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STRESS CATEGORIZATION IN NOZZLE TO PRESSURE VESSEL CONNECTION FINITE ELEMENT MODELS

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ABSTRACT

The ASME Boiler and Pressure Vessel Code, Section III, is the most important code for nuclear pressure vessel design. Its design criteria were developed to preclude the various pressure vessel failure modes through the so-called "Design by Analysis" method. In the "Design by Analysis" approach, also used in Section VIII, Division 2 of the Code, the design limits were established in correspondence to each failure mode. Thus, failure modes such as plastic collapse, excessive plastic deformation and incremental plastic deformation under cyclic loading (ratchetting) may be avoided by having limits on primary and secondary stresses. In order to perform the stress categorization to check results from finite element models, mainly from 3D solid finite element models, against the ASME Code stress limits, several research works have been conducted. This paper is included in this effort. A typical Pressurized Water Reactor (PWR) nozzle to pressure vessel connection subjected to internal pressure and concentrated loads was modeled with 3D solid finite elements in linear elastic and limit load analyses. Using some stress categorization approaches, the results from linear elastic and limit load analyses were compared to each other and also with results obtained by formulae for simple shell geometries. Based on the result comparison, some conclusions and recommendations on the type of finite element analysis (linear elastic or limit load) and on the stress categorization were addressed for the studied cases.

INTRODUCTION

The ASME Boiler and Pressure Vessel Code, Section III [1] is the most important code for nuclear pressure vessels design. The initial version of this code, issued in the early '60s, was innovative, introducing the so-called "Design by Analysis" approach where the design criteria were developed to preclude various possible pressure vessel failure modes. In this approach, also used in Section VIII, Division 2 of the Code [2], the design limits were established in correspondence to each failure mode to be avoided.

At the time the Code was first issued, the main pressure vessel design tool was the shell discontinuity stress analysis. This is one of the reasons for the separation of the stresses in the membrane and bending stress distributions, used as basis for the comparison with the stress limits of the Code.

Another important feature of the Code is the stress classification in primary, secondary and peak stresses. Among the various possible pressure vessel failure modes, the plastic collapse, the excessive plastic deformation, and the incremental plastic collapse under cyclic loading are very important to define the pressure vessel thickness and reinforcement. The first two are related to the primary stresses, mainly, and the last to the primary and secondary stresses.

So, the stress categorization, i. e., the stress separation and classification, is one of most important aspects in the design by analysis procedures. From the end of the '60s to now, with the advent and growing use of the finite element method (FEM) in the pressure vessels stress analysis, many works have been done [3-7] to categorize stresses from finite element models, mainly, from solid finite element models.

Then, in this work, the analyses were performed in a typical PWR nozzle to pressure vessel connection subjected to internal pressure and concentrated loads, using 3D solid finite elements. Adopting some stress categorization approaches, the results from linear elastic FEA (finite element analyses) are compared with limit load FEA and also with the results obtained by formulae for simple shell geometries.

Based on the comparison of the allowable loads and failure mode found in each analysis, some conclusions and recommendations are addressed for the studied cases. Those conclusions and recommendations, not indicated in previous papers on the subject, are related to the applicability of linear elastic FEA with suitable stress categorization to the design of nozzle to vessel connections under many sets of nozzle concentrated loads.

THE STUDIED NOZZLE TO PRESSURE VESSEL CONNECTION

Figure 1 and Table 1 show the geometry of the studied nozzle to pressure vessel connection , from Hechmer & Hollinger [8]. It is important to mention that the finite element models in this paper are not the same used in [8].

