

EVALUATION OF NOZZLE LOADS ON FLAT BOLTED CIRCULAR COVERS

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ABSTRACT

Current pressure vessel design codes do not provide for evaluation of nozzle loads on flat bolted covers. In practice, centrally placed nozzles on flat bolted covers are particularly common. Such nozzles to which piping is connected must be assessed for unacceptable piping reactions. This article provides for evaluating the nozzle loads, taking into account the internal pressure together with the external loads acting on the nozzle. The influence of the nozzle load on the flat cover with mating flange is considered. The approximate methodology described uses available formulas for flat plate structures from recognized sources.

Keywords: Flat bolted cover, Nozzle, Piping reactions, Nozzle load, Approximate methodology.

I. INTRODUCTION

The method is limited to evaluating the stresses around the centrally placed nozzle in the flat plate. The loads exerted on the nozzle that are included in the evaluation are successively the radial load **F** and the resulting bending moment **M**. Shear loads and the torsional moment are omitted because of their negligible influence on the stresses in the vicinity of the nozzle intersection. The degree of restraint of the considered flat plate also influences the intensity of the local stresses around the nozzle in the plate. The two extreme forms of the degree of restraint are "simply supported" and "clamped". In practice, an intermediate form will usually occur. However, the design codes for pressure vessels including ASME BPV Code Section VIII - Division 1(USA)[1],

PD 5500 (UK)[2] and EN 13445-3 (EU)[3] assume the simply supported condition when calculating bolted circular flat covers where the stress at the centre of the plate is decisive. For bolted flat ends, gross plastic deformation is prevented by limiting the bending stress in the centre of the end to a value of 1.5 times the nominal stress. This assures that the yield strength of the material will not be exceeded when the nominal design stress is limited by the material properties. For the evaluation of the loads on a centrally placed nozzle, we assume a "simply supported" cover which is in line with the strength calculation of the cover on internal pressure loading according to paragraph UG-34 equation (2)[1].

II. BACKGROUND and METHODOLOGY

Formulas for calculating the stresses around the nozzle caused by external loads are mainly taken from the book "Roark's Formulas for Stress & Strain" [4] and "The Complete Part Design Handbook" by E. Alfredo Campo [5]. The induced unit line load intensities for the radial load **F** and the resulting bending moment **M** are respectively:

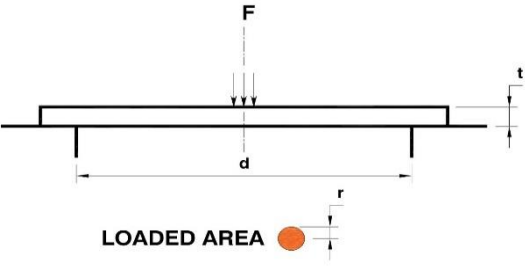

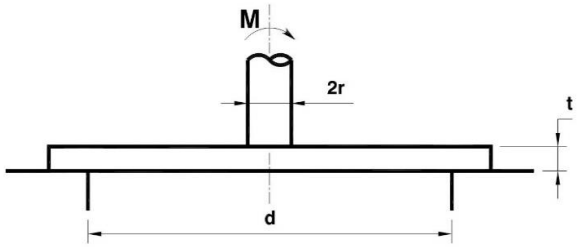
$$f_F = \frac{F}{2\pi r t} \text{ and } f_M = \frac{M}{2\pi r^2 t}.$$

Nomenclature of the symbols used is displayed below.

NOMENCLATURE					
F or F _E	Radial load	N	r	Outside radius nozzle	mm
M or M _E	Bending moment	Nmm	t	Thickness flat plate	mm
p	Internal pressure	MPa	d or G	Gasket load reaction diameter	mm
W	Total bolt force	N	C	Bolt circle diameter	mm
h _G	Gasket moment arm	mm	σ _{max,F}	Stress due to F	MPa
ν	Poisson's ratio	0.3	σ _{max,M}	Stress due to M	MPa

p_e	Equivalent pressure	MPa	$\sigma_{max,pe}$	Stress due to p_e	MPa
P_R	Flange rating pressure	MPa	P_D	Flange design pressure	MPa
t_n	Nominal nozzle thk's	mm	d_n	Internal nozzle diameter	mm

The framed representation below provides an overview of the relevant formulas with which the local stress due to the nozzle load around the nozzle intersection can be determined.

