

## Load Capacity Limits of Flanged Pressure Vessel Nozzles

**Walther Stikvoort\***

Wagnerlaan 37, 9402 SH Assen, The Netherlands

**\*Corresponding author:** Walther Stikvoort, Wagnerlaan 37, 9402 SH Assen, The Netherlands (NL), Tel: 31592347088; Email: stikvoort@ziggo.nl

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### Abstract

When designing pressure vessels, it is quite important to know the loads exerted on the nozzle by the connecting pipe work. However the piping reactions computed by the piping structural analysis are often not available at the vessel design stage. To overcome this problem, the pressure vessel must be exclusively designed for the internal design pressure, after which the permissible external loads for the nozzle- vessel intersection as well as for the nozzle-piping connection (flange) can be calculated. In this way the load limits and load capacity of the nozzle can be determined and are available at an early stage to the piping designer (pipe stress analyst). Successively it is the responsibility of the piping designer to ensure that the piping reactions are kept within the permissible load limits of the pressure vessel nozzle. The advantage of this approach is that the imposed loads does not necessitate thickening of the pressure-retaining shell of the vessel nor require additional reinforcing pads around the nozzle neck. Moreover it should be noted that by increasing the vessel shell thickness or adding a reinforcing pad, the nozzle becomes more rigid and therefore a better approximation to a fixed point or anchor thus effectively eliminating the advantage of any nozzle flexibility. This approach avoids remedial work of pressure vessels and/or pipework at late stages of a project, which for sure has negative impact on project costs and schedule.

**Keywords:** Pressure vessel; Nozzle; Piping reactions; Pipework; Flange; Individual loads; Load interaction

### Introduction

Many engineering contractors and operating companies provide the vessel designer and piping designer with standard nozzle loads. The basis of such standards are often unknown and the differences between them are often considerable. The standards are either in tabular form or in formula form. Practice shows that the use of such standards often leads to extra reinforcement around the nozzle neck and a higher flange rating and consequently increased costs. However extensive research has shown, that additional nozzle reinforcements and / or a higher flange rating on top of

what is required for internal pressure can practically be excluded and can therefore be ignored. The challenge is to determine the load limits of both the nozzle- vessel intersection and the nozzle-piping flange that the piping designer and / or pipe stress analyst must conform to.

### Practical Approach

Determining and assessing the stresses at a nozzle to vessel intersection due to internal pressure and external forces and moments is one of the most complex problems in pressure vessel design. That is why in this article we opted for a simplified calculation methodology that

applies to radial nozzles on a cylinder or sphere (and spherical part of torispherical head), eventually provided with a reinforcing pad for which the shear stress caused by transverse force and the torsional moment on the nozzle - shell intersection will not amount to more than 15% of the design stress  $f$ . The starting point for the nozzle-shell intersection is the so-called elastic shakedown criterion thus avoiding the possibility of repeated plastic cycling or ratcheting, whereby the sum of the pressure stress and the local stresses due to the external loads must be less than 3 times the design stress

$f$  or 2 times the yield stress  $S_y$ . Since the maximum pressure stress intensity can be set at  $2f$  ( $1.33 S_y$ ), remains  $1f$  ( $0.67 S_y$ ) for the sum of all external load stress intensities. This is conservative as the locations of their maximum stresses will surely not coincide.

To serve the vessel designer in his effort to calculate the permissible nozzle loads, all quantities, units and formulas are displayed in the forms shown below (Tables 1-4).

Maximum allowable individual loads Nozzle on cylinder w/o reinforcing pad	Maximum allowable individual loads Nozzle on sphere (or spherical part) w/o reinforcing pad
$F = f/6 C_{21}$	$F = f/1.75 C_{21}$
$M_l = f/1.5 C_{31}$	$M = f/1.75 C_{31}$
$M_c = f/1.15 C_{31} \cdot C_{41}$	
<b>Auxiliary Values</b>	
$C_{11} = (D_o - T)/(2T)$	
$C_{21} = (C_{11})^{0.5}/(\pi \cdot T \cdot D_n)$	
$C_{31} = 4(C_{11})^{0.5}/(\pi \cdot T \cdot D_n^2)$	
$C_{41} = (D_n/2T)^{0.5}$	

