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**Research Paper**

# Review of traditional tubular pad reinforced nozzles versus Long Welding Neck Flange nozzles in pressure vessels.

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***ABSTRACT:*** *When designing pressure vessels, the vessel mechanical engineer is confronted with the implementation of a suitable nozzle design, with a number of aspects to be considered. From the many available solutions prudent choices have to be made with regard to both functional and economic aspects of the design, in which facets that are related to manufacturing costs, availability, and inspectability play a major role. Moreover, the nozzle design must have sufficient pressure capacity and insight must be gained with regard to the external load capacity of the nozzle, because the nozzle should be capable to withstand piping reactions exerted by the connected piping. This paper focuses in particular on two types of nozzle designs that often occur in practice , namely: a traditional set- in or set-on flush nozzle consisting of a nozzle neck made from pipe welded to a welding neck flange whether or not provided with a reinforcing pad and a long forged welding neck flange used as a flush set-in or set-on integral nozzle. In the elaborated cases, specific attention have been paid to the pressure- and nozzle load capacity of the two investigated nozzle configurations. The criterion chosen here is an approximately equal allowable internal pressure for both nozzle configurations.*
***KEYWORDS****: pressure vessel, nozzle design, pressure capacity, traditional nozzle , Long Welding Neck Flange, nozzle load capacity*
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## I. INTRODUCTION

The use of long welding neck (abbreviated LWN) flanges for nozzles on pressure vessels offers many advantages over traditional tubular nozzles that usually consist of a nozzle neck fabricated from a pipe on which a welding neck flange is welded. To compensate for the weakening of the nozzle in the wall of the pressure vessel, a reinforcing pad must often be provided around the nozzle neck. Due to its construction, a long-welding neck flange has the advantage of having more inherent reinforcement than a traditional nozzle, through which a reinforcing pad can be avoided. Long welding neck flanges, can be defined as forged nozzles that meet the dimensional requirements of a flanged fitting given in ASME B16.5[1] but have a straight hub/neck. The neck inside diameter shall not be less than the nominal size of the flange, and the outside diameter of the neck and any nozzle reinforcement shall not exceed the diameter of the hub as specified in ASME B16.5 [1]. LWN flanges are quite similar to a standard welding neck flanges, but the “neck” is considerably longer. A LWN flange can be used as a nozzle in a cylindrical shell or spherical / torispherical / ellipsoidal shaped head. In addition, there are several types such as: LWN Heavy Barrel (HB) and the Equal LWN Barrel (E); they have a different shape and a thicker hub. Typical shapes of long (forged) welding neck flanges are depicted in Figure 1. The nomenclature of applied symbols pertaining to calculations is included in the Appendix.

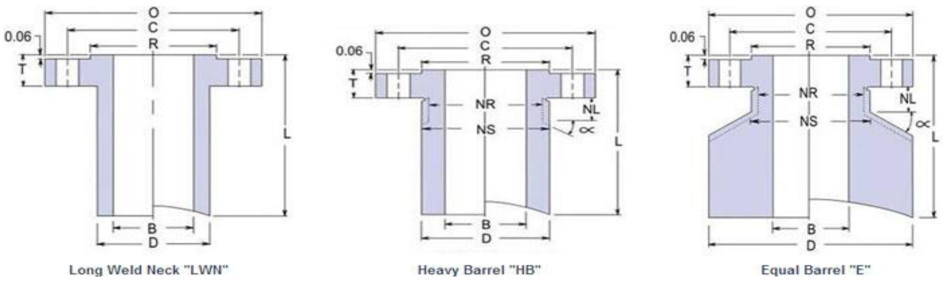


**Figure 1 Various Types of Long Forced Welding Neck Nozzles**

The picture is taken from a leaflet from Texas Flange & Fitting Supply Inc. The main dimensions of LWN flanges correspond to those for traditional welding neck flanges in accordance with ASME B16.5[1] with the exception of the "hub" shape and dimensions. The rated pressure of LWN flanges is identical to that of traditional flanges according to ASME B16.5 [1]. Typically is the actual bore size for regular long weld neck flanges (NOT heavy barrels) is the same as the nominal size of the flange. The various types of long welding neck flanges can be supplied on a global scale by many manufacturers and suppliers. In the following chapter, the pressure capacity of a traditional tubular nozzle is first determined, which is made up of a pipe part with welded-on welding neck flange without a reinforcing pad. Then, based on an identical internal pressure capacity, the implementation of two nozzle configurations will be compared. One nozzle is equipped with a long welding neck flange and the other as a traditional nozzle consisting of a pipe part (neck) with welded-on welding neck flange to which a reinforcing pad has been added. Figure 2 shows sketchy images of the nozzle configurations considered.

