Stress Evaluation of a Typical Vessel Nozzle Using PVRC 3D Stress Criteria: Guidelines for Application

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ABSTRACT

The stress linearization methodology recommended in the PVRC 3D Stress Criteria: Guidelines for Application (Hechmer and Hollinger, 1997) is used to evaluate the stresses in a three-dimensional *brick element* model of a typical refining or chemical plant thin walled nozzle. The results of the evaluation are compared with a 1996 analysis of the same nozzle using *plate elements*. The applicability of the Guidelines to routine nozzle analysis is discussed and a comparison is made to a previous evaluation proposal (Porter and Martens, 1996).

INTRODUCTION

The use of finite element (FE) analysis software for investigation of stresses in vessel nozzle-to-shell junctions is now economically practical for many design projects in the refining and chemical industries. The engineer's decision to use brick elements or shell elements for the investigation may have a bearing on the results. The use of brick elements will provide results that must be linearized by the engineer before comparison to the applicable ASME Boiler and Pressure Vessel Code (1995) allowable stress. A stress linearization methodology recommended for the evaluation of brick element FE results is provided by the PVRC in its recent publication by Hechmer and Hollinger (1997).

For the brick element investigation the authors decided to investigate a nozzle-to-shell junction problem that they had previously investigated using shell elements. (Porter and Martens, 1996). The authors' goal was to determine what variation in FE-reported stresses would result when comparing linearized brick element results to shell element results.

The engineering effort required to create, analyze and develop linearized stresses is considerable unless the FE code employed has the linearization routine built-in. The authors wanted to determine what effect the incorporation of stress linearization would have on the engineering effort involved in a finite element analysis investigation. The authors hoped to determine if this additional effort was justified for thin wall vessel applications, an important factor in the practicality of everyday engineering work.

NOMENCLATURE

Sm	= Design Stress Intensity Value (ASME, 1995)
Sy	= Yield Strength Value (ASME, 1995)
r	= Radius of nozzle
R	= Radius of vessel
t	= Wall thickness of nozzle
T	= Wall thickness of vessel

FE MODEL

The model used for evaluation in this paper consists of a 96" diameter, ½" thick vessel with a 24" diameter, ½" thick nozzle attached perpendicular to the centerline of the vessel. The vessel was reinforced at the nozzle intersection with a 42" diameter, ½" thick pad. The nozzle reinforcement design is per ASME Section VIII, Division 2 (ASME, 1995). The material is SA516-70. For this material at a temperature of 500 °F, the ASME Section VIII, Division 2 allowable properties are Sm= 20.5 Ksi and Sy= 30.7 Ksi. The basic geometry of the model is illustrated in Figure 1. This is the same nozzle that was examined using shell elements in the previously cited paper (Porter and Martens, 1996).

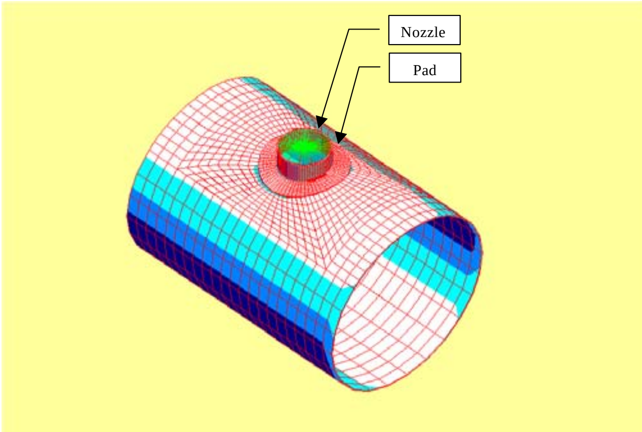

Figure 1 - Basic Model Geometry

Even though the vessel and nozzle may both be considered "thin" (r/t = 24, R/T = 48 in the pad area and R/T = 96 in the vessel in general), 3-D brick elements were used to model the vessel and nozzle. The use of brick elements to model this intersection increased the required effort to a level not generally justified in a typical refining or chemical plant design project. In order to provide a direct comparison with the previous analysis, the fillet weld typically used in such nozzles was not modeled. Although the radius of this weld is a factor in the peak stress levels, this geometry is not accounted for when using plate elements and, for purpose of comparison was not included with the brick elements. It should be noted that the nozzle example in appendix IV of the PVRC document (Hechmer and Hollinger, 1997) also omits the modeling of the weld radius. Including the weld radius in the model would likely have resulted in lower indicated stresses.

