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Yardstick for the evaluation of nozzle loads

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ABSTRACT: *The objective of this article is to provide both the pressure vessel design engineer and the piping stress engineer with a yardstick for the evaluation of nozzle loads. The yardstick enables the engineers involved in the design process to make a quick and reliable assessment of the piping reactions exerted by the connecting pipework on the nozzle of the pressure vessel. The nozzle load induced local stresses around the nozzle intersection are derived from the modified M.W. Kellogg's "Choking Model" equations [1].The yardstick also provides for the evaluation of the external loads acting on the nozzle flange, whereby the occurring loads (forces and moments) are converted into an equivalent pressure and successively evaluated against the rated pressure of the relevant flange, including the design pressure. Note that the original equation of M.W. Kellogg Company [2] for the determination of the equivalent pressure has been modified by including the so-called "Koves" factor [3], which results in a less conservative influence of the bending moment exerted on the flange relative to the equivalent pressure. In addition, an alternative approach is given which aims to eliminate the over-conservatism of the original Kellogg Equivalent Pressure Method and refers to reference [5]*
KEYWORDS : *nozzle loads, piping reactions, local stresses, equivalent pressure, bending moment*

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I. INTRODUCTION

The basis for this yardstick of determining local stresses in the vicinity of a nozzle intersection arises from the so-called "choking model" or "shrink ring" [6] approach that was first published by the M.W. Kellogg Company [1]. Many years later, this approach was evaluated using currently available numerical computations methods such as finite element analysis (FEA). As a result, adjustments were considered necessary to bring the results in line with the results obtained by applying finite element analysis. Ultimately, this led to the modified improved shrinkage technique (MIST) [4]. An overview of the local stress formulas and a simplified local load criterion are presented in section II. Figure 1 displays the nozzle configurations with the loads acting on the nozzles.

II. OVERVIEW of LOCAL STRESS FORMULA

Nozzle on cylindrical shell

Stress due to radial load (σ_{Fr}) [MPa]	$6.0\ F_r\ (\sqrt{R/T})/(2\pi\ r_oT)$
Stress due to longitudinal moment (σ_{Ml}) [MPa]	$1.5\ M_l\ (\sqrt{R/T})/(\pi\ r_o^2T)$
Stress due to circumferential moment (σ_{Mc}) [MPa]	$1.15\ M_c\ (\sqrt{R/T})\ (\sqrt{r_o/T})\ /\ (\pi\ r_o^2T)$

Nozzle on spherical shell

Stress due to radial load (σ_{Fr}) [MPa]	$1.75\ F_r\ (\sqrt{R/T})/(2\pi\ r_o\ T)$
Stress due to meridional moment (σ_{Mm}) [MPa]	$1.75\ M_m\ (\sqrt{R/T})/(\pi\ r_o^2T)$

Notation

F_r	Radial load (N)
M_l	Longitudinal moment (Nmm)
M_c	Circumferential moment (Nmm)
M_m	Meridional moment (Nmm)
R	Mean radius of vessel (mm)
T	Thickness of vessel (mm)
r_o	Outside nozzle radius (mm)

Shear loads and torsional moments are ignored since the pressure vessel integrity will not significantly be affected by such loads. Normally, the stress due to torsional and shear loads does not exceed 15% of the design stress.
The application range is limited to the following conditions:
$10 \le R/T \le 100$, $r_o/R \le 0.8$ and $t/T \ge 0.4$

Satisfying conditions:

Shell nozzle: $\sigma_{Fr} + \sigma_{Ml} + \sigma_{Mc} \le f$
Head nozzle: $\sigma_{Fr} + \sigma_{Mm} \le f$

The elastic shake - down criterion limits the total stress intensity to 3f . The maximum pressure stress intensity used to ensure the pressure integrity of a nozzle without external loadings is 2f. Hence (3f - 2f) = f is available for local loads due imposed external loadings. In case the nozzle reinforcement is more than required to compensate for the weakening effect, the pressure induced stress will be lower than 2f, hence, a larger stress intensity for the external load can be allowed. Directions for determining the pressure induced stresses are given in the Appendix. In order to cope with the fact that the local stresses in a relatively thin wall nozzle neck are higher than in the shell, f should be divided by a factor t/T for situation where t/T < 1.0 in the expression for the satisfying condition.