Overview of the formulas for the calculation of local stress at the nozzle - flat cover intersection																					
<p>Circular Plate, Centre Load (F), Simply Supported</p>  <p style="text-align: center;">LOADED AREA </p>	<p>Circular Plate, Centre Bending Moment (M), Simply Supported</p> 																				
$\sigma_{max,F} = \frac{3F}{2\pi t^2} \left[(1 + \nu) \ln\left(\frac{0.5 d}{r}\right) + 1 \right]$ <p>If $r < 0.5 t$ then r should be taken as: $\left(\sqrt{1.6 r^2 + t^2} \right) - 0.675 t$ </p> <p>Substituting $f_F = \frac{F}{2\pi r t}$ into the equation for $\sigma_{max,F}$ then we end up with:</p> $\sigma_{max,F} = \frac{3 f_F}{t} \left[(1 + \nu) \ln\left(\frac{0.5 d}{r}\right) + 1 \right]$ <p>Alternative formula for the calculation of the bending stress in the centre of a supported circular flat plate with a central concentrated load F on radius r.</p> <p>Formula taken from AISI, Committee of Steel Plate Engineering Data, Volume 2. (Imperial Units)</p> $\sigma_b = 1.43 \left[\log_{10} \left(\frac{R}{r} \right) + 0.334 + 0.06 \left(\frac{t}{R} \right)^2 \right] \frac{F}{t^2}$ <p>With: $R = 557.2/2 = 278.6 \text{ mm} = 10.9685 \text{ inch}$, $t = 57 \text{ mm} = 2.2441 \text{ inch}$, $r = 219.1/2 = 109.55 \text{ mm} = 4.313 \text{ inch}$, $F = 4450 \text{ N} = 1000.4 \text{ lbf}$ $\sigma_b = 210.7456 \text{ psi} = 1.453 \text{ MPa}$</p>	$\sigma_{max,M} = \frac{3M}{4\pi r t^2} \left[1 + (1 + \nu) \log \frac{4(0.5 d - r)}{K d} \right]$ <p>With: $K = \frac{0.1225 d^2}{(r+0.35 d)^2}$</p> <p>Substituting $f_M = \frac{M}{2\pi r^2 t}$ into the equation for $\sigma_{max,M}$ then we end up with:</p> $\sigma_{max,M} = \frac{1.5 f_M r}{t} \left[1 + (1 + \nu) \log \frac{4(0.5 d - r)}{K d} \right]$ <p>Alternative formula : $\sigma = 2 \frac{\beta M}{d t^2}$</p> <table border="1" style="width: 100%; text-align: center;"> <tr> <td>$\frac{2r}{d}$</td> <td>0.1</td> <td>0.15</td> <td>0.2</td> <td>0.25</td> </tr> <tr> <td>β</td> <td>9.478</td> <td>6.252</td> <td>4.621</td> <td>3.625</td> </tr> <tr> <td>$\frac{2r}{d}$</td> <td>0.3</td> <td>0.4</td> <td>0.5</td> <td>0.6</td> </tr> <tr> <td>β</td> <td>2.947</td> <td>2.062</td> <td>1.489</td> <td>1.067</td> </tr> </table> <p>Linear interpolation between the various $2r/d$ values is allowed.</p>	$\frac{2r}{d}$	0.1	0.15	0.2	0.25	β	9.478	6.252	4.621	3.625	$\frac{2r}{d}$	0.3	0.4	0.5	0.6	β	2.947	2.062	1.489	1.067
$\frac{2r}{d}$	0.1	0.15	0.2	0.25																	
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β	2.947	2.062	1.489	1.067																	

