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Design considerations for rectangular bolted full face flanged joints for surface condensers

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ABSTRACT

Rectangular bolted full face flanged joints are widely used in surface condensers within the power generation industry including the nuclear one. In order to design these components, it is necessary to analyze the flanged joint from the point of view of structural strength and leak tightness. This work presents an analytical procedure applied to a rectangular bolted flange to determine the thickness of the flange, the bolt stresses and leak tightness conditions. First, the proposed analytical procedure is validated by comparing its results with those from finite element analysis (FEA) using non-linear approach considering the behavior of the materials, gasket and contacts. In addition, the proposed procedure is applied to the design of a rectangular flanged joint of a steam surface condenser using two different gaskets: compressed non-asbestos fiber gasket and NBR elastomer gasket. The obtained results show a better performance of the NBR elastomer gasket in comparison with compressed fiber gasket: better sealing condition, reduction of the flange thickness and reduction of the bolt stresses. It is important to highlight there are practically no references of procedures for design of non-circular full face flanged joints.

Keywords: **Bolted Joint, Full face, Rectangular flange, Surface condenser.**

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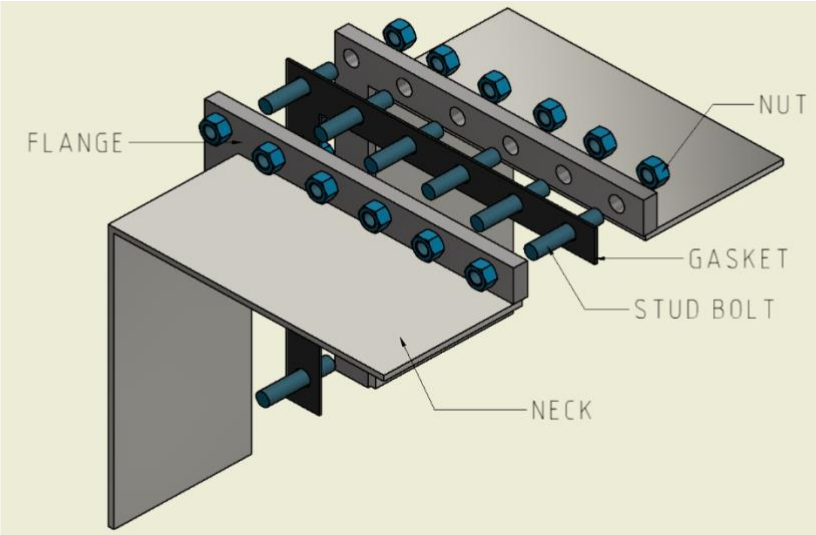
1. INTRODUCTION

Bolted flanged joints are widely used within all industrial sectors, mainly because their ability to assemble and disassemble. These joints must fulfill two primary functions: i) to guarantee structural strength and ii) to maintain the leak tightness of the joints [1].

This work proposes a procedure applied to rectangular bolted full face flanged joints, as shown in figure 1, which are used in several industrial equipment, like surface condensers in nuclear power generation systems, digesters, cyclones, chutes and air ducts [2]. This procedure seeks greater certainty in relation to the sealing requirements, since the procedure presented by the Enquiry Case 133 of PD 5500 [3], for example, underestimates the necessary bolt pretension to seating the gaskets.

It is important to notice that full face bolted flanged joints are more popular in low pressure applications and, also, for non-circular flanges which are difficult to seal with ring or strip gaskets. Despite the continuous use of such flanges, no design rules are contained in the most popular design code, the ASME VIII Division 1 [4].

Figure 1: *Rectangular bolted full-face flanged joint.*



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Appendices 2 and Y of ASME VIII Division 1 [4] cover flanges with a ring gasket located within the bolt circle and flanges with metal-to-metal contact outside the bolt circle, respectively.

The first one was developed in [5] and the second one was developed in [6]. Bolted joints used in conjunction with soft gaskets over the full face of the flange have no specific design rules, and the two mentioned appendices are not really suitable for such applications.

Full face gaskets are extensively used in the industry due to their simple and economical design, and low contact stress sealing requirements. The latter is generally achieved by the use of soft gaskets such as those based on rubber, elastomers, polytetrafluoroethylene (PTFE), and fibers. The design of full face flanges should minimize both separation at the bore and flange rotation. In the absence of a specific standard design procedure, full face flanges are sized by trial and error, or by an approximate extension of the Taylor Forge method [7].

