An Improved Approach for Performing Simplified Elastic-Plastic Fatigue Analysis

Technical Report

An Improved Approach for Performing Simplified Elastic-Plastic Fatigue Analysis

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REPORT SUMMARY

Current ASME Code rules require use of a strain correction factor (K_e) for fatigue analysis when the primary plus secondary stress intensity range exceeds the $3S_m$ limit. It is believed that K_e is very conservative in most applications, especially for cases involving thermal transients. An improved approach for performing elastic-plastic analysis is presented in this report.

Background

Current ASME Code (1995) design rules are based on elastic stress analysis. When the primary plus secondary stress intensity range exceeds $3S_m$, a strain correction factor (ratio of plastic strain to elastically calculated strain) is applied to the elastically calculated peak stress to account for the effect of additional strain due to plasticity. This factor includes significant notch effects due to mechanical loads, but is too conservative to apply to cases with minimal structural discontinuities. For cases without severe discontinuities, there should be some flexibility in the application of an alternative rule for the practical notch effect, instead of using a conservative bounding value of K_e.

Objectives

- To quantify the amount of conservatism in the ASME Section III, Class 1 fatigue analysis
- To develop a more realistic and simpler approach to account for strain due to plasticity

Approach

Investigators reviewed the procedures for performing ASME Code Class 1 fatigue analysis for vessels and piping. The background and basis for the current method for performing elastic-plastic analysis was documented. An alternate approach developed for the French RCC-M Code, described in Welding Research Council (WRC) Bulletin 361, was also reviewed and documented. Plastic analysis was conducted for both thermal and thermal/mechanical loading of some cylindrical components. Strain

correction factors calculated directly from elastic-plastic finite element analyses were compared to the strain correction factors calculated from the approaches in the WRC Bulletin 361 and the ASME Code.

Results

A unified approach was developed to calculate a more realistic and less conservative strain correction factor (K_e). This approach considers all of the effects of localized thermal loading, elastic follow-up, and notches in one formulation. It also simplifies the procedure by including the constitutive relation, $\sigma = A\varepsilon^n$, in the formulation directly. Analyses conducted in this study show that values of K_e can be reduced from above 3.0 to approximately 1.5-2.0.

EPRI Perspective

Design for fatigue is a major concern for any power or process facility. Accurate methods of engineering for fatigue are important for cost-effective design, for root cause failures, and for evaluating remaining fatigue life of plant designs. The work being done under the EPRI fatigue program continues to establish the technical justification to allow for reductions in the level of conservatism associated with current Code approaches. The results of this program can provide a basis to extend the calculated fatigue life of components.

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Interest Categories

Piping, reactor vessel & internals

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INTRODUCTION

Section III of the ASME Boiler and Pressure Vessel Code for Class 1 components [1] includes consideration of primary, secondary, and peak stresses. By meeting the stress intensity limits for primary loadings (for example, pressure, gravity, and mechanical loadings), gross rupture of pressure retaining components is avoided. By meeting limits on stress intensity ranges for all possible combinations of loading conditions, gross distortion and fatigue failure are avoided.

In performing stress analysis for Class 1 components, the usual approach is to perform linear elastic analysis. This approach is theoretically correct, but represents an approximation when the local stresses exceed the material yield stress. Section III recognizes this fact and provides stress intensity limits and alternate approaches for cases where the predicted stresses are above the yield strength of the material.

The Code has a procedure for performing a simplified elastic-plastic analysis when the range of primary plus secondary stress intensity exceeds three times the allowable stress intensity ($3S_m$). The Code also allows plastic analysis, but because of the effort required in applying this more complex analysis, it is rarely used in the initial design of Class 1 components.

In this report, the complete requirements for performing ASME Code Class 1 fatigue analysis for vessels and piping is reviewed. The background and basis for the current Code procedure for performing simplified elastic-plastic analysis is reviewed. In addition, the work reported in Welding Research Council Bulletin 361 [2] is reviewed, wherein an alternate approach is presented for performing elastic-plastic analysis.

To further evaluate the approach presented in WRC Bulletin 361, plastic analysis was conducted for both thermal and thermal / mechanical loading of some simplified cylindrical components. This analysis confirmed that there is considerable conservatism in the Section III, Class 1 approach, leading to a specific proposed alternative approach that should be offered to users of the ASME Code.

A draft of a Code Case is presented in Appendix A. The proposed approach would allow users to remove considerable conservation for cases involving high stress intensity ranges due to thermal stresses.

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REVIEW OF CURRENT CODE FATIGUE ANALYSIS PROCEDURES

For both vessel (NB-3200 / 3300) and piping (NB-3600) components, the requirements for stress acceptance are similar. These stress analysis requirements include the following steps:

- Show that the primary stress intensity limits are met for pressure, dead weight, and mechanical loadings. Vessel limits are provided in NB-3221 for general membrane, local membrane, and bending stress intensities (and their combinations). Piping limits are specified in NB-3652 (Equation 9).
- Show that secondary stress intensity ranges for all sets of Service Level A and B conditions are met. For vessels, the requirements are in NB-3222. The allowables are met if the range of stress intensity does not exceed three times the material allowable stress intensity ($3S_m$). However, special rules are contained in NB-3228.5 for exceeding the $3S_m$ limit. Similarly, piping requirements are in NB-3653, Equation 10, which includes a limit of $3S_m$. However, by meeting the requirements of Equations 12 and 13, and NB-3653.7, a simplified elastic plastic analysis using Equation 14 might be conducted for each stress intensity range pair not meeting the $3S_m$ limit.
- Perform a cumulative fatigue damage evaluation.