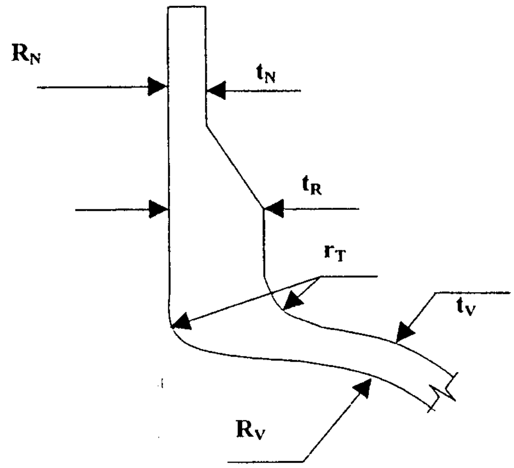


Figure 1: The studied nozzle to pressure vessel connection

Table 1: Dimensions (in mm) of the studied nozzle to pressure vessel connection

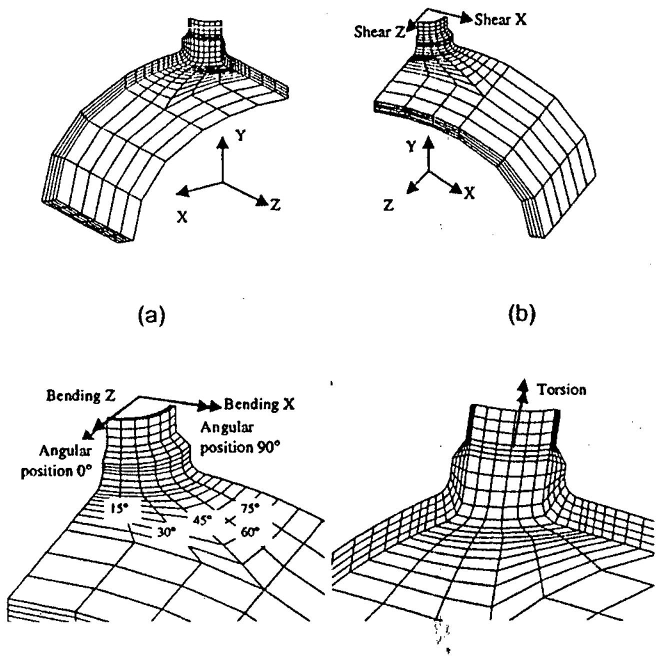
Vessel radius, R_V	*1016*
Vessel thickness, t_V	*98*
Nozzle radius, R_N	*130*
Nozzle thickness, t_N	*16*
Reinforcement thickness, t_R	*55*
Transition radii, r_T	*50*

The material is a pressure vessel ferritic steel with the following mechanical properties:

Young's modulus	E = 2.0091 E5 MPa
Poisson's ratio	$\nu = 0.3$
ASME Code Basic stress limit	$S_m = 174.7$ MPa
Yield stress	$S_y = 1.5\ S_m = 262$ MPa

THE FINITE ELEMENT MODELS

The finite element models were built using 3D solid finite elements with 20 nodes and 3 degrees-of-freedom per node of the element library of the commercial FEA program ANSYS [9]. The basic 3D model is ¼ of the complete 3D geometry. Symmetric and anti-symmetric boundary conditions were used according to the loads (internal pressure and single nozzle concentrated load). In Figure 2, some details of this basic model can be seen.

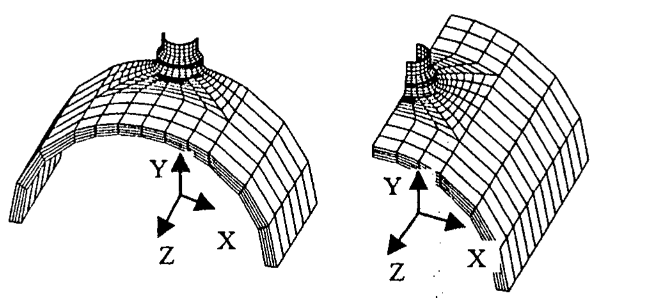


(a) (b)

(c) (d)

Figure 2: Details of the basic finite element model

For combinations of internal pressure plus nozzle concentrated loads, except torsion, the basic model was doubled according the combination symmetry. The models can be seen in Figure 3



(a) *XY plane symmetric combinations*

(b) *ZY plane symmetric combinations*

Figure 3: Finite element models for loading combinations

The nozzle length was established based on the distance $3\sqrt{R_N t_N} = 137$ mm. The end nozzle in the model is 200 mm (greater than 137 mm) far from the end of the primary nozzle thickness. Also, in the finite element models, the nozzle loads were adequately imposed as line loads, equivalent to the external concentrated loads, in the end nozzle. For the pressure evaluations, a line load was included, on the axial nozzle direction, to simulate the pressure acting on the piping. In the case of the shear loads, countering moments were also modeled to avoid spurious effects in the analyses from those shear loads moments.