According to [8], the use of full face flanges has been applied in flanged joints to reduce the moment applied to the flanges, especially when those have limitations in terms of thickness or material properties (strength limits).

For this type of joint, to guarantee the leak tightness, it is required many times that the minimum sealing gasket stress exceeds the region of the holes (figure 2), resulting in a large area of the joint to be compressed. Thus, it is necessary that a great force must be applied by the bolts, requiring a great bolt section area. Therefore, the use of this type of flange can become costly for equipment with high pressures, which makes this application common and convenient for low pressure one.

2. METHODOLOGY

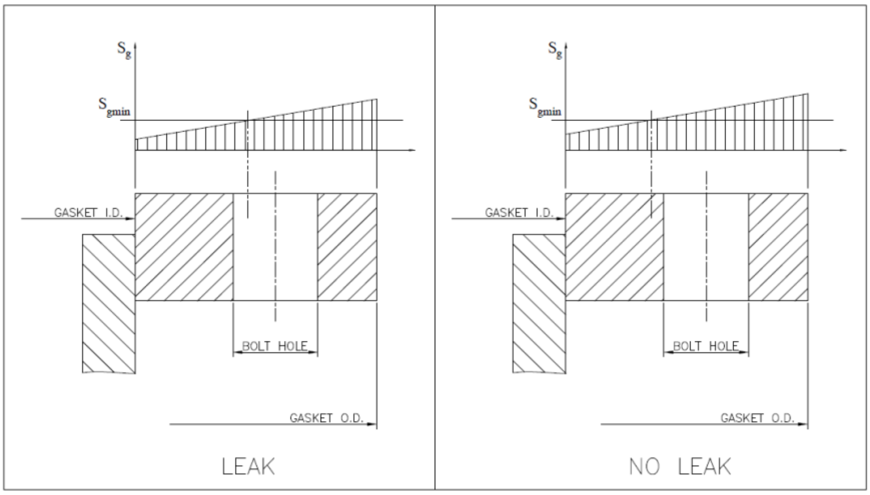
The proposed procedure of this paper is based on Enquiry Case 133 of PD 5500 [3] applied to rectangular bolted flanged joints. Two changes were done:

1) The area assumed to seating gasket is estimated as A1+A2-A3 (figure 3)

where A1 = area outside bolts lines, A2 = area of effective gasket inside bolts line and A3 = area of bolt holes

2) A factor considering a linear stress distribution in gasket region is applied to achieve seating stress in the limited line inside bolts lines (e.g., 5 mm near hole [9][3]).

Figure 2: *Leak and no leak conditions due gasket stress values and extension.*



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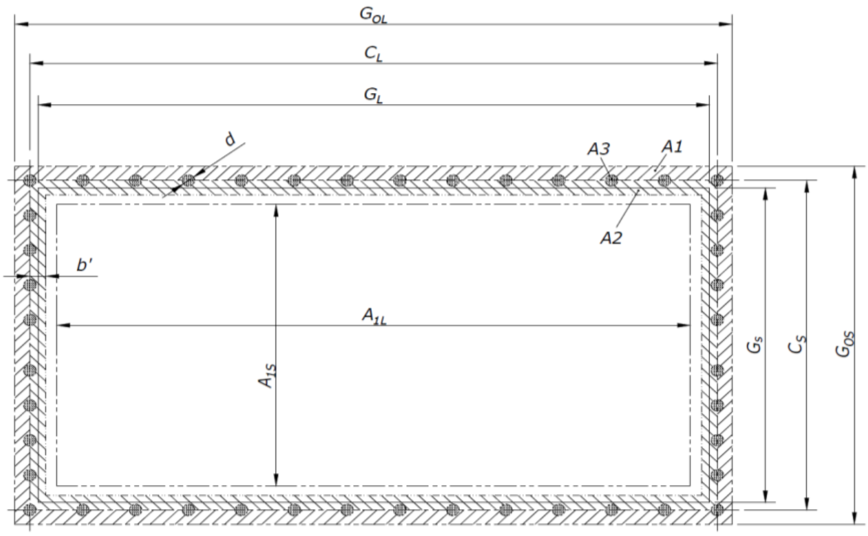
The results obtained from the proposed procedure were compared with other procedures already used in the industry, i.e., equivalent circular flange method with the Taylor-Forge full face flange method [10] and the Enquiry Case 133 of PD 5500 [3] and, also, with the results from nonlinear finite element analysis (FEA). Some conclusions and comments were addressed based on the comparisons.

