American Journal of Engineering Research (AJER)
e-ISSN: 2320-0847 p-ISSN : 2320-0936
Volume-9, Issue-3, pp-277-285
www.ajer.org
Research Paper Open Access

Review of Buffer Approach to Compensate Unknown Nozzle Loads

* Walther Stikvoort
Consultant Static Pressure Equipment and Structural Integrity
Wagnerlaan 37, 9402 SH, Assen, The Netherlands (NL)
** Corresponding Author: Walther Stikvoort*

ABSTRACT: *ANNEX V of EN 13445-3 (amendment A8: 2019) states that a buffer should be considered for unknown nozzle loads that relate to the opening design. That is, if the external loads at the nozzle are unknown, for example the pipe stress analysis was not available or was carried out to EN 13480-3, Annex Q (simplified pipe stress analysis), a buffer should be taken into account. A well structured view is provided of the proposed approach in Annex V and also examines the consequences of this for the design of the pressure vessel. In addition, available alternatives to the Annex V approach are put into perspective, so that a more nuanced insight is obtained.*

KEYWORDS: *nozzle loads, external loads, opening design, buffer.*

Date of submission: 13-03-2020 Date of acceptance: 28-03-2020

I. INTRODUCTION

The approach according to Annex V [2] of EN 13445-3 [1] first of all requires the calculation of the buffer quantity "bn", which depends on the shell thickness at the location of the nozzle intersection and the thickness of any reinforcing pad that is arranged around the nozzle.
The buffer should be calculated by:

$$bn = \max\left[\,0.3\,;\,1 - \frac{3}{e_{as} + e_p} - \frac{1}{(e_{as} + e_p - 3)^2}\,\right]$$

Successively the reduced cross-sectional area effective as reinforcement (shell stress loaded area) must be calculated with the following formulas:
Stress loaded cross-sectional area for set-in nozzle:

$$A_{fs} = l_s' \,.\, e_{c,s} \,.\, bn$$

Stress loaded cross-sectional area set-on nozzle:

$$A_{fs} = (l_s' + e_b)\,.\, e_{c,s} \,.\, bn$$

The A_{fs} values calculated in this way must be applied in the usual nozzle reinforcement (compensation) calculation according to EN 13445-3 and should comply with: $\Phi_P = P / P_{max} \leq 1.0$
Consult the appendix for more detailed information.

Symbol	Description	Unit
A_{fs}	Stress loaded cross-sectional area effective as reinforcement.	mm^2
bn	Buffer factor.	-

e_b	Effective thickness of nozzle (or mean thickness within the external length l_{bo} or internal length l_{bi} taken into account for reinforcement calculation.	mm
$e_{c,s}$	Assumed shell thickness of shell wall (see equation 9.5-2 of EN 13445-3) for checking of reinforcement of an opening. The thickness may be assumed by designer between the minimum required shell thickness e and the shell analysis thickness $e_{a,s}$. This assumed thickness shall then be used consistently in all requirements. NOTE: For $e_{c,s}$ the shell analysis thickness may be used always, but sometimes it may be advantageous to use a smaller assumed value to obtain smaller distances from adjacent shell discontinuities.	mm
e_p	Effective thickness of reinforcing plate taken into account for reinforcement calculation.	mm
$e_{a,s}$	Analysis thickness of shell wall or mean analysis thickness within the length l'_s and excluding the thickness of the reinforcing pad if fitted.	mm
l'_s	Effective length of shell for opening reinforcement.	mm

Figure 1 shows the variation of the buffer values "bn" as a function of the thickness (e_{as} + e_p) for the wall thickness domain between 5.525 and 50 mm. As the wall thickness increases, the "bn" value also increases. The strongest increase in the "bn" value occurs with relatively small wall thicknesses and proceeds almost asymptotically with large wall thicknesses.

