Validation of a Complex Pressure Vessel Integrity Assessment Using In-Service Data

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Dave Dewees, P.E.
Robert G. Brown, P.E.
The Equity Engineering Group, Inc.
Abstract

- Many advanced numerical and evaluation techniques are available for assessment of pressure equipment, and these techniques are often combined when dealing with the most challenging integrity problems. Examples range from non-linear Finite Element Analysis (FEA) and advanced material modeling to determination of critical flaw sizes to avoid sudden brittle fracture. Each of these techniques has uncertainty associated with it, which in isolation should be adequately understood. When these techniques are combined however, direct verification and validation (V&V) of the overall analysis and results are generally not possible. This presentation, rather than focusing on a micro-level V&V of individual parts of the analysis (which is of course important), offers a macro-level V&V based on extended in-service data of a series of process vessels subject to repeated thermal-mechanical cycling and cracking. Detailed simulation of weld residual stress, local post weld heat treatment, and operating thermal and mechanical cycling is used as input to fatigue, crack growth and fracture assessments, and the results compared with historical crack initiation and growth data over a 10 year operating history.
Presentation Overview

- Case study of 16 cyclic molecular sieve vessels in a gas plant (molesieves) used to absorb moisture from methane
- Vessels subjected to thermal and pressure cycling (1200 cycles/year), and repeated fatigue cracking
- API 579 level 3 FFS assessment to determine critical flaw dimensions and evaluate structural integrity of repairs
- Focus of this presentation is on analyses for crack initiation in bottom nozzles relative to available field data
Initial Operation

- ASME Section VIII Div. 1 vessels, cyclic service not considered in design
- Original nozzles were corner welded designs
- 4 inches of internal refractory lining in lieu of external insulation
  - Internal lining not included on the nozzle bores
  - Source of inherent thermal mismatch between the nozzle and head at operating temperature
- Nozzles were redesigned as butt-welded insert forgings and replaced within 10 years of service due to repeated fatigue cracking
Inspection Data

• Redesigned nozzles expected to have significantly improved fatigue resistance
• After replacement, operator continued to perform periodic inspection of (new) nozzles
• Over an operating period of \( \approx 10 \) years, inside surface breaking cracks were again found in the bottom nozzles of 14 of the 16 molesieves
  - Cracks were typically about 0.6 inches (15 mm) deep
  - Located in inside radius of the nozzle forging base material
  - All of the cracks spanned 360 degrees of circumference
• Over a period of several years, numerous repeat inspections performed
  - Results generally showed relatively slow and predictable crack growth
  - However, a few vessels had significant cracking to a maximum depth of 2.2 inches (56 mm)
Inspection Data

- Unfortunately, detailed crack growth data is not available
- Data should be interpreted as bounding fatigue crack initiation cycles
Numerical Analysis

• As mentioned, focus of this presentation is on the fatigue crack initiation aspect of the analyses
• WRS and LPWHT are discussed only briefly
• Three different fatigue analyses are performed:
  – Elastic analysis and ASME Code smooth bar fatigue
  – Elastic analysis with plasticity correction using modern strain based fatigue as implemented in fe-safe commercial software
  – Elastic-plastic analysis considering stresses due to welding, LPWHT and repeated service cycling, as input to new ASME Section VIII Division 2 elastic-plastic fatigue rules
Operating Details

- Pressure (green) and gas temperature (red) are not in phase
- Limiting part of cycle is sudden increase in gas temperature (520 F in 2 minutes)
Operating Analysis

- Sudden temperature increase causes nozzle to be much hotter than lined head (below)
- Results in very large compressive stresses
Operating Analysis

- Elastic stress history at nozzle radius shown below (blue) overlaid on pressure and temperature histories.
- **Elastic stress range normal to crack:** -64 ksi to 28 ksi, $S_a = 46$ ksi.
Elastic Fatigue Analysis

- Original Code fatigue data is presented as pseudo-stress, or actual strain amplitude multiplied by E (even if strain is elastic-plastic)
- The idea was that this will correspond to elastically calculated strains the majority of the time since the plastic region being fatigued should be forced to follow the surrounding elastic material
- Note that the stress in the notch from the elastic calculation would be incorrect

Figure from 1969 Criteria Document

\[ S = \frac{E}{4\sqrt{N}} \ln \left( \frac{100}{100 - A} \right) + B \]

- \( E \): elastic modulus (psi)
- \( N \): number of cycles to failure
- \( S \): strain amplitude times elastic modulus

**Figure 9. Fatigue Data - Carbon Steels.**
Elastic Fatigue Analysis

- Code curves have factors applied ("2 and 20"), but not a clear statistical basis
- Recent work has shown that the Design curve is similar to a $-3\sigma$ curve for the data on the last page, while the equation from the Criteria Document can be used as a mean curve:

![ASME Carbon Steel Fatigue Curves](image-url)
Elastic Code Fatigue

- This is the ASME smooth bar method
  - Plasticity is accounted for with a correction factor, $K_e$
  - Although the total stress range is quite high, the linearized stress for the section is quite close to the $3S = 64.8$ ksi limit and $K_e$ is negligible

- The allowable cycles for an alternating stress of 46 ksi is:
  - ASME Smooth Bar Design Curve: 5,477 cycles
  - ASME “Mean” Curve: 112,606 cycles
Modern Strain-Based Fatigue

- Strain-based fatigue procedures have continued to evolve since the introduction of the Code procedure in the 1960s.