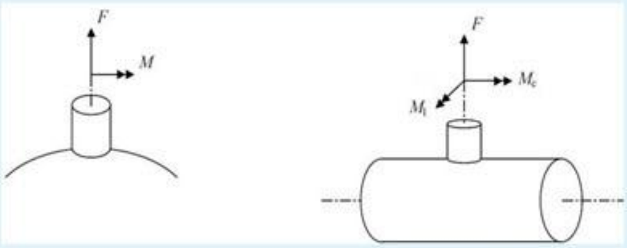

Figure 1: Nozzle configurations and loadings.

Key: F = Radial force; M = Meridional moment; M_c = Circumferential moment; M_l = Longitudinal moment

III. NOZZLES LOCATED on CYLINDRICAL SHELL

Simplified local load criterion for nozzles on cylindrical shells.

Nozzle load induced shell stresses are within acceptable limits if the inequality is satisfied:

$$\frac{3.0 \cdot R_n \cdot F_r + 1.5 \cdot M_l + 1.15 \cdot \sqrt{\frac{R_n}{T}} \cdot M_c}{\pi \cdot K} \le 1$$

Where:
F_r = radial load on the nozzle [N]
M_l = longitudinal moment on the nozzle [Nmm]
M_c = circumferential moment on the nozzle [Nmm]
R_n = outside radius of the nozzle [mm]
K = auxiliary quantity (moment-factor) [Nmm] :

$$K = \frac{(R_n \cdot T)^2 \cdot f}{\sqrt{R \cdot T}}$$

Where:
R = mean radius of the vessel [mm]
f = design stress as per applicable design code or standard [MPa]
T = thickness of the vessel [mm]

In case the nozzle is reinforced with a doubling plate or reinforcing pad then it is necessary to check the permitted external loads at two locations:

- at the nozzle/shell junction with R_n the outside nozzle radius and T the sum of the vessel thickness and the pad thickness.
- at the outer edge of the reinforcing pad with R_n the outer radius of the pad and T the vessel thickness.

IV. NOZZLES LOCATED on SPHERICAL SHELL

Simplified local load criterion for nozzles on spherical shells, in the spherical part of torispherical heads or in the central part of ellipsoidal heads.

Nozzle load induced shell stresses are within acceptable limits if the inequality is satisfied:

$$\frac{(F_r \cdot R_n) + 2 \cdot M_m}{3.6K} \leq 1.0$$

Where:

M_m = meridional bending moment on the nozzle [Nmm]

R = mean (spherical) radius of the shell [mm]
for ellipsoidal heads this is the equivalent spherical mean radius [mm]

For the remaining nomenclature, see section III.

In case the nozzle is reinforced with a doubling plate or reinforcing pad then the same procedure as in section III shall be applied.

V. EVALUATION of EXTERNAL FLANGE LOAD

For the operating/design condition the following must be satisfied:

$$P_d + [(4/\pi D_g^2)\,(F + 4M / D_{bc}.K_v)] \leq P_r$$

With : $K_v = 1 + [T_f^2 + (W_f - D_{bh}*)^2]/2.6\,T_f^2$ and $D_{bh}* = \max [\,D_{bh}\,(1 - D_i/1000)\,;\,0.5\,D_{bh}]$

Where:

P_d = internal design pressure (MPa)

P_r = rated flange pressure according ASME B16.5 or ASME 16.47 (MPa)

F = radial tension load (if compressive than ignore) (N)

M = resultant bending moment (Nmm)

D_g = mean gasket diameter (mm)

D_{bc} = bolt circle diameter (mm)

D_{bh} = bolt hole diameter (mm)

D_{bh}*= reduced bolt hole diameter (mm)

D_i = internal flange diameter (mm)

W_f = width of flange (OD flange - ID flange)/2 (mm)

T_f = flange thickness (mm)

K_v = "Koves" factor (-)

An alternative to the approach described above can be derived from reference [5].