Figure 1: Wall thickness dependent buffer values

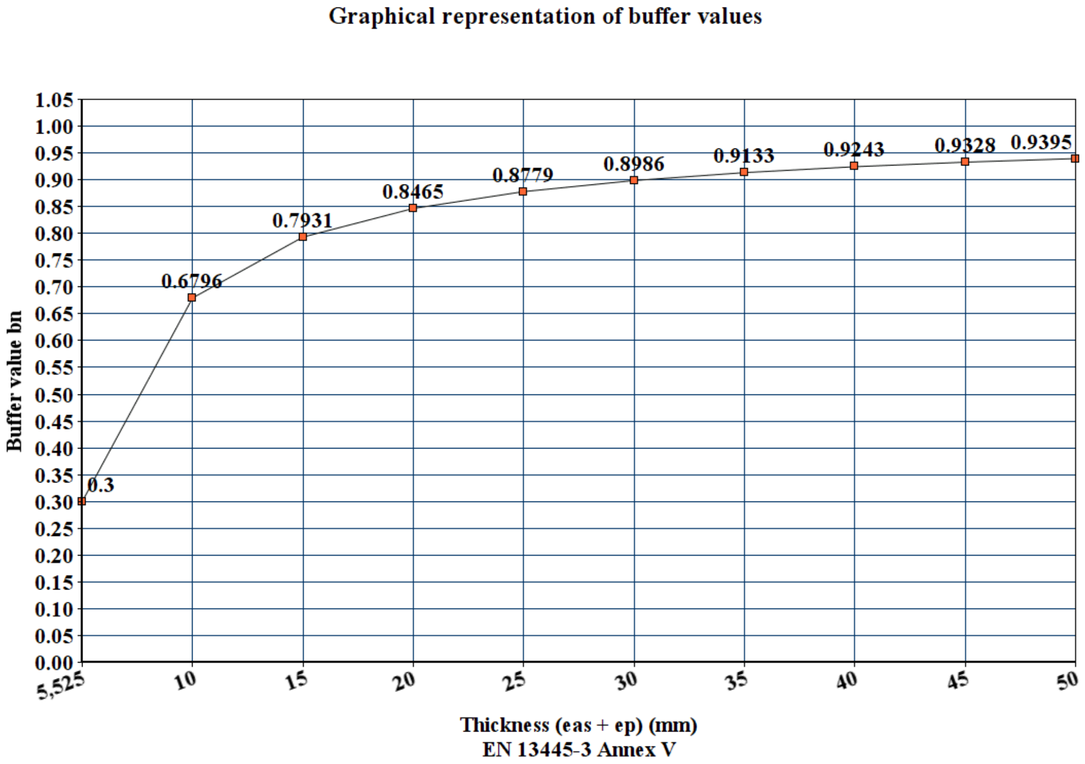


Table 1: Tabulated buffer values

e_{as} + e_p (mm)	bn	e_{as} + e_p (mm)	bn	e_{as} + e_p (mm)	bn
5.525	0.3	20	0.8465	35	0.9133
6	0.3889	21	0.8541	36	0.9157
7	0.5089	22	0.8609	37	0.9181
8	0.5850	23	0.8671	38	0.9202
9	0.6389	24	0.8727	39	0.9243
10	0.6796	25	0.8779	40	0.9243
11	0.7116	26	0.8827	41	0.9261

12	0.7149	27	0.8871	42	0.9279
13	0.7592	28	0.8913	43	0.9296
14	0.7774	29	0.8951	44	0.9312
15	0.7931	30	0.8986	45	0.9328
16	0.8066	31	0.9019	46	0.9342
17	0.8184	32	0.9051	47	0.9342
18	0.8289	33	0.9080	48	0.9370
19	0.8382	34	0.9107	49	0.9383
				50	0.9395

II. ELABORATED CASE

A pressure vessel is designed for an internal design pressure of 9 bar (0.9 MPa) and a design temperature of + 200° C. The vessel shell is made from A515 Grade 60 carbon steel and has an outside diameter of 1200 mm and a thickness of 8 mm. The vessel cylindrical shell is provided with a nozzle: NPS 12" (NB 300) Schedule Std. made from A 106 Grade B carbon steel. The nominal nozzle neck thickness is 9.53 mm and the outside diameter is 323.9 mm. Tolerance of nozzle neck is - 12.5 %. No reinforcing pad is added. Shell plate tolerance is set on zero and the corrosion allowance is nil. The throat dimension of the outside nozzle filled weld is 6 mm.The method used of determining both the internal pressure capacity (P_{max}) as well as the determination of the allowable individual loads ($F_{Z\ max}$, $M_{X\ max}$ and $M_{Y\ max}$) will be performed in accordance with EN 13445-3.