The final mesh employed in the model consisted of 64 elements around the periphery of the nozzle. The elements in the other dimensions were sized to maintain an aspect ratio of 2:1 or less. The mesh through the thickness was a minimum of three elements in the intersection area. The mesh density was increased in the region of highest stress so that the brick elements in this region had aspect ratios of nearly 1:1:1. This final mesh sized was based on a series of model runs with decreasing mesh size in the high stress region. The last doubling of the mesh density resulted in a stress increase of less than 5%. This was considered an acceptable level of convergence for the analysis. In other industries, the nuclear industry for example, tighter convergence criteria would likely be appropriate.

The model was loaded as follows:

Internal Pressure:	165 PSI
Force Y:	-6,480 LB
Moment X:	33,160 FT-LB
Moment Y:	38,250 FT-LB
Moment Z:	25,500 FT-LB

Note that the actual forces applied to the nozzle included a 74,644-LB load in the Y direction due to the pressure thrust on the nozzle. This thrust load must be added to the -6,480 LB load that was reported by the piping analysis. Note that the thrust load is not reported by piping analysis programs and is an often-overlooked load in the transfer of loads from the piping engineer to the vessel engineer. The use of a free body diagram to check the balance of the boundary loads will help prevent neglecting this load. All force, moment and thrust loads were applied to the model using a "spider web" of beam elements as described Martens *et al* (1996).

COMPUTED STRESSES

Figure 2 illustrates the stress intensity contours in the nozzle and shell. As would be expected, the highest stresses are indicated to be in a small region near the intersection of the nozzle and shell. The maximum indicated stress intensity is approximately 77,000 psi. Since this is a linear elastic analysis and the material yield is only 30,700 psi, the reported stress intensity is not an accurate number. A redistribution of the stresses will occur until equilibrium is achieved with the actual maximum stress being between the yield and ultimate strength of the material. The use of a nonlinear analysis is called for to evaluate the actual stresses. Nonetheless, the fictitious indicated

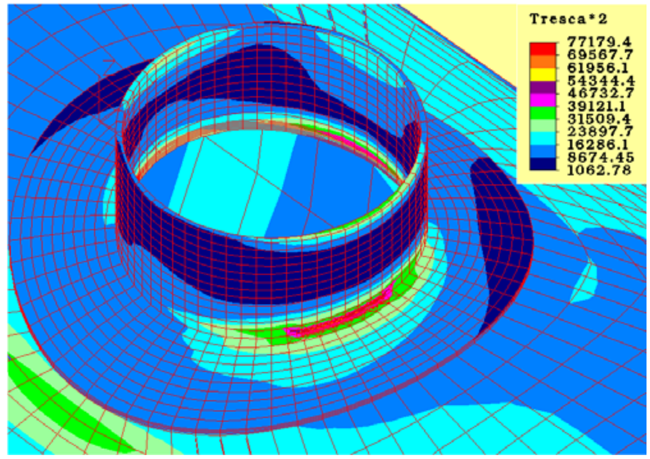

Figure 2 - Stress Intensity Contours

stresses from a linear elastic analysis are used in the evaluation of code compliance.

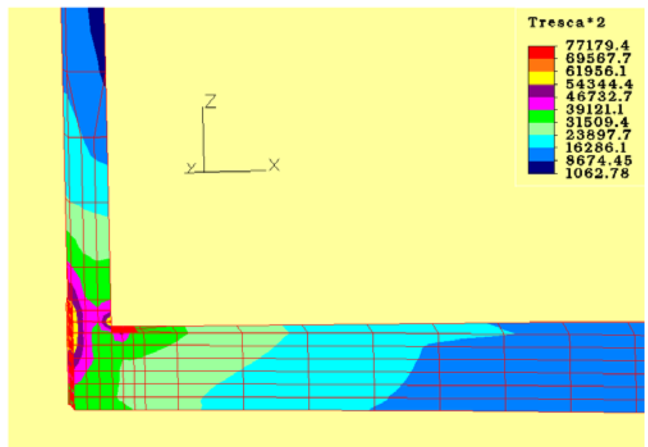

Figure 3 - Slice Through High Stress Region

Figure 3 illustrates a slice through the nozzle in the region of highest stress. It may be seen that the highest indicated stresses are on the inner and outer faces of the nozzle in the region where the nozzle intersects with the pad/shell. From the stress contours alone, the stress in the nozzle appears to range from 39,000-46,000 psi. In the pad/shell region, the stress appears to range from 32,000-39,000 psi. As would be expected, the stress intensity varies through the thickness of both the nozzle and pad/shell.

