytical calculation approach of bolted flange joints with full face gaskets close to [3]. Indeed it was taking into account the flexibility of the gasket and the bolt. Moreover it introduced the elastic interaction between bolt up and operating conditions as it is done in [3].

The aim of this article is to present a method based on [3], enabling to take the “FULL FACE” gasket case into account by introducing minor modifications of the existing method existing in [3].

The method described in this document involves 3 main steps. First, an homogeneous gasket (without any hole) equivalent in terms of reaction force to the real full face gasket (with bolt holes) is defined. This first step defines the external diameter of this homogeneous equivalent gasket assumed to have the same internal diameter than the real gasket. Then the “effective” dimensions of the homogeneous equivalent gasket are determined using the equations existing in [3]. Finally, the tightness criteria verification equations are modified, in order to take into account that only a part of the effective gasket width participates to the sealing behavior of the joint.

DEFINITION OF AN HOMOGENEOUS EQUIVALENT GASKET

The full face gasket is divided in three areas as shown in Figure1. An equivalent (in terms of reaction force) homogeneous gasket is defined. This equivalent gasket is assumed to have the same internal diameter (d_G1) than the real gasket. The aim of this part is to determine the external diameter (d_G4) of the equivalent gasket (Figure2). First of all, the reaction force on the different areas of the real gasket must be determined.

The document FD CR13642 [2] gives the background information of EN1591-1 [3]. It takes into account two mechanical behaviours for the gasket for the effective gasket width calculation. A first calculation is performed assuming a non-linear (the unloading gasket modulus depending on the initial gasket load) elastic behaviour. A second calculation is performed assuming a pure plastic behaviour. In this case, the gasket stress along the gasket width is uniformly equal to the maximal allowable gasket stress: Q_max,Y. Using the same philosophy, both cases are considered to determine the dimensions of the equivalent gasket in this method.

External area

Bolt holes area

Internal area

Figure 1 : “FULL FACE” Gasket areas division

Elastic case

At first the gasket width where the gasket is compressed (b_GQ) is determined knowing the total force on the gasket and the gasket unloading modulus dependence versus its initial load, as in [2]. (Figure3).

Figure 3 : Compressed gasket width

The gasket force on the hole part (zoneA) is calculated by substracting the force on the bolt holes surface (zoneC) to the force on an homogeneous gasket part with dimensions equal to those of the bolt hole part (zoneB) (see Figure4).

Figure 2 : Definition of the equivalent gasket
\[ F_{G_{zoneA}} = F_{G_{zoneB}} - F_{G_{zoneC}} \]  
(1)

**Figure 4 : Gasket force calculation on bolt hole part**

The gasket force on zoneB can be evaluated using equation (2) assuming the reaction force is located at diameter \( d_{F2} \). The value of reaction force diameter \( d_{F2} \) is determined using equation (5).

\[ F_{G_{zoneB}} = \pi \times d_{F1} \times \int_{s_{F1}}^{s_{F2}} Q(x)dx \]  
(2)

With

\[ x_{B1}^{el} = \left( d_{3} - d_{5} - d_{G2} \right) / 2 + b_{GQ} \]  
(3)

\[ x_{B2}^{el} = \left( d_{3} + d_{5} - d_{G2} \right) / 2 + b_{GQ} \]  
(4)

\[ \left( d_{F3}^{el} - d_{G2} + 2 \times b_{GQ} \right) \times \int_{s_{F3}}^{s_{F2}} Q(x)dx = \int_{s_{F3}}^{s_{F2}} xQ(x)dx \]  
(5)

The Gasket force on zoneC can be estimated by the equation (6), assuming that the gasket stress is uniform on the surfaces corresponding to the bolt holes.

\[ F_{G_{zoneC}} = \pi \times nB \times \frac{d_{2}^{2}}{4} \times Q\left(d_{2}/2 - \frac{d_{G2}}{2} - b_{GQ} \right) \]  
(6)

The equation (7) 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 zoneA. Using the equations (7 to 9), the value of \( d_{G3} \) can be determined.