Vessel and nozzle data

Outside diameter cylindrical shell (mm)	D_e	1200
Combined analysis thickness of the shell and reinforcing pad (mm)	e_c	8
Mean shell diameter at the opening (mm)	D	1192
Outside nozzle diameter (mm)	d_e	323.9
Nozzle analysis thickness (0.875 x 9.53) (mm)	e_b	8.34
Mean nozzle diameter (mm)	d	315.56

Design condition vessel

Internal design pressure (MPa)	P	0.9
Design temperature (°C)	ϑ_m	200
Corrosion allowance (mm)	ca	0
Thickness tolerance shell	tol	0
Thickness tolerance nozzle neck		12.5%

Materials and properties

Part	Material	Yield strength (MPa)	Tensile strength (MPa)	Yield strength @ [ϑ_m] (MPa)	Design stress [f] (MPa)
Cylindrical shell	A515 Gr.60	220	415	190	126.67 = f_s
Nozzle neck	A106 Gr. B	240	415	182	121.33 = f_{ob}

CASE#1

Determination of maximum allowable pressure

Maximum length of shell contributing to opening reinforcement:
$l_{so} = [(1200 - 8)\ 8]^{0.5} = 97.65$ mm
Length of nozzle contributing to reinforcement:
$l_{bo} = [(323.9 - 8.34)\ 8.34]^{0.5} = 51.30$ mm
Shell stress loaded area:
$A_{fs} = (97.65 \times 8) = 781.2$ mm^2
Nozzle stress loaded area:
$A_{fb} = (51.30 \times 8.34) + 8 \times 8.34 = 494.562$ mm^2
Weld stress loaded area:
$A_{fw} = 0.5 \times 6\sqrt{2} \times 6\sqrt{2} = 36$ mm^2
Pressure loaded area in shell:
$A_{ps} = 592\ (97.65 + 8.34 + 153.61) = 153683.2$ mm^2

Pressure loaded area in nozzle:
A_{pb} = (51.30 + 8) 153.61 = 9109.073 mm^2
Maximum allowable pressure:

$$P_{max} = \frac{(A_{fs} + A_{fw})\, f_s + A_{fb}\,.\, f_{ob}}{(A_{ps} + A_{pb}) + 0.5\,(A_{fs} + A_{fw} + A_{fb})}$$

$$P_{max} = \frac{(781.2+36)126.67 + 494.562 \times 121.33}{(153683.2+9109.073) + 0.5\,(781.2+36+494.562)} = 1.0 \text{ MPa}$$

Individual load ratio pertaining to the pressure capacity of the connection:
$\Phi_P = P / P_{max}$ = 0.9 / 1.0 = 0.9 < 1.0
Determination of maximum allowable individual nozzle loads:
Limitations:
$\lambda_c = d/(D.e_c)^{0.5}$ = 315.56 / (1192 x 8)$^{0.5}$ = 3.231 ≤ 10 ; d/D = 315.56 /1192 = 0.2647 ≤ 0.5
e_{as}/D = 8 /1192 = 0.00671 ≤ 0.1
Calculation factors:
$C_1 = \max[(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 1.81]$
C_1 = max [(0.60072181 + 0.95196257(3.231) + 0.0051957881(3.231)^2 - 0.001406381(3.231)^3 + 0(3.231)^4); 1.8]
C_1 = max [3.68375 ; 1.81] = 3.68375
$C_2 = \max[(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 4.90]$
C_2 = max [4.526315 + 0.064021889(3.231) + 0.15887638(3.231)^2 - 0.021419298(3.231)^3 + 0.0010350407(3.231)^4); 4.90]
C_2 = max [5.78233; 4.90] = 5.78233
$C_3 = \max[(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 4.90]$
e_b / e_c = 8.34/8 = 1.0425 > 0.5
for e_b / e_c > 0.5
C_3 = max [(4.8588639 + 2.1870887(3.231) + 1.4567053(3.231)^2 - 0.3316430(3.231)^3 + 0.0253850(3.231)^4); 4.90]
C_3 = max [18.71484; 4.9] = 18.71484