Hence the allowable pressure, that the sum of equivalent pressure P_E and design pressure P_D shall not exceed, is increased by a so-called “Moment Factor” F_M , that depends on the flange pressure class and nominal pipe size (NPS):

$P_E + P_D \leq P_R\,(1 + F_M)$

The combination of flange design pressure with external moment and external axial tensile force shall satisfy the following equation:

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

Nomenclature

M_E = External moment (Nmm)

F_E = External tensile axial force (N)

G = Gasket reaction diameter (mm)

P_R = Flange pressure rating at design temperature (MPa)

P_D = Flange design pressure at design temperature (MPa)
F_M = Moment factor (-) in accordance with Table 1 of code case 2901[6]

VI. RECOMMENDED READING

Reference [7] is aimed at providing the engineer with more insight regarding sound nozzle design of pressure vessels, where external loads exerted on the nozzle by the connecting pipework are of crucial importance.

VII. ELABORATED CASE

A pressure vessel is designed for an internal design pressure of 10 bar (1 MPa) and a design temperature of 200°C. The vessel shell and torispherical (korbbogen) head are both made from A515 Grade 60 carbon steel and has an outside diameter of 1200 mm and a minimum thickness of 10 mm. The cylindrical shell and head are both provided with an NPS 12" nozzle. The NPS 12" nozzle neck thickness is 9.5 mm and provided with a Class 150 welding neck flange made of A105 carbon steel complying to ASME B16.5. No reinforcing pads are added. The design stress for the A515 Grade 60 material is 126 MPa and for the nozzle necks 138 MPa. Thickness tolerance and corrosion allowance are neglected. The flange rated pressure is 13.8 bar @ 200°C. The nozzle flange is equipped with a spiral wound gasket according to ASME B16.20.
The nozzle load summary below defines the design nozzle loads applied by piping on pressure vessels nozzles which shall be designed to withstand these loads. All tabulated force and moment components may be +ve or –ve. The moments and forces shall be applied simultaneously.

Nozzle Load Summary

SHELL NOZZLE	F_r = 4500 N	M_l = 3500 Nm	M_c = 1150 Nm
HEAD NOZZLE	Fr = 3750 N	M_m = 4500 Nm	

Evaluation of local stresses around shell nozzle

$\sigma_{Fr} = 6.0\ F_r\ (\sqrt{R/T})/(2\pi r_o T) = 6.0 \times 4500\ (595/10)^{½} / (2 \times \pi \times 161.95 \times 10) = 20.47$ MPa
$\sigma_{Ml} = 1.5\ M_l\ (\sqrt{R/T})/(\ \pi r_o^2 T) = 1.5 \times 3500000\ (595/10)^{½} / (\pi \times 161.95^2 \times 10) = 49.15$ MPa
$\sigma_{Mc} = 1.15\ M_c (\sqrt{R/T})\ (\sqrt{r_o/T}) / (\pi\ r_o^2 T) = 1.15 \times 1150000\ (595/10)^{½} (\ 161.95\ /10)^{½} / (\pi \times 161.95^2 \times 10)$
$\sigma_{Mc} = 49.82$ MPa
$\sigma_{Fr} + \sigma_{Ml} + \sigma_{Mc} = 20.47 + 49.15 + 49.82 = 119.44$ MPa $< 126 \times 9.5/10 = 119.7$ MPa