\[ F_{G_{zoneA}} = \pi \times d_{F1} \times \int_{s_{F1}}^{s_{F2}} Q(x)dx \]  
(7)

\[ x_{B3}^{el} = \left( d_{G3} - d_{G2} \right) / 2 + b_{GQ} \]  
(8)

\[ \left( d_{F4}^{el} - d_{G2} + 2 \times b_{GQ} \right) \times \int_{s_{F4}}^{s_{F2}} Q(x)dx = \int_{s_{F4}}^{s_{F2}} xQ(x)dx \]  
(9)

The external diameter of the equivalent gasket in the elastic case \( d_{G3}^{el} \) is determined using the same approach as for the bolt hole part. The equation (10) is stating the equality of gasket force for the real and the equivalent gaskets on their external parts.

\[ \pi \times d_{F3} \times \int_{s_{F3}}^{s_{F2}} Q(x)dx = \pi \times d_{F4} \times \int_{s_{F4}}^{s_{F2}} Q(x)dx \]  
(10)

\[ x_{B4}^{el} = \left( d_{G4}^{el} - d_{G2} \right) / 2 + b_{GQ} \]  
(11)

\[ \left( d_{F3}^{el} - d_{G2} + 2 \times b_{GQ} \right) \times \int_{s_{F3}}^{s_{F2}} Q(x)dx = \int_{s_{F3}}^{s_{F2}} xQ(x)dx \]  
(12)

\[ \left( d_{F4}^{el} - d_{G2} + 2 \times b_{GQ} \right) \times \int_{s_{F4}}^{s_{F2}} Q(x)dx = \int_{s_{F4}}^{s_{F2}} xQ(x)dx \]  
(13)

**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 \( F_{G_{real}}^{pl} \) and the equivalent gasket \( F_{G_{equi}}^{pl} \) are given by the equations (14 and 15). The equivalent gasket is defined to have the equality of these two forces. Thus we finally get the value of the external diameter for the equivalent gasket given by equation (16).

\[ F_{G_{real}}^{pl} = Q_{max,Y} \times \left[ \pi \times d_{G2} \times b_{GQ} - \pi \times nB \times d_{s}^{2} / 4 \right] \]  
(14)

\[ F_{G_{equi}}^{pl} = \pi \times Q_{max,Y} \times d_{G2} \times \left( d_{G4}^{pl} - d_{G2} \right) \]  
(15)

\[ d_{G4}^{pl} = d_{G2} - \left( nB \times d_{s}^{2} / 4 \times d_{G2} \right) \]  
(16)

**Figure 5 : Equivalent gasket external diameter in plastic case**

**General case**

The external diameter of the equivalent gasket for the general case \( d_{G4}^{el} \) 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).
DETERMINATION OF THE EFFECTIVE DIMENSIONS OF THE EQUIVALENT GASKET

In this phase, the effective geometry of the equivalent homogeneous gasket is determined using the iterative process of [3]. 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 of [3].

This problem can be solved using another approach. All the equations concerning the iterative process to determine \( b_{Ge} \) can be gathered in one 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 \( n \).

\[
d_{Gi}(n+1) = f\left(d_{Gi}(n)\right)
\]

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.

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. \( \text{Figure 6} \). 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, \( Q_{seal} \). \( \text{Figure 7} \). For the plastic behaviour, the gasket stress is uniformly equal to \( Q_{max,Y} \) along the gasket.

\( \text{Figure 6 : Effective sealing and mechanical parts} \)

In order to assess 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. \( \text{Figure 8} \).

\( \text{Figure 7 : Average gasket stress distribution} \)

\( \text{Figure 8 : Elastic/plastic gasket behaviour combination} \)

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 (18)), comparing the average gasket stress on the effective gasket part to the gasket stress on the sealing part.

\[
K_{seal} = \frac{Q_{AGe}}{Q_{seal}}
\]

In [3], the tightness criteria are checked with equations (19) and (20), defining the required gasket force at assembly and in operating conditions, in order to insure that the minimal average gasket contact pressures \( (Q_{min} \text{ for assembly and } Q_{I} \text{ for operating condition}) \) are reached.

