Transactions on Engineering Sciences vol 10, © 1996 WIT Press, www.witpress.com, ISSN 1743-3533

FRP bolted flanged connections subjected to longitudinal bending moments

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Abstract

Bolted flanged connections for fiber reinforced plastic (FRP) pipes and pressure vessels are of great importance for any user of FRP material in fluid containment applications. At present, no dimensional standards or general design rules exist for FRP flanges. Most often, flanges are fabricated to dimensional standards for metallic flanges without questioning their applicability to FRP materials.

Flanges in FRP piping systems are often subjected to external, longitudinal bending moments of considerable magnitude. In piping systems, such bending moments are usually caused by thermal expansion of the piping components themselves, or by thermal expansion of pressure vessels and other equipment to which the piping system is connected. Such bending moments are of utmost importance on FRP piping systems due to the relative stiffness of most FRP materials.

None of the pertinent design codes, neither the ASME Code, Section X, Fiber-Reinforced Plastic Pressure Vessels, nor the ASME/ANSI B31.3 Code for Chemical Plant and Petroleum Refinery Piping, contain design rules for external bending moments on FRP flanges.

In previous papers by the same author, the effect of such external bending moments on metallic flanges were investigated and compared with a design criterion listed in the ASME Code, Section III, Nuclear Power Plant Equipment, based on an "equivalent internal pressure". An analysis of the stresses in the flange-bolt assembly due to external bending moments was proposed and results were compared with flange thicknesses using the equivalent internal pressure. Calculated stresses were also compared with results from strain gage measurements on a test pressure vessel.

To verify stress values in FRP flanges under external bending, a test rig is being used to perform strain gage measurements on a test pipe with a bolted flanged connection, subjected to internal pressure and longitudinal bending.

1 Introduction

Bolted flanged connections on pressure vessels and in piping systems are often subjected to bending moments of considerable magnitudes. On pressure vessels, longitudinal bending moments are mostly the result of external forces such as wind or seismic loads; in piping systems, such external moments are usually caused by thermal expansion.

Two interpretive literature surveys, one by Blach and Bazergui [1]*, the other by Cassidy and Kim [2], show that very little has been published on the problem of bolted flanged connections with external loads, even for metallic flanges. Most early researchers on bending moments on steel flanges concentrated their efforts on the behavior of bolts in such connections and assumed metal-to-metal contact between the flanges. The flange-gasket-bolt interaction which exists in externally loaded flanged connections was usually neglected.

Even for metallic flanges, design Codes such as the ASME BPV Code, Section VIII, Division 1 [3], or the ASME-ANSI B31.3 [4] Piping Code, do not contain rules for the design of bolted flanged connections with external loads. In the pertinent design codes for FRP flanges, very little can be found regarding this problem. The ASME Code, Section X, Fiber Reinforced Plastic Pressure Vessels [5], does not mention external loads on flanges. The ASME/ANSI B31.3, Chemical Plant and Petroleum Refinery Piping [4], Chapter VII, Nonmetallic Piping, covers this problem with a note only: "Nonmetallic flanges shall be adequate to develop the full rating of the joint and to withstand expected external loadings".

The only exception is the ASME BPV Code, Section III, Nuclear Power Plant Components, Division 1 [6], whose Subsections NB, NC, and ND contain a mandatory formula to convert external bending moments to an equivalent internal pressure to be added to the design pressure. The expression given is based on an arbitrary factor of $16/\pi$ and makes no allowance for different flange geometries such as the outside to inside diameter ratio "K".

In this paper an attempt is made to analyse the stresses produced in an FRP bolted flanged connection subject to bending moments and axial loads. Results are then compared with the criterion of the Nuclear Code [6], requiring an equivalent internal pressure. A simple design method is proposed which may be used with the Pressure Vessel or Piping Codes [4], [5].

(*) Numbers in square brackets [ ] indicate references listed at the end of this paper.

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2 Equivalent internal pressure of the nuclear code

The Subsections NB, NC, and ND of the ASME Code, Section III, Nuclear Power Plant Components [6], include a mandatory requirement for bolted connections subjected to external bending moments. This requirement expresses the bending moment in terms of an equivalent internal pressure which must be added to the design pressure of the bolted connection, namely

$$P_e = \frac{16M}{\pi G^3} \tag{1}$$

where "G" is the effective gasket diameter as specified in the Code.