- Essential characteristics of modern strain based fatigue are:
  - Strain-life equation
  - Cyclic stress-strain curve and Neuber Rule
  - Morrow mean stress correction
  - Critical plane approach

*Figure from Dowling*
Modern Strain-Based Fatigue

- Strain-life equation is constructed from an elastic strain-life curve and a plastic strain-life curve
- $2N_f$ is typical terminology, where $N_f$ is a half cycle and $2N_f$ is a complete cycle
- Cyclic stress-strain curve is obtained as part of the test and used to separate total strain into elastic and plastic parts for curve-fitting

\[ \varepsilon_a = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \]

Figure from [9]
Modern Strain-Based Fatigue

- Strain-life equation has been modified based on critical plane approach
  - Says that fatigue will initiate on the plane that has a critical combination of shear and opening stress
  - No equivalent stress used – shear and normal put directly into strain-life equation
- Morrow mean stress correction typically used for ductile metals under typical operating conditions

The conventional strain-life equation

\[ \frac{\Delta \varepsilon}{2} = \frac{\sigma'_f}{E} (2N_f)^b + \varepsilon'_f (2N_f)^c \]

is re-written with the shear strain amplitude and normal strain amplitude on the left hand side

\[ \frac{\Delta \gamma_{\text{max}}}{2} + \frac{\Delta \varepsilon_N}{2} = C_1 \frac{\sigma'_f}{E} (2N_f)^b + C_2 \varepsilon'_f (2N_f)^c \]

and the complete Brown-Miller strain-life equation is then

\[ \frac{\Delta \gamma_{\text{max}}}{2} + \frac{\Delta \varepsilon_N}{2} = 1.65 \frac{\sigma'_f}{E} (2N_f)^b + 1.75 \varepsilon'_f (2N_f)^c \]

The Brown-Miller equation can be modified by including the Morrow mean stress correction

\[ \frac{\Delta \gamma}{2} + \frac{\Delta \varepsilon_N}{2} = 1.65 \frac{\sigma'_f - \sigma_{N,m}}{E} (2N_f)^b + 1.75 \varepsilon'_f (2N_f)^c \]
Modern Strain-Based Results

- Exact same elastic stress analysis is used with the commercial software fe-safe
- Brown-Miller with Morrow mean stress correction as just described, along with plasticity accounted for by using the Neuber correction
  - fe-safe baseline: 74,440 cycles
  - fe-safe (surf. finish prec. forged): 10,036 cycles

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<tr>
<th>Method</th>
<th>Cycles</th>
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<td>fe-safe (Kt=1)</td>
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<td>ASME “Best Fit”</td>
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<tr>
<td>fe-safe (surf finish prec. forged)</td>
<td>10,036</td>
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<tr>
<td>ASME Design Curve</td>
<td>6,373</td>
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</table>
Elastic-Plastic Analysis

- Elastic stresses are beyond yield strength
- Compressive overload will lead to tensile residual stresses
- Cyclic elastic-plastic analysis is ideally suited to more realistically predict stress and strain ranges

\[ K_{e_{ss}} \cdot \left( \frac{\Delta p}{2} \right)^{n_{e_{ss}}} = \frac{C}{\gamma} \tanh \left( \gamma \frac{\Delta p}{2} \right) + \sigma_o \]

\[ d\alpha = \sum_{i=1}^{4} \frac{C_i}{\sigma_o} (\sigma - \alpha) \, dp - \gamma_i \cdot \alpha_i \cdot dp \]

\[ \varepsilon_{\text{ia}} = \frac{\sigma_a}{E} + \left( \frac{\sigma_a}{K_{e_{ss}}} \right) \frac{1}{n_{e_{ss}}} \]

\[ \varepsilon_{\text{ir}} = \frac{\sigma_r}{E} + 2 \left( \frac{\sigma_r}{2K_{e_{ss}}} \right) \frac{1}{n_{e_{ss}}} \]
Elastic-Plastic Analysis

- Concern over effectiveness of LPWHT performed during nozzle replacement
- For this reason, detailed WRS and LPWHT analysis were also performed
- This analysis primarily helped in assessing critical flaw sizes and fracture margins, but also provides a starting point for the cyclic analysis of the operating conditions
Weld Residual Stress Analysis

- Before LPWHT analysis can be performed, the initial (welding) stresses must be determined.
- Basic parameters used are I=28 A, E=145 V, v=3 ipm and process efficiency ≈ 0.80%
- Weld bead shapes estimated in part from macros.
Weld Residual Stress Analysis

- Stress transverse to welding direction
- Stress parallel to welding direction
LPWHT Analysis

- Internal refractory is broken out with a jack hammer to access nozzles
- Typically, only a limited amount of the cast refractory is broken out, which makes effective stress relief a challenge
LPWHT Analysis