Maximum allowable axial force:
$F_{Z,max} = f.\, e_c^2\, .\, C_1$ = 126.67 x 8^2 x 3.68375 = 29863.7 N ≈ 29.9 kN
Maximum allowable circumferential moment:
$M_{X,max} = f.\, e_c^2(d/4).C_2$ = 126.67 x 8^2 x (315.56 / 4) x 5.78233 = 3698099.30 Nmm ≈ 3.7 kNm
Maximum allowable longitudinal moment:
$M_{Y,max} = f.\, e_c^2(d/4).C_3$ = 126.67 x 8^2 x (315.56 / 4) x 18.71484 = 11969108.9 Nmm ≈ 12 kNm

Summary of maximum allowable individual loads @ nozzle-cylindrical shell intersection

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	29.9	3.7	12

Just for comparison reasons find below the results from WRC 297 [9] analysis

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	24.6	3.3	12.2

CASE#2

Application of Annex V of EN 13445 - 3
Buffer factor bn = 0.585
$A_{fs} = l'_s\, .\, e_{c,s}\, .\, bn$ = 97.65 x 8 x 0.585 = 457 mm^2 = Reduced stress loaded cross-sectional area

$$P_{max} = \frac{(457+36)126.67 + 494.562 \times 121.33}{(153683.2+9109.073) + 0.5\,(457+36+494.562)} = 0.75 \text{ MPa}$$

Φ_P = 0.9 / 0.75 = 1.2 > 1.0

To achieve a Φ_P of ≤ 1.0 , a thicker shell or thicker nozzle neck is required.
First we will opt for a thicker nozzle neck, namely Schedule XS (12.7 mm) instead of Std (9.53 mm).
The net wall thickness of the nozzle neck becomes: 12.7 x 0.875 = 11.1125 mm
$l_{bo} = [(323.9 - 11.1125)\ 11.1125]^{0.5} = 58.96$ mm
$A_{fb} = (58.96 \times 11.1125) + 8 \times 11.1125 = 744.1\ mm^2$
$A_{ps} = 592\ (97.65 + 11.1125 + 150.8375) = 153683.2\ mm^2$
$A_{pb} = (\ 58.96 + 8\ )\ 150.8375 = 10100\ mm^2$

$$P_{max} = \frac{(\ 457 + 36)126.67 + 744.1 \times 121.33}{(\ 153683.2 + 10100) + 0.5\ (457 + 36 + 744.1)} = 0.929 \text{ MPa}$$

$\Phi_P = 0.9 / 0.929 = 0.9688 < 1.0 \Rightarrow$ condition satisfied

Limitations associated with changed nozzle neck thickness:
$\lambda_c = d/(D.e_c)^{0.5} = 312.7875 / (1192 \times 8)^{0.5} = 3.203 \leq 10$; $d/D = 312.7875 / 1192 = 0.2624 \leq 0.5$
$e_{as}/D = 8 / 1192 = 0.00671 \leq 0.1$
Because of the small differences with the previously calculated parameters for the thinner nozzle neck, it can be said that the calculated allowable individual nozzle loads are hardly subject to change. This has been demonstrated in the displayed summary below. The deviation for this case is less than 2%.

Summary of maximum allowable individual loads @ nozzle-cylindrical shell intersection

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	29.6	3.7	11.8

Just for comparison reasons find below the results from WRC 297 [9] analysis

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	23.6	3.2	12

The alternative option is that the wall thickness of the nozzle neck remains nominally 9.53 mm, but with an increased shell thickness. We assume a shell thickness of 10 mm. The calculation proceeds as follows:

CASE#3

Determination of maximum allowable pressure

Maximum length of shell contributing to opening reinforcement:
$l_{so} = [(1200 - 10)\ 10]^{0.5} = 109.09$ mm
Length of nozzle contributing to reinforcement:
$l_{bo} = [(323.9 - 8.34)\ 8.34]^{0.5} = 51.30$ mm
Reduced shell stress loaded area:
$A_{fs} = (109.09 \times 10\ )\ 0.6796 = 741.36\ mm^2 \Rightarrow$ with bn = 0.6796
Nozzle stress loaded area:
$A_{fb} = (51.30 \times 8.34) + 10 \times 8.34 = 511.24\ mm^2$
Weld stress loaded area:
$A_{fw} = 0.5 \times 6\sqrt{2} \times 6\sqrt{2} = 36\ mm^2$
Pressure loaded area in shell:
$A_{ps} = 590\ (109.09 + 8.34 + 153.61) = 159913.6\ mm^2$
Pressure loaded area in nozzle:
$A_{pb} = (\ 51.30 + 10\ )\ 153.61 = 9416.293\ mm^2$
Maximum allowable pressure:

$$P_{max} = \frac{(\ A_{fs} + A_{fw}\ )\ f_s\ +\ A_{fb}\ .\ f_{ob}}{(\ A_{ps} + A_{pb}\ ) +\ 0.5\ (A_{fs} + A_{fw}\ +\ A_{fb}\ )}$$

$$P_{max} = \frac{(\ 741.36 + 36)126.67 + 511.24 \times 121.33}{(\ 159913.6 + 9416.293) + 0.5\ (741.36 + 36 + 511.24)} = 0.944 \text{ MPa}$$

Individual load ratio pertaining to the pressure capacity of the connection:
$\Phi_P = P / P_{max} = 0.9/ 0.944 = 0.953 < 1.0$
Determination of maximum allowable individual nozzle loads:
Limitations:
$\lambda_c = d/(D.e_c)^{0.5} = 315.56 / (1192 \times 10)^{0.5} = 2.893 \leq 10$; $d/D = 315.56 /1190 = 0.2652 \leq 0.5$
$e_{as}/D = 10 /1190 = 0.0084 \leq 0.1$
Calculation factors:
$C_1 = \max[(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 1.81]$
$C_1 = \max [(0.60072181 + 0.95196257(2.893) + 0.0051957881(2.893)^2 - 0.001406381(2.893)^3 + 0(2.893)^4); 1.8]$
$C_1 = \max [3.36393 ; 1.81] = 3.36393$
$C_2 = \max [(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 4.90]$
$C_2 = \max [4.526315 + 0.064021889(2.893) + 0.15887638(2.893)^2 - 0.021419298(2.893)^3 + 0.0010350407(2.893)^4); 4.90]$
$C_2 = \max [5.59497; 4.90] = 5.59497$
$C_3 = \max[(a_0 + a_1.\lambda_c + a_2.\lambda_c^2 + a_3.\lambda_c^3 + a_4.\lambda_c^4); 4.90]$
$e_b / e_c = 8.34/10 = 0.834 > 0.5$
for $e_b / e_c > 0.5$
$C_3 = \max [(4.8588639 + 2.1870887(2.893) + 1.4567053(2.893)^2 - 0.3316430(2.893)^3 + 0.0253850(2.893)^4); 4.90]$
$C_3 = \max [17.12482; 4.9] = 17.12482$

Maximum allowable axial force:
$F_{Z,max} = f. e_c^2 . C_1 = 126.67 \times 10^2 \times 3.36393= 42610 \text{ N} \approx 42.6 \text{ kN}$
Maximum allowable circumferential moment:
$M_{X,max} = f. e_c^2(d/4).C_2 = 126.67 \times 10^2 \times (315.56 / 4) \times 5.59497= 5591000 \text{ Nmm} \approx 5.59 \text{ kNm}$
Maximum allowable longitudinal moment:
$M_{Y,max} = f. e_c^2(d/4).C_3 = 126.67 \times 10^2 \times (315.56 / 4) \times 17.12482= 17112000 \text{ Nmm} \approx 17.1 \text{ kNm}$

Summary of maximum allowable individual loads @ nozzle-cylindrical shell intersection