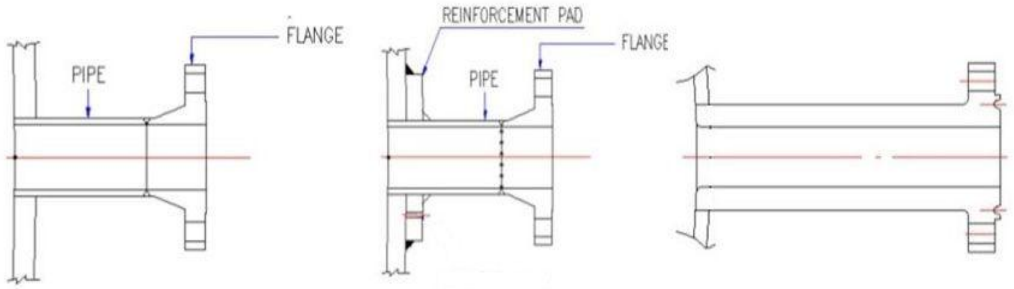


**Figure 2 Considered pressure vessel nozzle configurations**

## II. PRESSURE INTEGRITY INVESTIGATION OF LWN FLANGE NOZZLE VERSUS TRADITIONAL TUBULAR NOZZLE

The 'Pressure-Area Method' will be applied to determine the internal pressure capacity of the relevant nozzle configuration, because this one is well known and widely used in various recognized design codes for unfired pressure vessels like: EN 13445[2], PD 5500[3], AD 2000[4] and ASME Section VIII Division 2[5]**.**The pressure-area method is based on ensuring that the reactive force provided by the material is greater than, or equal to, the load from the pressure. The former is the sum of the product of the average membrane stress in each component and its stress loaded cross-sectional area (see Figure 3 for visualization). The latter is the sum of the product of the pressure and the pressure loaded cross-sectional areas. The key element in applying the pressure area method is to determine the dimensions of the reinforcing zone, i.e., the length of the shell, height of the nozzle and reinforcing pad dimensions (if reinforcing pad is provided), that resist the applied pressure.

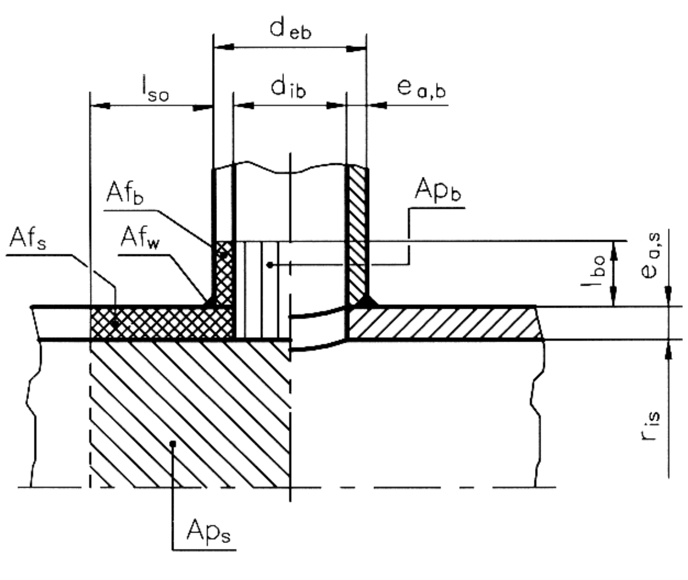


**Figure 3: Configuration of set-on nozzle (derived from EN 13445-3)**

The general equation for the reinforcement by set-on nozzle connection of an isolated opening is given by:

$(Af_s + Af_w)(f_s - 0.5P) + Af_b\,(f_{ob} - 0.5P) \geq P(Ap_s + Ap_b)$

If nominal design stresses for the materials of shell and nozzle are equal ( $f_s = f_{ob} = f$ ) the maximum permissible pressure is given by:

$$P_{max} = \frac{(Af_s + Af_w + Af_b)f}{(Ap_s + Ap_b) + 0.5\,(Af_s + Af_w + Af_b)}$$

