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RESEARCH ARTICLE

STANDARDIZED NOZZLE LOADS FOR THE INITIAL PRESSURE VESSEL DESIGN

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ABSTRACT

Pipework exerts forces and moments on a vessel due to thermal expansion or contraction, internal pressure and dead weight. On a project, pipework design usually occurs after the order of the vessel has been placed. Hence standard loads are specified as provision against the actual loads calculated later in the design of the pipework. This guideline specifies an approach that the pressure vessel designer can apply assuming that the loads act on the nozzles of pressure vessels that are imposed by the connected pipe system. These loads should be included in the mechanical design of pressure vessels during the procurement phase of a project. The Piping Stress Engineer shall ensure that the loads on the nozzles of the pressure vessels are within these values.

Key words: Pipework, nozzles, Forces and moments, Pressure vessels, Standard nozzle loads, Procurement phase.

INTRODUCTION

Since actual nozzle loads are often not available in the initial vessel design stage, pressure vessels should be designed to withstand a set of standard nozzle loadings (forces and moments). These standard loadings are for design purposes only and have no other significance. They are necessary because actual nozzle loadings computed by analysis of the piping system attached to the nozzle are often not available at the procurement phase of the vessel. Therefore, to enable the vessel design to be finalised, standard loadings must be assumed for design purposes. Subsequently the piping system needs to be designed so that these standard loadings are not exceeded. When the application of the standard nozzle loadings necessitate thickening of the pressure-retaining shell of the vessel, the imposed loads shall be reviewed with the aim of reducing the design loads. It may be possible to predict loadings for design purposes which are substantially lower than the standard nozzle loadings. Where actual loadings cannot be incorporated without increasing the pressure-retaining shell thickness of the vessel, the Piping Stress Engineer should ensure that the piping structural analysis used to generate the loads takes into account the flexibility of the vessel nozzle. Simple piping structural analysis normally assumes vessel nozzles to be fixed points, i.e. anchors. Methods for predicting nozzle flexibility and the incorporation of this flexibility into piping stress analysis may substantially reduce the computed loadings, particularly if they are self-limiting. Vessel 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.

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It should be noted that by increasing the vessel 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.

Scope

The application of the described approach is limited to pressure vessels which are predominantly statically loaded. It is considered that this is the case, if the strain or loading rate is less than 0.1 s^{-1}.

Standardized nozzle loads

The equations below, may serve to provide guidance on expected forces and moments (Piping Reactions) likely to be induced from external pipework, in addition to the internal or external pressure.

Equations to estimate the magnitude of nozzle loads (see NOTE):

$$M = \{[4 \times (DN - 25)^{1.4} + 2 \times 10^{-5} \times PN \times (DN)^{2.7}]10^{-3}\}[kNm]$$
$$F = \{[7.5 \times (DN)^{1.2} + 0.1 \times PN \times (DN)^{1.2}]10^{-3}\}\ [kN]$$

NOTE: The equations originate from NORSOK STANDARD R-001:2017 Mechanical Equipment; Clause 5.1.6. [1][2]

The above displayed equations derive a force and a moment based on pressure rating and nozzle size, to be applied in specified directions for vessel design to whatever code. The same loads are used as upper limit values in pipe stress analysis. This permits safe parallel engineering of piping and vessels and avoids delays and prevents discussion between different engineering disciplines.

Standard nozzle load matrix

Attention must be paid to the matters addressed below concerning the application and interpretation of Table 1.

1. For nozzles loadings use Table 1
2. The following notes apply to Table 1
   a. Values listed are not applicable to pressure vessels constructed of non-ferrous materials, austenitic stainless steel less than 6 mm thick or non-metallic materials.
   b. Allowable nozzle loads for pressure vessels constructed of non-ferrous materials, austenitic stainless steel less than 6 mm thick or non-metallic materials shall be calculated individually.
   c. Loads on nozzles without flanges shall use the applicable table for which the appropriate pressure class would apply.
   d. All forces and moments shall be taken as acting at the intersection of the nozzle centreline axis at the mid-thickness of the vessel shell or head.
   e. The moments 'M' and the forces 'F' shall be applied simultaneously in:
      - Two perpendicular directions at the right angle to the axis of pipe or in the plane tangent to the pressure retaining part at the nozzle-to-shell interface;
      - Direction perpendicular to the above plane.

   f. Forces and moments may act in any combination and direction concurrently with internal or external pressure unless otherwise specified. Moreover the effect of both positive and negative values of the forces shall be evaluated.
   g. The designer shall account for the worst condition of combined forces, moments and pressure.
   h. The effect of transverse force and the torsional moment shall be ignored.