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	42.6	5.6	17.1

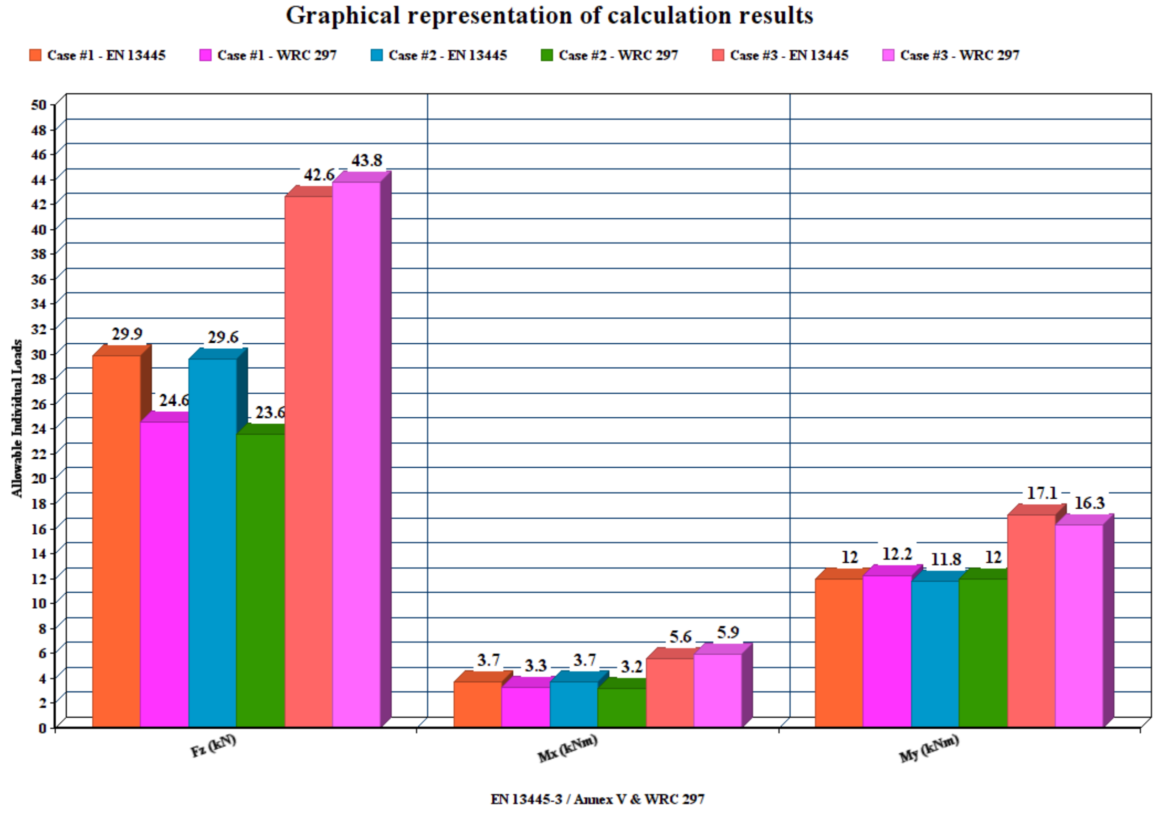
Just for comparison reasons find below the results from WRC 297 [9] analysis

Loading	**$F_{Z,max}$ (kN)**	**$M_{X,max}$ (kNm)**	**$M_{Y,max}$ (kNm)**
Intensity	43.8	5.9	16.3

Table 2: Overview calculation results

Loads	**Basic design CASE #1**	**Including Annex V Increased neck thickness CASE #2**	**Including Annex V Increased shell thickness CASE #3**
	e_c = 8 mm	**e_c = 8 mm**	**e_c = 10 mm**
	e_b = 8.34 mm	**e_b = 11.1125 mm**	**e_b = 8.34 mm**
P (bar)	9	9	9
P_{max}	10	9.29	9.44
F_Z (kN) EN 13445-3	29.9	29.6	42.6
F_Z (kN) WRC 297	24.6	23.6	43.8
M_X (kNm) EN 13445-3	3.7	3.7	5.6
M_X (kNm) WRC 297	3.3	3.2	5.9
M_Y (kNm) EN 13445-3	12.0	11.8	17.1
M_Y (kNm) WRC 297	12.2	12.0	16.3

Figure 2: Allowable Individual Loads



Key: F_Z = Allowable Individual Axial Load ; M_X = Allowable Individual Circumferential Moment;
M_Y = Allowable Individual Longitudinal Moment