Boundary limits of reinforcement zones:

$l_{so} = [(2r_{is} + e_{a,s})\,e_{a,s}]^{1/2}$ respectively $l_{bo} = [(d_{eb} - e_{a,b})\,e_{a,b}]^{1/2}$

The following data applies to the nozzle that consists of an NPS 10 "(NB 250) class 150 long welding neck flange: Nozzle neck O.D = 304.8 mm, Nozzle bore = 254 mm, LWN flange hub thickness = 25.4 mm, O.D cylindrical shell = 1200 mm, Shell thickness = 16 mm, I.D cylindrical shell = 1168 mm, Weld throat thickness = 6 mm, Nominal design stresses ( $f_s = f_{ob} = f$ ) are set at 138 MPa.

Corrosion allowance and wall thickness tolerances have been omitted.

**Calculation**

$l_{so} = [(2r_{is} + e_{a,s})\,e_{a,s}]^{1/2} = [(1168 + 16)\,16]^{1/2} = 137.637$ mm

$l_{bo} = [(d_{eb} - e_{a,b})\,e_{a,b}]^{1/2} = [304.8 - 25.4)\,25.4]^{1/2} = 84.242$ mm

$Af_s = 137.637 \times 16 = 2202.19\ mm^2$

$Af_w = 6 \times 6 = 36\ mm^2$

$Af_b = 84.242 \times 25.4 = 2139.75\ mm^2$

$Ap_s = (137.637 + 0.5 \times 304.8)\ 0.5 \times 1168 = 169381.6\ mm^2$

$Ap_b = (84.242 + 16)\ 0.5 \times 254 = 12730.73\ mm^2$

$P_{max} = [( 2202.19 + 36 + 2139.75)\ 138] / [(169381.6 + 12730.73) + 0.5(2202.19 + 36 + 2139.75)]$

$P_{max} = 3.278$ MPa = 32.78 bar

Class 150 flange rating is the limiting factor here!

The equation for $P_{max}$ applicable for an isolated conventional nozzle in a cylindrical shell with added reinforcing pad and $f_s = f_{ob} = f_p = f$ is given by:

$$P_{max} = \frac{\left(Af_s + Af_w + Af_b + Af_p\right)f}{(Ap_s + Ap_b) + 0.5\,(Af_s + Af_w + Af_b + Af_p)}$$

The following data applies to the conventional nozzle that consists of an NPS 10 "(NB 250) Schedule 40 pipe welded to a class 150 welding neck flange and provided with a reinforcing pad.

Nozzle neck O.D = 273 mm, Nozzle neck thickness = 9.27 mm Nozzle I.D = 254.46 mm, O.D cylindrical shell = 1200 mm, Shell thickness = 16 mm, I.D cylindrical shell = 1168 mm, Pad thickness = 16 mm, Pad width = 72 mm, Weld throat thicknesses = 6 mm, Nominal design stresses ( $f_s = f_{ob} = f_p = f$ ) are set at 138 MPa

Corrosion allowance and wall thickness tolerances have been omitted.

**Calculation**

$l_{so} = [(2r_{is} + e_{a,s})\,e_{a,s}]^{1/2} = [(1168 + 16)\,16]^{1/2} = 137.637$ mm

$l_{bo} = [(d_{eb} - e_{a,b})\,e_{a,b}]^{1/2} = [273 - 9.27)\,9.27]^{1/2} = 49.44$ mm

$Af_s = 137.637 \times 16 = 2202.19\ mm^2$

$Af_w = (6 \times 6)2 = 72\ mm^2$

$Af_b = (49.44 + 16)\ 9.27 = 606.63\ mm^2$

$Af_p = 72 \times 16 = 1152\ mm^2$

$Ap_s = (137.637 + 0.5 \times 273)\ 0.5 \times 1168 = 160096\ mm^2$

$Ap_b = (49.44 + 16)\ 0.5 \times 254.46 = 8325.93\ mm^2$

$P_{max} = [( 2202.19 + 72 + 606.63 + 1152)138] / [(160096 + 8325.93) + 0.5(2202.19 + 72 + 606.63 + 1152)]$

$P_{max} = 3.265$ MPa = 32.65 bar ~ 32.78 bar

Class 150 flange rating is the limiting factor here!