Using the design rules for bolted flanged connections given in [3] and [6], the flange operating moment for a flange with ring gasket may be written in a way similar to equation (1)

$$M_O = H_D h_D + H_G h_G + H_T h_T$$

$$M_O = \frac{\pi}{4} B^2 P h_D + 2b\pi G m P h_G + \frac{\pi}{4}(G^2 - B^2) P h_T$$

$$M_O = \frac{\pi G^3}{16}\left(\frac{4B^2}{G^3}(h_D - h_T) + \frac{32bmh_G}{G^2} + \frac{4h_T}{G}\right)P$$

$$\frac{16M_O}{PG^3\pi} = \lambda \tag{2}$$

where

$$\lambda = \frac{4B^2}{G^3}(h_D - h_T) + \frac{32bmh_G}{G^2} + \frac{4h_T}{G} \tag{3}$$

The nomenclature used in (3) is the same as the one used in both, Sections III [6] and VIII [3] of the Code. It can be seen that the factor λ is independent of moment and pressure and depends only on the geometry of flange and gasket.

Comparing (1) and (2)

$$\frac{P_e G^3}{M} = \frac{16}{\pi} \qquad \frac{PG^3}{M_O} = \frac{16}{\pi\lambda}$$

it can be seen that for compatibility between internal and equivalent external pressure, λ should be unity, or should at least be constant. This, however, is not the case, as can be seen in Table 1 which gives values of λ for some standard flanges per ASME-ANSI B16.5, Class 150 [7], which are used extensively for FRP pipe flanges.

Table 1 shows that the values of λ differ for each pipe size.

Nom. Pipe Size	K*	λ
4	2.848	2.186
6	2.061	1.612
8	1.879	1.282
10	1.747	1.011
12	1.708	.909
14	1.736	1.025
16	1.672	.892
18	1.623	.787
20	1.584	.725
24	1.584	.670

(*) K is based on STD pipe wall thickness

Table 1, Factors K and λ

3 Eccentric flange load design method

To describe the effects of eccentric loading on a bolted flanged connection, the model shown in Fig. 1 is used. The flange bolt area is assumed to be evenly distributed along the bolt circle "C", the gasket compression area is taken along the effective gasket diameter "G", with an effective width of "2b", using the Code nomenclature of [3] and [5]. For simplicity in establishing geometric relations, the gasket area is placed along the bolt circle and compensated in width.

A stiffness reduction factor "n" is used to relate the relative stiffness of the gasket to the flange bolts. Numerical values of "n" are difficult to find in literature. Schwaigerer [8] gives some values for certain gasket materials. Values vary from 1.2 for solid metal gaskets to 20 for soft mineral fiber composition gaskets.

From geometric relations of Figure 1, parameters can be calculated, based on a variation of α or "k", relating bolt side (tension) and gasket side (compression) stresses. From external loads such as bending moment and axial force, the location of the neutral axis can be found and the bolt and gasket stresses evaluated. An iterative procedure to find "k" can be found in [9].

For a given "k", found by iteration, parameters "z", "i" and "j" are calculated, also parameters C_S and C_C as shown below:

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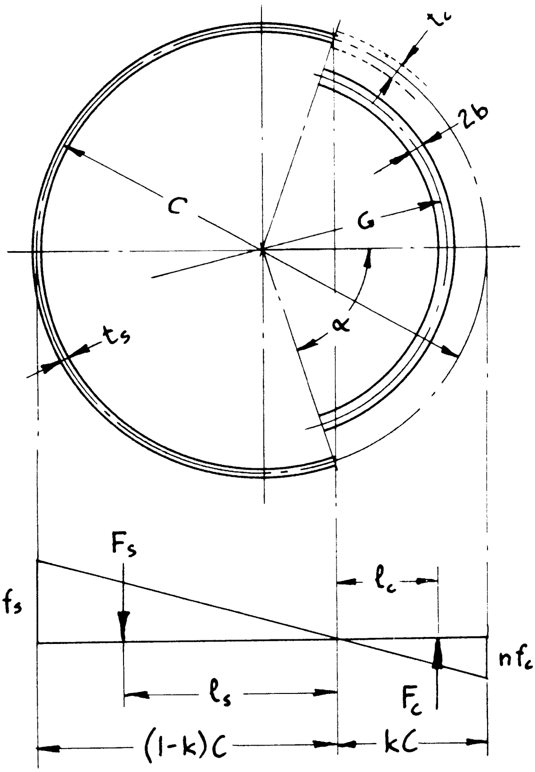