- Thermal Analysis is performed, and then results applied to ending state of WRS analysis
- LPWHT is simulated with Omega creep model (API 579) and CREEP user subroutine

\[
\log_{10} \dot{\varepsilon}_{co} = -\left( A_o + \Delta \sigma_{\text{ref}}^{\text{sr}} + \frac{1}{460 + T} \right) \left[ A_1 + A_2 \log_{10} \sigma_{\text{ref}}^{\text{sr}} + A_3 \log_{10} \sigma_{\text{ref}}^2 + A_4 \log_{10} \sigma_{\text{ref}}^3 \right]
\]

\[
\log_{10} \Omega = B_o + \Delta \sigma_{\text{ref}}^{\text{cr}} + \frac{1}{460 + T} \left[ B_1 + B_2 \log_{10} \sigma_{\text{ref}}^{\text{cr}} + B_3 \log_{10} \sigma_{\text{ref}}^2 + B_4 \log_{10} \sigma_{\text{ref}}^3 \right]
\]

\[
n_{BN} = -\left( \frac{1}{460 + T} \right) \left[ A_2 + 2A_3 \log_{10} \sigma_{\text{ref}}^{\text{cr}} + 3A_4 \left( \log_{10} \sigma_{\text{ref}}^{\text{cr}} \right)^2 \right]
\]

\[D' = \Omega m \dot{\varepsilon}_{co}, \quad D = \int_0^t D' \, dt\]
Effect of LPWHT

After Welding

Stress Transverse to Welding Direction

After LPWHT
Effect of LPWHT

- **Maximum Transverse Stress During Operation:**

  - Elastic
  - Elastic-Plastic
  - Elastic-Plastic w/Residual Stress
Operating Stress History

Transverse Stress-Strain History For 1st 40 Cycles at Nozzle Radius
No WRS
WRS

Stress Normal to Cut Line (Local S22, psi)
Distance from Inside/Radius Surface (in.)

Change in Through-Wall Opening Stress Distribution with Cycling

Stress at nozzle radius decreases with cycling to stable value
Code Elastic-Plastic Fatigue

- **E-P Method: Driving Force** – Alternating stress computed from equivalent total (i.e. elastic + plastic) strain

\[ S_{alt,k} = \frac{1}{2} E_{yf} \cdot \Delta \varepsilon_{eff,k} \]

Effective strain range

\[ \Delta \varepsilon_{eff,k} = \frac{\Delta S_{p,k}}{E_{ya,k}} + \Delta \varepsilon_{peq,k} \]

\[ \Delta S_{p,k} = \frac{1}{\sqrt{2}} \left[ \Delta \sigma_{11,k} - \Delta \sigma_{22,k} + \Delta \sigma_{11,k} - \Delta \sigma_{33,k} \right]^2 + \Delta \sigma_{22,k} - \Delta \sigma_{33,k}^2 \]

\[ \Delta \varepsilon_{peq,k} = \frac{\sqrt{2}}{3} \left[ \Delta \varepsilon_{11,k} - \Delta \varepsilon_{22,k} + \Delta \varepsilon_{22,k} - \Delta \varepsilon_{33,k} \right]^2 + \left( \Delta \varepsilon_{33,k} - \Delta \varepsilon_{11,k} \right)^2 + 6 \left( \Delta \varepsilon_{12,k} + \Delta \varepsilon_{23,k} + \Delta \varepsilon_{31,k} \right) \]

Definition

\[ E_{yf} \] - value of modulus of elasticity on the fatigue curve being utilized

- **E-P Method: Resistance** – Fatigue curve, same as elastic method
Code Elastic-Plastic Fatigue

- Stable cycle shown below - from analysis considering residual stress, but identical to ranges from plastic analysis only
- From data below and formulas on the last slide, the elastic plastic alternating stress (or really pseudo-strain) is 33.2 ksi
  - ASME Smooth Bar Design Curve: **15,717 cycles**
  - ASME “Mean” Curve: **500,000 cycles**

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Summary

• The mean curve data seems to grossly over-estimate life in this case.
• Environmental effects were not considered as the region in question is covered by cast refractory.
• Using the more reasonable predictions gives the following summary:
  - Elastic strain-based fatigue: 5,477 cycles
  - Modern strain-based fatigue: 10,036 cycles
  - Cycle-by-cycle elastic-plastic fatigue: 15,717 cycles
• In this case, the simplest analysis seems to have given the conservative prediction.
• The modern strain based results seem to match the data best.
• Further detail will be presented on this case in an upcoming series of WRC Bulletins.
Further Information

• More complete information can be found in PVP2011-57657, “Case History Using Advanced Analysis To Evaluate Fitness-for-service Of Cyclic Vessels In The Petrochemical Industry,” R. Brown and D. Dewees

• More details on cyclic plasticity modeling used here in PVP2010-25641, “Application of Elastic-Plastic Design Data in the New ASME B&PV Code Section VIII, Division”
Questions?

Dave Dewees
email: djdewees@equityeng.com
20600 Chagrin Blvd. • Suite 1200
Shaker Heights, OH 44122  USA
Phone: 216-283-9519 • Fax: 216-283-6022
www.equityeng.com