PVP2015-45408: A Comparison of Design by Analysis Techniques for Evaluating Nozzle-to-Shell Junctions per ASME Section VIII Division 2

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INTRODUCTION

- Part 5 of ASME Section VIII Division 2 offers several design by analysis (DBA) techniques for evaluating pressure retaining equipment for Code compliance using detailed computational stress analysis results.
- These procedures can be used to check components for protection against multiple failure modes including plastic collapse, local failure, buckling, and cyclic loading.
- In particular, this study investigates the use of these methods for evaluating nozzle-to-shell junctions subjected to internal pressure and nozzle end loads.
  - Specifically, elastic stress analysis, limit load analysis, and elastic-plastic stress analysis are utilized to check for protection against plastic collapse, and computational results for a given load case are compared.
- Additionally, the twice elastic slope method for evaluating protection against plastic collapse is utilized as an alternate failure criterion to supplement elastic-plastic analysis results.
- The goal of these comparisons is to highlight the difference between elastic stress checks and the non-linear analysis methodologies outlined in ASME Section VIII Division 2.
- Finally, commentary on the applicability of performing the Code-mandated check for protection against ratcheting for vessels that do not operate in cyclic service is provided.
SECTION VIII DIVISION 2 DBA TECHNIQUES

- **Elastic Stress Analysis:**
  - Linear superposition can be employed when using elastic stress analysis because the method does not account for geometric or material non-linearity.
  - Once stresses are linearized, they are characterized as either primary or secondary and delineated into equivalent stresses that are membrane (average through a section), membrane plus bending, and peak stress.
  - Classified elastic stresses are compared to specified limits ($S_{PL}$, $S_{PS}$, etc.).

- **Limit Load Analysis:**
  - Utilizes elastic-perfectly plastic material models, strain-displacement relations that are consistent with small displacement theory (nonlinear geometric effects are not considered).
  - Displacements and strains achieved prior to non-convergence (achieving static equilibrium) have no physical meaning, and a pseudo-yield value equal to 1.5 times the allowable stress at temperature is employed in the FEA.
  - The maximum allowable load is established by taking two-thirds of the maximum load achieved directly prior to non-convergence.

- **Elastic-Plastic Analysis:**
  - Incorporates elastic-plastic material model (including strain hardening effects) into numerical analysis and accounts for geometric nonlinearity (large displacement theory).
  - The allowable load is established by applying a design factor to the calculated collapse load, or the load that satisfies equilibrium prior to non-convergence. The current design factor in Section VIII Division 2 is 2.4 (for Section VIII Division 1 it is 3.5).
TWICE ELASTIC SLOPE METHOD

- The twice elastic slope method was incorporated into the 1974 Edition of ASME Boiler and Pressure Vessel Code Section III, Division 1 and the 1977 Edition of ASME Section VIII Division 2.

- This method establishes a maximum permitted plastic pressure at the intercept of a line drawn from the origin of a pressure-deformation curve at a slope of twice the value of the slope of the elastic portion of the curve.

- This methodology is outlined in a Figure on the next slide as shown in the 2004 Edition of Section VIII Division 2. In this figure, the angle $\phi$ is twice the angle $\theta$ (corresponding to the elastic portion of the curve).

- The collapse load is then defined where the collapse limit line intersects the pressure-deformation curve. The maximum permitted load is taken as two-thirds of this collapse load (similar to the calculation of maximum load in a limit load assessment).

- This method utilizes a consistent design margin (two-thirds) based on the calculated collapse load; this removes the dependence of tensile margin (based on the Code of construction) on maximum permitted loads.

- Calculated collapse load is dependent on the load-displacement (or strain) extraction location on the FEA model. Care must be taken to ensure the location that is plastically collapsing is chosen.
TWICE ELASTIC SLOPE METHOD

![Graph showing the twice elastic slope method with labels for regression line, collapse limit line, collapse load point, test collapse load, and maximum principal strain or displacement load.]

- Regression line
- Collapse limit line
- Collapse load point
- Test collapse load
- Maximum principal strain or displacement load

- \( \phi \)
- \( \theta \)
COMMENTARY ON RATCHETING

• Ratcheting is defined as progressive or incremental accumulation of plasticity in a component; component must cycle to ratchet.

• The following limit must be satisfied:

\[ \Delta S_{n,k} = P_L + P_B + Q \leq S_{PS}, \]

• From 5.5.1.6 of Section VIII Division 2:
  
  "Protection against ratcheting shall be considered for all operating loads listed in the User’s Design Specification and shall be performed even if the fatigue screening criteria are satisfied."

• This implies that the \( S_{PS} \) limit on \( P+Q \) stress must be met even if the vessel does not operate in cyclic service.

• The \( P+Q \) check is sensible for vessels operating in a cyclic environment, but can be limiting from a design standpoint for non-cyclic equipment.

• The \( P+Q \) limit is a distortion check that is intended to provide protection against progressive plastic deformation (ratcheting).

