2021 International Nuclear Atlantic Conference – INAC 2021
Virtual meeting, Brazil, November 29 – December 2, 2021

ISBN: 978-65-594-1256-3

DESIGN CONSIDERATIONS FOR RECTANGULAR BOLTED FULL FACE FLANGED JOINTS FOR SURFACE CONDENSERS

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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 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 within many 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 PD5500[3], for example, underestimates the necessary bolt pretension to seating the gasket.
It is important to notice that full face bolted flanged connections are more popular in low pressure applications 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 ASME BPVC Code [4].
Appendices 2 and Y of the ASME code Sec. VIII [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 (FFFs) 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]. In this case, it is at the designer's discretion and judgment to apply the design formulas from both Appendix 2 [4] and the stiffness requirements.
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, resulting in a large area of the joint to be compressed. Thus, it is necessary that a great force is applied by the bolts, requiring a great bolt section area. Therefore, the use of this type of flange can become costly for applications with high pressures, which makes this application common and convenient for low pressures equipment.

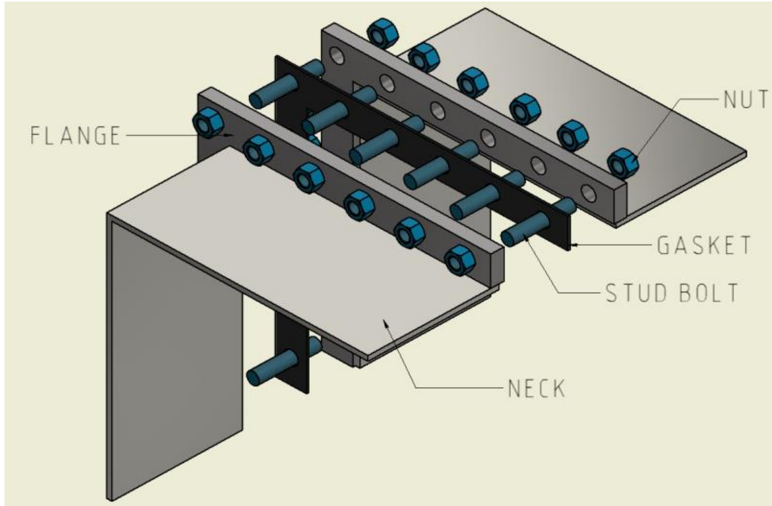

Figure 1: rectangular bolted full-face flanged joint

2. Methodology

The proposed procedure by this paper is based on Enquery 133 of PD 5500 [3] applied to rectangular bolted flanged joints and two changes were done:

1) The area assumed to seating gasket is calculated as A1+A2-A3, 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 was applied to achieve seating stress in the limited line inside bolts lines (5 mm near hole).

The results obtained from the proposed procedure were compared with previous procedures already used in the industry: equivalent circular flange method with the Taylor-Forge full face flange method [9] and the Enquery case 133 of PD5500 [3] and with the results from finite element analysis. Some conclusions and comments were addressed based on the comparisons.

3. 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 [10]. 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.

4. Results and Discussion

For comparison purpose between the proposed procedure and previous existing procedures, the results are organized in table I for compressed non-asbestos fiber gasket and for the NBR elastomer gasket. The name of each column refers to the procedure used: equivalent circular flange with Taylor forge method (**ECTF**) [9], the procedure presented in the enquiry case 133 of PD5500 (**133/5500**) [3] and lastly the results presented by the proposed procedure by this work (**Proposed**).

The finite element model used the symmetry presented by the joint and was discretized with solid elements (figure 2). Flange materials were modeled with elastic-plastic behavior, bolts material was modeled with linear elastic material, and gaskets materials were modeled with a loading and unloading curve characteristic of each material.

