Alternative Approaches for ASME Code Simplified Elastic-Plastic Analysis

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NRC Public Meeting on Fatigue Research and Related ASME Activities

Rockville, MD
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Background

- NRC RG 1.207 requires the use of multipliers ($F_{en}$) on the cumulative usage factors (CUF) to account for the effects of environmentally assisted fatigue (EAF)
  - Application of RG-1.207 can increase the calculated CUF significantly, making it difficult to meet the CUF limits for new plants and plants with license renewal.
  - This can be exacerbated when higher number of cycles associated with flexible operation are considered.
  - In reality, there has been no field experience of cracking attributed to EAF; in the few cases where there has been cracking, it has been due to high cycle fatigue where EAF is not a factor.
  - On the other hand, EAF test data show a strong environmental effect; this is not consistent with the good field performance. While the $F_{en}$ factors in RG 1.207 are consistent with test data, they still do not reflect the good EAF field performance.
Background (cont.)

- One way of addressing the EAF problem is to examine the original CUF (without $F_{en}$) which may be over-conservative.
  - Justify a lower CUF in the original analysis so that the fatigue usage multiplied by $F_{en}$ is still acceptable.

- The use of the ASME Code simplified elastic-plastic analysis (NB-3228.5 or NG-3228.5) is often the biggest source of conservatism in fatigue analysis.
  - The focus of the EPRI project is to develop Alternative Approaches for ASME Code Simplified Elastic-Plastic Analysis.

- There are two ways to update the high fatigue usage:
  - Use new elastic-plastic (EP) analysis; an expensive option that requires new finite element analysis; difficult to apply for piping. Also, the Code does not provide explicit rules on how EP analysis is performed.
  - Propose a more realistic approach as an alternative to the NB-3228.5 (or NG-3228.5) rules for the Code simplified elastic plastic analysis.
Basis for Current Code $K_e$ Equation

- Developed originally by Langer almost 50 years ago based on a simple model for a cantilever beam and tapered bar
- Bounded by the proposed $K_e = 1/n$ where $n$ is the strain hardening coefficient
- Simple formulation, but overly conservative, especially for carbon steel and low alloy steel with low strain hardening coefficient
- Subsequently Modified with input from Tagart
**Current Code $K_e$ Equation**

- $K_e = 1$ for $S_n \leq 3S_m$
- $K_e = 1 + \frac{(1-n)}{n(m-1)} \left\{ \frac{s_n}{3S_m} - 1 \right\}$ for $3S_m \leq S_n \leq 3mS_m$
- $K_e = 1/n$ for $S_n \geq 3mS_m$

<table>
<thead>
<tr>
<th>Materials</th>
<th>m</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon Steel</td>
<td>3.0</td>
<td>0.2</td>
</tr>
<tr>
<td>Low Alloy Steel</td>
<td>2.0</td>
<td>0.2</td>
</tr>
<tr>
<td>Austenitic Stainless Steel</td>
<td>1.7</td>
<td>0.3</td>
</tr>
<tr>
<td>Ni-Cr-Fe (Alloy 600)</td>
<td>1.7</td>
<td>0.3</td>
</tr>
</tbody>
</table>

- Comparison with the results from elastic plastic analysis show the conservatism in the Code $K_e$ value
- A new approach that preserves the simplicity of the Code approach, but results in a more realistic CUF value is needed
Conservatism in the Code $K_e$ – Elastic-Plastic Analysis

- Comparison of the Code $K_e$ value with the results of elastic-plastic analysis show that the Code value is conservative by a factor of two or higher.
- The higher Code $K_e$ can result in an overestimate of 20-100 in fatigue usage.
- More realistic Code $K_e$ factors can be significant in addressing license renewal and RG 1.207 EAF challenges.
Conservatism in the Code $K_e$ Equation – Air Test data

- Tests have also been done on notched carbon steel and stainless steel specimens in air to compare the Code $K_e$ value with values based on test data.
- Results confirm that the Code $K_e$ values are conservative by factors well in excess of 2.