Fig. 1, Geometry of Eccentrically Loaded Flange

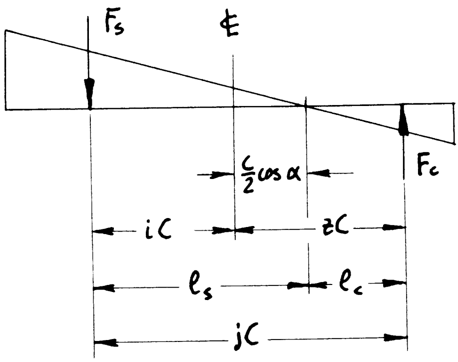

Fig. 2, Moment Arms

$$C_S = 2\left[\frac{(\pi-\alpha)\cos\alpha - \sin\alpha}{1+\cos\alpha}\right] \tag{4}$$

$$C_C = 2\left[\frac{\sin\alpha - \alpha\cos\alpha}{1-\cos\alpha}\right] \tag{5}$$

$$\ell_S = \frac{C}{2}\left[\frac{(\pi-\alpha)\cos^2\alpha + 3/2\sin\alpha\cos\alpha + 1/2\,(\pi-\alpha)}{(\pi-\alpha)\cos\alpha + \sin\alpha}\right] \tag{6}$$

$$\ell_C = \frac{C}{2}\left[\frac{\alpha\cos^2\alpha - 3/2\sin\alpha\cos\alpha + \alpha/2}{\sin\alpha - \alpha\cos\alpha}\right] \tag{7}$$

$$z = \frac{\ell_C}{C} + \frac{\cos\alpha}{2} \tag{8}$$

$$i = \frac{\ell_S}{C} - \frac{\cos\alpha}{2} \tag{9}$$

$$j = \frac{\ell_S + \ell_C}{C} \tag{10}$$

Stresses are then calculated. For moment loading only, without axial force, from equilibrium

$$M = f_S j C \qquad F_C = F_S$$

$$f_S = \frac{2\pi M}{C_S A_S j C} \tag{11}$$

$$f_C = \frac{2M}{C_C t_C j C^2} \tag{12}$$

$$pn = \frac{nA_S}{2bG\pi} \tag{13}$$

$$k = \frac{1}{1 + C_C G / C_S (pn) C} \tag{14}$$

A "k" is found by iteration, then parameters "z", "i", "j", C_S and C_C

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are calculated with sufficient accuracy (3 decimal places). The stresses in bolts and gasket can now be obtained using (11) and (12).

For a combined moment plus axial force, a similar procedure, using a dimensionless eccentricity in the calculation oparameters and stresses, is given in Reference [9].

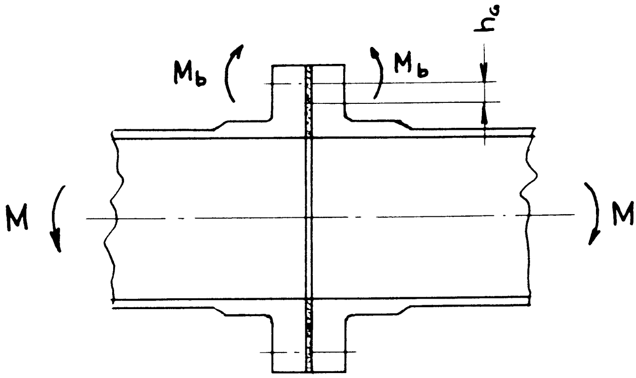

Fig. 3, Flange Bending Moment

The maximum flange bending moment due to the external loads can now be calculated. From Figure 3 it can be seen that the maximum flange bending occurs at the tension (bolt) side of the bolted connection. On the compression (gasket) side, the effects of the external bending moment are subtractive of the effects of internal pressure, hence will not govern the flange design. Thus

$$M_b = H_b h_G = f_S A_S h_G \tag{15}$$

$$P_e = \frac{16\, M_b}{G^3 \pi \lambda} \tag{16}$$

Equation (16) gives an equivalent internal pressure, in most cases smaller than the pressure obtained using equation (1), for the following reasons: for standard pipe flanges, the flange bending moment M_b is smaller than the external moment M; also, for many pipe sizes and gasket types, it is larger than unity. The flange bending moment M_b can be larger than the external moment only in the case of large flange moment arms.