In case the Class 150 welding neck flange is replaced by a class 300 welding neck flange, the allowable internal pressure is determined solely by the nozzle neck including reinforcing pad.

In addition to the above considerations, we will calculate the allowable pressure whereby we omit the reinforcing pad, so that we gain more insight into the influence of the reinforcing pad on the allowable internal pressure.

$P_{max} = [( 2202.19 + 36+ 606.63)138] / [(160096 + 8325.93) + 0.5(2202.19 + 36 + 606.63)] = 2.31$MPa = 23.1bar

The Class 150 flange rating is still limiting for the allowable internal pressure for this nozzle connection. By omitting the reinforcing pad, the permissible internal pressure in this case is reduced by approximately 30%. It should be noted that the reinforcement efficiency of a reinforcing pad is set at 100% by almost all recognized codes, despite the fact that in practice there is no real integral connection with the shell. Experience shows, that there is almost always a gap between the reinforcing pad and the shell. For that reason it is recommended in such a case to introduce a 75% reinforcement efficiency for the reinforcing pad which means that $Af_p$ should be multiplied by a factor of 0.75 to be on the safe side. However, this reinforcement efficiency has not been applied in the treated case!

As a follow-up to previous considerations, we are replacing the Class 150 -10" LWN flange with a Class 300 LWN flange with the following hub (neck) dimensions:
Nozzle neck O.D = 320.5 mm, Nozzle bore = 254 mm, LWN flange hub thickness = 33.25 mm

**Calculation**

Limitation of effective thickness ratio for nozzles can be obtained from Figure 9.4-14 of EN 13445-3. The permissible ratio depend on: $d_{ib} / 2r_{is}$ = 254/1168 = 0.217465 which leads to a permissible $e_b / e_{a,s}$ ratio of 2. This means that the effective thickness of the nozzle neck up to 2 times the shell thickness (2 x 16 = 32 mm) may be taken into account. Consequently, the imaginary outer diameter of the nozzle neck becomes 254 + 2 x 32 = 318 mm. The left over neck thickness of 33.25 - 32 = 1.25 mm is furthermore not taken into account as a contribution to the nozzle reinforcement. Theoretically, the remaining thickness of 1.25 mm can still be considered as an imaginary reinforcement ring with a maximum height of 1.5 times the shell thickness, so in this case 1.5 x 16 = 24 mm. This provides a contribution to $Af_p$ of only 24 x 1.25 = 30 mm$^2$, but has been further left out of consideration in order to be conservative.

$l_{so} = [(2r_{is} + e_{a,s})\, e_{a,s}]^{½} = [(1168 + 16)\, 16]^{½} = 137.637$ mm

$l_{bo} = [(d_{eb} - e_{a,b})\, e_{a,b}]^{½} = [318 - 32)\, 32]^{½} = 95.67$ mm

$Af_s = 137.637 \times 16 = 2202.19$ mm$^2$

$Af_w = 6 \times 6 = 36$ mm$^2$

$Af_b = 95.67 \times 32 = 3061.3$ mm$^2$

$Ap_s = (137.637 + 0.5 \times 320.5)\, 0.5 \times 1168 = 173966$ mm$^2$

$Ap_b = (95.67 + 16)\, 0.5 \times 254 = 14182.1$ mm$^2$

$P_{max} = [( 2202.19 + 36 + 3061.3)\, 138] / [(173966 + 14182.1) + 0.5(2202.19 + 36 + 3061.3)]$

$P_{max} = 3.833$ MPa = 38.33 bar

Class 300 flange rating is not the limiting factor here!

Maximum allowable working pressure for the cylindrical shell without nozzle:

$$\boxed{\mathrm{MAWP} = \frac{2fd}{De - d} = \frac{2 \times 138 \times 16}{1200 - 16} = 3.73\ \mathrm{MPa} = 37.3\ \mathrm{bar}}$$

i.e. undisturbed cylinder (w/o nozzle) determines the permissible internal pressure.