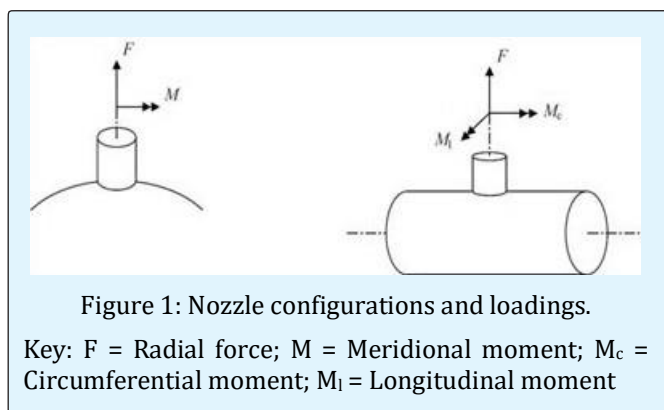
Table 1: Individual loads nozzle w/o reinforcing pad.

Maximum allowable individual loads Nozzle on cylinder with reinforcing pad (At the transition between vessel and reinforcing pad)	Maximum allowable individual loads Nozzle on sphere (or spherical part) with reinforcing pad (At the transition between vessel and reinforcing pad)
$F = f/6 C_{22}$	$F = f/1.75 C_{22}$
$M_l = f/1.5 C_{32}$	$M = f/1.75 C_{32}$
$M_c = f/1.15 C_{32} \cdot C_{42}$	
<b>Auxiliary Values</b>	
$C_{12} = (D_o - T)/(2T)$	
$C_{22} = (C_{12})^{0.5}/(\pi \cdot T \cdot D_{pad})$	
$C_{32} = 4(C_{12})^{0.5}/(\pi \cdot T \cdot D_{pad}^2)$	
$C_{42} = (D_{pad}/2T)^{0.5}$	

Table 2: Individual loads nozzle with reinforcing pad (At the transition between vessel and reinforcing pad).

Maximum allowable individual loads Nozzle on cylinder with reinforcing pad (Adjacent to the nozzle neck)	Maximum allowable individual loads Nozzle on sphere (or spherical part) with reinforcing pad (Adjacent to the nozzle neck)
$F = f/6 C_{21}$	$F = f/1.75 C_{21}$
$M_l = f/1.5 C_{31}$	$M = f/1.75 C_{31}$
$M_c = f/1.15 C_{31} \cdot C_{41}$	
<b>Auxiliary Values</b>	
$C_{11} = [D_o - (T + T_{pad})]/[2(T + T_{pad})]$	
$C_{21} = (C_{11})^{0.5}/[\pi(T + T_{pad})D_n]$	
$C_{31} = 4(C_{11})^{0.5}/[\pi(T + T_{pad})D_n^2]$	
$C_{41} = [D_n/2(T + T_{pad})]^{0.5}$	

Table 3: Individual loads nozzle with reinforcing pad (Adjacent to the nozzle neck).



### Remark

In case of very thin nozzle necks the stresses in the nozzle neck can be higher than in the vessel shell. It is recommended to avoid thin nozzle necks whenever possible. If this is not possible for practical reasons, it is recommended to reduce the allowable individual loads by a factor of  $C_n$ . The factor  $C_n$  accounts for the higher stresses in the nozzle neck with respect to the vessel wall. For nozzle neck thicknesses less than the shell thickness

$C_n$  becomes equal to the ratio of shell thickness and nozzle neck thickness.

Maximum allowable individual flange loads
$F = (P_r - P_d) (\pi/4) G^2$
$M = (P_r - P_d) (\pi/16) G^2 \cdot C \cdot K_f$
Auxiliary Value
$K_f = 1 + [t^2 + (W - d^*_{h})^2 / 2.6 t^2]$

Table 4: Nozzle Individual flange loads.

The individual permissible forces and moments (which can be applied independently on a nozzle) follow from the application of the procedure described above. Subsequently, the vessel designer append the data of the nozzle load capacity to the requisition or otherwise the manufacturers quotation in a format that corresponds to Table 5, which can consecutively be made available to the pipe stress analyst for further evaluation and judgment.

Note that in the case of a nozzle provided with a reinforcing pad, the lowest calculated value must be entered in the Table 5.

Magnitude of Individual Allowable Nozzle Load Capabilities					
Nozzle	Nozzle / Cylinder Intersection			Flange Facing	
	F [N]	$M_l$ [Nmm]	$M_c$ [Nmm]	F [N]	M [Nmm]
N2					
Nozzle	Nozzle / Head / Sphere Intersection			Flange Facing	
	F [N]	M [Nmm]		F [N]	M [Nmm]
N1					

Table 5: Nozzle load capabilities of Nozzle intersection and Flange facing.