2.1 Design By Analysis

For Class 1 vessel design, the rules of NB-3200, Design by Analysis, are provided. NB-3200 is based on the maximum shear stress theory of failure. Stress limits are based upon the computed values of stress intensity (defined as twice the maximum shear stress) that exist at a point in the structure.

For design loadings, vessels must satisfy the limits of NB-3221.1, which include the following:

- The primary membrane stress intensity (average across the section) must not exceed S_m.
- The local membrane stress intensity (with limitation on extent defined in NB-3213.10) must not exceed $1.5S_m$.
- The membrane + bending stress intensity must not exceed $1.5S_m$.
- Any special stress limits (NB-3227), such as bearing, pure shear, etc., must be met.

For Service Level A and B (normally expected) loading conditions, the requirement is to evaluate the ranges of stress intensity between various loading conditions and to demonstrate acceptability for cyclic operation. Both primary and secondary (for example, thermal) loadings must be considered as follows:

- The primary plus secondary stress intensity range at any location across the thickness of a section must not exceed $3S_m$, except that special rules for exceeding $3S_m$ are included in NB-3228.5.
- Thermal expansion stresses (excluding effects of local discontinuities) at any location across the thickness of a section must not exceed $3S_m$.
- The primary plus secondary plus peak stress intensity ranges at any location across the thickness of a section must be evaluated in a cumulative fatigue usage evaluation per NB-3222.4. Rules are provided to demonstrate that a component might be exempted from fatigue evaluation if the cyclic operating conditions meet certain criteria.
- Thermal stress ratchet must be evaluated per the requirements of NB-3222.5. This limit is specified as a limit on the linear or parabolic variation of temperature through the wall of a vessel that is a function of the maximum membrane stress due to pressure.

Since analysis is generally conducted using linear elastic methods, NB-3227.6 contains special rules for computing stresses that exceed the yield strength of the material. Specifically,

- For all stresses other than fatigue allowables, it is acceptable to calculate the stresses on an elastic basis.
- For comparison to fatigue allowables, it is acceptable to calculate stresses on an elastic basis, except those due to local thermal stresses. For local thermal stresses, the Code requires that a modified value of Poisson’s ratio (ν') be used to account for excess plastic straining that can occur above the material yield stress:

$\nu' = 0.5 - 0.2\,(S_y / S_a)$ (but not less than 0.3)

where:

S_y = yield strength at the mean temperature of the cycle

S_a = value obtained from the applicable fatigue curve for the specified number of cycles of the condition being considered.

Alternate means are provided in NB-3228 for qualifying vessels through use of plastic analysis. For primary stress limits, the design is acceptable if it can be shown by using limit analysis, experimental evaluation, or plastic analysis that the specified loadings do not exceed 2/3 of the collapse load, as defined by the Code. In each case, the Code requires that the effects of localized high strains that might affect fatigue, ratcheting, or buckling behavior be evaluated. In the case of thermal stress ratchet (NB-3222.5) and progressive distortion of non-integral connections (NB-3227.3), NB-3228.4 states that limits on local membrane stress, total secondary stress, thermal ratchet and progressive distortion need not be met, provided that a plastic analysis shows that shakedown occurs.

NB-3228.5 contains the rules for performing a simplified elastic-plastic analysis when the range of primary plus secondary stress intensity exceeds $3S_m$. This analysis can be used in lieu of a full plastic analysis and requires that the following be met:

- The range of primary membrane plus bending stress intensity, excluding thermal bending stresses, must be less than $3S_m$.
- The value of S_{alt} used in entering the applicable fatigue curve must be multiplied by the factor K_e, where:

$K_e = 1.0$ for $S_n \le 3S_m$

$K_e = 1.0 + \dfrac{1+n}{n(m-1)}\left(S_n / 3S_m - 1\right)$ for $3S_m < S_n < 3mS_m$

$K_e = 1/n$ for $S_n \ge 3mS_m$

where

S_n = range of primary plus secondary stress intensity, psi

m,n = material parameters from Table NB-3228.5(b)-1 (See Table 2-1)

- If this procedure is used, the Poisson ratio adjustment of NB-3227.6 need not be used.
- Thermal stress ratchet requirements of NB-3222.5 must be met.
- The ratio of the minimum yield strength to minimum tensile strength must be less than 0.8.

Table 2-1
Simplified Elastic Plastic Analysis Material Parameters

Material	m	n	T_{max}, F
Carbon Steel	3.0	0.2	700
Low-Alloy Steel	2.0	0.2	700
Martensitic Stainless Steel	2.0	0.2	700
Austenitic Stainless Steel	1.7	0.3	800
Nickel-Chrome-Iron	1.7	0.3	800
Nickel-Copper	1.7	0.3	800

Figure 2-1 shows a plot of K_e as a function of the ratio of $S_n / 3S_m$ for the materials shown.