It should be realized that the structural flange behaviour is influenced by the acting nozzle loads. The most obvious solution to account for this effect in the flange/cover calculation is to convert the nozzle loading into an equivalent pressure. The equivalent internal pressure can be calculated from the following expressions:

$p_e = p + \frac{4F}{\pi d^2} + \frac{16 M}{\pi d^2 C}$ or from the well known and more conservative "Kellogg" formula for the converted equivalent pressure: $p_e = p + \frac{4F_E}{\pi G^2} + \frac{16 M_E}{\pi G^3}$. For the flanges the following condition should be satisfied: $p_e \leq P_R$

Note that F or F_E can be omitted in case this force is **not** jacking apart the cover - mating flange connection.

As per[1] paragraph UG-44(b) the following expression must be satisfied for the nozzle flange loads:

$$16 M_E + 4 F_E G \leq \pi G^3 [(P_R - P_D) + F_M P_R]$$

Maximum stress in the middle of the flat cover due to internal pressure and edge moment derived from equation (2) of UG-34 [1]

$$\sigma_{\max,pe} = \frac{3(3+\nu)}{8} \frac{p_e \left(\frac{d}{2}\right)^2}{t^2} + \frac{6 W h_G}{\pi d t^2}$$

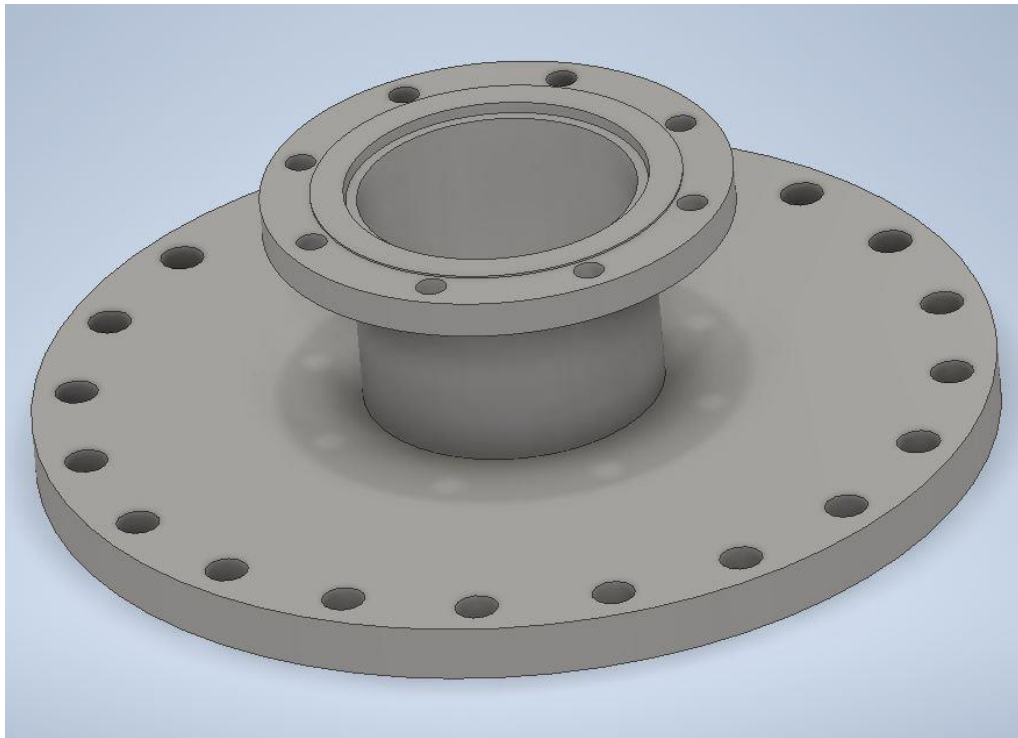


Figure 1: Illustration of typical flat cover configuration with centrally located nozzle

III. DISCUSSION

In the case of a circular flat bolted cover fitted with a centrally located flanged nozzle that is exposed to both internal pressure and piping reactions acting on the nozzle, the following aspects should be considered:

- The pressure acting on the circular plate
 - Local stresses at the vicinity of the nozzle as a result of piping reactions acting on the nozzle
 - The influence of the piping reactions on the structural flange behaviour of the circular cover plate attached to the mating flange
 - The weakening effect of the nozzle on the circular flat plate
 - The edge moment caused by the flange bolts
 - Imposed nozzle loads on flanged nozzle
- These aspects are addressed in Section II.