**Ke Formulation in WRC-361**

- WRC-361 was one of the first efforts to examine the NB-3228.5 Code rules and offered alternate methods to determine Ke.
- The Ke formulation in WRC-361 considers the following:
  - Effect of Poisson’s ratio during plastic behavior. This is addressed by developing an equivalent \( \nu^* = 0.5 - \frac{E_s}{E} (0.5 - \nu) \) and determining the ratio of stress intensity for elastic and elastic plastic behavior under strain controlled (e.g. thermal) loading. The stress intensity ratio is:
    \[
    K_{\nu} = \frac{S_{int}^{Plastic}}{S_{int}^{Elastic}} = \frac{1 - \nu}{1 - \nu^*}.
    \]
    For \( \nu = 0.3 \), the maximum value of \( K_{\nu} \) is 1.4.
  - Elastic follow-up during mechanical load cycling; this is evaluated using the present Code Ke equation.
  - Notch strain redistribution based on Neuber analysis; the additional notch factor (over and above the stress concentration factor, \( K_T \)) is \( K_n = K_T \left(\frac{1-n}{1+n}\right) \).
- The effective Ke value for the first two factors is determined by a weighted average of \( K_{\nu} \) and \( K_e \). This is then multiplied by the notch factor \( K_n \).
Application of WRC-361: French RCC-M Code

- There are several industry codes (e.g. EPRI Report TR-107533 in 1998) that attempt to adjust the Code conservatism; all based on WRC-361 concepts.
- The RCC-M code includes the $K_v$ factor (Poisson’s ratio effect)
  \[ S_a = 0.5(K_e^{\text{mech}}S_p^{\text{mech}} + K_e^{\text{ther}}S_p^{\text{ther}}) \]
  \[ K_e^{\text{ther}} = 1.86 \left[ 1 - \frac{1}{1.66+S_n/S_m} \right] \text{ but } \geq 1 \]
- $K_n$ (Neuber notch effect) is included in RCC-MR for high temperature reactors but not in RCC-M (for PWRs).
- Some disadvantages:
  - $K_e$ correction even below 3$S_m$
  - $S_p^{\text{mech}}$ and $S_p^{\text{ther}}$ are new stress terms that need new stress analysis
  - Potential Discontinuity in $K_e$ at 3$S_m$
- British Code similar to RCC-M
Code Case N-779

- Extends the work by Deardorf in EPRI Report TR-107533
- Combines contributions from three sources
  - Mechanical loading multiplied by the Code $K_e$
  - Thermal load multiplied by $K_{\nu}$
  - Notch effect by including $K_n$
- Requires determination of stresses not currently available in ASME Code stress reports
- Somewhat difficult to use
- Identified as “Unacceptable” in RG 1.193
Why is a New $K_e$ Relationship Needed?

- Most of the existing ‘improved’ $K_e$ formulations require new stress analysis; if new analysis is performed, we might as well use new elastic-plastic (EP) analysis. However the Code does not specify rules for performing EP analysis; wide range of methods used.
- Some of the $K_e$ factors may be more conservative than the Code value for stress ranges below $3S_m$ where the current rules are adequate.
- There may be a discontinuity in $K_e$ at $S_n=3S_m$ in many of the proposals.
- Some of the new $K_e$ expressions may require new analysis and may be somewhat complex (e.g. Code Case 779).
- We need a new approach that uses existing information in current ASME Code stress reports and retains the simplicity of the current code but without the excessive conservatism.
  - The new approach should apply to pressure vessel components (NB-3200) as well as piping (NB-3600).
  - It should cover common structural materials- austenitic stainless steel, nickel based alloys, carbon steel and low alloy steel.
Proposed New $K_e$ Formulation (w/o $K_n$ Neuber Factor)

