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Research Article

Pressure-Induced Stresses at Nozzle–Shell Intersections in Pressure Vessels: An SCF-Based Analytical Approach

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Abstract - *Accurate assessment of local stresses at nozzle–shell intersections is essential for the safe design of pressure vessels, as these regions are highly susceptible to stress concentrations under combined internal pressure and external loading. Existing design methodologies, such as WRC Bulletins, primarily address externally applied loads and do not explicitly account for pressure-induced stress intensification. This study presents empirical formulations for stress concentration factors (SCFs) to quantify pressure-induced stresses at nozzle–shell intersections in cylindrical and spherical shells, including configurations with reinforcing pads. The proposed SCFs utilize important geometric parameters and are designed to be used directly in engineering calculations, allowing them to be integrated with existing WRC-based processes via stress superposition. An example representative gives an illustration of how the method can be applied and illustrates the role of pressure-induced stresses in the overall stress condition, which may have a considerable impact on the available margin of external nozzle loads. Validation is carried out by comparative evaluation with settled analytical techniques, such as the Rules for Pressure Vessels (RfPV) and a revised Decock-based model.The findings reveal that the RfPV-based methodology offers balanced and physically consistent estimates, whereas the developed empirical methodology offers conservative estimates. The suggested methodology provides a viable model of integrating the effects of pressure-induced stress into the common nozzle design and testing. It is suitable for preliminary design, verification and screening engineering but more detailed numerical analysis is advisable in complex or critical applications.*

Keywords - *Pressure Vessels, Nozzle-Shell Intersection, Stress Concentration Factor, Internal Pressure, Reinforcing Pad, WRC Methods, Structural Integrity.*

I. INTRODUCTION

The design of pressure vessels requires accurate evaluation of local stresses at nozzle–shell intersections, where geometric discontinuities result in significant stress concentrations. These regions are often critical for structural integrity, particularly under combined internal pressure and external piping loads. The standard design procedures, including WRC Bulletin 107 [1], WRC Bulletin 297 [2], and WRC Bulletin 368 [3], offer a standardized approach to analyzing local stresses caused by external forces and moments on nozzles. Such techniques are common in engineering practice; but they tend to deal with externally induced stresses only and explicitly do not deal with the stress intensification caused by pressure at nozzle-shell intersections.

Previous analytical studies, including Dekker and Stikvoort [4], have investigated stress distributions in intersecting cylindrical shells under internal pressure. While these provide valuable theoretical insight, their complexity limits direct application in routine design. In engineering practice, combined loading conditions involve considering both internal pressure and external nozzle loads. Where there exists no clear direction in WRC procedures, simplified assumptions are frequently employed, and this can result in either over-conservative or non-conservative designs. To address this limitation, this paper will suggest empirical SCF equations that will utilize the essential geometric parameters such as nozzle diameter, shell thickness, and reinforcing pad geometry. The formulations are meant to be extensions of the available WRC-based techniques by stress superposition and retention of simplicity in engineering application. The scope is limited to cylindrical and spherical shells within defined geometric ranges. The formulations are based on analytical considerations and calibrated observations and are intended for conservative design use.

II. METHODOLOGY AND FORMULAS

A. Stress Concentration Factors (SCFs)

Stress concentration factors (SCFs) are also presented to explain pressure-related stresses at nozzle-shell intersections. The formulations provided below are empirical and shall be applied within the specified geometric limits.

Table 1. SCF Formulations

Shell Type	Repad	SCF Expression	Validity Range
Cylindrical	Yes	$SCF = \left[2.2 + 5\left(\frac{d}{D}\right)\right].[1.1 - 0.1\left(\frac{t_n}{t_s}\right)].[1.2 - 0.2\left(\frac{b}{\sqrt{R.t_s}}\right)\sqrt{(\frac{t_p}{t_s})}]$	$0.1 \leq \frac{d}{D} \leq 0.5$; $0.2 \leq \frac{t_n}{t_s} \leq 1.0$; $0.2 \leq \frac{b}{\sqrt{R.t_s}} \leq 1.0$; $0.5 \leq \frac{t_p}{t_s} \leq 1.5$
Cylindrical	No	$SCF = \left[2.2 + 5\left(\frac{d}{D}\right)\right].[1.1 - 0.1\left(\frac{t_n}{t_s}\right)]$	
Spherical	Yes	$SCF = [1.8 + 4(\frac{d}{D})].[1.1 - 0.1\left(\frac{t_n}{t_s}\right)][1.15 - 0.15\left(\frac{b}{\sqrt{R.t_s}}\right)\sqrt{(\frac{t_p}{t_s})}]$	
Spherical	No	$SCF = \left[1.8 + 4\left(\frac{d}{D}\right)\right].[1.1 - 0.1\left(\frac{t_n}{t_s}\right)]$	