Figure 3: *Assumed areas of proposed procedure and rectangular gasket dimensions*



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3. PROPOSED PROCEDURE

The proposed procedure is shown below where the parts in bold refer to the modifications made in Enquiry Case 133 of PD 5500 [3] formulation. The proposed procedure is divided into 4 steps for better understanding:

Step 1: Gasket Details

Gasket width

$$b'_o = min\ [(G_{0L} - C_L); (C_L - A_{1L})] \tag{1}$$

Effective gasket width

$$\boldsymbol{b' = max[\ 4\sqrt{b'_0}; d+5]} \tag{2}$$

Long side length of gasket reaction

$$G_L = C_L - b' \tag{3}$$

Short side length of gasket reaction

$$G_S = C_S - b' \tag{4}$$

Step 2: Forces, moments arms and moment calculation

Linear stress distribution on gasket along flange face

$$\boldsymbol{K = \frac{G_{0L} - A_{1L}}{2(C_L - B_L - d - 5)}} \tag{5}$$

Required gasket compression force

$$H_G = 4b'(G_L + G_S)mp\boldsymbol{K} \tag{6}$$

Gasket Reaction arm (figure 4)

$$\boldsymbol{h_G = b'/2} \tag{7}$$

Hydrostatic Force

$$H = p(C_L - d)(C_S - d) \tag{8}$$

Hydrostatic force applied by shell junction

$$H_D = pB_LB_S \tag{9}$$

Hydrostatic force on flange face

$$H_T = H - H_D \tag{10}$$

Application arm of H_D force

$$h_D = (C_L - B_L - g_1)/2 \tag{11}$$

Application arm of H_T force

$$h_T = (C_L - G_L)/2 \tag{12}$$

Moment acting on the flange

$$M = H_Dh_D + H_Gh_G + H_Th_T \tag{13}$$

Arm of h_r force outside bolts line

$$h_R = \frac{G_{0L} - C_L - d}{4} + \frac{d}{2} \tag{14}$$

Reaction force outside bolt line

$$H_R = M/h_R \tag{15}$$

Step 3: Bolt loads and strength

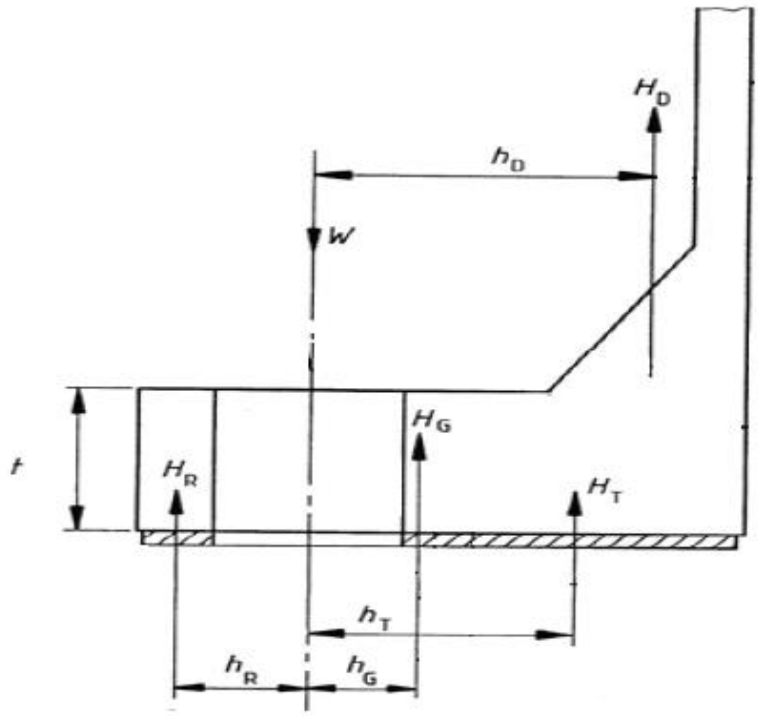
Minimum required bolts load for operation condition

$$W_{m1} = H_G + H + H_R \tag{16}$$

Minimum required bolts load for gasket seating condition

$$\boldsymbol{W_{m2} = [G_{0L}G_{0S} - C_LC_S + 2(G_L+G_S)b' - nd^2\pi/4]Ky} \tag{17}$$

Figure 4: *Forces and arms*



Source: Adapted from PD5500 [3]