- Follows the WRC-361 method of using a weighted average approach for the thermal and mechanical load stresses
  - $K_e^* = K_v \frac{S_{n\text{ therm}}}{S_n} + K_e \frac{S_{n \text{ mech}}}{S_n}$
    - $S_{n \text{ mech}} =$ Mechanical load: $P+Q$-Thermal Bending
    - $S_{n \text{ therm}} =$ Thermal Load: Thermal Bending (TB)
  - $K_v$ is conservatively assumed to be 1.4 (corresponding to $v=0.3$)
  - $K_e^* = 1.4(1 - R) + K_e R$ for $3S_m \leq P + Q \leq 3mS_m$; $R = \frac{P+Q-TB}{S_n}$, but not higher than $K_e$.
  - Eliminates the discontinuity at $S_n = 3S_m$

- The proposed $K_e$ equation applied to piping also except that $R$ is defined as:
  - $R = \frac{P+Q-TB}{S_n} = \frac{\text{Equation 13 of NB-3653}}{\text{Equation 10 of NB-3653}}$
  - $S_n = P+Q = \text{Equation 10 of NB-3653}$
  - $P+Q-TB = \text{Equation 13 of NB-3653}$

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Proposed $K_{e^*}$ Factor for Low Alloy Steel

Ke* for Low Alloy Steel

- ASME Code
- $P+Q-TB=1Sm$
- $P+Q-TB=2Sm$
- $P+Q-TB=3Sm$
- $P+Q-TB=0$ (i.e. all TB)

Sn/Sm vs. Ke* graph showing various curves for different TB values.
Proposed $K_{e*}$ Factors for Stainless and Carbon Steels

Ke* for Stainless Steel

Ke* for Carbon Steel
Consideration of Notch Effects

- WRC-361 recognizes Poisson’s ratio effects and elastic follow-up effects by using a weighted approach of thermal and mechanical stresses and multiplies it by a notch factor based on Neuber analysis.

- WRC-361 specifies a Notch factor (over and above the standard stress concentration factor $K_T$ used in elastic analysis)
  - The notch factor is given by: $K_n = K_T \left( \frac{1-n}{n+1} \right)
  - Depends on the strain hardening exponent $n$ (equal to 0.3 for stainless steel and 0.2 for carbon and low alloy steel)

- The notch effects are first described here using Neuber Analysis. Since many Codes (e.g. RCC-M code) do not explicitly include the notch factor, example EP analysis is performed to determine whether there is a need to add the notch factor $K_n$. 
**Notch Stress-Strain**

- Analysis performed for stainless steel and carbon steel
- Remote stress and strain: $S, e$
- Local stress and strain: $\sigma, \varepsilon$
- Example considers the case $K_T=2$

\[
K_t^2 = K_o K_e
\]
\[
K_t^2 = \frac{\sigma \varepsilon}{S e}
\]
\[
K_t^2 = \frac{\sigma \varepsilon E}{S S}
\]
Neuber Notch Analysis Approach

- Neuber analysis relates the stress and strain at the notch to the global stress and strain

\[
\frac{(K_s S)^2}{E} = \frac{\sigma \varepsilon}{\text{applied load}} = \frac{\sigma \varepsilon}{\text{notch response}}
\]

\[
K_t^2 = K_\sigma K_\varepsilon
\]

\[
K_t^2 = \frac{\sigma}{S} \varepsilon
\]

\[
K_t^2 = \frac{\sigma \varepsilon E}{S S}
\]

- A bilinear stress strain curve was fitted to the power law with \( n = 0.3 \) for stainless steel and \( S_y = 50 \text{ ksi} \) (close to \( 3S_m = 50.7 \text{ ksi} \) for stainless steel at 550°F)
Example Results for Stainless Steel $K_T=2$

\[ K_n = K_T^{\frac{1-n}{n+1}} = 1.45 \]
Verification Problems for the Proposed $K_e^*$

- The objective of the verification problems was twofold:
  - Compare the prediction of the $K_e^*$ equation with the results of elastic plastic analysis for unnotched geometry
  - Determine whether the additional notch factor $K_n$ is needed for the evaluation of components with stress concentration factor (SCF), $K_T$
    - $K_n$ is the additional factor over and above the $K_T$ and accounts for local yielding in the SCF region
- Examples include notched and unnotched locations with a combined of mechanical (P+Q-TB) and thermal bending (TB)
  - Bettis stepped pipe test (no notch)
  - Notched ($K_T=2.9$) beam (both notch and unnotched locations) evaluated by Adams at KAPL
  - Axial groove in a pipe ($K_T=3$) with mechanical and thermal loading
  - Taper location in a pipe ($K_T=1.6$)
Options Considered for Notches