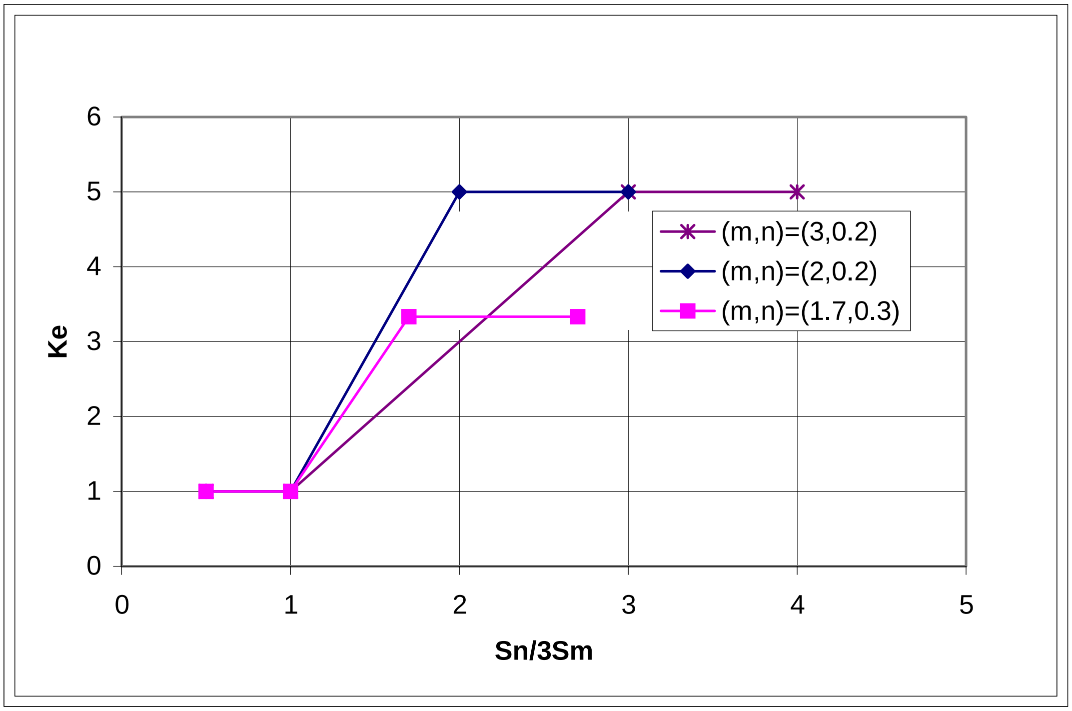

Figure 2-1
Simplified Elastic-Plastic K_e Factor

2.2 Piping Analysis

Similar to the requirements for vessels, the design requirements for Section III, Class 1 for piping components are based on the maximum shear stress theory. The design is considered to be acceptable if the design passes a series of equations for the various loadings to which the component is exposed. The introduction to Reference 3 includes a discussion of the Class 1 piping design criteria and philosophy.

A primary stress intensity limit is provided to show that the design is acceptable for load-controlled (primary) loadings. The primary stress intensity limit is satisfied if Equation 9 is met:

$$B_1 \frac{PD_o}{2t} + B_2 \frac{D_o}{2I} M_i \leq 1.5S_m \qquad \text{(Section III, Cl. 1, Eq. 9)}$$

where:

B_1, B_2 = primary stress indices for the specific product under investigation

P	=	design pressure, psi
D_o	=	outside diameter of pipe, in.
t	=	nominal wall thickness of product, in.
I	=	moment of inertia, in^4
M_i	=	resultant moment due to a combination of Design Mechanical Loads, in-lb.
S_m	=	basic allowable design stress intensity value, psi

For loading conditions classified as Service Level B, the above equation must also be met, except that the stress intensity limit might be increased from 1.5 S_m to 1.8 S_m using Service Level B coincident pressure and moments.

The remainder of the equations for Service Levels A and B are provided to assure satisfactory behavior for cyclic operation. To satisfy the range of primary plus secondary stress intensity (which will assure that shakedown occurs and that excessive distortion does not occur), Equation 10 is provided. The calculation of the stress intensity range is based upon the effect of changes which occur in mechanical or thermal loadings that take place as the system goes from one load set condition to any other. Equation 10 must be evaluated for all pairs of load sets:

$$S_n = C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o}{2I} M_i + C_3 E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right| \leq 3S_m \qquad \text{(Section III, Cl. 1, Eq. 10)}$$

where:

C_1, C_2, C_3	=	secondary stress indices for the specific component under investigation
D_o, t, I, S_m	=	as defined for Equation 9

P_o = range of service pressure, psi

M_i = resultant range of moment that occurs when the system goes from one service load set to another, in-lb.

E_{ab} = average modulus of elasticity of the two sides of material or structural discontinuities at room temperature, psi

α_a, α_b = room temperature coefficient of thermal expansion on side a and side b of a structural or material discontinuity, in / in-°F

T_a, T_b = range of average temperatures on side a and side b of a structural discontinuity, when the system goes from one service load set to another, °F. (Specific rules are provided to define the distance over which the average temperature is determined.)

If Equation 10 cannot be satisfied for all pairs of load sets, the alternative analysis of NB-3653.6 or evaluation using NB-3200 might still permit qualifying the component. Only those pairs of load sets that do not satisfy Equation 10 need to be considered.

The fatigue resistance of the component is assessed by evaluating the range of peak stress intensity. For every pair of load sets, the range of peak stress intensity is calculated using Equation 11:

$$S_p = K_1 C_1 \frac{P_o D_o}{2t} + K_2 C_2 \frac{D_o}{2I} M_i + K_3 C_3 E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right|$$

$$+ \frac{1}{2(1-\nu)} K_3 E\alpha \left| \Delta T_1 \right| + \frac{1}{1-\nu} E\alpha \left| \Delta T_2 \right| \qquad \text{(Section III, Cl. 1, Eq. 11)}$$

where:

K_1, K_2, K_3 = local stress indices for the specific component under investigation

$E\alpha$ = modulus of elasticity (E) times the mean coefficient of thermal expansion (α), both at room temperature, psi °F

ΔT_1 = range of the temperature difference for each load set combination between the temperature of the outside surface T_o and the temperature of the inside surface T_i of the piping product assuming a moment generating equivalent linear temperature distribution, °F

ΔT_2 = range for that portion of the non-linear thermal gradient through the wall thickness not included in ΔT_1, °F

As stated above, NB-3653.6 contains special rules that can be used when equation 10 exceeds $3S_m$:

(a) Equation 12 shall be met:

$$S_e = C_2 \frac{D_o}{2I} M_i^* \leq 3S_m \qquad \text{(Section III, Cl. 1, Eq. 12)}$$

where:

S_e = nominal value of expansion stress, psi

M_i^* = similar to M_i in Equation 10, except that it includes only moments due to thermal expansion and thermal anchor movements, in-lb.