IV. APPROACH

1. Evaluate the imposed loads on the nozzle flange according to paragraph UG-44(b) of [1].
2. Calculate the equivalent pressure of the mating flange to the circular bolted cover taking the nozzle loadings into account.
3. Evaluate the imposed loads on the mating flange according to paragraph UG-44(b) of [1].
4. Determine the bolt load W of the mating flange considering the internal design pressure plus the equivalent pressure of the mating flange.
5. Calculate the required thickness of the flat bolted cover according to paragraph UG-34 [1] for the applicable configuration taking into account the design pressure and bolt load W .
6. Determine the required reinforcement in the flat bolted cover to compensate for the weakening effect of the nozzle opening according to section UG-39[1].
7. Calculate the local stresses around the nozzle taking into account both the internal pressure and the external loads F and M applied to the nozzle.
8. Evaluate the total locally occurring stress around the nozzle taking into account the actual thickness of the bolted flat cover adjacent to the nozzle.

This approach will be further elaborated in the worked example.

V. WORKED EXAMPLE

The assembly (see Figure 2) is composed of the following parts:

- A. Welding neck flange NPS 20" RF Class 150 made from forged A105 material and equipped with a Spiral Wound Gasket (SS) according to ASME B16.20[7].
- B. A circular flat bolted cover is connected to the flange mentioned under A and is also made of forged A105 material.
- C. A (set-on type) NPS 8" Schedule XS with a stand-out of 200 mm is welded into the heart of the bolted flat cover. The material of this nozzle is A 106 Grade B.
- D. The nozzle is equipped with an NPS 8" RF welding neck flange Class 150 and fitted with a Spiral Wound Gasket (SS) according to ASME B16.20 [7] to which pipework is connected. The flange material is forged A105 material

The piping reactions acting on the NPS 8" nozzle are a radial tensile force of 4.45 kN and a resulting bending moment of 8500 Nm. The internal design pressure is 5 bar and the design temperature is 150 C. Rated flange pressure 15.8 bar @ 150°C according ASME B16.5 [8]. Corrosion allowance: nil.