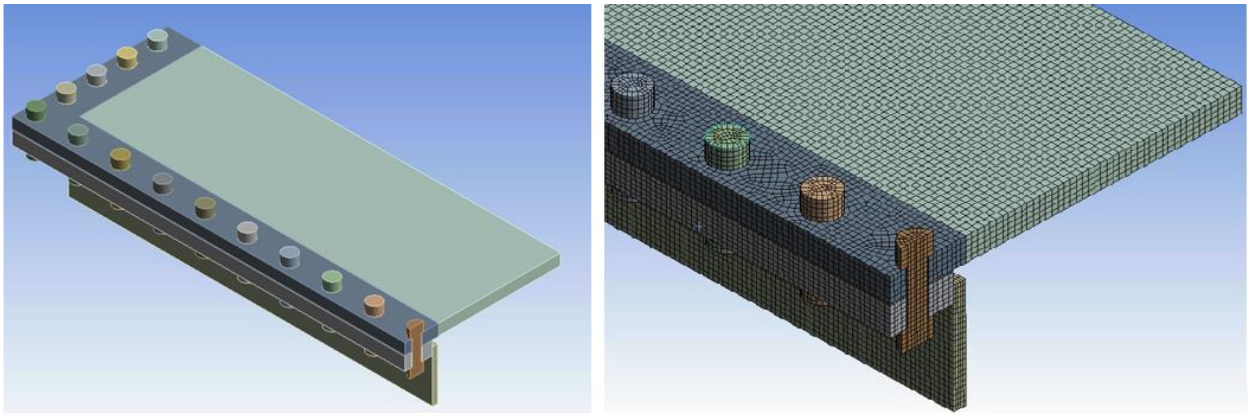

Figure 2: FEA Model

Table I: Results

Compressed non-asbestos fiber: Minimum gasket seating stress (y) = 24,132 MPa			
Item	**ECTF**	**133/5500**	**Proposed**
Bolt Load	7 866 kN	6 792 kN	19 318 kN
Bolt Stress	209,9 MPa	181,2 MPa	515,49 MPa
Required Flange Thickness	36,66 mm	38,93 mm	38,93 mm
NBR elastomer: Minimum gasket seating stress (y) = 1,4 MPa			
Bolt Load	456,4 kN	691,1 kN	1121 kN
Bolt Stress	12,18 MPa	18,44 MPa	29,91 MPa
Required Flange Thickness	28,61 mm	23,36 mm	23,36 mm

Bolt stress was also analyzed in the finite element analysis. The acceptance criterion used in the bolt is described in paragraph 5.2.2 and 5.7 of ASME VIII Div. 2 [11]. In the proposed method, the bolt stress (515,49 MPa) exceeds the allowable stress and therefore is unsatisfactory for this application. This same unsatisfactory result was obtained in FEA, where the bolts stress is above the allowable limits. For CS2 the stresses are below the allowable limits and show satisfactory results for both criterion (proposed procedure and FEA).

For finite element analysis, the paragraph 5.2.4 of the ASME VIII Div. 2 [11] was applied to verify the acceptability of the flange. The obtained stress values are within the code allowable limits. Therefore, the thickness of the flange is satisfactory according to the design by analysis and also by the proposed method (CS1 required 38,93 mm and the finish thickness of flange is 47 mm).

Regarding the gasket seating stress, the results obtained in the FEA were taken with a distance of 5 mm from the bolt hole. For the analysis carried out with bolt pretension calculated in accordance with **133/5500**, a stress value below that required to seating the gasket material was observed: in CS1 the bolt pretension value calculated by the **133/5500** is 6,792 kN and when carrying out the analysis, the gasket stress near bolt holes showed is only 8,787 MPa. This compression gasket stress being below that required for seating the gasket material (24,132 MPa). Gasket seating stress observed in FEA when applying the bolt pretension calculated by the proposed procedure was slightly higher than minimum gasket seating stress (24,21 MPa). In CS2 the gasket stress observed in FEA, when applying the bolt pretension calculated by **133/5500** was 0,867 MPa (below the required gasket seating stress) and the stress observed when apply the bolt pretension calculated by **proposed** procedure was exactly the minimum gasket seating stress (1,4 MPa).

4. Conclusions

Based on the presented values, it is concluded that the proposed procedure presents an excellent agreement of the required bolt pretension, complying with gasket sealing requirements.

From a practical point of view and beholding that the pretension value calculated by the proposed procedure is related to the minimum required, it is recommended that a pretension of up to 1,5 greater be used in order to increase the compression stress on the gasket. It is also worth that it is necessary to follow a correct torqueing procedure for the flanged gasket joint to achieve uniformity in the gasket stress.

It can also be observed that for CS1, the joint configuration is not satisfactory for use, since bolts used will have a stress value above the allowable. To apply the gasket material used in CS1 the joint must be modified, with a significant increase in available area of bolt sections (increase number of bolts and/or increase bolt diameter). Therefore, it is evident that FF type joints should use softer gaskets such as CS2 (rubber).

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