ELASTIC FEA AND LIMIT LOAD FEA

In this work several analyses were performed for different loads and loading combinations as can be seen in Table 2

Table 2: Load and loading combinations

1	*Internal pressure*
2	*Nozzle concentrated load - Shear X*
3	*Nozzle concentrated load - Shear Z*
4	*Nozzle concentrated load - Bending X*
5	*Nozzle concentrated load – Bending Z*
6	*Nozzle concentrated load – Torsion*
7	*Internal pressure + Nozzle concentrated load – Shear X*
8	*Internal pressure + Nozzle concentrated load – Shear Z*
9	*Internal pressure + Nozzle concentrated load – Bending X*
10	*Internal pressure + Nozzle concentrated load – Bending Z*
11	*Internal pressure + Nozzle concentrated load – Torsion*

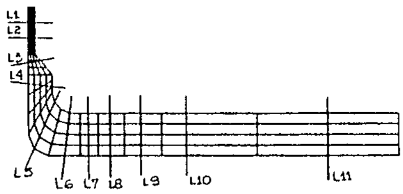
X and Z directions according to Figures 2 and 3

When there was just a single load applied, in elastic FEA, the analyses were performed with unit loads. The results were post processed to categorize the stresses. First, the linearization was done following the procedure called "linearization by a line" [5] using the lines in several angular positions as showed in Figure 4 for angular positions 0° and 90°.

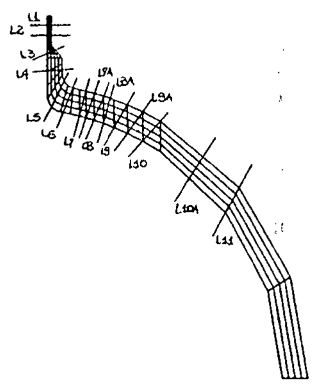
The linearization procedure used is that available in the ANSYS program [9] for 3D solid elements, based on the references Kroenke [3] and Kroenke et al. [4] and recommended in Hechmer & Hollinger [7].

The next step was the classification of stresses in the categories P_m (general primary membrane stress), P_L (local primary membrane stress), P_b (primary bending stress) and Q (secondary stress), following the ASME Code, considering the type of load and the location under evaluation. After that, some lines were eliminated from the analysis using the criteria for line validation showed in Hechmer & Hollinger [7]. Thus, the critical lines may be those located in the nozzle shell and in the vessel shell (like L1 and L10 in Figure 4). According to this, it may not be necessary a mesh refinement in the nozzle-to-vessel transition. Finally, one maximum load was defined in each location by a comparison of the obtained stress with the ASME Code limit. The limiting load was achieved as the minimum load among the values found. Also, the failure mode associated with it was found by evaluating what stress category indicated the limiting load and where was its location in the structure.

For the loading combinations of internal pressure plus nozzle concentrated loads, the internal pressure was fixed as the design pressure of 12.3 MPa and the procedure above indicated was the same, i. e., the analysis were performed with internal pressure of 12.3 MPa plus unit loads and, at the end, the allowable nozzle concentrated load was found.



(a) Angular position 0°



(b) Angular position 90°

Figure 4: Chosen lines for stress linearization

Limit load FEA was performed using the perfect plasticity material model in ANSYS program [9]. For a single load applied, it was monotonically increased until the non-convergence was achieved in the finite element solution. The asymptotic behavior of a typical load-displacement curve was also checked. Thus, the maximum load was found and its associated failure mode. Using the ASME Code limits, the allowable load was defined as 2/3 of the maximum load. For the loading combinations of internal pressure plus nozzle concentrated loads, the internal pressure was fixed as the design pressure of 12.3 MPa and the procedure was the same.