The engineering contractor is ultimately responsible for ensuring that the individual load ratios as well as the interaction of loadings at the nozzle-shell intersection are assessed and comply with the following linear 'Load Interaction Rule' for the nozzle vessel interface:  $F_{\text{actual}}/F + M_{l,\text{actual}}/M_l + M_{c,\text{actual}}/M_c \leq 1.0$ . Note that the actual forces and moments should be derived from the formal pipe stress analysis as performed by the engineering contractors pipe stress engineer/analyst.

In addition flange loadings shall be assessed according to the linear 'Load Interaction Rule', for which the following shall be satisfied:  $F_{\text{actual}}/F + M_{\text{actual}}/M$ .

The results of above mentioned nozzle load assessment, at both the nozzle-shell intersection and the

flange facing, shall be included in the formal pipe stress report.

### Discussion

Where actual loadings cannot be incorporated without increasing the pressure- retaining shell thickness of the vessel, the Engineering Contractor should ensure that the piping structural analysis (Pipe Stress Analysis) used to generate the loads takes into account the flexibility of the vessel nozzle at its intersection. Simple piping structural analysis normally assumes vessel nozzles to be fixed points, i.e. anchors. A method for predicting nozzle flexibility and the incorporation of this flexibility into piping stress analysis and hence substantially reducing the magnitude of piping reactions are often part of the

pipe stress analysis software package.

Vessel shell thicknesses should not be increased for nozzle loads until it has been shown that, even allowing for nozzle flexibility, the vessel will be over stressed. It should be noted that by increasing the vessel shell thickness the nozzle becomes more rigid and therefore a better approximation to a fixed point or anchor thereby, in practice, reducing the benefit of any nozzle flexibility which may limit actual applied loadings. It should be emphasized that for relatively thin nozzle necks the magnitude of stress in the nozzle neck becomes critical and hence needs careful consideration.

The value of nozzle loadings used for design purposes shall be clearly stated on the vessel drawings for future reference.

## Worked Example

A pressure vessel is designed for an internal design pressure of 30 bar (3 MPa) and a design temperature of 275°C. The vessel shell is made from A516 Grade 65 carbon steel and has a outside diameter of 1600 mm and a thickness of 20 mm. The vessel cylindrical shell is provided with a forged nozzle: NPS 12" Long Welding Neck flange Class 300 made of A350 LF2 carbon steel. The nozzle neck thickness is 35 mm and the outside neck diameter is 375 mm. No reinforcing pad is added. The design stress for the cylindrical shell is 129.33 MPa and for the nozzle neck (LWN flange) 132.67 MPa (Table 6-9).

Auxiliary Values	
$C_{11} = (D_o - T)/(2T) = (1600 - 20)/(2 \times 20) = 39.5$	
$C_{21} = (C_{11})^{0.5}/(\pi \cdot T \cdot D_n) = (39.5)^{0.5}/(\pi \times 20 \times 375) = 0.00026674$	
$C_{31} = 4(C_{11})^{0.5}/(\pi \cdot T \cdot D_n^2) = 4(39.5)^{0.5}/(\pi \times 20 \times 375^2) = 0.000002845$	
$C_{41} = (D_n/2T)^{0.5} = (375/2 \times 20)^{0.5} = 3.0619$	

Table 6: Determination of auxiliary quantities.

f	Design stress	129.33 MPa
P <sub>d</sub>	Internal design pressure	3 MPa
P <sub>r</sub>	Rated pressure (ASME B16.5 or ASME B16.47)	4.085 MPa
A	Outside diameter flange	520 mm
B	Inside diameter flange	305 mm
C	Bolt circle diameter	450.8 mm
t	Flange thickness	49.3 mm
G	Effective sealing diameter	357.3 mm
W	Flange width = 0.5 (A - B)	107.5 mm
d <sub>h</sub>	Bolt hole diameter	31.75 mm
d* <sub>h</sub>	max [d <sub>h</sub> (1 - B/1000) ; 0.5 d <sub>h</sub> ]	22.066 mm

Table 7: Flange data.

$$K_f = 1 + [t^2 + (W - d_h^*)^2/2.6 t^2] = 1 + [49.3^2 + (107.5 - 22.066)^2/2.6 \times 49.3^2] = 2.54$$

Allowable individual flange loads	
$F = (P_r - P_d) (\pi/4) G^2 = (4.085 - 3)(\pi/4) 357.3^2 = 108789 \text{ N}$	
$M = (P_r - P_d) (\pi/16) G^2 \cdot C \cdot K_f = (4.085 - 3)(\pi/16) 357.3^2 \times 450.8 \times 2.54 = 31141769 \text{ Nmm}$	

Table 8: Determination of Allowable individual flange loads.