## III. INVESTIGATION OF NOZZLE LOAD CAPACITY

Another aspect that must be taken into consideration is the nozzle load capacity.
Two cases will be taken into consideration for this, namely:
**Case # 1**: Class 300 - 10" LWN flange used as nozzle in a cylindrical shell with O.D = 1200 mm and a thickness of 16 mm.
**Case #2**: Traditional nozzle consisting a Class 300 W.N flange welded on a NPS 10"- Schedule 40 pipe (O.D 273 x 9.27 nominal wall thickness) and a reinforcing pad 72 mm width and 16 mm thick.
When determining the nozzle load capacity for the selected cases, use will be made of the recently published article in the Petroleum & Petrochemical Engineering Journal; entitled "Load Capacity Limits of Flanged Pressure Vessel Nozzles [6]

**Elaboration Case #1**
**Formula overview adopted from [6]**

| **Maximum allowable individual loads Nozzle on cylinder w/o reinforcing pad** | **Numerical elaboration** |
|---|---|
| $F = f / (6\, C_{21})$ | F = 138/ (6 x 0.000377575) = 60915 N |
| $M_l = f / (1.5\, C_{31})$ | $M_l$ = 138/ (1.5 x 0.000004712) = 19524618 Nmm |
| $M_c = f / (1.15\, C_{31}.\, C_{41})$ | $M_c$ = 138 / (1.15 x 0.000004712 x 3.16475) = 8047047 Nmm |
| **Auxiliary values** | **Auxiliary values** |
| $C_{11} = (D_o - T) / (2T)$ | $C_{11}$ = (1200 -16) / (2 x16) = 37 |
| $C_{21} = (C_{11})^{0.5} / (\pi.T.\, D_n)$ | $C_{21} = (37)^{0.5} / (\pi \times 16 \times 320.5) = 0.000377575$ |
| $C_{31} = 4(C_{11})^{0.5} / (\pi.T.\, D_n^2)$ | $C_{31} = 4(37)^{0.5} / (\pi \times 16 \times 320.5^2) = 0.000004712$ |
| $C_{41} = (D_n / 2T)^{0.5}$ | $C_{41} = (320.5 / 2 \times 16)^{0.5} = 3.16475$ |
| **Maximum allowable individual flange loads** | **Numerical elaboration** |
| $F = (P_r - P_d)\, (\pi/4)\, G^2$ | $F = (4.285 - 3.73)\, (\pi/4)\, 301.36^2 = 39587$ N; Case #1<br>$F = (4.285 - 3.265)\, (\pi/4)\, 301.36^2 = 72755$ N; Case #2 |

| | |
|---|---|
| $M = (P_r - P_d)(\pi/16)\,G^2.C.K_f$ | $M = (4.285 - 3.73)(\pi/16)\,301.36^2 \times 387.4 \times 2.38$<br>M = 9124953 Nmm; Case #1<br>$M = (4.285 - 3.265)(\pi/16)\,301.36^2 \times 387.4 \times 2.38$<br>M = 16770185 Nmm; Case #2 |
| **Auxiliary value**<br>$K_f = 1 + [t^2 + (W - d^*_h)^2/2.6\,t^2]$ | **Auxiliary value**<br>$K_f = 1 + [46.1^2 + (95.5 - 21.31695)^2/2.6 \times 46.1^2]$<br>$K_f = 2.38$ where: $d^*_h = \max[d_h(1 - Di/1000);\ 0.5d_h] = \max[28.575(1 - 254/1000);\ 0.5 \times 28.575] = 21.31695$ |

Note: $P_r$ = rated pressure according ASME B16.5 for group 1.1 material A350 LF2 / A105 @ 225°C

**Elaboration Case #2**

**Formula overview adopted from [6]**

| **Maximum allowable individual loads Nozzle on cylinder with reinforcing pad (Adjacent to the nozzle neck)** | **Numerical elaboration** |
|---|---|
| $F = f/(6\,C_{21})$ | F = 138/ (6 x 0.000155657) = 147761 N |
| $M_l = f/(1.5\,C_{31})$ | $M_l$ = 138 / (1.5 x 0.000002281) = 40333187 Nmm |
| $M_c = f/(1.15\,C_{31}.C_{41})$ | $M_c$ = 138 / (1.15 x 0.000002281 x 2.06534) = 25472080 Nmm |
| **Auxiliary values** | **Auxiliary values** |
| $C_{11} = [D_o - (T+T_{pad})]/[2(T+T_{pad})]$ | $C_{11} = [1200 - (16+16)]/[2(16+16)] = 18.25$ |
| $C_{21} = (C_{11})^{0.5}/[\pi(T+T_{pad})D_n]$ | $C_{21} = (18.25)^{0.5}/[\pi(16+16)273] = 0.000155657$ |
| $C_{31} = 4(C_{11})^{0.5}/[\pi(T+T_{pad})D_n^2]$ | $C_{31} = 4(18.25)^{0.5}/[\pi(16+16)273^2] = 0.000002281$ |
| $C_{41} = [D_n/2\,(T+T_{pad})]^{0.5}$ | $C_{41} = [273/2\,(16+16)]^{0.5} = 2.06534$ |