FORMULAE

Internal Pressure

If the collapse occurs on the cylindrical shell, the collapse pressure p_c is:

$$p_c = \frac{t_v Sy}{R + 0.5\, t_v} \qquad (1)$$

where R and t_v are the internal radius and the thickness of the vessel.

Nozzle Concentrated Loads

The Tresca criterion was assumed ($\tau_{max} = 0.5\ S_y$) and that the collapse would occur in the pipe, i. e., where the loads were applied. The ASME Code factor of 2/3 was also used to obtain the allowable load.

Shear Force

The maximum shear stress in pipe section under a shear load C is

$$\tau_{max} = \frac{C}{A_{cis}} \qquad (2)$$

where $A_{cis} = 0.5A$ and A is the pipe cross section.

Bending Moment

The maximum stress in a pipe section under a bending moment M is

$$\sigma = \frac{MD}{2I} \qquad (3)$$

where D is the external diameter and I is the moment of inertia of the pipe section.

Torsion Moment

Considering a torsion moment T and the maximum shear stress limit of 0.6 S_y (ASME [1]) for this case the maximum torsion moment is

$$T_{max} = 0.6\ Sy\ w_t \qquad (4)$$

where w_t is the polar modulus of the pipe section.

Internal Pressure plus Nozzle Concentrated Loads

In all cases of load combination the Tresca criterion was assumed ($\tau_{max} = 0.5\ S_y$) and that the collapse would occur in the pipe, i. e., where the concentrated loads were applied. The ASME Code factor of 2/3 was also used to obtain the allowable nozzle concentrated load.

Internal Pressure plus Shear Force

With the expression (5) and knowing the internal pressure p, the maximum value of the shear load C may be found imposing $\tau_{max} = 0.5\ S_y$.

$$\tau_{max} = \frac{1}{2}\sqrt{\left(\frac{pD}{4t}\right)^2 + 4\left(\frac{C}{0.5A}\right)^2} \qquad (5)$$

Internal Pressure plus Bending Moment

Using the expression (6) and knowing the internal pressure p, the maximum value of the bending moment M may be found imposing $\tau_{max} = f\ 0.5\ S_y$, where f is the shape factor of the pipe section assuming the collapse would occur due to bending

$$\sigma = 0.5\frac{pD}{2t} + \frac{MD}{2I} \qquad (6)$$

Internal Pressure plus Torsion Moment

Using the expression (7) and knowing the internal pressure p, the maximum value of the torsion moment may be found imposing $\tau_{max} = 0.5\ S_y$.

$$\tau_{max} = \sqrt{\left(\frac{pD}{8t}\right)^2 + \left(\frac{T}{w_t}\right)^2} \qquad (7)$$

RESULTS

Internal Pressure

The obtained results for internal pressure are shown in Table 3.

Table 3: Allowable Pressures

Analysis	*Allowable Pressure (MPa)*
Limit load FEA	*15.83*
Elastic FEA	*15.53*
By Formulae	*16.07*

In all cases the failure mode found was the plastic collapse in the vessel shell.

Nozzle Concentrated Loads

The obtained results for nozzle concentrated loads are shown in Table 4.

Table 4: Allowable Nozzle Concentrated Loads

Analysis	*Shear X (N)*	*Shear Z (N)*	*Bending X (N mm)*	*Bending Z (N mm)*	*Torsion (N mm)*
Limit load FEA	*6.09 E5*	*6.22 E5*	*1.64 E8*	*1.64 E8*	*1.90 E8*
Elastic FEA	*5.39 E5*	*5.39 E5*	*1.43 E8*	*1.43 E8*	*1.64 E8*
By formulae	*6.06 E5*	*6.06 E5*	*1.59 E8*	*1.58 E8*	*1.90 E8*

X and Z directions according to Figures 2 and 3

In all cases the failure mode found was the plastic collapse in the nozzle.