Notes

- $\frac{d}{D}$: outside nozzle diameter / mean shell diameter
- $\frac{t_n}{t_s}$: nozzle neck thickness / shell thickness
- $\frac{t_p}{t_s}$: pad thickness / shell thickness
- $\frac{b}{\sqrt{R.t_s}}$: width of repad / $\sqrt{(\text{mean shell radius x shell thickness})}$ = dimensionless pad width parameter
- $R = D/2$

These expressions are empirical and should not be used outside the stated ranges. Note that calibration of the formula have been applied against RfPV sheet 1141[5] within the defined geometric range.

Table 2. Stress Formula for Intersecting Nozzles

Shell Type	Stress Formula for Undisturbed Shells	Stress Formula For Disturbed Shells
Cylindrical	$f_{cyl} = \frac{P_d}{\ln\left(\frac{D_{o;cyl}}{D_{i;cyl}}\right)}$	$f_{intersection} = SCF.\, f_{cyl}$
Spherical	$f_{sph} = \frac{P_d}{2\ln\left(\frac{D_{o;sph}}{D_{i;sph}}\right)}$	$f_{intersection} = SCF\, .\, f_{sph}$

Nomenclature

- f_{cyl} = Membrane stress cylindrical shell
- f_{sph} = Membrane stress spherical shell
- P_d = Internal pressure
- $D_{o;cyl}$ = Outside diameter of cylindrical shell

- $D_{i;cyl}$ = Inside diameter of cylindrical shell
- $D_{o;sph}$ = Outside diameter of spherical shell
- $D_{i;sph}$ = Inside diameter of spherical shell
- d = Outside diameter nozzle neck
- D = Mean shell diameter
- t_s = shell thickness
- t_p = pad thickness
- R = Mean shell radius
- ln = Natural logarithm
- SCF = Stress concentration factor corresponding to the case
- $f_{intersection}$ = Stress at nozzle intersection including SCF

Units: Dimensions in mm; stresses and pressures in MPa.

III. DISCUSSION

Assessment of nozzle loading requires a clear understanding of the allowable stress margin available for externally applied piping loads. During his employment at NAM a joint venture between Shell plc and ExxonMobil the author played a key role in developing the engineering specification for pressure vessels (NSS 12-D-4-05) [9], which is a supplement to the "Rules for Pressure Vessels" [5]. This specification establishes acceptance criteria on the imposed piping reactions at the junction of the nozzle and shell. The principle behind this is that this junction should be reinforced to ensure that stresses caused by internal pressure do not surpass 2 x the design stress whilst being consistent with the requirements of the relevant code or standard.

When the elastic shakedown criterion is applied, with a limit of 3 x the design stress, it is assumed the residual margin is 1 x the design stress, which may be used by stresses due to external piping loads on the nozzle. This methodology of assessment is described in detail in Section 3.6 of NSS 12-D-4-05 [9]. Accordingly, the stress caused by internal pressure must be deducted from this limit to determine the remaining capacity available for external nozzle loads.

The previously proposed expression:

$$SCF_p = 3 + K\left(\frac{d}{D}\right)^2\left(\frac{T}{t}\right)^{0.5}\Phi_p \,.\, \Psi_b$$

where the correction factors are defined as:

$$\Phi_p = \text{pad thickness factor} = 1 + 0.7\left(\frac{t_p}{T}\right) - 0.15\left(\frac{t_p}{T}\right)^2$$

$$\Psi_b = \text{pad width factor} = \frac{1}{1+0.6\frac{b}{\sqrt{R.T}}}$$

with:
T = t_s and t = t_n
K = geometric constant:
K = 2.2 - 3.0 for nozzle on cylindrical shell
K = 1.8 - 2.2 for nozzle on spherical shell

In the absence of a reinforcing pad, both correction factors reduce to unity : Φ_p =1.0 and Ψ_b =1.0 was evaluated, with correction factors for pad thickness and width. However, practical application has shown that this formulation does not consistently yield reliable or physically realistic results. Its sensitivity to geometric ratios can cause overestimation and underestimation of the effects of stress concentration effects.