The bolts load required is

$$W_m = max(W_{m1};W_{m2}) \tag{18}$$

The one bolt load required is

$$\boldsymbol{w = W_m / n_b} \tag{19}$$

And the bolt stress

$$S_{b'} = w / A_b \tag{20}$$

Step 4: Flange thickness

Required flange thickness for operation condition

$$t_1 = \sqrt{\frac{6M}{S_{FO}[2(C_L + C_S) - nd]}} \tag{21}$$

Required flange thickness for maximum distance between bolts

$$t_2 = \frac{(P_{bmax} - 2d_b)(m+0,5)}{6\left(\frac{E}{200000}\right)^{0,25}} \tag{22}$$

The required flange thickness for the full face bolted flanged joint is

$$t = max(t_1;t_2) \tag{23}$$

4. APPLICATION OF THE PROPOSED PROCEDURE IN A CASE STUDY

The main rectangular flange of a surface condenser was adopted as a case study of this work. This joint is subjected to a design pressure of 98,1 kPa and design temperature of 150 °C. The material of the neck and flange is SA-516 70 and the bolts are manufactured of SA-193 B7, both materials according to ASME II [11]. Two types of gasket materials were analyzed:

1) CS1: Compressed non-asbestos fiber gasket with minimum seating gasket stress (y) of 24,13 MPa and maintenance factor (m) of 2;

2) CS2: NBR Elastomer with minimum seating gasket stress (y) of 1,4 MPa and maintenance factor (m) of 1.

Table 1 shows the complete input data for CS1 calculation, Table 2 shows CS2 gasket inputs (the other data are the same of Table 1) and Table 3 shows the output for both cases

Table 1: Flange joint and CS1 input data.

Symbol	Value	Unit	Description
p	0,0981	MPa	Design pressure
C_L	2754	mm	Length of hole center line parallel to the long side
C_S	1040	mm	Length of hole center line parallel to the short side
A_{1L}	2638	mm	Internal long side gasket length
B_L	2638	mm	Internal long side flange length
B_S	928	mm	Internal short side flange length
d	36	mm	Bolt hole diameter
n	54		Number of bolts
A_b	694	mm^2	Bolt tensile stress area
g_1	19	mm	Neck thickness
G_{0L}	2840	mm	External long side gasket length
G_{0S}	1130	mm	External short side gasket length
y	24,13	MPa	Minimum gasket seating stress
m	2		Gasket maintenance factor
P_{bmax}	160	mm	Maximum spacing between bolts
P_{bmin}	110	mm	Minimum spacing between bolts
d_b	33	mm	External diameter of bolt
E	195 000	MPa	Flange material modulus of elasticity
S_{FO}	138	MPa	Flange allowable stress

Table 2: CS2 gasket input data.

Symbol	Value	Unit	Description
y	1,4	MPa	Minimum gasket seating stress
m	1		Gasket maintenance factor

Table 3: CS1 and CS2 outputs.

Symbol	CS1 Result	CS2 Result
b'_o	*86 mm*	*86 mm*
b'	*41 mm*	*41 mm*
G_L	*2713 mm*	*2713 mm*
G_S	*999 mm*	*999 mm*

K	*1,347*	*1,347*
H	*267 702 N*	*267 702 N*
H_G	*160 846 N*	*80 423 N*
H_D	*240 155 N*	*240 155 N*
H_T	*27 547 N*	*27 547 N*
h_D	*48,5 mm*	*48,5 mm*
h_G	*20,5 mm*	*20,5 mm*
h_T	*20,5 mm*	*20,5 mm*
M	*15 509 588 N.mm*	*13 860 914 N.mm*
h_R	*30,5 mm*	*30,5 mm*
H_R	*508 511 N*	*454 456 N*
W_{m1}	*937 060 N*	*802 582 N*
W_{m2}	*19 318 576 N*	*1 120 753 N*
W_m	*19 318 576 N*	*1 120 753 N*
w	*357 751 N*	*20 755 N*
$S_{b'}$	*515,5 MPa*	*29,91 MPa*
t_1	*10,93 mm*	*10,33 mm*
t_2	*39,42 mm*	*23,65 mm*
t	*39,42 mm*	*23,65 mm*