- $K_n$ is assumed to be constant and equal to 1.0; i.e. no additional notch strain factor over and above $K_T$
- Linear variation from 1.0 at $S_n=3S_m$ and linear variation from 1.0 at $S_n=3S_m$ to $K_n$ at $3S_m<S_n<3S_m$
Example Problem: Bettis Stepped Pipe Test (SS)

- Test performed by Bettis to evaluate Environmental Fatigue effects in Piping
- Cycling from 100° to 650° F every four minutes; pressure held constant at 2500 psi
- Thermal analysis and elastic plastic stress analysis results published by Jones et al at Bettis (ASME PVP 2004-2748)

<table>
<thead>
<tr>
<th>Thickness, inch</th>
<th>P+Q, ksi</th>
<th>Elastic Strain Amplitude, %</th>
<th>E-P analysis strain amplitude %</th>
<th>$K_S$ based on Elastic Plastic analysis</th>
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<td>0.179</td>
<td>79.8</td>
<td>0.30</td>
<td>0.37</td>
<td>1.25</td>
</tr>
</tbody>
</table>

Transient:
- 100°F – 650°F in 3 seconds
- Hold at 650°F for 3 minutes 57 seconds
- 650°F – 100°F in 3 seconds
- Hold at 100°F for 3 minutes 57 seconds
- Repeat
- Pressure held constant at 2500 psi
Ke Comparison: Elastic Plastic Analysis vs. Proposed $K_e^*$

$$K_e^* = \text{Lower of:}$$

$$1.4 \frac{TB}{S_n} + K_e \frac{P+Q-TB}{S_n} \text{ for } 3S_m \leq P + Q \leq 3mS_m \text{ or } K_e$$
Notched Beam Example (Adams – KAPL)

Low Alloy Steel Beam with Notch ($K_T=2.9$) subjected to Thermal Bending and Axial Loading
Notched Beam: Comparison with EP Analysis

Notched Beam Analysis (Adams)

- Code Ke
- Proposed Ke TB 100%
- Proposed Ke TB=51.6%
- Ke from EP Analysis_TB=100%(Notch)"
- Ke from EP Analysis_TB=51.6%(Notch)
- Ke from EP Analysis_TB=100%(No notch)
- Ke EP Analysis_TB=51.6% (no notch)
Example Problem: Axial Groove in a Pipe

Stainless Steel Pipe (\(K_T = 3\))

- Stress, ksi: 0 to 100
- Strain: 0 to 0.01

**Stainless Steel**

- Young's Modulus: \(E = 25 \times 10^6 \text{ psi}\)
- Poisson's Ratio: \(\nu = 0.3\)

**Case 1:**
- \(P_r = 2000 \text{ psi}\)
- \(T_{\text{metal}} = 550°F\)
- \(T_{\text{in}} = 100°F\)
- \(\Delta T = 450°F\)

**Case 2:**
- \(P_r = 1000 \text{ psi}\)
- \(T_{\text{metal}} = 550°F\)
- \(T_{\text{in}} = 100°F\)
- \(\Delta T = 450°F\)

**Case 3:**
- \(P_r = 2000 \text{ psi}\)
- \(T_{\text{metal}} = 350°F\)
- \(T_{\text{in}} = 100°F\)
- \(\Delta T = 250°F\)

**Case 4:**
- \(P_r = 0 \text{ psi}\)
- \(T_{\text{metal}} = 550°F\)
- \(T_{\text{in}} = 100°F\)
- \(\Delta T = 450°F\)

- **a)** Elastic Analysis
- **b)** Elastic-Plastic Analysis; Assume Bilinear stress strain curve, Kinematic hardening