In contrast, the SCF obtained using the Rules for Pressure Vessels (RfPV)[5], specifically Sheet D 1141 with reinforcement efficiency of Sheet D 0501, however, is much closer to that of the numerical analyses. In this method: SCF = 2.5/z, where z denotes the reinforcement efficiency (or strength reduction coefficient). The method is intrinsically consistent, and the overall contribution of shell, nozzle and reinforcing pad geometry is consistently and physically meaningful manner. The comparison reveals that the RfPV-based formulation makes more consistent and realistic estimates, especially when the configurations involve reinforcing pads. This observation is the foundation of revised SCF expressions that are presented in this study.

IV. WORKED EXAMPLE

A representative example is presented to demonstrate the application of the proposed method and to compare different SCF formulations.

A. Pressure Vessel and Nozzle Data

- **Pressure Vessel**: Outside cylindrical shell diameter = 1200 mm, wall thickness = 14.2 mm (net)
- **Nozzle**: Set-in flush nozzle, external diameter = 323.8 mm, neck thickness = 9.6125 mm (net)
- **Reinforcing Pad**: Width = 80 mm, thickness = 15.7 mm (net)

B. Design Conditions

- Internal pressure = 25 bar (2.5 MPa)
- Temperature = 150°C

C. Materials and Design Stresses

- **Vessel shell & reinforcing pad**: A515 Grade 70, design stress = 154.667 MPa
- **Nozzle neck**: A106 Grade B, design stress = 142.667 MPa

D. Analysis Objectives

- Determine the stress concentration factor (SCF)
- Calculate the pressure-induced stress at the nozzle intersection
- Determine the available stress for external nozzle loads

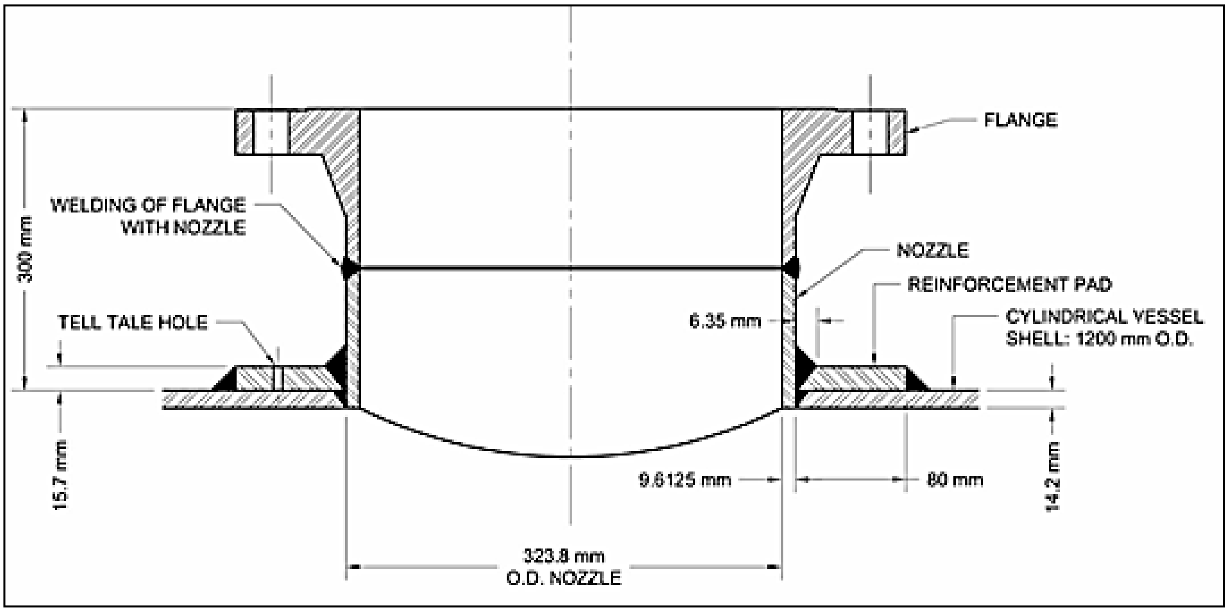

Figure 1. Typical Nozzle Configuration Illustrating a Flush Set-in Nozzle with a Reinforcing Pad, in Accordance with the Worked Example