Magnitude of Individual Allowable Nozzle Load Capabilities					
Nozzle	Nozzle / Cylinder Intersection			Flange Facing	
	F [N]	M <sub>I</sub> [Nmm]	M <sub>c</sub> [Nmm]	F [N]	M [Nmm]
N2	80809 Approx.81 kN	30305800 Approx. 30 kNm	12910057 Approx.13 kNm	108789 Approx.109 kN	31141769 Approx. 31 kNm

Table 9: Nozzle load summary showing calculation results of allowable individual loads.

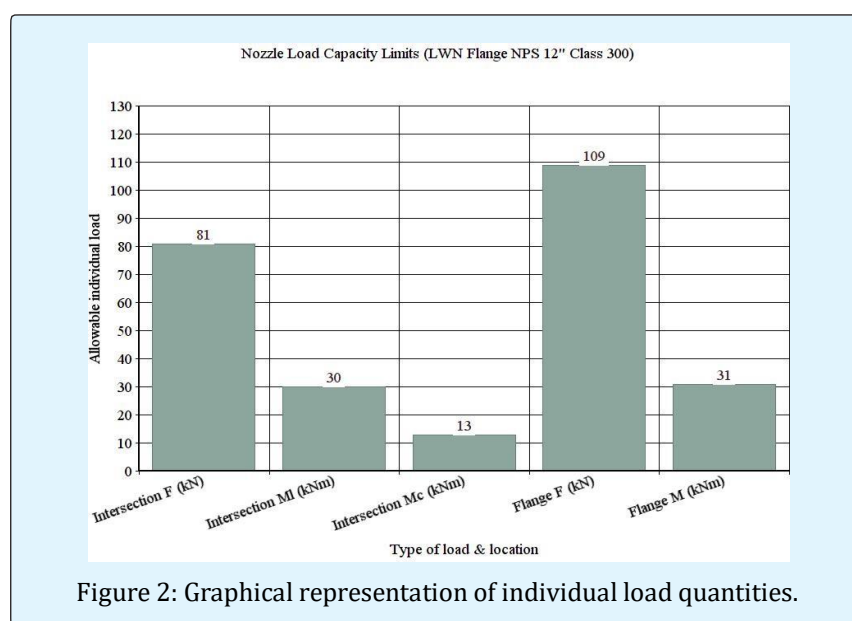


Figure 2: Graphical representation of individual load quantities.

The pipe stress analyst must demonstrate in its stress report that the piping reactions comply with the Load Interaction Rule for both the vessel nozzle intersection and connecting nozzle flange.

## Conclusion

The method provided is aimed for predominantly static

loaded pressure vessels, whereby the design temperature lies below the creep range for the applied materials. It provides a solution to a vexing pressure vessel - piping interface problem to which vessel - and piping designers often are confronted. Prudent application of the method has proved to be safe and reliable.

## Nomenclature

Symbol	Description	Unit
F	Force	N
M <sub>I</sub>	Moment	Nmm
M <sub>c</sub>	Moment	Nmm
M	Moment	Nmm
f	Design stress	MPa
D <sub>o</sub>	Outside diameter shell/sphere/spherical part	mm
T	Wall thickness shell/sphere/spherical part	mm
D <sub>n</sub>	Outside diameter nozzle neck	mm
D <sub>pad</sub>	Outside diameter reinforcing pad	mm
P <sub>d</sub>	Internal design pressure	MPa

$P_r$	Rated pressure (ASME B16.5 or ASME B16.47)	MPa
A	Outside diameter flange	mm
B	Inside diameter flange	mm
C	Bolt circle diameter	mm
t	Flange thickness	mm
G	Effective sealing diameter	mm
W	Flange width = 0.5 (A - B)	mm
$K_f$	'Koves' factor	-
$d_h$	Bolt hole diameter	mm
$d^*_h$	$\max [d_h(1 - B/1000) ; 0.5 d_h]$	mm

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