• If the vessel in question does not cycle in service (or if the fatigue screening criteria provided in the Code is satisfied), damage from cyclic loading is not necessarily a viable failure mechanism. Thus, the applicability of this check is open to some interpretation.
EXAMPLE PROBLEM

• Design by Analysis is employed to validate the design of a vessel subjected to internal pressure and prescribed nozzle loads.
• This vessel is not designed for cyclic service (commercial nuclear vessel).
• This example considers a NPS 4-inch nozzle intersecting a 2:1 elliptical head with a reinforcing pad.
• The following loads are independently assessed:
  – Internal pressure
  – Nozzle bending moment
  – Nozzle axial force
• The following methods of analysis are performed and compared:
  – Elastic stress analysis (Part 5.2.2)
  – Limit load analysis (Part 5.2.3)
  – Elastic-plastic analysis (Part 5.2.4)
• The applicability of a P+Q check at the nozzle-to-head junction is discussed for the elastic stress analysis.
• The twice-elastic slope method is investigated for the elastic-plastic analysis to provide a potential bound on overall deflection.
  – Further discussion on the twice elastic slope method is available in WRC Bulletins 163, 219, 230, 254, 255, 364, and 414.
finite element model

- 3D finite element model constructed using solid, quadratic elements.
- 4-inch nozzle:
  - NPS 4 schedule 40
  - 304 stainless steel
  - SA-516-70 re-pad
- 1-inch inlet nozzle intersects 4-inch nozzle with re-pad.
- 2:1 elliptical head:
  - 20-inch OD
  - SA-516-70
- Nozzle loads applied via a kinematic coupling at the end of the nozzle.
- Coupling translates loads but permits radial dilation.
ALLOWABLE STRESSES

- Nozzle material specification is SA-312 TP304L seamless pipe.
- ASME section VIII Division 1 (2013) allowable stress:
  - At ambient and design temperature, allowable stress is 16.7 ksi (based on 90% yield).
- The following stress checks are performed as specified in the Code:
  - \( P_L < 1.5S_{PL} \)
  - \( P_L + P_B + Q < S_{PS} \)
- Where \( S \) is the allowable stress at design temperature (16.7 ksi) and \( S_{PS} \) is three times \( S \) at the average temperature during operational cycle.
  - \( 1.5S = 25.0 \) ksi
  - \( S_{PS} = 50.1 \) ksi
- Linearized stress through the nozzle neck are the most limiting due to nozzle neck thickness compared to head + re-pad thickness.
ELASTIC ANALYSIS: INTERNAL PRESSURE

- Contour of elastic von Mises stresses at the nozzle junction are shown for the maximum permitted 642 psi internal pressure.
- This is limited by local primary membrane stress ($P_L$) through the thickness of the nozzle neck at the junction with the reinforcing pad.
LIMIT LOAD ANALYSIS: INTERNAL PRESSURE

- Based on limit load analysis, the internal pressure achieved prior to non-convergence is 1,000 psi.
- The location that fails to achieve equilibrium first is the nozzle-to-head junction as shown.
- This pressure achieved prior to non-convergence/1.5 gives an allowable pressure of 667 psi.

Displacement Prior to Non-Convergence

![Displacement Prior to Non-Convergence](image)

Load vs. Displacement

![Load vs. Displacement](image)
Based on elastic-plastic analysis, the internal pressure achieved prior to non-convergence is 2,200 psi (dividing by 2.4 = 917 psi).

Limiting displacements using the twice elastic slope method gives a maximum internal pressure of 1,475 psi.

Twice elastic slope margin on collapse pressure = 1.5

This collapse pressure/1.5 gives an allowable pressure of 983 psi.
Elastically scaling nozzle axial force until $P + Q$ limit ($S_{ps}$) is achieved gives an allowable axial force of 14,700 lbf.

- Contour of longitudinal stress in nozzle is shown below for axial force.
- Contour scale capped at $P/A$ hand calculation value in tension and compression.
- Stresses removed from discontinuities match hand calculation.
LIMIT LOAD ANALYSIS: AXIAL FORCE

- Based on limit load analysis, the axial force achieved prior to non-convergence is 38,100 lbf.
- The load achieved prior to non-convergence/1.5 gives an allowable axial force of 25,400 lbf.
- Yielding occurs in entire nozzle cross-section prior to non-convergence.
ELASTIC-PLASTIC ANALYSIS: AXIAL FORCE

- Based on elastic-plastic analysis, the axial force achieved prior to non-convergence is 129,800 lbf (divided by 2.4 = 54,000 psi).
- Limiting displacements using the twice elastic slope method gives a maximum axial force of 68,000 lbf.
- Twice elastic slope margin on collapse pressure = 1.5
- The collapse load/1.5 gives an allowable axial force of 45,300 lbf.

Plastic Strain Prior to Non-Convergence

Displacement vs. Load

Force = 68,000 lbf

Twice Elastic Slope
ELASTIC ANALYSIS: BENDING MOMENT

- Contour of longitudinal stress in nozzle is shown below for bending moment only.