5. FINITE ELEMENT MODEL

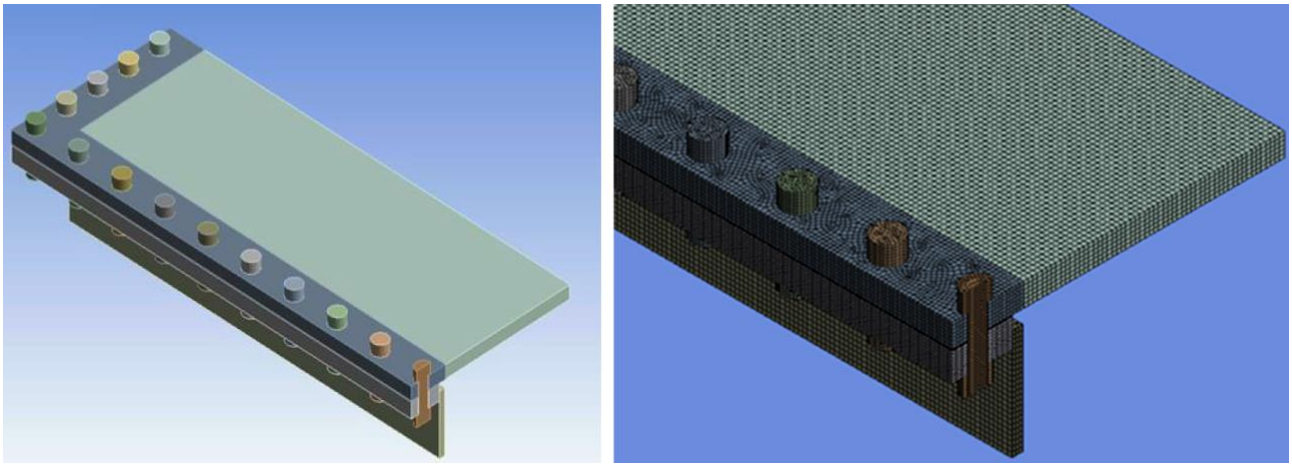
The finite element model (figure 5) used the symmetry presented by the joint and was discretized with solid elements according to table 4. Flanges and neck materials were modeled with elastic-plastic behavior according ASME VIII Division 2 item 3-D [12] (figure 6 shows below the stress-strain curve), bolts material was modeled with linear elastic material (Elastic modulus are 191 GPa and 184 GPa, for ambient and design temperature, respectively), and gaskets materials were modeled with a loading and unloading characteristic curve of each material (figure 7).

The 3D geometry used for numeric model can also be seen in figure 5, the simplified bolts were modeled together with the nuts and without the thread detail, having a diameter corresponding to the stress area according to ASME PCC-1[13]. The nuts were simplified by cylinders, without the hexagonal parts. Weld details have been omitted.

Three frictionless contacts were applied to constrain the model, one on each plane of symmetry and one on the under end of the neck. The contacts were configurated according to table 5 below. The loads of model were inputted in two steps:

1) Installation: Bolt pretension and body temperature of 20 ºC (ambient).
2) Operation: Internal surface design pressure of 98,1 kPa and body temperature of 150 °C (design).

Figure 5: *FEA Model*

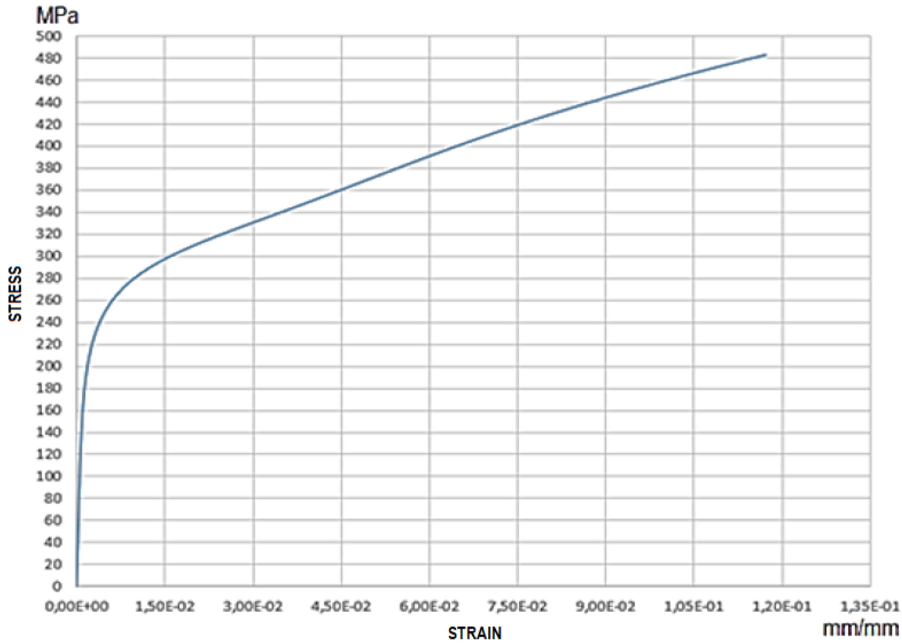


Source: Author

Table 4: Mesh.