- Assembly condition \( (I = 0) \)
  \[
  FG0_{min} = A_{ge} \times Q_{min}
  \]
  \[
  FGI_{min} = \max\{A_{ge} \times Q_{I}; -\left(F_{Gy} + F_{Hy}\right)\}
  \]

In the case of “FULL FACE” gaskets, \( Q_{seal} \) must be greater than \( Q_{min} \) at assembly condition and greater than \( Q_{I} \) 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 \( (I = 0) \)
  \[
  FG0_{min} = A_{ge} \times K_{seal} \times Q_{min}
  \]
  \[
  FGI_{min} = \max\{A_{ge} \times K_{seal} \times Q_{I}; -\left(F_{Gy} + F_{Hy}\right)\}
  \]
APPLICATION ON SEVERAL STANDARD BOLTED FLANGE CONNECTIONS

Definition of the assemblies

Dimensions following NF EN1092-1 [11]
- PN6 DN200/DN600
- PN10 DN200/DN600/DN1000/DN1600/DN2000
- PN16 DN1000/DN1600/DN2000
- PN25 DN200/DN1000/DN1600/DN2000
- PN40 DN100/DN600

Material
- Flanges : E=210000MPa, nominal design stress= 210 MPa (bolt up), 170 MPa (service)
- Bolts : E=210000MPa, nominal design stress = 427 MPa (bolt up), 300 MPa (service)
- Gasket : (values from [4])
  - \( E_G = E_0 + K_1 \cdot Q \) with 5 studied cases :
    - Case 1 2 3 4 5
      - \( E_0 \) 13 100 200 200 200
      - \( K_1 \) 0 0 0 10 15
  - \( Q_{min} = 0.5 \text{ MPa} \) et \( Q_I = 0.9 \times P \) ([4])
  - Gasket creep factor gc=1
  - \( Q_{max,Y} = 28 \text{ MPa} \) for all the situations ([4])

Results

The required bolt up load is lower (ratio going from 7 to 86 %) than the value required by using the actual method (max (WA; WP)). (Figure10)

In [3], load ratios are calculated for all the elements of the bolted joint (flanges/bolt/gasket). All the load ratios must 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. (Figure11)

APPLICATION OF THE METHOD ON A NON-STANDARD BOLTED FLANGE CONNECTION

Dimensions

The considered gasketed joints involves two integral flanges (internal diameter 2800), with a collar, and a nominal pressure of 25 bar.

Material data
- Flanges : E=210000MPa, nominal design stress= 210 MPa (bolt up), 170 MPa (service)
- Bolts : E=210000MPa, nominal design stress = 427 MPa (bolt up), 300 MPa (service)
- Gasket :
  - Qmin= 0.5 MPa et QI=0.9*P ([4])
  - Gasket creep factor gc=1
  - Qmax,Y =28MPa for all the situations ([4])

Bolt up and service (internal pressure variation between 2 and 25 bar at room temperature without additional external force) are studied.
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 12. We can see that the initial required bolt load is much lower with the new method, especially for the soft gaskets. In fact, the gasket mechanical behaviour is not taken into account in the existing method.

![Figure 12: Initial required bolt load comparison](image1)

The width participating to the sealing behaviour varies between 20 and 50% of the width of the real gasket (Figure 13). 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 table 1 of [3]. (Figure 14)

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 [3], the effective gasket width value is considered equal between bolt up and subsequent situations. Thus we get a huge difference (2.5 to 120 mm) between the gasket widths given by both methods.

CONCLUSION

The method detailed in this document is based on the determination of an homogeneous equivalent gasket. Then the existing equations of actual EN1591-1 [3] 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.

The method introduced above enables to take the full face gaskets into account without deeply modifying the equations of [3]. 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 FE modelling and experiments, this new method could easily be introduced in the [3] without deep modifications.

REFERENCES


[2]: CEN TC 74, 2000, FD CR 13642: Flanges and their joints - Design rules for gasketed circular flange connections - Background information


[5]: CEN TC 269, 2002, EN12953-3: Shell boilers - Part 3: Design and calculation for pressure parts