(b) To avoid thermal stress ratchet, it must be shown that the range of ΔT_1 cannot exceed that calculated per NB-3653.7:

$$\Delta T_1 \leq \frac{y' S_y}{0.7E\alpha} C_4$$

where:

y' = 3.33, 2.00, 1.20, and 0.80 for x = 0.3, 0.5, 0.7, and 0.8, respectively

x = $(PD_o / 2t)\,(1 / S_y)$

P = maximum pressure for the set of conditions under consideration, psi

C_4 = 1.1 for ferritic material

= 1.3 for austenitic material

$E\alpha$ = as defined for Equation 11, psi / °F

S_y = material yield strength value, psi, taken at average fluid temperature of the transients under consideration

(c) The primary plus secondary membrane plus bending stress intensity, excluding thermal bending and thermal expansion stresses, shall not exceed $3S_m$. This requirement is satisfied by meeting Equation 13:

$$C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o M_i}{2I} + C_3' E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right| \leq 3S_m \quad \text{(Section III, Cl. 1, Eq. 13)}$$

where:

M_i = moment as defined for Equation 9, in-lb, and all other terms as previously described

C_3' = stress index (NB-3680)

(d) If these conditions are met, the value of S_{alt} shall be calculated by Equation 14:

$$S_{alt} = K_e \frac{S_p}{2} \quad \text{(Section III, Cl. 1, Eq. 14)}$$

where:

S_{alt} = alternating stress intensity, psi

S_p = peak stress intensity range calculated by Equation 11, psi

K_e = 1.0 for $S_n \leq 3S_m$

= $1.0 + [(1 - n) / n(m - 1)](S_n / 3S_m - 1)$, for $3\,S_m < S_n < 3mS_m$

= $1 / n$, for $S_n \geq 3mS_m$

S_n = primary plus secondary stress intensity value calculated in Equation 10, psi

m,n = material parameters provided in Table NB-3228.5(b)-1

The alternating stress intensity (S_{alt}) for all load sets is computed as one-half of the peak stress ranges calculated from Equation 11, or by the alternate approach of Equation 14 if Equation 10 is not met. The fatigue analysis is then performed per the requirements of NB-3222.4(e)(5) using the applicable Code fatigue curve and the design number of cycles for each loading from the design specification. Criteria are provided in NB-3630 that allow Class 1 piping to be designed to Class 2 rules, with no fatigue evaluation, if the cyclic operating conditions are limited.

It should be noted that for ASME Section III Code editions prior to the Summer 1979 Addenda, Equation 10 contained an additional term. In these earlier Code editions, the ΔT_1 term of the peak stress Equation 11 was also included in Equation 10:

$$S_n = C_1 \frac{P_o D_o}{2t} + C_2 \frac{D_o}{2I} M_i + C_3 E_{ab} \left| \alpha_a T_a - \alpha_b T_b \right|$$

$$+ \frac{E\alpha \left| \Delta T_1 \right|}{2(1-\nu)} \leq 3S_m \qquad \text{(Section III, Cl. 1, Eq. 10)}$$

Addition of this term frequently increased the stress intensity range, S_n, above $3S_m$. When this occurred, Equations 12 and 13 had to be met and the fatigue analysis had to be conducted using a relatively high K_e factor, increasing the alternating stresses used in the fatigue analysis. The ASME Section III Working Group on Piping Design justified that this was over-conservative and modified the equation accordingly, starting with the Summer 1979 Addenda. However, most current nuclear plants designed to ASME Section III were designed to the earlier version of the Code.

3

SUMMARY OF APPROACH DEVELOPED FOR FRENCH RCC-M

WRC Bulletin 361 [2] presents studies made for preparing the evolution of the French RCC-M Pressure Vessel Code. A key element of WRC Bulletin 361 is an alternate approach for evaluating K_e for local thermal stresses.

The approach for thermal stresses is similar to, but different from, the approach taken in NB-3227.6. Given that stresses are computed on an elastic basis, there needs to be a correction factor applied if the strain range exceeds twice the yield stress value. To accomplish this, a factor K_ν is developed that accounts for the incompressibility of metals for strains above the yield stress. Two approaches were considered:

1. Equivalent strain yielding criteria

$$K_\nu = (1-\nu)/(1+\nu) \times (1+\nu^*)/(1-\nu^*)$$

and,

2. Tresca yielding criteria

$$K_\nu = (1-\nu)/(1-\nu^*)$$

where:

ν = linear elastic Poisson's ratio (normally 0.3)

ν^* = effective Poisson's ratio considering plasticity

The effective Poisson's ratio, ν^* is developed based on the secant modulus of elasticity, E_s, for a cyclic stress-strain curve (expressed in total stress and strain range coordinates).

$$\nu^* = 0.5 - (0.5 - \nu)(E_s / E)$$

where:

E = modulus of elasticity

E_s = secant modulus including effects of plasticity

The secant modulus is determined by dividing the plastic stress intensity range by the plastic strain range for a load set pair. Thus, if the stress intensity range is less than $3S_m$ (remaining elastic), $E_s = E$, and $\nu^* = \nu$. When the stress intensity range includes same plasticity, the effective modulus of elasticity reduces, leading to an increase in ν^*. This is shown in Figure 3-1.