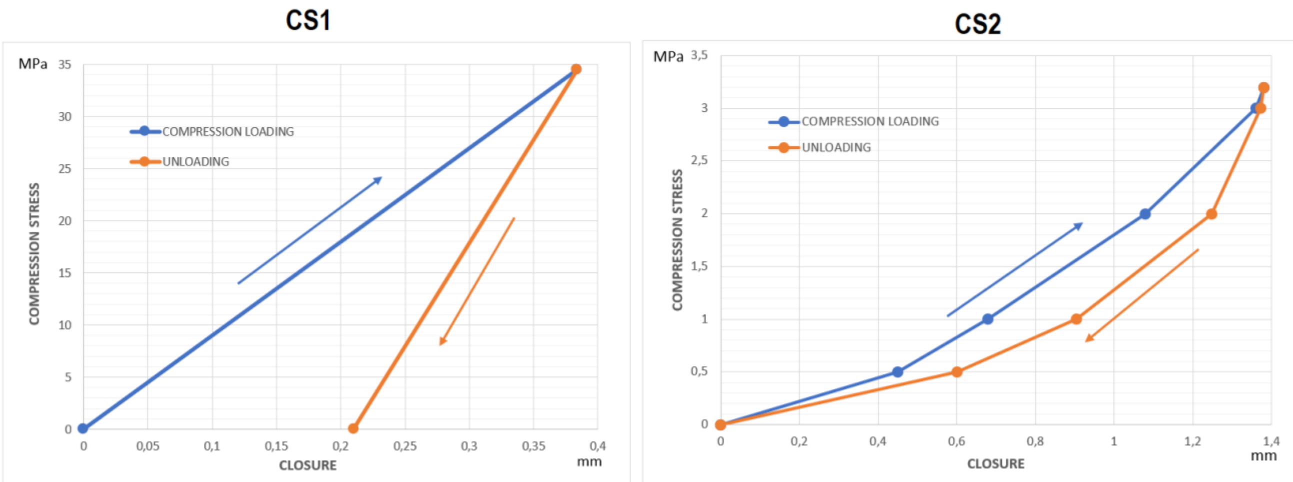
Component	Mesh and Element
Neck	*Hexagonal mesh with 7 elements (2,3 mm) through neck thickness and ratio of 1:3:3 (1 thickness, 3 height, 3 width). Element type: Solid185®[14]*
Flange	*Hex dominant mesh with 10 elements (4,7 mm) through flange thickness and ratio of 1:1:1. Element type: Solid185®[14]*
Blind flange	*Near bolt holes, hex dominant mesh with 7 elements (6,7 mm) through blind flange thickness and ratio 1:1:1. Far bolt holes, hexagonal mesh with 6 elements (7,8 mm) through blind flange thickness and ratio of 1:1:1. Element type: Solid185®[14]*
Bolts	*Hex dominant mesh, element size of 5 mm. Element type: Solid185®[14]*
Gasket	*Hex dominant mesh, element size of 2 mm. Element type: Inter185®[14]*

Figure 6: *SA-516 70 stress-strain curve*



Source: Author

Figure 7: *Gasket loading and unloading curve*



Source: CS1 according [15] and CS2 according [16]

Geometrically and Materially Nonlinear Analysis was performed using the software Ansys Workbench release 2021 R1 [14].

Table 5: Contacts.

Contact	Configuration
Flange-neck	*Bonded*
Flamge-nut	*Bonded*
Blind flange-nut	*Bonded*
Gasket-flange	*Frictional (1)*
Gasket-blind flange	*Frictional (1)*