Evaluation of local stresses around head nozzle

$\sigma_{Fr} = 1.75\ F_r\ (\sqrt{R/T})/(2\pi r_o T) = 1.75 \times 3750\ (965/10)^{½} / (2 \times \pi \times 161.95 \times 10) = 6.34$ MPa
$\sigma_{Mm} = 1.75\ M_m\ (\sqrt{R/T})/(\ \pi_{ro}^2 T) = 1.75 \times 4500000\ (965/10)^{½} / (\pi \times 161.95^2 \times 10) = 93.89$ MPa
$\sigma_{Fr} + \sigma_{Mm} = 6.34 + 93.89 = 100.23$ MPa $< 126 \times 9.5/10 = 119.7$ MPa

Evaluation of loads on shell nozzle

Auxiliary quantity (moment-factor): $K = \dfrac{(R_n \cdot T)^2 \cdot f}{\sqrt{R \cdot T}}$ [Nmm]

$K = [(323.9/2 \times 10)^2\ 126] /\{[(1200 - 10)/2]\ 10\}^{½} = 4284241.7$ Nmm

$$\frac{3.0 \cdot R_n \cdot F_r + 1.5 \cdot M_l + 1.15 \cdot \sqrt{\frac{R_n}{T}} \cdot M_c}{\pi \cdot K} \leq 1$$

$[3.0 \times 161.95 \times 4500 + 1.5 \times 3500000 + 1.15\ (161.95/10)^{½} \times 1150000] /(\pi \times 4284241.7) = 0.948 < 1.0$
∴ Condition is met!

Flange Data (Dimensions in mm)

O.D.	I.D.	G	D_g	D_{bc}	D_{bh}	D_i	W_f	T_f
485	304.9	357.5	355.8	431.8	25.4	304.9	90.05	30.2

Evaluation of shell nozzle flange

$D_{bh}* = \max [\ D_{bh} (1 - D_i/1000)\ ; 0.5\ D_{bh}]$
$D_{bh}* = \max [\ 25.4\ (1 - 304.9/1000)\ ; 0.5 \times 25.4] = \max [17.65\ ; 12.7] = 17.65$ mm
$K_v = 1 + [T_f^2 + (W_f - D_{bh}*)^2]/2.6\ T_f^2 = 1 + [30.2^2 + (90.05 - 17.65)^2\ ] / 2.6 \times 30.2^2 = 3.595$
$P_d + [(4/\pi D_g^2)\ (F + 4M / D_{bc} \times K_v)] \leq P_r$
$1 + [(4/\pi \times 355.8^2)\ (4500 + 4 \times (3500000^2 + 1150000^2)^{½} / 431.8 \times 3.595)] = 1.14 < 1.38$
∴ Condition is met!

Alternative evaluation

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

F_M is 1.2 according Table 1 0f ASME Code Case 2901 and

$M_E = (3500000^2 + 1150000^2)^{½} =$ 3684087 Nmm

$16 \times 3684087 + 4 \times 4500 \times 357.5 \leq \pi \times 357.5^3[(1.38 - 1) + 1.2 \times 1.38]$

$65380398 \leq 292250851.2$ $\rightarrow$ The condition is amply met! (Ratio: 4.47)

Evaluation of loads on head nozzle

Auxiliary quantity (moment-factor): $K = \frac{(R_n \cdot T)^2 \cdot f}{\sqrt{R \cdot T}}$ [Nmm]

$K = ([(323.9/2 \times 10)^2\ 126] / [(0.8 \times 1200 + 0.5 \times 10)]\ 10\}^{½} = 3364099.2$ Nmm

$$\frac{(F_r \cdot R_n) + 2 \cdot M_m}{3.6K} \leq 1.0$$

$[(3750 \times 323.9/2) + 2 \times 4500000] / 3.6 \times 3364099.2 = 0.793 \leq 1.0$

$\therefore$ Condition is met!