**Formula overview adopted from [6]**

| **Maximum allowable individual loads Nozzle on cylinder with reinforcing pad (At the transition between vessel and reinforcing pad)** | **Numerical elaboration** |
|---|---|
| $F = f/(6\,C_{22})$ | F = 138 / (6 x 0.0002902) = 79256 N |
| $M_l = f/(1.5\,C_{32})$ | $M_l$ =138 / (1.5 x 0.000002784)<br>= 33045977 Nmm |
| $M_c = f/(1.15\,C_{32}.C_{42})$ | $M_c$ = 138 / (1.15 x 0.000002784 x 3.61)<br>= 11940013 Nmm |
| **Auxiliary Values** | **Auxiliary Values** |
| $C_{12} = (D_o - T)/(2T)$ | $C_{12} = (1200 - 16)/(2 \times 16) = 37$ |
| $C_{22} = (C_{12})^{0.5}/(\pi.\,T.\,D_{pad})$ | $C_{22} = (37)^{0.5}/(\pi \times 16 \times 417) = 0.0002902$ |
| $C_{32} = 4(C_{12})^{0.5}/(\pi.\,T.\,D_{pad}^2)$ | $C_{32} = 4(37)^{0.5}/(\pi \times 16 \times 417^2) = 0.000002784$ |
| $C_{42} = (D_{pad}/2T)^{0.5}$ | $C_{42} = (417/2 \times 16)^{0.5} = 3.61$ |

**Summarized results of investigated nozzle configurations**

| **Concerns** | **Configuration** | | **Cylindrical shell** |
|---|---|---|---|
| Case #1 | 10" (NB250) LWN Flange Class 300<br>Bore: 254 mm<br>Hub thickness: 33.25 mm<br>No reinforcing pad | | O.D. 1200 mm x 16 mm thick |
| Case #2 | W.N. Flange Class 300<br>10" (NB 250) pipe<br>Nozzle neck: Sched.40 (9.27 mm)<br>Reinforcing pad width: 72 mm<br>Reinforcing pad thickness: 16 mm | | O.D. 1200 mm x 16 mm thick |
| **Concerns** | **Permissible internal pressure Nozzle vs undisturbed cylindrical shell** | | **Maximum allowable individual flange loads** |
| Case #1 | 38.33 bar / 37.3 bar | | F = 39587 N (Ignore if compressive ); M = 9124953 Nmm |
| Case #2 | 32.65 bar / 37.3 bar | | F = 72755 N (Ignore if compressive ); M = 16770185 Nmm |
| **Concerns** | **Maximum allowable individual radial load** | **Maximum allowable individual longitudinal moment** | **Maximum allowable individual circumferential moment** |
| **Load** | **F** | **$M_l$** | **$M_c$** |
| Case #1 | 60915 N | 19524618 Nmm | 8047047 Nmm |
| Case #2 [1] | 147761 N | 40333187 Nmm | 25472080 Nmm |
| Case #2 [2] | 79256 N | 33045977 Nmm | 11940013 Nmm |

Notes: [1] Adjacent to the nozzle neck
[2] At the transition between vessel shell and reinforcing pad

**Decisive nozzle loads and internal pressure (rounded off figures)**

| Case | Allowable individual radial load F (kN) | Allowable individual longitudinal moment $M_l$ (kNm) | Allowable individual circumferential moment $M_c$ (kNm) | Allowable internal pressure P (bar) |
|---|---|---|---|---|
| Case #1 | 60.9 | 19.5 | 8.0 | 37.3 |
| Case #2 | 79.2 | 33 | 11.9 | 32.65 |

## IV. DISCUSSION

The load exerted on the nozzle flange can be a limiting factor, therefore sufficient attention must be paid to it. With regard to the permissible loads on the nozzle, apart from the flange, it can be concluded that for the case under consideration the location at the transition from reinforcing pad to cylindrical shell is determining.