- Elastically scaling nozzle bending moment load until P + Q limit \( S_{PS} \) is achieved gives a bending moment of 21,300 in-lbf.
- Contour scale capped at Mc/I hand calculation value in tension and compression.
- Stresses removed from discontinuities match hand calculation.
- Worst case circumferential location is considered (stresses attenuate moving around the nozzle circumference).
LIMIT LOAD ANALYSIS: BENDING MOMENT

- Based on limit load analysis, the bending moment load achieved prior to non-convergence is 63,000 in-lbf.
- The load achieved prior to non-convergence/1.5 gives an allowable bending moment of 42,000 in-lbf.
- Plasticity in the junction becomes widespread prior to non-convergence.
ELASTIC-PLASTIC ANALYSIS: BENDING MOMENT

- Based on elastic-plastic analysis, the bending moment achieved prior to non-convergence is 88,600 in-lbf (dividing by 2.4 = 36,900 psi).
- Limiting displacements using the twice elastic slope method gives a maximum bending moment of 65,000 in-lbf.
- Twice elastic slope margin on collapse pressure = 1.5
- This load/1.5 gives an allowable bending moment of 43,300 in-lbf.
SUMMARY OF RESULTS

- A comparison of allowable loads based on the three analysis methods discussed herein is shown.
  - Elastic stress analysis (Part 5.2.2)
  - Limit load analysis (Part 5.2.3)
  - Elastic-plastic analysis (Part 5.2.4)

- The permissible load obtained from the twice-elastic slope method is compared to the permissible load using the load prior to non-convergence divided by the Div. 2 elastic-plastic margin of 2.4.
  - Results are in reasonable agreement.

- These results show that limit load and elastic-plastic analysis provide significantly higher allowable loads, especially for nozzle loads.

<table>
<thead>
<tr>
<th>DBA Technique</th>
<th>Internal Pressure (psi)</th>
<th>Axial Force (lbf.)</th>
<th>Bending Moment (in-lbf.)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic Analysis</td>
<td>642</td>
<td>14,700</td>
<td>21,300</td>
</tr>
<tr>
<td>Limit Load Analysis</td>
<td>667</td>
<td>25,400</td>
<td>42,000</td>
</tr>
<tr>
<td>Elastic-Plastic Analysis (Collapse Load/2.4)</td>
<td>917</td>
<td>54,000</td>
<td>36,900</td>
</tr>
<tr>
<td>Elastic-Plastic Analysis (Twice Elastic Slope)</td>
<td>983</td>
<td>45,300</td>
<td>43,300</td>
</tr>
</tbody>
</table>
SUMMARY OF RESULTS (CONTINUED)

The diagram shows the normalized allowable load for different loading conditions: Internal Pressure, Bending Moment, and Axial Force, under various analyses:
- Elastic Stress Analysis
- Limit Load Analysis
- Elastic-Plastic Analysis (2x Elastic Slope Method)
- Elastic-Plastic Analysis (Collapse Load/2.4)
PERSPECTIVE ON MANDATORY RATCHETING CHECK

• In this case, because the vessel does not cycle in service (fatigue screening criteria are satisfied), limiting the design based on this elastic stress check for supplemental loading may be overly conservative.

• As discussed previously, strictly enforcing this ratcheting check for all types of loading (even if fatigue damage is not a viable failure mode) can be overly restrictive.

• For cases where nozzle bending moments or shear forces dominate, exceeding the $S_{PS}$ limit does not necessarily imply that the nozzle-to-shell junction will ratchet.

• In fact, unless the loading is significant enough to cause widespread plasticity around the circumference of the nozzle, ratcheting will most likely not occur because of the restraint of the surrounding material that experiences attenuated stress levels.

• Some practical limit on $(P_L + P_B + Q)$ should be enforced for elastic stress analysis to ensure that gross plastic deformation does not occur. However, instead of mandating that the $S_{PS}$ limit be satisfied for all cases, it may be more appropriate for the analyst to employ judgment to evaluate supplemental loads acting on nozzles, lugs, or other support members:
  – Particularly, if the vessel does not operate in cyclic service, and the Code fatigue screening criteria is fulfilled (i.e., ratcheting is not a viable failure mode).
OVERVIEW AND CONCLUSIONS

• The elastic stress limits outlined in ASME Section VIII Division 2 at nozzle-to-shell junctions are investigated using an example geometry.

• The following load cases are compared:
  – Internal pressure
  – Nozzle bending moment
  – Nozzle axial force

• The following analysis methods are compared:
  – Elastic stress analysis
  – Limit load analysis
  – Elastic-plastic analysis (current approach vs. 2x elastic slope method)

• The P+Q check (compared to $S_{ps}$ limit) can be very limiting from a DBA standpoint for vessels that do not operate in cyclic service and particularly for nozzle loads.

• Limit load and elastic-plastic analysis methods reduce the conservatism associated with the Code elastic stress criteria, and the twice elastic slope method is a reasonable alternative to current elastic-plastic analysis.

• Elastic-plastic analysis gives significantly higher allowable nozzle loads than the elastic P+Q check permits because failure occurs when plasticity is present in a large section of the nozzle neck and not just a local section.