STRESS LINEARIZATION

In order to derive stress intensity levels for the nozzle and pad/shell that can be compared to the ASME criteria, the method outlined in the recent PVRC document (Hechmer and Hollinger, 1997) was employed. This method requires a five-step process:

First, one must select Stress Classification Lines (SCL) in the region of interest. These lines are ideally normal to both the inside and outside surfaces. Other criteria for the evaluation of SCLs are discussed in the document by Hechmer and Hollinger (1997).

Once the SCLs are selected, the component stresses (Sxx, Syy, Szz, Sxy, Syz and Szx) must be obtained at nodal points along the SCL. Additionally, where possible, the radial (Rs), axial (As) and hoop (Hs) components of the stress are also compiled.

Using the process outlined in the PVRC document, these component stresses are linearized to obtain the membrane and bending components along the SCLs.[1]

The linearized component stresses are then combined to yield a composite set of linearized component stresses representing the SCL. Several options for combining the component stresses are presented in the PVRC document. In this paper, membrane plus bending were used for the hoop and axial components along with the membrane alone for the remaining components.

Finally, the linearized stress intensity (as well as the principal stresses and von Mises stress) is computed from these component stresses.

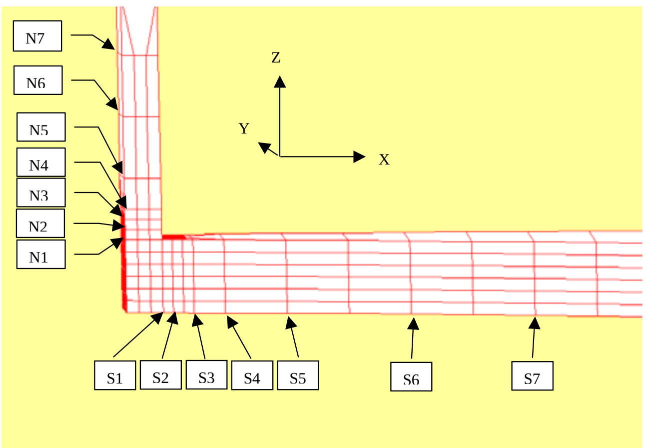

Figure 4 - Stress Classification Lines (SCL's)

Hechmer and Hollinger (1997) recommend that the SCLs selected be adjacent to the point of intersection for a model such as the 90-degree nozzle-to-shell intersection used in this paper. Lines N1 and S1 in Figure 4 represent these SCLs. In order to compare the 3-D brick results with the previous shell model, six additional SCLs on both the nozzle (N2-N7) and shell (S2-S7) were employed.

LINEARIZED RESULTS

Figure 5 illustrates the stress intensity plotted at each of the SCLs as a function of a normalized distance parameter. This normalized distance parameter has been used to facilitate comparison with the previous analysis by Porter and Martens (1996). In addition, the stress intensity computed using shell elements with one of the codes examined in the previous paper is indicated.

[1] The details of this linearization process were presented by Broyles (1997). Bibel and Kovach (1990) presented a somewhat less complicated procedure for the linearization of the stresses. For this model, the two processes yielded reasonably close, but not identical, membrane and bending stresses. The primary differences were in the bending stress components.

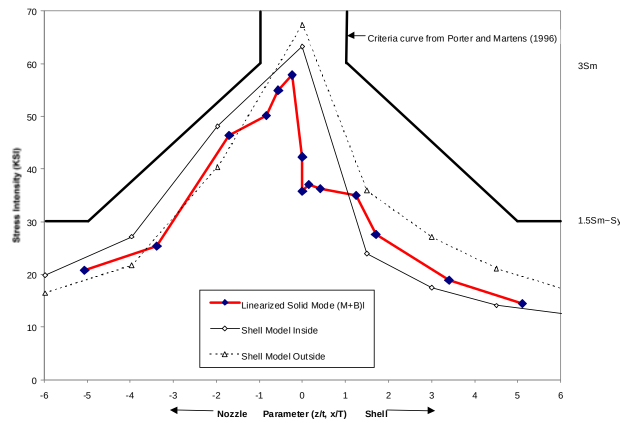

Figure 5 - Linearized Solid Model Stresses

The overall agreement between the shell solution and the 3-D brick solution seems to be qualitatively good. Quantitatively, there seems to be a significant degree of disagreement, especially in the nozzle. Since the shell element used in this example did not include shear deflection, it is possible that this could be the reason for the higher indicated stresses on the nozzle side. The pressure-only linearized stress intensity at the intersection as determined with the 3-D elements (27,064 psi. on the shell and 30,925 psi. on the nozzle) is somewhat lower than that computed with the shell elements.