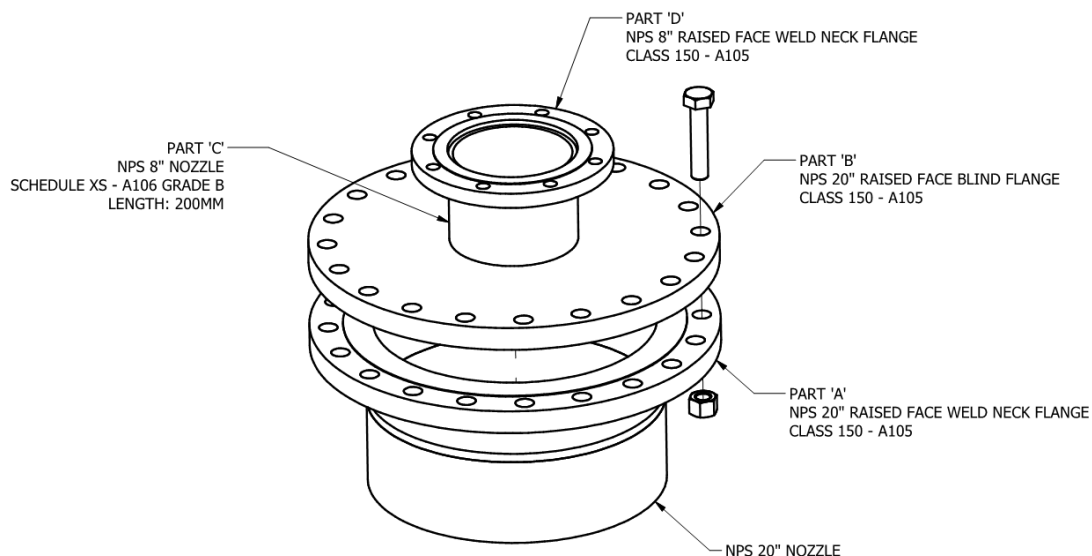


Figure 2: Blind assembly

VI. EVALUATIONS

- **Checking nozzle loads on NPS 8" WN RF Flange**

Determination of gasket reaction diameter "G"

OD Gasket = 263.7 mm ; Bead = 1.5 mm ; Effective OD Gasket: $263.7 - 2 \times 1.5 = 260.7$ mm

ID Gasket = 233.7 mm ; Gasket contact width $N = (260.7 - 233.7) 0.5 = 13.5$ mm

$b_0 = 13.5/2 = 6.75$ mm ; $b = 2.52\sqrt{6.75} = 6.55$ mm ; $G = 260.7 - 2 \times 6.55 = 247.6$ mm

Satisfying condition: $16 M_E + 4 F_E G \leq \pi G^3 [(P_R - P_D) + F_M P_R]$ as per paragraph UG – 44(b). [1]

For Class 150 $\Rightarrow F_M = 1.2$

$$16 \times 8500000 + 4 \times 4450 \times 247.6 \leq \pi \times 247.6^3 [(1.58 - 0.5) + 1.2 \times 1.58]$$

$$140407280 \leq 141917098 \Rightarrow \text{Condition satisfied}$$

- **Checking nozzle loads on NPS 20" WN RF Flange**

Determination of gasket reaction diameter "G"

OD Gasket = 577.9 mm ; Bead = 1.5 mm ; Effective OD Gasket: $577.9 - 2 \times 1.5 = 574.9$ mm

ID Gasket = 525.5 mm ; Gasket contact width $N = (574.9 - 525.5) 0.5 = 24.7$ mm

$b_0 = 24.7/2 = 12.35$ mm ; $b = 2.52\sqrt{12.35} = 8.85$ mm ; $G = 574.9 - 2 \times 8.85 = 557.2$ mm

Satisfying condition: $16 M_E + 4 F_E G \leq \pi G^3 [(P_R - P_D) + F_M P_R]$ as per paragraph UG – 44(b). [1]

For Class 150 $\Rightarrow F_M = 1.2$

$$16 \times 8500000 + 4 \times 4450 \times 557.2 \leq \pi \times 557.2^3 [(1.58 - 0.5) + 1.2 \times 1.58]$$

$$145918160 \leq 161395099 \Rightarrow \text{Condition satisfied}$$

Note:

An approximate calculation has also been performed according to WRC Bulletin 107 [6] assuming a sphere with a very large diameter tending to a flat plate and fitted with an NPS 8 Sch. 80 (XS) nozzle on it. The maximum calculated stress was 32.87 MPa.

- **Determination of the bolt force W for the NPS 20" WN flange connecting the circular flat cover**

The nozzle loads that are exerted on the NPS 8" Class 150 WNRF-A105 flange also act on the connection of the flat cover with the mating NPS 20" Class 150 WNRF - A105 flange.

To calculate the bolt force (W) of this connection, it is first necessary to determine the equivalent pressure using the so-called Kellogg equivalent pressure method.

$$p_e = p + \frac{4F_E}{\pi G^2} + \frac{16 M_E}{\pi G^3} = 0.5 + \frac{4 \times 4450}{\pi \times 557.2^2} + \frac{16 \times 8500000}{\pi \times 557.2^3} = 0.7685 \text{ MPa}$$

$$W_{m1} = \frac{\pi}{4} \cdot G^2 \cdot p_e + (2b \cdot \pi \cdot G \cdot m \cdot p_e)$$

$$W_{m1} = \frac{\pi}{4} \times 557.2^2 \times 0.7685 + (2 \times 8.85 \times \pi \times 557.2 \times 3 \times 0.7685) = 258827 \text{ N}$$

$$W_{m2} = \pi \cdot b \cdot G \cdot y = \pi \times 8.85 \times 557.2 \times 69 = 1068940 \text{ N}$$