Calculation 1

$$\text{SCF} = \left[2.2 + 5\left(\frac{d}{D}\right)\right].\,[1.1 - 0.1\left(\frac{t_n}{t_s}\right)].\,[1.2 - 0.2\left(\frac{b}{\sqrt{R\,.t_s}}\right)\sqrt{(\frac{t_p}{t_s})}]$$

$$\text{SCF} = \left[2.2 + 5\left(\frac{323.8}{1185.8}\right)\right].\,[1.1 - 0.1\left(\frac{9.6125}{14.2}\right)].\,[1.2 - 0.2\left(\frac{80}{\sqrt{592.9 \times 14.2}}\right)\sqrt{(\frac{15.7}{14.2})}] = 3.7418$$

Calculation 2

$$f_{cyl} = \frac{P_d}{\ln\left(\frac{D_{o;cyl}}{D_{i;cyl}}\right)}$$

$$f_{cyl} = \frac{2.5}{\ln\left(\frac{1200}{1171.6}\right)} = 104.379 \text{ MPa}$$

$f_{intersection}$ = SCF. f_{cyl} = 3.7418 x 104.379 = 390.565 MPa

Calculation 3

Available stress for external nozzle loads = 3 x 154.667 - 390.565 = 73.436 MPa.

Check of Recommended Geometric Range

$$0.1 \le \frac{d}{D} \le 0.5 \rightarrow \frac{d}{D} = \frac{323.8}{1185.8} = 0.273$$

$$0.2 \le \frac{t_n}{t_s} \le 1.0 \rightarrow \frac{t_n}{t_s} = \frac{9.6125}{14.2} = 0.677$$

$$0.2 \le \frac{b}{\sqrt{R.t_s}} \le 1.0 \rightarrow \frac{b}{\sqrt{R.t_s}} = \frac{80}{\sqrt{592.9 \times 14.2}} = 0.872$$

$$0.5 \le \frac{t_p}{t_s} \le 1.5 \rightarrow \frac{t_p}{t_s} = \frac{15.7}{14.2} = 1.106$$

All parameters fall within the recommended ranges.

A detailed analytical verification in accordance with the Pressure Vessel Rules (RfPV)[5] is provided in the annex. Moreover, a stress concentration factor (SCF) formulation is provided, which is based on a simplified version of the expression of Decock [7]. This process in some conditions transforms the thickness of the plate of reinforcement along with the thickness of the wall into a similar wall thickness. Further elaboration on this approach is also included.

V. VALIDATION AND COMPARATIVE ASSESSMENT

To evaluate the reliability of the proposed SCF formulations, the results obtained in the worked example are compared with established analytical approaches, namely the Rules for Pressure Vessels (RfPV) method [5] and a variation of the same, using the Decock method, to assess the reliability of the proposed SCF formulations. Finite element analysis (FEA) is not part of this work; however, both reference methods are common and have been previously shown to show a good agreement with numeric results.

For the representative case presented in Section IV, the following stress concentration factors (SCFs) were obtained:

- Proposed empirical formulation: SCF = 3.74
- RfPV-based formulation: SCF = 3.24
- Modified Decock-based formulation: SCF ≈ 4.03–4.05

The corresponding pressure-induced stresses at the nozzle–shell intersection are:

- Empirical formulation: 390.6 MPa
- RfPV-based method: 338.2 MPa
- Modified Decock formulation: ≈ 421–423 MPa

The comparison shows that the modified Decock formulation yields the highest SCF and corresponding pressure-induced stress, followed by the proposed empirical formulation, while the RfPV method provides the lowest values. This trend reflects the different underlying modelling assumptions:

- The RfPV method incorporates reinforcement efficiency through a global parameter (z), leading to stable and moderated stress predictions.

- The empirical formulation provides conservative estimates through simplified geometric relationships.
- The modified Decock formulation introduces an equivalent shell thickness, which strongly influences stiffness representation and can lead to higher stress concentration depending on how teq is defined.

The findings also indicate that the Decock-based methodology is extremely sensitive to the definition of equivalent shell thickness, but differences between other teq formulations create only small changes in SCF values in this case.