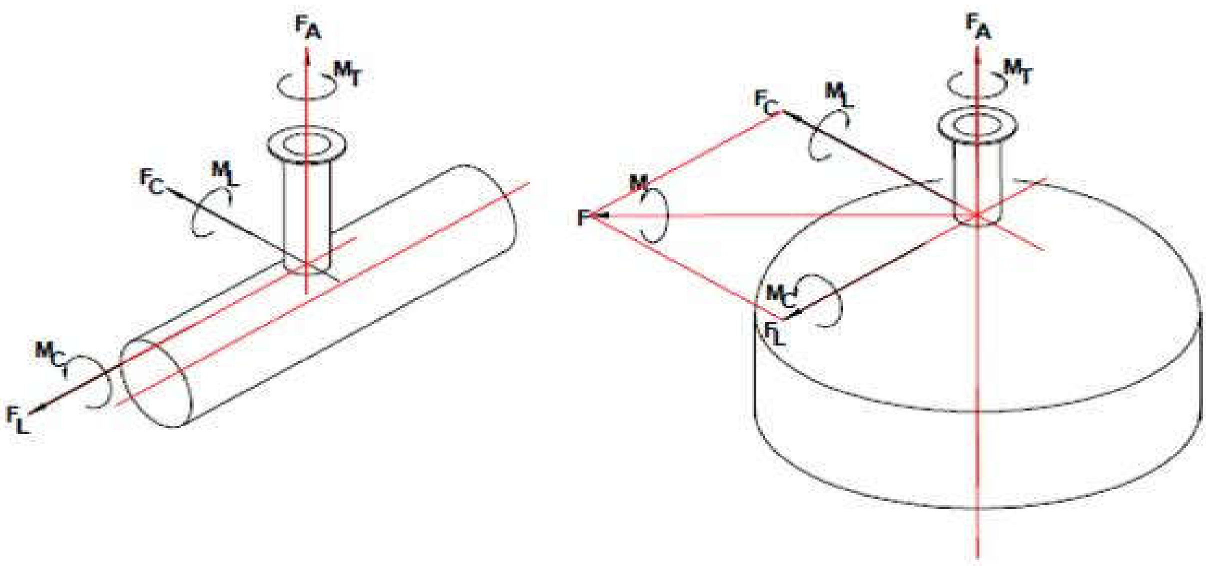
Notation

DN - Diameter Nominal, Nominal Diameter [mm]
NPS - Nominal Pipe Size [inch]
PN - Pressure Number [bar]
M - Moment [kNm]
F - Radial force [kN]

Table 1. Loading Matrix

Flange Class: ASME B16.5		150		300		600		900		1500		2500	
Flange Pressure Number, PN		20		50		100		150		250		420	
NPS	DN	M	F	M	F	M	F	M	F	M	F	M	F
2	50	0.38	1.04	0.40	1.37	0.44	1.91	0.48	2.46	0.56	3.55	0.69	5.41
3	80	1.15	1.83	1.23	2.40	1.37	3.36	1.51	4.32	1.78	6.25	2.25	9.51
4	100	1.79	2.39	1.94	3.14	2.19	4.40	2.44	5.65	2.94	8.16	3.80	12.43
6	150	3.75	3.88	4.20	5.11	4.95	7.15	5.70	9.19	7.20	13.28	9.75	20.23
8	200	6.18	5.48	7.16	7.21	8.79	10.10	10.42	12.98	13.69	18.76	19.24	28.57
10	250	9.05	7.17	10.84	9.43	13.82	13.20	16.80	16.97	22.76	24.51	32.90	37.34
12	300	12.35	8.92	15.28	11.73	20.16	16.43	25.04	21.12	34.79	30.51	51.38	46.47
14	350	16.10	10.73	20.54	14.12	27.93	19.77	35.33	25.41	50.12	36.71	N.A	N.A
16	400	20.30	12.59	26.66	16.57	37.27	23.20	47.88	29.83	69.09	43.09	N.A	N.A
18	450	24.96	14.51	33.71	19.09	48.29	26.72	62.87	34.36	92.02	49.63	N.A	N.A
20	500	30.11	16.46	41.73	21.66	61.11	30.33	80.48	38.99	119.23	56.32	N.A	N.A
24	600	41.89	20.49	54.00	26.96	92.61	37.74	124.30	48.52	187.70	70.09	N.A	N.A

All loads acting on nozzles placed on both cylindrical shell and dished head are displayed in the figure below.