III. DISCUSSION

One may wonder whether the inclusion of Annex V in EN 13445-3/A8 makes sense. Annex V aims to be an aid in designing nozzles in pressure vessels, while the nozzle loads are still unknown. In order to achieve this, a so-called buffer factor has been introduced that leads to a reduction of the shell stress loaded area (A_{fs}). The degree of reduction depends exclusively on the wall thickness of the pressure vessel (including an eventual installed reinforcing pad) at the location of the nozzle intersection. The smaller the wall thickness, the greater the reduction of A_{fs}. It appears that in the domain of the smallest wall thickness the buffer factor is the smallest, so the reducing effect on the shell stress loaded area is greatest (see Figure 1). The purpose of applying such a buffer factor is to ensure that a certain amount of over-design is created so that external loads (piping reactions) can be better accommodated. However, this is not apparent from the elaborated case. In the case of small wall thicknesses at the nozzle intersection, there is in fact a more or less flexible connection, as a result of which the piping reactions will in reality be substantially lower than in the case of a rigid connection with larger wall thicknesses. In addition, EN 13445-3 [1] clauses 16.4 and 16.5 offers the possibility to determine the load capacity of the nozzles and thus provide insight into the allowable nozzle loads for a specific nozzle configuration. Most design agencies and manufacturers of pressure equipment usually have validated software at their disposal. However, one must also realize that the piping loads exerted on the nozzle flange can be decisive instead of the local stresses around the nozzle intersection. It is recommended to consult references [3] to [8], so that more insight can be gained regarding the necessity of applying Annex V. Experience has shown, however, that preference should be given to a pressure vessel designed exclusively to withstand internal pressure for which the individually allowable nozzle loads can be determined according to conventional methods. The piping system connecting to the nozzle must be routed in such a way that the piping reactions exerted on the nozzle can be adjusted against the individually allowable nozzle loads by making use of a so-called Load Interaction Rule. The bottom line is that the connecting piping system must be designed in such a way that the load limits of the relevant pressure vessel nozzle are met.

IV. CONCLUSIONS

It appears that when applying Annex V where the so-called buffer factor is introduced, this has consequences for the pressure vessel. In order to be able to guarantee sufficient internal pressure capacity at a reduced stress loaded area, it is necessary to increase the original nozzle neck thickness or a choice can be made to maintain the original nozzle neck thickness and to increase the shell wall thickness. Of course, applying a reinforcing pad around the nozzle is also possible, whereby the original shell thickness and nozzle neck thickness can remain unchanged. By applying Annex V, changes to the pressure vessel are in any case necessary, which will have a cost-increasing effect. If we look at Table 2, we hardly see any differences in the allowable individual nozzle loads between the basic design case and the case with the thickened nozzle neck. However, if we compare the basic case with the case where the shell thickness has been increased by 25% to 10 mm, then the differences in allowable individual nozzle loads are considerable. The question arises as to whether such an increase in allowable individual nozzle loads is necessary in view of the fact that the nozzle flexibility with a thicker shell wall increases and the piping reactions (nozzle loads) will also actually increase. In practical cases, it is generally sufficient to design the pressure vessel and nozzle configuration as economically as possible, whereby no over-design is created in connection with external nozzle loads. In the references one can find sufficient guidelines which have proven themselves successfully in practice over many years.

REFERENCES

[1]. EN 13445 "Unfired Pressure vessels" Part 3 - Design ;issue 5:2018
[2]. EN 13445 "Unfired Pressure Vessels" Part 3 - Design ; Addendum A8;2019
[3]. Walther Stikvoort (2018) "Load Capacity Limits of Flanged Pressure Vessel Nozzles" Pet Petro Chem Eng 2(3): 1-6.
[4]. Walther Stikvoort. "Determination of nozzle loads to facilitate the initial pressure vessel design" - Chemical Engineering, October 26, 2018.(online publication)
[5]. Walther Stikvoort "Standardized nozzle loads for the initial pressure vessel design" Int.J.Inno.Sci.Res.Vol.7, No:08, pp.1239-1242, August,2018.
[6]. Walther Stikvoort "Yardstick for the evaluation of nozzle loads" American Journal of Engineering Research (AJER), Vol.8, Issue 3, 2019, pp. 293-298.
[7]. Walther Stikvoort "Overview of different approaches for completing the nozzle design of pressure vessels" International Journal of Engineering Development and Research (IJEDR), 2018, Volume 6, Issue 4 ISSN: 2321-9939.
[8]. Walther Stikvoort "Review of traditional tubular pad reinforced nozzles versus Long Welding Neck Flange nozzles in pressure vessels" Journal of Research in Mechanical Engineering, Volume 5 - Issue 1 (2019) pp: 01-07.
[9]. J. L. Mershon " Local Stresses on Cylindrical Shells due to Loadings on Nozzles"-WRC ; Welding Research Council, 1987.