The observed trend:
SCF_Decock ≥ SCF_empirical ≥ SCF_RfPV indicates that methods incorporating equivalent thickness may predict higher local stress intensification, while RfPV provides a more balanced estimate. Although validation is limited to analytical comparison, the similarities of the trends prove the relevance of the offered method to preliminary design and verification. In more critical applications or geometries that are not within the recommended ranges, additional verification on more detailed methods (e.g., FEA) is advised. It should be noted that the RfPV reinforcement efficiency parameter z intrinsically loads the shell, nozzle, and pad in a smeared manner, which mitigates the peak stress prediction.

VI. CONCLUSIONS

This study presents a practical analytical method to assess stresses caused by pressure at nozzle-shell intersections with the help of SCFs. The technique builds upon the traditional design practice, explicitly introducing the stress intensification imposed by pressures, which is not explicitly considered in standard WRC procedures. Existing SCF formulations from the literature do not consistently provide reliable predictions across a broad range of geometric configurations. Conversely, SCFs based on the RfPV[5] have good agreement with numerical studies and offer a more practical foundation on which to base engineering. The formulations suggested make it possible to directly compute local stresses and integrate with WRC- based methods, thus making it possible to integrate with existing WRC-based methods easily using stress superposition. The example of work shows that stresses due to pressure may cause a substantial lowering of the permissible margin of external nozzle loads. The method is particularly suited for preliminary design, verification, and engineering screening . However, for applications involving complex geometries or loading conditions, detailed numerical analysis is recommended.

VII. ACKNOWLEDGEMENT

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ANNEX

VESSEL INPUT DATA			
Symbol	**Description**		**Unit**
D_o	Outside diameter shell	1200	mm
D_i	Inside diameter shell	1171.6	mm
t_s	Shell thickness	14.2	mm
t_n	Nozzle neck thickness	9.6125	mm
d_n	Outside nozzle diameter	323.8	mm
d_i	Inside nozzle diameter	304.575	mm
L_p	Width of reinforcing pad	80	mm
t_p	Thickness of reinforcing pad	15.7	mm
K_s	Shell factor	1.0	-
K_n	Nozzle factor	1.0	-
K_p	Repad factor	0.75	-

Calculation

In this worked example, the pressure–area method as presented in reference [6] has been applied.

$$\lambda = \frac{d_i}{\sqrt{(D_o - t_s)t_s}} = \frac{304.575}{\sqrt{(1200-14.2)14.2}} = 2.347$$

$$L_n = k_n\ \sqrt{(d_n - t_n)t_n} = 1.0\sqrt{(323.8 - 9.6125)9.6125} = 54.956\ \text{mm}$$

$$L_s = k_s\ \sqrt{(D_o - t_s)t_s} = 1.0\sqrt{(1200 - 14.2)14.2} = 129.763\ \text{mm} \Rightarrow \lambda = 2.347 \Rightarrow k_s = 1.0$$

$$A_p = \frac{D_i}{2}\left(L_s + \frac{d_n}{2}\right) + \frac{d_i}{2}(L_n + t_s)$$

$$A_p = \frac{1176.6}{2}\left(129.763 + \frac{323.8}{2}\right) + \frac{304.575}{2}(54.956 + 14.2) = 182116.937\ \text{mm}^2$$

$$A_f = t_s\ .L_s + (L_n + t_s)t_n\ .\left(\frac{f_n}{f_s}\right) + k_p\ (L_p\ .t_p)\left(\frac{f_p}{f_s}\right)$$

$$A_f = 14.2 \text{ x } 129.763 + (54.956 + 14.2)9.6125.\left(\frac{142.667}{154.667}\right) + 0.75\ (80 \text{ x } 15.7)\left(\frac{154.667}{154.667}\right) = 3397.82\ \text{mm}^2$$

$$z = C\,.\frac{D_i+d}{d}\cdot\frac{A_f}{2 \text{ x } A_p + A_f} = 1.0\ \frac{1171.6+14.2}{14.2}\cdot\frac{3397.82}{2 \text{ x } 182116.937 + 3397.82} = 0.7718$$

$$SCF = \frac{2.5}{z} = \frac{2.5}{0.7718} = 3.24 \rightarrow \text{Pressure stress } \sigma_p = SCF.\ P_d\,\frac{D_i+d}{2d}$$

$$\text{with } P_d = 2.5\ \text{MPa} \rightarrow \sigma_p = 3.24 \text{ x } 2.5\ \frac{1171.6+14.2}{2 \text{ x } 14.2} = 338.2\ \text{MPa}$$

Available stress for external nozzle loads = 3 x 154.667 - 338.2 = 125.8 MPa.