It was recognized that there is a distinction between non-displacement-limited mechanical loading effects and local displacement-limited thermal loadings effects. To account for the effects of mechanical loadings, the proposed approach included a modified strain correction factor $K_e^{'}$.

$$K_e' = K_\nu (S_{p,therm} / S_p) + K_e (S_{n,mech} / S_n)$$

where:

K_ν = correction factor accounting for Poisson's ratio variation

S_p = total primary-plus-secondary-plus-peak stress intensity range

$S_{p,therm}$ = thermal primary-plus-secondary-plus-peak stress intensity range

K_e = simplified elastic-plastic correction factor from ASME Code

S_n = total primary-plus-secondary-stress intensity range

$S_{n,mech}$ = mechanical primary-plus-secondary-stress intensity range

In the above, $S_{p,therm}$ would apply to the localized thermal stresses and presumably not to the overall piping thermal expansion stresses. The fatigue analysis would be conducted (per Code requirements) using K_e' instead of K_e.

In addition to the thermal-plus-mechanical loading effects, strain concentration can occur at structural discontinuities. Hence, an additional local correction using a Neuber's Rule approach was also suggested in WRC Bulletin 361.

From elastic analysis, the primary-plus-secondary stress intensity range is calculated as S_n. The corresponding elastic strain range ($\Delta\varepsilon_{en}$) according to Hook's law is S_n / E. When S_n is greater than $3S_m$, $\Delta\varepsilon_{en}$ should be enlarged due to effects like K_e'. However, WRC 361 suggests that the thermal part of the K_e' be changed to $K_\nu \times (S_{n,mech} / S_n)$ and called K_e^*. The enlarged strain range is then $\Delta\varepsilon_n = K_e^* \times \Delta\varepsilon_{en}$ and the corresponding plastic stress range is $\Delta\sigma_n$. These are the plastic stress and strain without strain concentration.

WRC 361 chose to use the Neuber's rule to estimate the effects of plastic stress and strain concentration, that is, $K_\varepsilon \times K_\sigma = K_T^2$. Here, the theoretical or elastic stress concentration factor $K_T = S_p / S_n$ is assumed to be equal to the geometric mean value of the plastic strain concentration factor (K_ε) and the plastic stress concentration factor (K_σ). Hence, the actual plastic strain ($\Delta\varepsilon_{ep}$) and stress ($\Delta\sigma_{ep}$) range after strain concentration is calculated by:

$$\Delta\sigma_{ep} \times \Delta\varepsilon_{ep} = \Delta\sigma_n \times \Delta\varepsilon_n \times K_T^2$$

If it is assumed that the material elastic-plastic relationship $\sigma = A\varepsilon^n$ holds for $\Delta\sigma_{ep}$ and $\Delta\varepsilon_{ep}$, the plastic strain range, $\Delta\varepsilon_{ep}$, might be calculated directly. WRC 361 suggests that $\Delta\varepsilon_{ep}$ should be compared to the strain range associated with the Code fatigue curve.