The indicated stresses in the 3-D brick model on the nozzle side exhibit a pattern that was noted in Appendix IV of the PVRC document: the stress just away from the recommended SCL is higher than the stress at the recommended SCL (see Figure 5). In this case, the rise is rather dramatic, going from approximately 40,000 psi on the recommended SCL to approximately 58,000 psi at the next line. From inspection of Figure 5, the indicated linearized stress intensity does not fall back to the level of the recommended SCL until nearly 2 nozzle thicknesses away from the intersection. Unless this increased stress level can be shown to be an artifact of the FE process, it would seem that the recommended SCL is not always conservative. The nearly 50% difference in the linearized stresses at the various SCLs would seem to be of concern.

Porter and Martens (1996) proposed a criteria curve for evaluating the stress intensity and the ASME Code (1995). This criteria curve (indicated on Figure 5) established limits based on stress intensity as a function of the distance parameter away from the intersection. Based on the linearized stress intensities as computed from the 3-D brick model, the general shape of the proposed criteria curve would seem to be appropriate.

INTERPRETATION OF RESULTS

In order to interpret the linearized results, it is necessary to review Appendix 4 of ASME Section VIII, Div 2 (1995). Porter and Martens (1996) summarized Appendix 4 of ASME Section VIII, Div 2 as follows:

Using the design load, the general membrane stress (P_M), which excludes "discontinuities and concentrations," should be limited to kSm. If the factor k is unity, then the general membrane stress is limited to the allowable for the material and temperature involved.

Using the design load, the primary stress (P_M + P_L + P_B), which is due solely to mechanical loads and excludes stress concentrations, is limited to 1.5 kSm. Again, if k is unity, then the limit is 1.5 times the allowable. Note that the local membrane portion of the primary stress (P_L) "considers discontinuities" while the bending portion (P_B) excludes discontinuities.

Using the operating load, the combination of primary plus secondary membrane plus Bending (P_L + P_B + Q) is limited to 3Sm. Although this stress may be caused by mechanical or any other load, "local stress concentrations" are to be excluded.

Finally, the "Peak" (F) stress (that due to a stress concentration or "Certain thermal stresses which may cause fatigue but not distortion of vessel shape" due to operating loads) is limited by Sa, which is obtained from the fatigue curves, Figs. 5-110.1, 5-110.2 and 5-110.3.

The PVRC document (Hechmer and Hollinger, 1997) states directly that P_B is not to be evaluated in the vicinity of the shell/nozzle intersection. Based on the shape of the stress vs. distance from the intersection curve in Figure 5, any distance within 5 times the thickness of the section would seem to be "in the vicinity" of the intersection. Thus, P_B and the corresponding 1.5Sm limit do not apply for this analysis.

The summary from Porter and Martens (1996) states that P_B does not apply at the intersection, while P_L does. The PVRC document states flatly that "All membrane stresses from pressure and pipe loads at a nozzle-shell juncture are treated as primary (P_L); bending stresses are treated as secondary (Q)." Thus, in order to assess this nozzle in accordance with the Code, it would seem that we must separate the membrane from the bending.