(1) Friction coefficient = 0,25

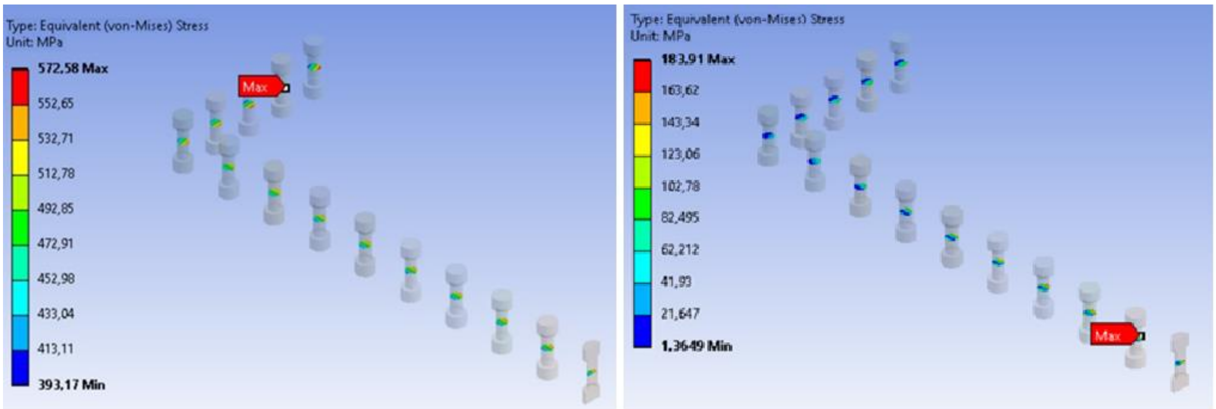
6. RESULTS AND DISCUSSION

For comparison purpose between the proposed procedure and previous existing procedures, the results are organized in Table 6 for CS1 gasket and for CS2 gasket. The name of each column refers to the procedure used: equivalent circular flange with Taylor forge method (ECTF) [10], the procedure presented in the Enquiry case 133 of PD 5500 (133/5500) [3] and lastly the results presented by the proposed procedure by this work (Proposed).

Bolt stresses were also analyzed in the FEA. First, the bolt section that presented the highest stress value was located and then its acceptance criteria were verified (see figure 8 where bolts with highest section stress are indicated). The acceptance criterion used for the bolts is described in paragraph 5.2.2 and 5.7 of ASME VIII Division 2 [12]. In the proposed procedure, the maximum bolt stress (515,5 MPa) exceeds the allowable stress and therefore is unsatisfactory for this application. This same unsatisfactory result was obtained in FEA, where the maximum bolts stress is above the allowable limits. For CS2 the stresses are below the allowable limits and show satisfactory results for both approaches (proposed procedure and FEA).

The paragraph 5.2.4 of the ASME VIII Division 2 [12] was applied to check the acceptability of the flange in the finite element analysis. The obtained stress values are within the code allowable limits. Therefore, the thickness of the flange is acceptable according to the design by analysis and also by the proposed procedure (for CS1 case, the required flange thickness is 39,42 mm and for CS2 case the required flange thickness is 23,65 mm. The finish flange thickness is 47 mm).

Figure 8: *Stresses in bolts sections for CS1 (left) and CS2 (right)*



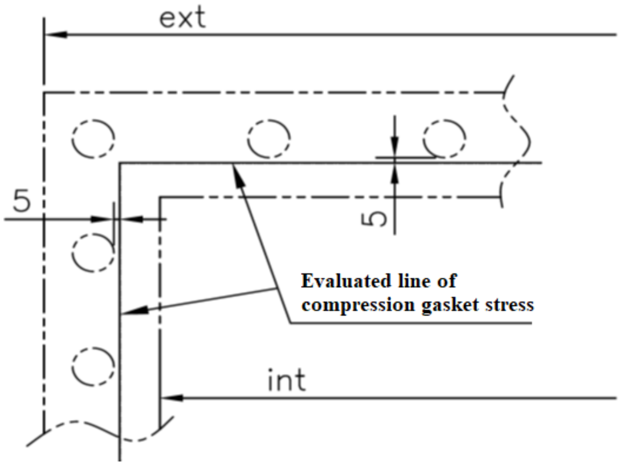
Source: Author

Table 6: Results Comparison.

CS1: Minimum gasket seating stress (y) = 24,13 MPa			
Item	**ECTF**	**133/5500**	**Proposed**
Bolt Load	145,67 kN	125,79 kN	357,75 kN
Bolt Stress	210,0 MPa	181,2 MPa	515,5 MPa
Required Flange Thickness	34,75 mm	39,42 mm	39,42 mm
CS2: Minimum gasket seating stress (y) = 1,4 MPa			
Item	**ECTF**	**133/5500**	**Proposed**
Bolt Load	8,4518 kN	12,799 kN	20,755 kN
Bolt Stress	12,18 MPa	18,44 MPa	29,91 MPa
Required Flange Thickness	28,30 mm	23,65 mm	23,65 mm