**Pipe Dimensions:**
- 24 in. diameter
- 2 in. thickness

**Material Properties:**
- Young's Modulus: \(E = 25 \times 10^6 \text{ psi}\)
- Thermal Expansion Coefficient: \(\alpha = 9 \times 10^{-6} \text{ /°F}\)
- Reference Temperature: \(T_{\text{ref}} = 70°F\)
- Pipe ID: 24 in.
- Axial semi-circular notch: ¼ inch radius
Axial Groove Strain: Elastic Plastic Analysis
Axial Groove: Comparison with Elastic Plastic Analysis
Example Problem: Tapered Shoulder in a Pipe

<table>
<thead>
<tr>
<th>Cases</th>
<th>Pressure</th>
<th>Ti</th>
<th>To</th>
<th>Delta T</th>
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<tr>
<td></td>
<td>psi</td>
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<td>Deg F</td>
<td>Deg F</td>
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<tr>
<td>D</td>
<td>12000</td>
<td>100</td>
<td>100</td>
<td>0</td>
</tr>
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</table>
Tapered Shoulder in Pipe: Elastic Plastic Analysis
Tapered Shoulder: Comparison with Elastic Plastic Analysis
Consideration of the Need for the Notch factor

- The EP analyses for the four example problems described here suggests that the $K_e^*$ equation (without the notch factor) proposed here bounds the results of the EP analysis for the cases described here.

- This suggests that no explicit $K_n$ application is needed.

- There other justifications for not including the $K_n$ factor:
  - Cyclic yield strength in in many materials (e.g. SS) is higher than $3S_m$
  - The theoretical stress concentration factor $K_T$ is already included in the elastic analysis.
  - There are other conservatisms (e.g. $K_v$ of 1.4, use of the Code $K_e$ for all other loads including mechanical and thermal membrane loading)

**Recommendation: No $K_n$ application**
Summary and Recommendations

- The existing ASME Code simplified procedures for elastic plastic analysis have been shown to be overly conservative.

- Several proposals have been made for modification of the ASME Code procedures but these proposals have been complicated and required new stress analyses.

- A new proposal has been developed that:
  - Has been shown to be conservative relative to elastic-plastic analysis.
  - Considers Poisson’s ratio and elastic follow up \( (K_e) \) effects.
  - Requires no new stress analysis.
  - No application of Neuber notch factors over and above \( K_T \).
  - Has the potential to reduce CUF values.
  - Will in most cases offset the need for elastic-plastic analyses.
Next Step

Submit a Code Case for consideration by the ASME Code

Proposed Code Case N-XXX

Alternative Rules for Simplified Elastic Plastic Analysis

Section III, Division 1

**Inquiry:** What alternatives to NB-3228.5 in Section III, Subsection NB or NG-3228.5, Section III, Subsection NG, may be used for simplified elastic plastic analysis of components when the $3S_m$ limit on the range of primary plus secondary stress is exceeded?

**Reply:** It is the opinion of the committee that, as an alternative to NB-3228.5 in Section III, Subsection NB or NG-3228.5, Section III, Subsection NG, the $3S_m$ limit on the range of primary plus secondary stress intensity, $P+Q$, may be exceeded provided that the requirements of (a) through (c) below are met.

a) The range of primary plus secondary membrane plus bending stress intensity $(P+Q)$, excluding thermal bending stresses (TB), shall be $\leq 3S_m$.

b) The value of $S_a$ used for entering the design fatigue curve is multiplied by the factor $K_e^*$, where

$$K_e^* = 1 \text{ for } S_n \leq 3S_m$$

$$= \text{ Smaller of } K_e \text{ and } \left\{1.4(1-R) + K_eR\right\} \text{ for } S_n < 3S_m \leq 3mS_m$$

$$= \left\{(1.4(1-R) + (1/n)R\right\} \text{ for } S_n > 3mS_m$$

where $R = (P+Q-TB)/(P+Q)$ and $K_e$, $m$ and $n$ are defined in NB-3228.5 or NG-3228.5 of Section III Subsection NB or NG respectively.

c) The requirements of (c) through (f) of NB-3228.5 or NG-3228.5 are met.
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