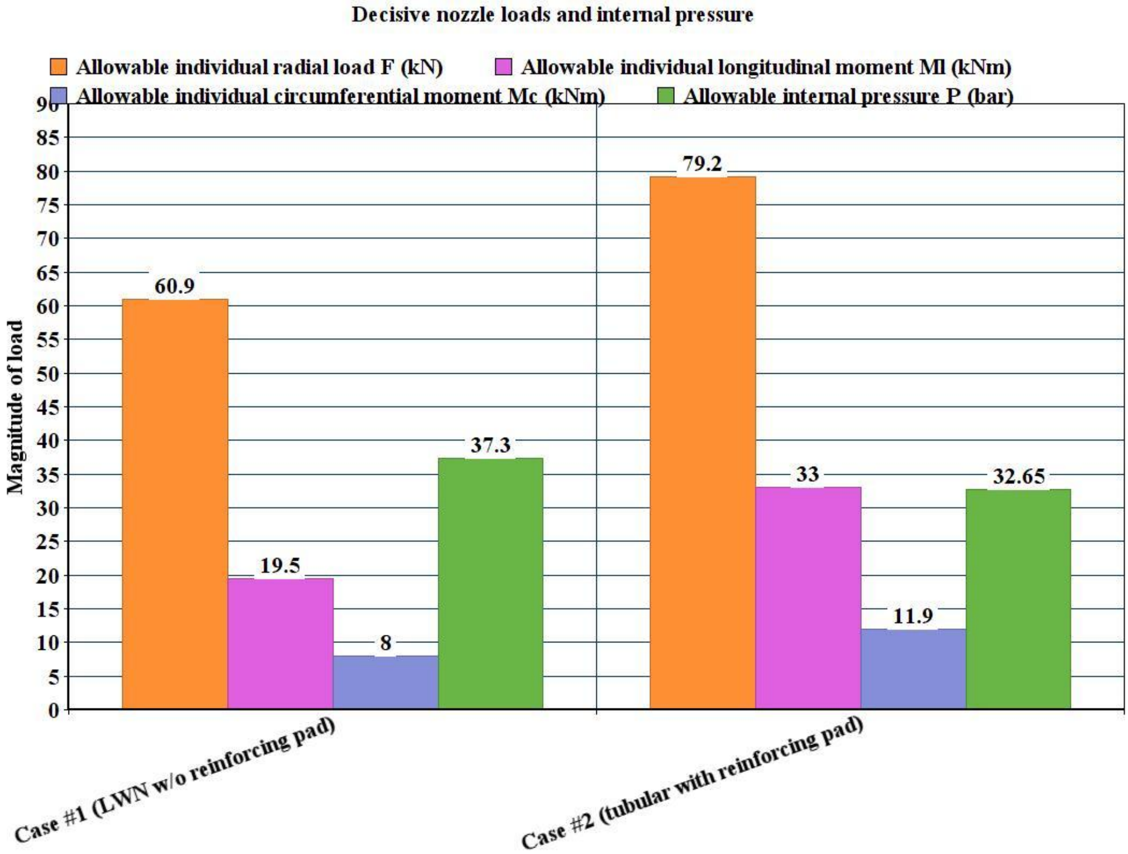


**Figure 4 Graphical representation of allowable loads for considered nozzle configuration cases**

## V. CONCLUSIONS

Although this research has only been carried out on a limited scale, the use of LWN flanges as a nozzle is by far the preferred choice from a structural point of view. It should be noted here that there is considerably less welding work and better inspection options compared to a traditional tubular nozzle fitted with W.N. flange and reinforcing pad. In low temperature applications, LWN flanges are preferable to traditional tubular nozzles by the fact that there is less welding work, and as a result, less chance of defects and fewer stress raisers. In terms of load capacity of the nozzle, it can be stated that pressure integrity can be assured at all times when LWN type nozzles are used, also in view of the number of LWN shapes in such applications. The use of reinforcing pads in the case of traditional tubular nozzles can often be restrictive and, moreover, often more expensive and difficult to inspect thoroughly. The sensitivity to brittle fracture is greater with pad reinforced nozzles due to the presence of a gap between shell and reinforcing pad. The reinforcing pad and the underlying shell course cannot be considered as an entirely integral part. With high temperature applications reinforcement pads should be avoided anyway. This also applies to pressure vessels that are subject to non-predominantly static loads and thus prone to fatigue damage. With regard to the flanges and their load-bearing capacity, there is actually no difference if the same internal pressure regimes are assumed.