Internal Pressure plus Nozzle Concentrated Loads

Applying a design internal pressure of 12.3 MPa, the results obtained for nozzle concentrated loads are shown in Table 5.

Table 5: Allowable Nozzle Concentrated Loads with an Internal Pressure of 12.3 MPa

Analysis	*Shear X (N)*	*Shear Z (N)*	*Bending X (N mm)*	*Bending Z (N mm)*	*Torsion (N mm)*
Limit load FEA	*4.83 E5*	*4.83 E5*	*1.52 E8*	*1.52 E8*	*(*)*
Elastic FEA	*4.79 E5*	*4.66 E5*	*1.27 E8*	*1.23 E8*	*1.42 E8*
By formulae	*5.74 E5*	*5.74 E5*	*1.62 E8*	*1.62 E8*	*1.50 E8*

(*) not processed due to computational limitations
X and Z directions according to Figures 2 and 3

In all cases the failure mode found was the plastic collapse in the nozzle.

CONCLUSIONS AND COMMENTS

This paper deals with the areas indicated in Pressure Vessel Research Council project *3D Stress Criteria* [7], mainly those related to failure mechanisms, stress categories and locations for evaluation.

The studied geometry, modeled with 3D solid finite elements, is of great interest due to the difficulty to apply in a straight way the procedures of stress categorization of the ASME Code. The question to be answered is using *design criteria based on shell stresses, how to check stresses in parts of a vessel that are not shells?*

In this case (radial nozzle-cylinder connection), from the results comparison in terms of the allowable loads it can be said that:

1. For internal pressure (see Table 3), if the nozzle, the vessel shell, and the reinforcement are adequately sized for internal pressure according the minimum ASME Code requirements, the failure mode is the plastic collapse in the vessel shell. This confirms the conclusions of many previous works such as Pastor & Hechmer [10], where the general primary membrane stress P_m defines the failure mode on the vessel shell far from the connection to the nozzle. It is not necessary to use finite element models to find the allowable pressure once elastic FEA and limit load FEA give the same result from formulae.
2. For nozzle concentrated loads (see Table 4), applied only one each time, and again, if the nozzle, the vessel shell, and the reinforcement are adequately sized for internal pressure according the minimum ASME Code requirements, the failure mode is the plastic collapse in the pipe. There is a good agreement between the allowable loads obtained from limit load FEA, elastic FEA and by formulae, controlled by P_m on the pipe. The elastic FEA allowable loads are 10-15% smaller than those from limit load FEA and by formulae are.
3. For loading combinations of internal pressure plus nozzle concentrated loads (see Table 5), the latter applied only one each time, and again, if the nozzle, the vessel shell, and the reinforcement are adequately sized for internal pressure according the minimum ASME Code requirements, the failure mode is the plastic collapse in the pipe. The agreement also is good as in the previous case. The elastic FEA allowable loads are up to 20 % smaller and the results by formulae are up to 20 % greater than those from the limit load FEA.
4. In all cases, the influence of local plus bending primary stresses and of the primary plus secondary stresses in the geometry transitions does not indicate that the failure modes of excessive plastic deformation or incremental plastic deformation must occur.
5. From the limit load FEA or from the elastic FEA, the plastic collapse and the excessive plastic deformation were not found in the nozzle piping transition, imposing the special stress limits of the ASME Code related to this location.
6. It is very hard to perform a limit load FEA for a load combination of the internal pressure plus all nozzle concentrated loads to compare with elastic FEA or formulae. But, if the nozzle, the vessel shell, and the reinforcement are adequately sized for internal pressure according to the minimum ASME Code requirements, it can be expected that similar agreement in terms of allowable loads may be achieved.

So, the use of elastic FEA for nozzle to vessel connections, following the recommendations and procedures described and referenced in this work, may be acceptable in the design of pressure vessels, where there are many sets of nozzle concentrated loads in the design, service and test conditions. The use of formulae is also useful with simple geometries.

It is important to mention that some conclusions addressed may be considered conservative because the effects of material hardening were not included in the analyses.

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