Cylindrical shell Dished end

For the purpose of this approach, the following forces and moments as indicated in the above figure are of importance:

$F_A = F$; $M_C = M_L = M$

The loads F_C, F_L and M_T may be left out of consideration in this approach.

Assessment of standardized nozzle flange in case of flanged nozzles

For a standard flange conforming ASME B16.5 or ASME B16.47 Series A the following condition should be satisfied:

$$\boxed{P + [4 / \pi \,.\, G^2]\,[F + (4 \,.\, M / C \,.\, K_f)\,] \leq P_{RATING}}$$

The above mentioned expression originate from the well-known M.W. Kellogg equivalent pressure approach which assumes that the action of the moment and forces is equivalent to the action of the pressure, which produces a gasket stress that it the same as the gasket stress produced by the force and moment. However the original expression has been adjusted with the so-called Koves-factor (K_f). For those interested in the Koves - method, I refer to ref. [3] and [4].

Note that in case F is compressive, than F may be left out of consideration in the above expression!

$$K_f = 1 + \{e^2 + [0.5(A\text{-}B) - d_{hred}]^2\} / 2.6 \times e^2$$
$$d_{hred} = \max\,[d_h\,(1 - B/1000)\,;\,0.5 \times d_h\,]$$

with:

Symbols and abbreviations

A	Outside diameter of the flange (mm)
B	Inside diameter of flange (mm)
C	Bolt pitch circle diameter (mm)
d_h	Diameter of bolt holes (mm)
d_{hred}	Reduced bolt hole diameter (mm)
e	Flange thickness (mm)
F	External force (N)
G	Diameter of gasket load reaction (mm)
K_f	Koves factor (-)
M	External moment (Nmm)
P	Internal pressure (MPa)
P_{RATING}	Rated pressure according ASME B16.5 or ASME B16.47 (MPa)

The above mentioned evaluation procedure which include the 'Koves' factor may be waived when the particular nozzle flange connection is not subject to significant external loading. Significant external loading is considered to be a combination of loads and moments that, when converted to an equivalent pressure $P_e = 4/\pi.G^2\,(F + 4M/G)$, are greater than 50% of the flange rated design pressure at design temperature. For the evaluation of external loads on standard flanged joints, use can also be made of the approach described in ASME BPVC.CC.BPV-Supp.4 Case 2901[5]. ASME Section III - both NB-3658 and NC-3658 provides a method for evaluating flanges conforming ASME B16.5 with high-strength bolting. The satisfying condition for static loading is:

$M_{fs} \leq 21.7\,(S_y / 250)\,C.A_b$

where:

M_{fs} = bending moment (Nmm)
S_y = yield strength of flange material at design temperature (MPa)
C = bolt circle diameter (mm)
A_b = total cross-sectional area of bolts at root of thread or section of least diameter under stress (mm^2)
Note that $S_y/250$ shall not be greater than unity.

Conclusion

This paper offers a practical solution to deal with external loads on nozzles of pressure vessels. Use is made of authoritative and recognized sources, which is beneficial to its reliability. It enables the vessel design engineer to carry out an prudent design at an early stage of a project in which the nozzle loads to be taken into account are crucial. Moreover, it gives the piping stress engineer sufficient information within which limits the piping reactions must remain that are exerted on the nozzle.

Practical application and further elaboration

A pressure vessel must be designed for a design pressure of 65 bar (6.5 MPa) and a design temperature of 225 °C. The pressure vessel is made of carbon steel and is equipped with an NPS 16 "(NB 400) Class 600 Welding Neck Flange made of A105 material and welded to a NPS 16" Schedule 80 nozzle neck. The nominal nozzle neck thickness is 21.44 mm. Which nozzle load should be taken into account during the initial design phase of the pressure vessel? In addition, check that the nozzle flanges can withstand these loads.

According to Table 1, a moment 'M' of 37.27 kNm (37270000 Nmm) and a radial force 'F' of 23.20 kN (23200 N) should be taken into account. However the moments 'M' and the forces 'F' shall be applied simultaneously in:

- Two perpendicular directions at the right angle to the axis of pipe or in the plane tangent to the pressure retaining part at the nozzle-to-shell interface;
- Direction perpendicular to the above plane.