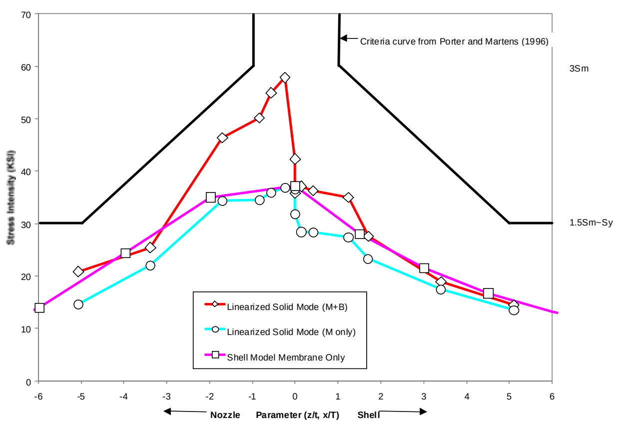

Figure 6 - Membrane and Bending Stress Intensity

Figure 6 illustrates the Stress Intensity computed using both membrane and bending (the upper curve) and membrane only (the lower curve). From Figure 6, we would conclude that the 1.5 Sm criteria for P_L (membrane only) is not met on the nozzle SCLs within about two thicknesses of the junction and on the shell SCL at the junction (the recommended SCL). The 3Sm criteria for P_L + Q is met at all SCLs. Since the P_L criteria were not met, however, this nozzle would not meet the code requirements.

DISCUSSION OF RESULTS

Assuming that the FE nodal grid is constructed so as to provide nodal stresses on the SCLs, stress linearization is not an overly arduous process. Some of the commercial codes have the linearization process built into the post processor. However, based on the differences observed between the two published linearization schemes, caution is advised. In an earlier paper, Porter and Martens (1997) pointed out that differing element formulations, even within the same FE code, could lead to differing results. Little, if any, guidance is available to the engineer as to which formulation to use. This same type of problem seems to possible with linearization methodologies.

In the case of a thin walled vessel and nozzle, as examined in this analysis, it is not at all clear that the use of the computationally more expensive elements results in better data. On this basis, there is no clear justification for the use of 3-D brick elements for such thin walled vessels.

The nozzle investigated was designed to meet the ASME (1995) design and reinforcing criteria. When analyzed using WRC 107 (Wichman *et al*, 1977), this nozzle does meet the code requirements, as demonstrated in the previous paper by Porter and Martens (1996). In addition, this nozzle and the associated loading are very typical of real nozzles that have been in successful operation for many years. That the results of this analysis, based on the PVRC document procedure, indicate that this nozzle does not meet the code requirements could indicate that the use of solid elements and stress linearization procedures might not be applicable for thin walled vessels. It is worth noting that when there are no pipe loads on the nozzle, the linearized stress intensities on the nozzle and shell are approximately 31,000 and 27,000 psi respectively. Thus, with pressure only, this nozzle apparently does not meet the code requirements. If the weld radius had been included in the model, the indicated stresses would have likely been lower. Based upon experience, however, it is the authors' opinion that including the weld radius would not have changed the results of this analysis so far as compliance with the code using the linearization procedure is concerned.

If we compare the membrane-only stress intensity computed with the shell elements and by the linearized 3-D brick elements (Figure 6), we see good agreement. Thus, it seems likely that the membrane stress in the vicinity of the junction is on the order of 35,000 psi. This being the case, it seems inappropriate to apply the $P_L < 1.5$Sm criteria "in the vicinity of the junction." Based on the data in this analysis, within 5 thicknesses of the junction would seem to be "in the vicinity." Therefore, we would conclude that P_L could exceed 1.5Sm within 5 thicknesses of the junction and still meet the code requirements. In reality, within 5 thicknesses of the junction the correct criteria of 3Sm would appear to be more appropriate.

CONCLUSIONS

- The SCL location recommended in the PVRC document (Hechmer and Hollinger, 1997) may not be conservative. In this particular investigation, SCLs nearby the shell to nozzle junction indicated considerably higher stresses. However, this maybe an artifact the exclusion of the weld radius, in an attempt to follow the model example in the PVRC guideline document.

- Based on a comparison of the brick element model results in this analysis and the plate element results previously reported by Porter and Martens (1996), no clear justification for the use of brick elements in the evaluation of thin walled vessels in refining or chemical plant applications is apparent. The use of the ASME Section VIII design criteria for establishing the nozzle design and the use of a plate element model to further investigate the nozzle piping imposed loading appears to be a suitable approach for most of these applications.
- The use of plate element finite element analysis models can provide the design engineer with the ability to develop spring rates for a nozzle suitable for utilization with piping design software, thereby improving the accuracy of the nozzle piping loadings. These piping loads can be investigated using the same finite element spring rate model, as an improvement over a WRC 107 analysis. Where the stresses are found to be excessive, it may be more practical to alter the nozzle or piping design to reduce the stresses rather than proceed with a 3-D brick element or non-linear finite element analysis.
- Since the PVRC document only addresses the evaluation of shell element models in passing, it seems that we are still lacking a clear guideline for the evaluation of finite element analyses when plate elements are employed.

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