The flange contains 20 x 1 1/8" UN bolts, material A193 B7 with $S_a = 172$ MPa

The total bolt root area is $20 \times 470 = 9400 \text{ mm}^2 = A_b$

$$\text{The required bolt area } A_m = \frac{1068940}{172} = 6214.8 \text{ mm}^2$$

$$W = \frac{(A_m + A_b) S_a}{2} = \frac{(6214.8 + 9400) 172}{2} = 1342873 \text{ N}$$

$$h_G = (C - G) 0.5 = (635 - 557.2) 0.5 = 38.9 \text{ mm}$$

- **Determination of required thickness of circular flat bolted cover fitted with nozzle**

Minimum required thickness of circular flat bolted cover according [1] follows from equation (2) of UG-34

$$t = d \sqrt{\frac{CP}{SE} + \frac{1.9 W h_G}{SE d^3}} = d \sqrt{\frac{0.3 \times 0.5}{138 \times 1.0} + \frac{1.9 \times 1342873 \times 38.9}{138 \times 1.0 \times 557.2^3}} = 40.35 \text{ mm}$$

No additional reinforcement is required if the thickness is greater than $\sqrt{2}$ times t resulted in $t = 57.06$ mm, provided that the reinforcement zone of the nozzle in the flat plate remains within diameter d.

Compensation of the nozzle weakening in the circular flat bolted plate can be calculated according to paragraph UG-39 [1].

Note reinforcement is only required for operating conditions not bolting up.

$S_n = 118$ MPa for A106 Grade B material and $S_v = 138$ MPa for A105 material

$f_{r1} = S_n / S_v = 118 / 138 = 0.855$; $t_n = 12.7$ mm; $d_n = 219.1 - 2 \times 12.7 = 193.7$ mm

Required nozzle thickness : $t_{rn} = \frac{d_n}{2} \left[\exp \left(\frac{p}{S_n \cdot E} \right) - 1 \right] = \frac{193.7}{2} \left[\exp \left(\frac{0.5}{118 \times 1.0} \right) - 1 \right] = 0.411$ mm

Cross-sectional area of reinforcement: $A_r = 0.5 d_n t = 0.5 \times 193.7 \times 40.35 = 3907.9$ mm²

Reinforcement area nozzle neck: $A_n = 5 \times t_n (t_n - t_{rn}) f_{r1} = 5 \times 12.7 (12.7 - 0.411) 0.855 = 667.2$ mm²

Required area to be compensated as excess thickness in the flat plate:

$A_e = A_r - A_n = 3907.9 - 667.2 = 3240.7$ mm²

Limiting diameter of the reinforcing pad is : $2 \times d_n = 2 \times 193.7 = 387.4$ mm = D_p

Required thickness of circular flat cover follows from:

$A_e = 3240.7 = (t_{req'd} \cdot d - t) (D_p - d_n) \Rightarrow 3240.7 = (t_{req'd} \cdot d - 40.35) (387.4 - 193.7) \Rightarrow t_{req'd} = 57$ mm

Weld area attaching the nozzle to the flat cover has been neglected.