Elaboration of the Modified Expression According to [7]

The stresses are evaluated at the nozzle edge, with the equivalent shell thickness initially defined as $t_{eq} = t_s + t_p$. However, when a reinforcing pad of width: $L_p < \sqrt{(D_o - t_s)(t_p + t_s)}$ is applied, the equivalent thickness t_{eq} should be determined as:

$$\boxed{t_{eq} = t_s + \min\left(\frac{t_p\,L_p}{\sqrt{(D_o - t_s)(t_s+t_p)}}\ ;\ (t_p)\ ;\ \min\left(\frac{f_2}{f_s}, 1.0\right)\right)}$$

The stress concentration factor (SCF) is then given by:

$$SCF = \frac{2+2\frac{d}{D}\sqrt{\frac{d.t_n}{D.t_{eq}}} + 1.25\frac{d}{D}\sqrt{\frac{D}{t_{eq}}}}{1+\frac{t_n}{t_{eq}}\sqrt{\frac{d.t_n}{D.t_{eq}}}}$$

Where (notation & physical meaning):

- t_{eq}= equivalent shell thickness at the nozzle edge (mm).
- t_n= nozzle neck thickness (mm).
- t_s= shell thickness at the nozzle location (mm).
- t_p= reinforcement pad thickness (mm).
- L_p= width of reinforcing pad (mm).
- D = mean shell diameter (mm).
- d = mean nozzle diameter (mm)

- $f_2$and f_s= material strength factors (MPa) — the ratio f_2/f_slimits the reinforcement contribution if $f_2 < f_s$.
- The inner $\min\left(\frac{f_2}{f_s}, 1.0\right)$ ensures the contribution does not exceed the reinforcement's permitted strength ratio.

Calculation

$$L_p = 80 \text{ mm} < \sqrt{(D_o - t_s)(t_p + t_s)} = \sqrt{(1200 - 14.2)(15.7 + 14.2)} = 188.3 \text{ mm}$$

$$t_{eq} = t_s + \min\left(\frac{t_p \,.\, L_p}{\sqrt{(D_o - t_s)(t_s + t_p)}} \; ; (t_p) \; ; \min\left(\frac{f_2}{f_s}, 1.0\right)\right)$$

$$t_{eq} = 14.2 + \min\left(\frac{15.7 \text{ x } 80}{\sqrt{(1200 - 14.2)(14.2 + 15.7)}} \; ; (15.7) \; ; \min\left(\frac{154.667}{154.667}, 1.0\right)\right) = 20.87 \text{ mm}$$

$$SCF = \frac{2 + 2\frac{d}{D}\sqrt{\frac{d.t_n}{D.t_{eq}}} + 1.25\frac{d}{D}\sqrt{\frac{D}{t_{eq}}}}{1 + \frac{t_n}{t_{eq}}\sqrt{\frac{d.t_n}{D.t_{eq}}}} = \frac{2 + 2\frac{314.1875}{1185.8}\sqrt{\frac{314.1875 \text{ x } 9.6125}{1185.8 \text{ x } 20.87}} + 1.25\frac{314.1875}{1185.8}\sqrt{\frac{1185.8}{20.87}}}{1 + \frac{9.6125}{20.87}\sqrt{\frac{314.1875 \text{ x } 9.6125}{1185.8 \text{ x } 20.87}}} = 4.0327$$

Pressure stress is : SCF x f_{cyl} = 4.0327 x 104.379 = 420.93 MPa

Available stress for external nozzle loads = 3 x 154.667 - 420.93 = 43.1 MPa

In the literature [8], it is noted that the reinforcing pad and the vessel shell should not be treated as an integral structure. Instead, an effective pad thickness should be used which is effective in capturing the sum of the bending stiffness of an independent pad and shell and yet is also conservative to local membrane stresses. The total thickness of the vessel shell and reinforcing pad may be determined as the following equation:

$t_{eq} = (t_s{}^{2.5} + t_p{}^{2.5})^{0.4}$ For the case in question, this leads to: $t_{eq} = 19.764$ mm