Remark

For flanges normally only the external axial tensile force should be considered, since this force will taking apart the flange pairs, whilst a compression axial force does not and thus can be set on zero. For the control of external flange loads, the necessary flange parameters are included in the following table:

P	Internal design pressure	6.5 MPa
P_{RATING}	Rated pressure (ASME B16.5	8.575 MPa
A	Outside diameter flange	685 mm
B	Inside diameter flange	363.52 mm
e	Flange thickness	76.2 mm
G	Diameter of gasket load reaction	442.985 mm
d_h	Bolt hole diameter	41.275 mm
C	Bolt circle diameter	603.2 mm
d_{hred}	Effective bolt hole diameter Max $[d_h\,(1 - B/1000)\,;\,0.5\,d_h]$	24.5 mm

First check

'Koves' factor: $K_f = 1 + \{e^2 + [0.5(A\text{-}B) - d_{hred}]^2\} / 2.6 \times e^2$
$K_f = 1 + \{76.2^2 + [0.5(685 - 363.52) - 24.5]^2\} / 2.6 \times 76.2^2 = 2.614$
Satisfying condition: $P + [4 / \pi \,.\, G^2]\,[F + (4 \,.\, M / C \,.\, K_f)\,] \leq P_{RATING}$
$6.5 + [\,4/\pi.442.985^2]\,[\,23200 + (4 \times 52707739 / 603.2 \times 2.614) = 7.518 < 8.575 \rightarrow$ OK!

Summary of nozzle loads pertaining to the NPS 16" Class 600 flanged nozzle located on cylindrical shell as well as on spherical part of dished head of this pressure vessel:

Summary of Nozzle and Flange Loads					
	Nozzle / Cylinder Intersection			Flange Facing	
Nozzle	F_r Radial load [N]	M_l Longitudinal moment [Nmm]	M_c Circumferential moment [Nmm]	F_{axial} Axial or radial load [N]	$M_{bending}$ Resultant bending moment [Nmm]
N1 NPS 16" Class 600	23200	37270000	37270000	23200 See remark	√2 x M √2 x 37270000 = 52707739
	Nozzle / Head Intersection			Flange Facing	
Nozzle	F_r Radial load [N]	M_m Meridional moment [Nmm]		F_{axial} Axial or radial load [N]	$M_{bending}$ Resultant bending moment [Nmm]
N2 NPS 16" Class 600	23200	√2 x M √2 x 37270000 = 52707739		23200 See remark	√2 x 37270000 = 52707739

The following checks have only been performed for comparison.

2nd check (optional)

$P_e = 4/\pi.G^2 (F + 4M/G) < 0.5\ P_{RATING} \rightarrow 4/\pi \times 442.985^2 (23200 + 4 \times 52707739/442.985)$

$P_e = 3.2385$ MPa $< 0.5 \times 8.575 = 4.2875$ MPa $\rightarrow$ Flange loading not to be considered as significant!

3rd check according[5] (optional): The following equation determines whether the applied external loads are acceptable or not.

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

where:

M_E = External moment
F_E = External tensile axial force
G = Gasket reaction diameter
P_R = Flange pressure rating at design temperature
P_D = Flange design pressure at design temperature
F_M = Moment factor, in accordance with the table stated below

Table - Flange moment factor (F_M)

Flange standard	Flange pressure class					
	150	300	600	900	1500	2500
ASME B16.5 ≤ NPS 12	1.2	0.5	0.5	0.5	0.5	0.5
ASME B16.5 12 < NPS ≤ 24	1.2	0.5	0.5	0.3	0.3	-
ASME B16.47 Series A	0.6	0.1	0.1	0.1	-	

Note that for sustained load cases the values should be divided by two

$16M_E + 4F_E\,G \leq \pi G^3[(P_R - P_D) + F_M P_R] =$
$16 \times 52707739 + 4 \times 23200 \times 442.985 \leq \pi \times 442.985^3\ [(8.575 - 6.5) + 0.5 \times 8.575]$
$884432832 < 1737579685 \rightarrow$ OK

4th check according ASME BPVC.III-1.NB-3658.1-2015 resp. ASME BPVC.III-1.NC-3 658.3-2015 (optional)

$S_y = 209$ MPa; $C = 603.2$ mm; $A_b = 20 \times 906.45 = 18129\ mm^2$ (for 20 x 1½"-8UN bolts)
$M_{fs} = 21.7\ (209/250)\ 603.2 \times 18129 = 198381511$ Nmm > 52707739 Nmm

Condition not satisfied! However this method can be considered as over-conservative for non-critical piping systems.

REFERENCES

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