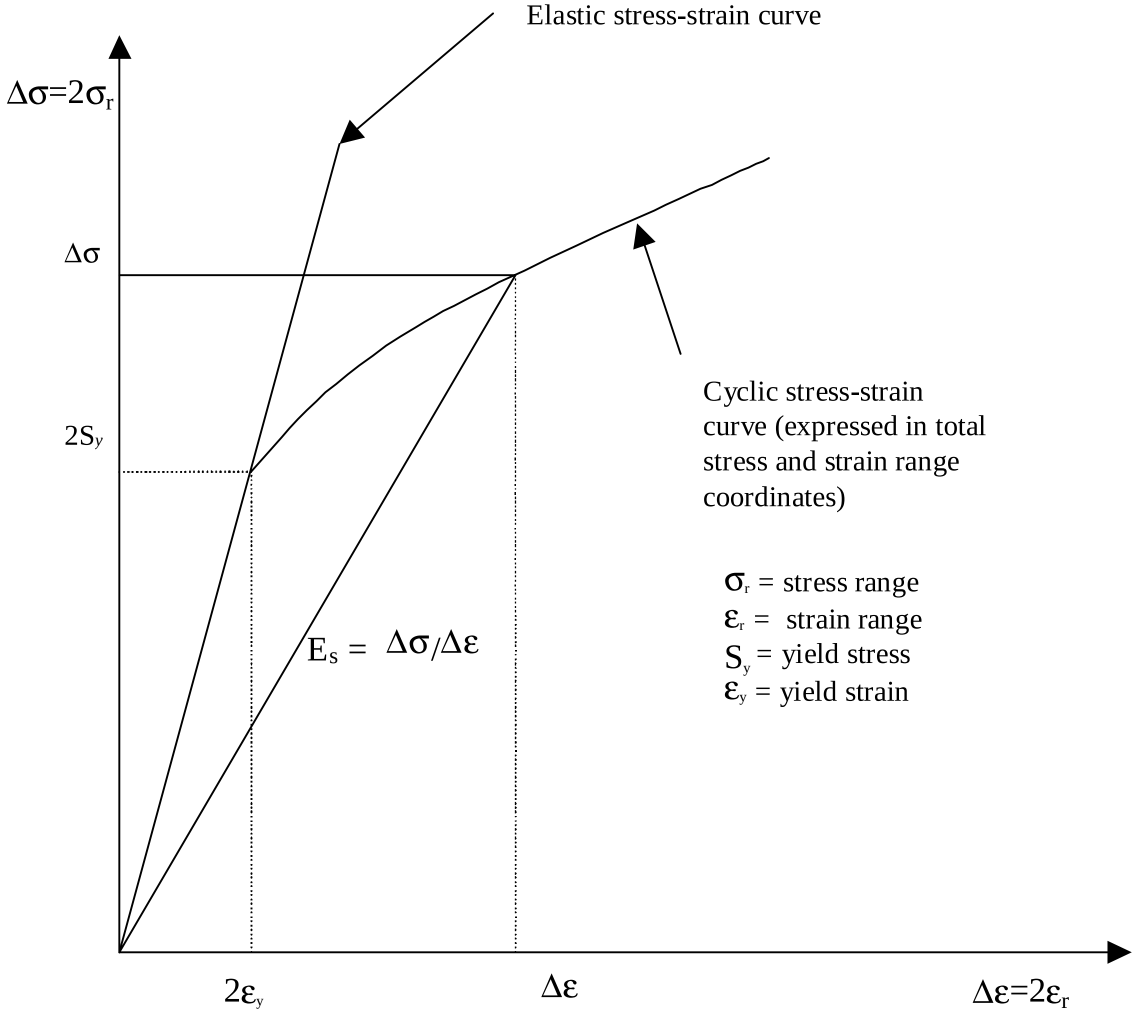

Figure 3-1
Definition of Secant Modulus

4

PLASTIC ANALYSIS AND COMPARISON TO AN ALTERNATE SIMPLIFIED APPROACH

In this section, the strain correction factor, $K_e = \Delta\varepsilon_{ep} / \Delta\varepsilon_e$, calculated directly from elastic-plastic finite element analyses will be compared to strain correction factors calculated from the simplified elastic-plastic approach in the current ASME Code. In addition, comparisons will be made to the alternate simplified approach discussed in the previous section, as developed in WRC Bulletin 361[2]. Though the alternate approach is based on the procedure in WRC 361, it is a unified approach considering all of the effects of localized thermal loading, elastic follow-up, and notches in one formulation. It simplifies the procedure by including the constitutive relation, $\sigma = A\varepsilon^n$, in the formulation directly.

4.1 Sample Problems and Finite Element Model Description

Three finite element models of axisymmetric structures under pressure and thermal loads have been analyzed using the ANSYS finite element program [3]. The three models are a straight pipe (Figure 4-1), a pipe with inner radius transition (Figure 4-2, identified as a transition pipe in the following text), and a pipe with outer radius transition (Figure 4-3, identified as a nozzle in the following text).

The straight pipe is an 8-inch Schedule 160 pipe with inner radius of 3.4065 inches, outer radius of 4.3125 inches, and wall thickness of 0.906 inches. The transition pipe is an 8-inch Schedule 160 pipe transition to 8-inch Schedule 80 pipe, that is, inner radius transition from 3.4065 inches to 3.8125 inches (or thickness change from 0.906 inches to 0.5 inches with outer radius unchanged at 4.3125 inches). The transition length to thickness ratio in both cases has been taken as 3:1. The nozzle has the same geometry as the thin section of the transition pipe but the transition is on the outer radius, that is, straight on the inner radius and transition on the outer radius (see Figure 4-3).

The material properties used are for typical stainless steel. The Young's modulus (E) is 28.3×10^6 psi. The Poisson's ratio (ν) is 0.3. The yield stress (σ_y) is 30 ksi. For plastic analysis, the stress-strain relation of $\sigma = A\varepsilon^n$ is used. For stainless steel, the material parameter n is taken from Table NB-3228.5(b)-1 of the ASME Code as 0.3. Assuming $\sigma_y = E\varepsilon_y$, the material constant A is equal to 234,168 psi.

To evaluate the effects of significant thermal / mechanical load cycling, a postulated transient was evaluated with the pressure and temperature histories on the inner of the pipe defined in Figure 4-4. The magnitude of the maximum internal pressure was set to generate hoop membrane stresses near the level of the allowable stress intensity of the material (S_m = 20 ksi for typical stainless steel). Temperature variation ranges were assumed to be from 200°F to 600°F. A sudden rising and dropping of inner temperatures (thermal shock) were specified, and stress intensity ranges (S_n) larger than $3S_m$ were expected. The pressure and temperature histories were arbitrarily assumed to be out-of-phase so that maximum S_n values could be physically achieved.

Coupled-field solid elements (PLANE13 in ANSYS) were used to construct the finite element models (Figures 4-1 to 4-3). Temperatures and displacements were solved simultaneously by these types of elements. The inner temperature was prescribed as in Figure 4-4. A heat transfer coefficient of 2,000 Btu / hr-ft^2-°F was assumed on the pipe inside surface. All the other boundaries were thermally insulated. For plastic behavior, discrete points on the $\sigma = A\varepsilon^n$ curve were input to ANSYS to define the kinematic strain hardening relations as shown in Table 4-1.