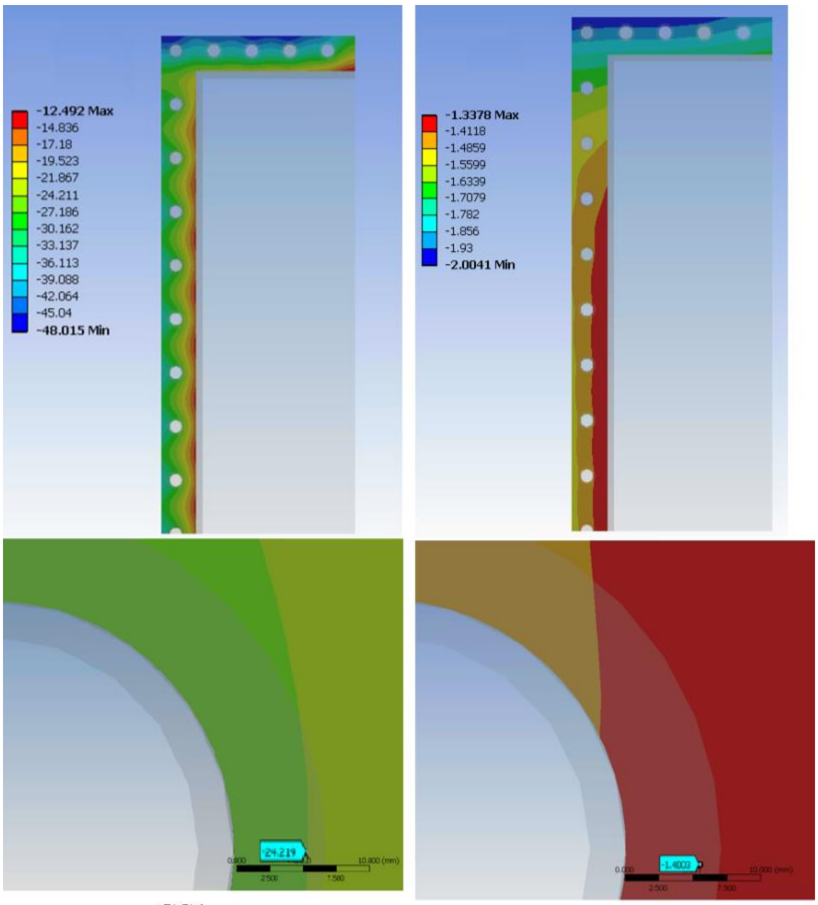
Regarding the gasket compression stress, the evaluated results were taken with 5 mm from the bolt holes (as illustrated in figure 9). For the FEA carried out with bolt pretension calculated in accordance with Enquiry case 133 of PD 5500 [3], the gasket stress observed was lower than the required seating stress of the gasket material. The compression gasket stress for CS1 case observed was 8,787 MPa and for CS2 case this value was 0,867 MPa (remembering the minimum values required are 24,13 MPa and 1,4 MPa respectively). Otherwise, for FEA carried out with bolt pretension calculated by proposed procedure, the observed gasket stress was close to the minimum required stress (see figure 10). CS1 case reached 24,21 MPa (slightly above the required gasket stress) and CS2 case reached 1,4 MPa (exactly the required stress).

Figure 9: *Evaluated line of gasket compression stress*



Source: Author

Figure 10: *Gasket Stress*



CS1 **CS2**

Source: Author

7. CONCLUSION

Based on the results obtained from the proposed procedure and from FEA, it can be concluded that there was an excellent agreement between them. The bolt pretension calculated by proposed procedure presents an exact value of the required seating gasket stress.

According to proposed procedure, it can also be observed that the bolted joint of the CS1 case presented an unsatisfactory design result, since the obtained bolt stresses are greater than the allowable limits. The results presented for CS1 case demonstrate that the bolted joint requires to be modified, with a significant increase of bolt section area (increase number of bolts and increase bolts diameter). The CS2 case, on the other hand, presented a satisfactory design result, with all stresses values bellow the allowable limits. Therefore, as expected, full face bolted flanged joint presents better results when using a softer gasket (as the elastomer used in CS2 case).

The pretension bolt load value calculated by the proposed procedure refers to the minimum required by the gasket material seating. Therefore, from the installation point of view, it is recommended to use a bolt torquing that applies a pretension of up to 50% greater than the bolt pretension calculated by the proposed procedure, in order to increase the compression gasket stress. It is also worth mentioning that it is necessary to follow a correct torquing procedure to achieve uniformity in the gasket stress.

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