## REFERENCES


[1]. ASME B16.5 - 2017 "Pipe Flanges and Fittings NPS ½ through NPS 24 standard"
[2]. EN 13445 - Unfired pressure vessels - Part 3: Design (2018)
[3]. PD 5500:2018 - Specification for unfired fusion welded pressure vessels
[4]. AD 2000 - Merkblatt B9: 2010 "Openings in cylindrical, conical and spherical shells"
[5]. ASME BPVC Section VIII - Division 2 : 2017 "Alternative Rules"
[6]. Walther Stikvoort (2018) "Load Capacity Limits of Flanged Pressure Vessel Nozzles" Pet Petro Chem Eng J 2(3):1-6.


## APPENDIX

### Nomenclature of applied symbols

| Symbol | Description | Unit |
|---|---|---|
| Af | Stress loaded cross-sectional area effective as reinforcement | $mm^2$ |
| $Af_s$ | Cross - sectional area of shell within the compensation limits | $mm^2$ |
| $Af_b$ | Cross - sectional area of branch within the compensation limits | $mm^2$ |
| $Af_w$ | Cross - sectional area of fillet weld between nozzle (or plate) and shell | $mm^2$ |
| $Af_p$ | Cross - sectional area of pad within the compensation limits | $mm^2$ |
| Ap | Pressure loaded area | $mm^2$ |
| $Ap_s$ | Ap of the shell for the length (see Figure 3) | $mm^2$ |
| $Ap_b$ | Ap of the branch for the length (see Figure 3) | $mm^2$ |
| $C_{11}$,$C_{12}$,$C_{41}$,$C_{42}$ | Auxiliary values | - |
| $C_{21}$,$C_{22}$ | Auxiliary values | $1/mm^2$ |
| $C_{31}$,$C_{32}$ | Auxiliary values | $1/mm^3$ |
| C | Bolt circle diameter of flange | mm |
| d | Wall thickness of shell | mm |
| $d_{eb}$ | External diameter of a nozzle fitted in a shell | mm |
| $D_e$ | Outside diameter of cylindrical shell | mm |
| $D_o$ | Outside diameter of shell | mm |
| $D_n$ | Outside diameter nozzle neck | mm |
| $D_{pad}$ | Outside diameter of reinforcing pad | mm |
| $d_h$ | Bolt hole diameter of flange | mm |
| $d^*_h$ | Reduced bolt hole diameter | mm |
| $e_{a,s}$ | Thickness of cylindrical shell | mm |
| $e_{a,b}$ | Thickness of branch or nozzle neck | mm |
| $e_{a,p}$ | Thickness of reinforcing pad | mm |
| f | Nominal design stress cylindrical shell | MPa |
| $f_s$ | Nominal design stress of shell material | MPa |
| $f_{ob}$ | Nominal design stress of branch/nozzle material | MPa |
| $f_p$ | Nominal design stress of reinforcing pad material | MPa |
| F | Allowable individual radial load | N |
| G | Diameter at location of gasket load reaction | mm |
| $K_f$ | 'Koves' factor flange | - |
| $l_{so}$ | Shell boundary limit of reinforcement zone | mm |
| $l_{so}$ | Branch boundary limit of reinforcement zone | mm |
| M | Allowable individual flange moment | Nmm |
| $M_l$ | Allowable individual longitudinal moment | Nmm |
| $M_c$ | Allowable individual circumferential moment | Nmm |
| MAWP | Maximum Allowable Working Pressure | MPa |
| $P_{max}$ | Maximum Allowable Pressure | MPa |
| $P_r$ | Rated pressure according ASME B16.5 | MPa |
| $P_d$ | Internal design pressure | MPa |
| $r_{is}$ | Inside radius cylindrical shell | mm |
| T | Shell thickness | mm |
| $T_{pad}$ | Thickness of reinforcing pad | mm |
| t | Flange thickness | mm |
| W | Width of flange (OD flange - ID flange)/2 | mm |

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