4.2 Load Cases

To calculate the ratio between plastically calculated and elastically calculated strains, $K_e = \Delta\varepsilon_{ep} / \Delta\varepsilon_e$, finite element analysis using plastic material properties and elastic material properties was performed separately for all of the combinations of pressure and thermal loads. The thermal transient ranges selected were ΔT = 200°F, 325°F, 450°F, and 600°F. A pressure range (ΔP) was selected as 0 to 5,320 psi for the straight pipe, and 0 to 2,625 psi for the transition pipe and the nozzle, so that primary membrane stresses were close to S_m. Table 4-2 shows the load cases analyzed to calculate K_e directly from its definition and to implement the alternate simplified approach. The locations where strain values are used to calculate the K_e are on the inner surface, at the mid-point of the straight pipe section and at the locations where thickness changes for the transition pipe and the nozzle, as shown in Figures 4-1 to 4-3.

4.3 Post-Processing of Finite Element Analysis Results

The procedures of post-processing to calculate K_e by the ASME Code definition, and by using the alternate simplified approach, are discussed separately as follows.

4.3.1 K_e Calculated by Definition

The strain correction factor due to plasticity is defined as $K_e = \Delta\varepsilon_{ep} / \Delta\varepsilon_e$. Here, $\Delta\varepsilon_e$ is the equivalent elastic strain range directly from the elastic finite element analysis. The numerator nominator, $\Delta\varepsilon_{ep}$, is the equivalent strain range calculated from the plastic analysis. The equivalent strain definition consistent with Tresca criteria is

$$\Delta\varepsilon_{eq} = \max \frac{1}{1+\nu}\left|\Delta\varepsilon_i - \Delta\varepsilon_j\right|$$

The equivalent strain definition consistent with von Mises criteria is

$$\Delta\varepsilon_{eq} = \tfrac{\sqrt{2}}{2(1+\nu)}[(\Delta\varepsilon_x - \Delta\varepsilon_y)^2 + (\Delta\varepsilon_y - \Delta\varepsilon_z)^2 + (\Delta\varepsilon_z - \Delta\varepsilon_x)^2 + \frac{3}{2}(\Delta\gamma_{xy}^2 + \Delta\gamma_{yz}^2 + \Delta\gamma_{zx}^2)]^{1/2}$$

They both have Poisson's ratio included in the formulation. The denominator $\Delta\varepsilon_e$ is the strain range calculated from an elastic analysis. The same formula and the same Poisson's ratio should be used in calculating the equivalent strain ranges from the results of the elastic and the plastic analysis so that $K_e = 1.0$ in the elastic range.

In the plastic analysis, the calculated elastic strain components and plastic strain components from ANSYS were added up algebraically to obtain the total strain components. The time points selected covered at least one complete cycle after the strain response established shakedown to a stable cyclic range. In the cases analyzed, four cycles were usually sufficient to get a stable strain range. Figures 4-5 to 4-7 are examples of the total strain shakedown at the thin section in Figure 4-2, for transients in Figure 4-4 with $\Delta T = 600°F$.

4.3.2. K_e Using Simplified Approach

The simplified approach described in WRC Bulletin 361 has three kinds of strain correction factor calculations: (1) Poisson's effect correction, (2) combined correction for Poisson's effect and elastic follow-up, and (3) combined correction for Poisson's effect, elastic follow-up and strain concentrations.

If the rules in WRC 361 are implemented directly, there is more than one set of formulas for different structure and load configurations. The suggested method in this study is to use only one procedure for all three cases mentioned in WRC 361. It follows the general procedure in case (3) of WRC 361, but with some modifications as described below.

The procedures in (2) and (3) in WRC 361 both considered combined K_v (local thermal plastic effects) and K_e (elastic follow-up effects) for global correction. However, they are combined artificially with different weightings and the calculation of the K_v's is different in the two cases. A more conservative and convenient option will be used in a common approach, regardless of whether or not there is strain concentration.

WRC 361 used Figure 3-1 in its calculation of the secant modulus (E_s). However, the coordinates in the plot were the total stress and strain ranges, which can be tabulated from the given stress-strain relation of $\sigma = A\varepsilon^n$. Assuming that half of the range of the equivalent stress and strain will follow the $\sigma = A\varepsilon^n$ rule, the stress-strain relation might be conveniently substituted directly into the Neuber's rule (Section 3.0) to derive a single formula for all three types of corrections in WRC 361.

In addition, the stress concentration factor K_T (referred to in Section 3) used in the Neuber's rule is mechanically relevant only for the notch effect, as stated in WRC 361. The proposed method in this study will include specific stress terms that should be used in the formulation for the effects of stress concentration.

The procedure of the proposed method to calculate the strain correction factor is summarized as follows:

Step 1: Determine the material constant A of $\sigma = A\varepsilon^n$, assuming the strain at yield $\varepsilon_y = S_y / E$ and $S_y = A\varepsilon_y^n$. Here, n is from Table NB-3228.5(b)-1 in ASME Code. Hence the stress-strain relation becomes

$$\sigma = \frac{S_y}{(S_y / E)^n}\varepsilon^n = S_y\left(\frac{\varepsilon}{\varepsilon_y}\right)^n$$

The values of S_y and E might also be taken from the ASME Code material tables.