If we successively substitute this equivalent thickness of 19.764 into the formula for SCF, the following is obtained: SCF = 4.049

Pressure stress is : SCF x f_{cyl} = 4.049 x 104.379 = 422.6 MPa

Available stress for external nozzle loads = 3 x 154.667 - 422.6 = 41.379MPa

VIII. DISCUSSION OF RESULTS

The results clearly demonstrate that the selection of the stress concentration factor (SCF) formulation has a decisive influence on the predicted pressure-induced stresses at the nozzle–shell intersection and, consequently, on the allowable capacity for external nozzle loads. The modified Decock-based formulation yields the highest SCFs (≈ 4.03–4.05) and, therefore, the highest pressure-induced stresses (≈ 421–423 MPa). This results in the most restrictive condition, with the available stress for external loading reduced to approximately 41–43 MPa. This behaviour is directly linked to the formulation's sensitivity to the definition of the equivalent shell thickness (t_{eq}), which governs the local stiffness representation. Though this approach provides a refined capture of stiffness effects, it may cause high stress intensifications and should be used with strong verification of its assumptions and validity limits.

The proposed empirical formulation produces slightly lower SCFs (≈ 3.74) and corresponding pressure-induced stresses (≈ 391 MPa), resulting in an available stress of approximately 73 MPa. This indicates a consistently conservative prediction, while remaining less restrictive than the Decock-based approach. The formulation does this in simplified geometric relations which is done to intentionally bias the results to safety within the stated applicability range. The RfPV-based method provides the lowest SCF (≈ 3.24) and corresponding pressure-induced stress (≈ 338 MPa), yielding a significantly higher available stress of approximately 126 MPa. This method offers a balanced and physically consistent representation of structural behaviour by incorporating reinforcement efficiency through the parameter z. As a result, it reflects a more realistic redistribution of stresses between the shell, nozzle, and reinforcing pad.

The observed relationship between the methods can therefore be summarized as:

SCF_Decock ≥ SCF_empirical ≥ SCF_RfPV

This trend highlights the influence of modelling assumptions:

- Methods based on equivalent thickness concepts (Decock) tend to increase predicted stress intensification due to their sensitivity to local stiffness representation.
- Empirical formulations provide conservative estimates through simplified parameterization.
- RfPV-based approaches moderate stress predictions by incorporating global reinforcement behaviour.

It is important to note that the variation in available stress from approximately 41 MPa to 126 MPa is substantial and has direct implications for design decisions. Depending on the selected method, the allowable external loading capacity may differ by a factor of up to three. This underscores the necessity of selecting an appropriate method aligned with the design objective (e.g., conservative screening versus optimized design).

Furthermore, there are insignificant variations in values of SCF between the different equivalent thickness formulations compared in the Decock-based approach. This fact underlies the strength of the concept of equivalent thickness, however the absolute stress levels are fragile to the definition of it. In general, the findings validate that stresses due to pressure are a predominant factor in the local stress state at the nozzle-shell intersections and need to be taken into account when evaluating designs. The failure to take these effects into consideration may result in a major overstatement of the allowable margin of external loads. The suggested SCF-based methodology offers a convenient and coherent model on how to include stresses due to pressure into the engineering process. To do preliminary design and verification, the empirical formulation can provide a simple and conservative tool. To obtain more sophisticated evaluations, RfPV method is suggested to be used because it represents structural behaviour in a balanced way. The modified Decock formulation may be applied for advanced evaluations, provided that its assumptions are carefully validated.

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AUTOBIOGRAPHY

	**Walther Stikvoort** is an independent consultant specializing in the design, mechanical behaviour, and structural integrity of pressure vessels. He has extensive experience in pressure vessel engineering, particularly in the areas of nozzle–shell connections, local stress analysis, and compliance with design codes and standards. Throughout his career, he has held senior and executive positions at Kellogg Continental and NAM, and has provided consultancy services to NRG, GLT PLUS, and P3 Engineering. His work includes contributions to key design aspects such as the evaluation of piping reactions on pressure equipment and the interaction between connected piping systems and pressure vessels. His research focuses on the development of simplified analytical methods, including stress concentration factor (SCF)-based approaches, aimed at improving design accuracy and complementing established procedures such as WRC methods and related standards.