4 Full face gasket flange design

In many instances, FRP flanges are fabricated to dimensions taken from flange standards for metallic flanges. The ASME Code, Section X, recommends that flanges to ANSI B16.5, Steel Flanges [7], be used in nozzles on FRP pressure Vessels. Section X [5] also contains design rules for full face gasketed bolted flanges in Article RD-11.

While, for simplicity, early FRP flange applications used the geometry for steel flanges, it soon became obvious that flanges with ring gaskets were not really suitable for FRP material, due to the high bending stresses induced in the flange. Hence the full face gasket, once used extensivly with cast iron flanges, became the gasket of choice for FRP flanges.

For full face gasketed flanges, no generally recognized design rules are contained in the pertinent codes for metallic pressure vessels or piping. While such flanges were always used for certain specialized applications, it was left to designers to qualify these flanges for given pressure-temperature conditions. It became customary to use a design method by the Taylor-Forge Company [10], in which a full face gasket is simulated by two ring gaskets, one lying inside, the other outside the bolt circle. Using this method, very heavy flanges are obtained.

A new design method, based on an elastic analysis was proposed by Blach et al. [11], also by Blach & Hoa [12], and compared with experimental data and results of the Taylor-Forge method. It was shown that the Taylor-Forge method yields thicknesses very much on the safe side, while results using [11] are closer to actually measured stresses. In [12], the serious problem of pull-back on hand lay-up flanges is also treated.

In the following, a numerical example is given in which an equivalent internal pressure is calculated using the method of this paper. This equivalent internal pressure, added to the design pressure of the flange, is used to calculate the required thickness of the FRP flange, employing both, the Taylor-Forge method [10] and the method by Blach et al. [11].

5 Numerical example

An FRP flange of 300 mm diameter, dimensionally equivalent to ANSI B16.5, Class PN-20, is subjected to the following operating conditions:

Design Code	ASME BPV Code, Sect. X
Design Pressure	700 kPa
Design Temperature	150°C
Gasket	3 mm compressed asbestos
Flange Material	Glass fiber reinforced Phenolic resin
Bending Moment	4,600 N-m
Axial Force	1,200 N

For the 300 mm, Class PN-20 flange, twelve bolts M-24, with a stress area of 353 mm^2 are used on a bolt circle of 432 mm. For an outside diameter of 485 mm and a full face gasket 92.5 mm wide, "G" is calculated to be 400 mm (see Ref. [1]) and used in (1) to obtain the equivalent internal pressure of 367 kPa. This equivalent pressure is then added to the design pressure and a combined pressure of 717 kPa is used in the design of the flange.

This total pressure is more than twice the required internal pressure and, using the design method prescribed in the ASME Code, Section X, the "Taylor-Forge" method, which, for a design pressure of 350 kPa, requires a flange thickness of 48 mm. For the combined pressure of 717 kPa, the flange thickness increases to 56 mm.

However, using the method described in this paper, the following parameters are obtained:

$$
\begin{aligned}
k &= .5048 \\
C_S &= 1.898 \\
C_C &= 2.011 \\
z &= .392 \\
j &= .785
\end{aligned}
$$

Bolt stress, flange bending moment, and equivalent internal pressure due to the external bending moment are now calculated.

$$
\begin{aligned}
f_S &= 10.7 \text{ MPa} \\
M_b &= 1800 \text{ N·m} \\
P_e &= 143 \text{ kPa} \\
P &= 350 + 143 = 493 \text{ kPa}
\end{aligned}
$$

The total internal pressure calculated using the method of this paper requires a flange thickness of 50 mm in order to remain within the allowable stresses of the ASME Code.

6 Conclusions

The empirical formula given in the ASME Code, Section III, to convert an external bending moment into an equivalent internal pressure, does not take into account the flange geometry, nor the characteristics of the gasket used. In Reference [9] it was shown that for some standard pipe flanges, the effects of external bending moments are less than the formula given in the Code; however, for certain flange geometries, especially for those with a low "K" value (narrow flanges), the external bending moment may have a greater influence on the flange than the one calculated using the Code formula.

The design procedure proposed may be used in FRP flange applications where external bending moments and axial forces are present, be it on pressure vessels per Section X of the ASME BPV Code, or in the Piping Code ASME-ANSI B31.3.

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