APPENDIX

Figure A.1: Illustration of pressure - and stress loaded areas pertaining to flush set-in nozzle configuration

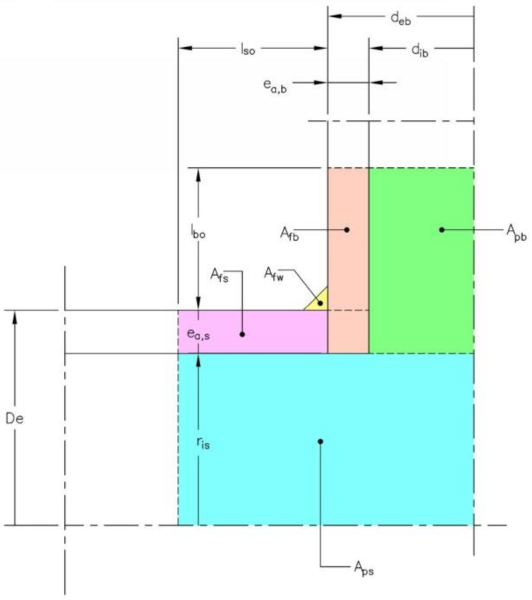


Table A.1: Notations and formulas of isolated opening reinforced by branch according figure A.1

Symbol	Description	Unit
d_{eb}	External diameter of a nozzle fitted in a shell	mm
$e_{a,b}$	Analysis thickness of nozzle	mm
d_{ib}	Internal diameter of a nozzle fitted in shell	mm
D_e	External diameter of a cylindrical shell	mm
$e_{a,s}$	Analysis thickness of shell	mm
r_{is}	Inside radius of curvature of the shell at the opening centre	mm
l_{so}	Maximum length of shell contributing to opening reinforcement $= [(D_e - e_{a,s})\, e_{a,s}]^{0.5}$	mm
l_{bo}	Maximum length of nozzle outside the shell for reinforcement $= [(d_{eb} - e_{a,b})\, e_{a,b}]^{0.5}$	mm
e_{fw}	Throat thickness of filled weld	mm
A_{ps}	Pressure loaded area within shell $= r_{is}\,(l_{so} + 0.5\, d_{eb})$	mm^2
A_{pb}	Pressure loaded area within branch $= 0.5\, d_{ib}\,(l_{bo} + e_{a,s})$	mm^2
A_{fb}	Cross-sectional area of branch within compensation limits $= e_{a,b} \,.\, (l_{bo} + e_{a,s})$	mm^2
A_{fs}	Cross-sectional area of shell within compensation limits $= e_{a,s} \,.\, l_{so}$	mm^2
A_{fw}	Cross-sectional area of filled weld between nozzle and shell $= e_{fw}^{\ 2}$	mm^2
A_f	Stress loaded cross-sectional area effective as reinforcement $= A_{fs} + A_{fb} + A_{fw}$	mm^2
A_p	Pressure loaded area $= A_{ps} + A_{pb} = r_{is}\,(l_{so} + 0.5\, d_{eb}) + 0.5\, d_{ib}\,(l_{bo} + e_{a,s})$	mm^2
f_s	Nominal design stress of shell material	MPa
f_b	Nominal design stress of nozzle material	MPa
For $f_b < f_s$ and $f_{ob} = \min(f_s\, ;\, f_b)$ the following applies: $P_{max} = \frac{(A_{fs} + A_{fw})\, f_s + A_{fb}\,.\, f_{ob}}{(A_{ps} + A_{pb}) + 0.5\,(A_{fs} + A_{fw} + A_{fb})}$ = Maximum allowable pressure (MPa)		