Evaluation of head nozzle flange

Since head nozzle flange is identical to the shell nozzle flange, K_v amount to 3.595

$P_d + [(4/\pi D_g^2)\ (F + 4M / D_{bc} \times K_v)] \leq P_r$

$1 + [(4/\pi \times 355.8^2)\ (3750 + 4 \times (4500000) / 431.8 \times 3.595)] = 1.154 < 1.38$

$\therefore$ Condition is met!

Alternative evaluation

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

F_M is 1.2 according Table 1 0f ASME Code Case 2901 and

$16 \times 4500000 + 4 \times 3750 \times 357.5 \leq \pi \times 357.5^3[(1.38 - 1) + 1.2 \times 1.38]$

$77362500 \leq 292250851.2$ $\rightarrow$ The condition is amply met! (Ratio: 3.778)

VIII. CONCLUSIONS

This yardstick forms a standard calculation basis that the author has successfully applied for more than 20 years when evaluating nozzle loads on different pressure vessels. A feature of this approach is the ease with which insight can be obtained about the load capacity of flanged nozzles on pressure vessels. Moreover, this approach is extremely suitable for processing in a spreadsheet, so that quick results can be achieved. The results obtained with this yardstick correspond perfectly with those obtained from numerical (FEA) computations, which should inspires confidence by the user of this yardstick. It can certainly compete with the methodologies described in WRC bulletins # 107 respectively # 297, which are rather limited in their application because of geometric restrictions. Moreover, the WRC methods does not cover stress from internal pressure. The technology incorporated in the Yardstick gives a significant improvement over the limitations inherent in WRC 107/297 methods.

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APPENDIX

The pressure stress for flush nozzles in cylindrical shells can be determined by the following empirical equations:

For relatively thin wall nozzle necks ($t/T \leq 1.0$):

$\sigma_P = PD/2T\ [2 + d/D\ (d\ t\ /D\ T)^{½} + 1.25\ d/D\ (D/T)^{½}\ ]\ /\ [1+ t/T\ (d\ t\ /D\ T)^{½}]$

For relatively thick wall nozzle necks ($t/T > 1.0$):

$\sigma_P = PD/2T\ [2.5 + 1.716\ d/D\ (d\ t\ /D\ T)^{½} + 0.907\ d/D\ (D/T)^{½}\ ]\ /\ [1+ 0.94\ t/T\ (d\ t\ /D\ T)^{½}]$

Where:

d = mean nozzle diameter (mm)

D = mean cylindrical shell diameter (mm)

t = nozzle neck thickness (mm)

T = thickness of cylindrical shell (mm)

P = internal pressure (MPa)

σ_P = pressure stress (MPa)

In clause G.2.5 of PD 5500 [10] information can be found to determine the pressure-induced stress for nozzles in spherical shells. However, there are also other authoritative sources [11][12] where information about this subject can be found. Determining the pressure stress for pad reinforced nozzles is more complicated and it is recommended to consult recognized design codes or sources about this.

Calculation of the pressure induced stress of the shell nozzle from the elaborated case:

$\sigma_P = PD/2T\ [2 + d/D\ (d\ t\ /D\ T)^{½} + 1.25\ d/D\ (D/T)^{½}\ ]\ /\ [1+ t/T\ (d\ t\ /D\ T)^{½}]$

With: P = 1 MPa, T = 10 mm, D = 1190 mm, t = 9.5 mm and d = 314.4 mm follows $\sigma_P = 231.2$ MPa

It turns out that: $\sigma_P < 2 \times f = 2 \times 126 = 252$ MPa and that the ratio $\sigma_P/2f$ becomes 0.917 which is satisfactory.

$\sigma_P + \sigma_{Fr} + \sigma_{Ml} + \sigma_{Mc} = 231.2 + 20.47 + 49.15 + 49.82 = 350.64$ MPa $< 3\ f = 3 \times 126 = 378$ MPa

Ratio $(\sigma_P + \sigma_{Fr} + \sigma_{Ml} + \sigma_{Mc})\ /\ 3\ f = 350.64/378 = 0.928 < 1.0$

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