- **Maximum occurring stress in the centre of the flat cover taking into account equivalent pressure and edge moment**

$$\sigma_{\max,pe} = \frac{3(3+\nu) P_e \left(\frac{d}{2}\right)^2}{8 t^2} + \frac{6 W h_G}{\pi d t^2} = \frac{3(3+0.3) \cdot 0.7685 \left(\frac{557.2}{2}\right)^2}{8 \cdot 57^2} + \frac{6 \times 1342873 \times 38.9}{\pi \times 557.2 \times 57^2} = 62.6825 \text{ MPa}$$

- **Stress in flat bolted cover due to radial nozzle load 'F'**

$$f_F = \frac{F}{2\pi r t} = \frac{4450}{2\pi \times 0.5 \times 219.1 \times 57} = 0.11342 \text{ N/mm}$$

$$\sigma_{\max,F} = \frac{3 f_F}{t} \left[(1 + \nu) \ln \left(\frac{0.5 d}{r} \right) + 1 \right] = \frac{3 \times 0.11342}{57} \left[(1 + 0.3) \ln \left(\frac{0.5 \times 557.2}{0.5 \times 219.1} \right) + 1 \right] = 1 \text{ MPa}$$

- **Stress in flat bolted cover due to bending moment 'M' on nozzle**

$$f_M = \frac{M}{2\pi r^2 t} = \frac{8500000}{2\pi \times (0.5 \times 219.1)^2 \times 57} = 1.9776 \text{ Nmm/mm} ; K = \frac{0.1225 d^2}{(r+0.35 d)^2} = \frac{0.1225 \times 557.2^2}{(0.5 \times 219.1 + 0.35 \times 557.2)^2} = 0.41$$

$$\sigma_{\max,M} = \frac{1.5 f_{M,r}}{t} \left[1 + (1 + \nu) \log \frac{4(0.5 d - r)}{K d} \right]$$

$$\sigma_{\max,M} = \frac{1.5 \times 1.9776 \times 0.5 \times 219.1}{57} \left[1 + (1 + 0.3) \log \frac{4(0.5 \times 557.2 - 0.5 \times 219.1)}{0.41 \times 557.2} \right] = 9.194 \text{ MPa}$$

Alternative formula: $\sigma = 2 \frac{\beta M}{dt^2} \Rightarrow 2r/d = 219.1/557.2 = 0.3932 \Rightarrow \beta = 2.0625$

$$\sigma = 2 \frac{\beta M}{dt^2} = 2 \frac{2.0625 \times 8500000}{557.2 \times 57^2} = 19.37 \text{ MPa}$$

- **Stress evaluation**

Total bending stress in the flat plate : $\sigma_{\max,pe} + \sigma_{\max,F} + \sigma_{\max,M} = 62.6825 + 1 + 9.194 = 72.8765$ MPa

When applying the alternative formulas for **M** and **F**: $62.6825 + 1.453 + 19.37 = 83.5055$ MPa

Stress ratios: $72.8765 / 138 = 0.52 < 1.0$ respectively $83.5055 / 138 = 0.605 < 1.0 \Rightarrow$ Passed

Taking into account the stress approximated by applying WRC Bulletin 107, the stress ratio becomes:
 $(62.6825 + 32.87) / 138 = 95.5525 / 138 = 0.692 \Rightarrow$ Passed

VII. CONCLUSIONS

Nozzle loads that the nozzle flange can withstand pose barely threat to the structural integrity of a circular flat bolted cover. Depending on the chosen method for assessing or evaluating the nozzle loads, a considerable spread can be seen in the calculation results. However, this does not mean that extra thickness is necessary for the flat cover that is subjected to nozzle loads that are considered permissible for the nozzle flange. This should be a satisfactory thought for the design engineer.

ACKNOWLEDGEMENT

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VIII. REFERENCES

- [1] ASME BPV Code Section VIII - Division 1: 2021
- [2] PD 5500: 2021 + A2: 2022; Specification for unfired pressure vessels
- [3] EN 13445-3: 2021; Unfired pressure vessels Part 3: Design
- [4] Roark's Formulas for Stress & Strain; McGraw -Hill: 2020
- [5] The Complete Part Design Handbook, by E. Alfredo Campo: 2006
- [6] WRC - Bulletin 107 "Local Stresses in Spherical and Cylindrical Shells due to External Loadings"; 1965
- [7] ASME B16.20 - 2017 "Metallic Gaskets for Pipe Flanges"
- [8] ASME B16.5 - 2020 "Pipe Flanges and Flanged Fittings: NPS 1/2 " through NPS 24 "