Step 2: Calculate the following peak parameters elastically:

S_p = Total primary-plus-secondary-plus-peak stress intensity range

$S_{p,t}$ = Total secondary-plus-peak (excluding thermal expansion) stress intensity range due to local thermal transients

Step 3: Calculate the following elastically:

S_n = primary-plus-secondary stress intensity range

$S_{n,t}$ = secondary stress intensity range due to local thermal transients

$S_{n,m}$ = primary-plus-secondary stress range due to mechanical loads

$S_{n,m}+S_{p,t}$ = stress intensity range by combining $S_{n,m}$ and $S_{p,t}$ at component level

Step 4: Calculate E_s, and ν^*

Since $E_s = \Delta\sigma \dfrac{E}{S_{n,m}+S_{p,t}}$ (Figure 3-1) and $\Delta\sigma/2 = \dfrac{S_y}{(S_y/E)^n}\left(\dfrac{(S_{n,m}+S_{p,t})/2}{E}\right)^n$ (Step 1), for $2S_y = 3S_m$, then $E_s = \left(\dfrac{3S_m}{S_{n,m}+S_{p,t}}\right)^{1-n} \times E$, but not larger than E. Note that $S_{n,m} + S_{p,t}$ is used, instead of S_n in case 3. The $S_{n,m} + S_{p,t}$ is equal to S_p for the case with thermal loads only.

Assuming that the volume variation is the same in the elastic and the plastic analyses, then E_s, and ν^* are related to E and ν by the relation of $\dfrac{1-2\nu^*}{E_s} = \dfrac{1-2\nu}{E}$. For $\nu = 0.3$ in elasticity,

$$\nu^* = 0.5 - 0.2\left[\frac{3S_m}{S_{n,m}+S_{p,t}}\right]^{1-n}$$

but not less than 0.3.

Step 5: Calculate K_ν

According to WRC 361, $K_\nu = \dfrac{(1-\nu)(1+\nu^*)}{(1+\nu)(1-\nu^*)}$, based on von Mises criteria,

or $K_\nu = \dfrac{1-\nu}{1-\nu^*}$, based on Tresca criteria.

As it will be demonstrated later, the former one is more conservative and is used in the unified approach proposed herein.

Step 6: Compute

$$K_e^{'} = K_\nu \ x \ \frac{S_{p,t}}{S_{n,m} + S_{p,t}} + K_e \ x \ \frac{S_{n,m}}{S_n}$$

Here K_e is the multiplying factor in NB-3228.5.

Step 7: Compute the plastic strain range ($\Delta\varepsilon_n$) including the thermal peak but excluding the notch effect.

$$\Delta\varepsilon_n = K_e^{'}\left(\frac{S_{n,m} + S_{p,t}}{E}\right)$$

Step 8: Compute stress / strain concentration factor due to notch effect,

$$K_T = \frac{S_p}{S_{n,m} + S_{p,t}} = \frac{\text{peak stress intensity (with notch effect)}}{\text{total stress intensity (without notch effect)}}$$

Step 9: Compute the total plastic strain range ($\Delta\varepsilon_{ep}$) based on Neuber's rule.

Neuber's rule states that $\Delta\sigma_{ep} \cdot \Delta\varepsilon_{ep} = \Delta\sigma_n \cdot \Delta\varepsilon_n \cdot K_T^2$. Here, $\Delta\sigma_{ep}$ and $\Delta\varepsilon_{ep}$ are the total plastic range values of stress and strain, and $\Delta\sigma_n$ and $\Delta\varepsilon_n$ are plastic ranges without notch effects. As $\Delta\sigma_{ep}/2$ and $\Delta\varepsilon_{ep}/2$, as well as $\Delta\sigma_n/2$ and $\Delta\varepsilon_n/2$, should follow the stress-strain relations described in step 1, $\Delta\varepsilon_{ep}$ and $\Delta\varepsilon_n$ are related by

$$\Delta\varepsilon_{ep} = K_T^{\frac{2}{n+1}} \Delta\varepsilon_n$$

Step 10: Calculate the final strain correction factor as

$$K_{ep} = \frac{\Delta\varepsilon_{ep}}{S_p / E} = \left(\frac{S_p}{S_{n,m} + S_{p,t}} \right)^{\left(\frac{1-n}{1+n}\right)} K_e^{'} = (K_T)^{\left(\frac{1-n}{1+n}\right)} K_e^{'}$$

Alternately check fatigue curve with the alternating stress intensity range $S_{alt} = (\Delta\varepsilon_{ep} / 2)E_{fatigue\ curve}$. Here, $E_{fatigue\ curve}$ is the Young's modulus corresponding to the fatigue curves in Appendix I of the ASME code.

4.4 Results and Discussion

The strain correction factors, by definition and by the unified method, are plotted against the normalized primary plus secondary stress intensity range, $S_n / 3S_m$. The K_e curve defined by NB-3228.3, Simplified Elastic-Plastic Analysis, will also be plotted for comparison.

4.4.1 Thermal Loads Only—K_v Evaluation

The global strain correction factors (K_e by definition and K_v) calculated for the cases with thermal loads only for the straight pipe, the transition pipe, and the nozzle are plotted in Figures 4-8 to 4-13.

In Figure 4-8, the finite element results (K_e by definition) are compared with K_v's based on von Mises and Tresca criteria. The results by Tresca criteria are not always conservative. Hence, the equivalent strain consistent with the von Mises criteria will be used in all of the following calculations.

When comparing K_v curves with the K_e curve in the ASME code (Figure 4-9), the differences in the curves of Figure 4-8 are small. Figures 4-10 to 4-13 are K_v curves for the transition pipe and the nozzle. The calculated K_v's are conservative in all of the cases.

4.4.2 Strain Correction Factor by Unified Method for Thermal Plus Mechanical Loads

The strain correction factors calculated by the unified method for both thermal and mechanical loads are shown in Figures 4-14 to 4-18.

Figure 4-14 is for a straight pipe. The finite element results showed that the strain correction factor should increase with $S_n / 3S_m$, while the results from the simplified methods did not. This is because the increases in $S_n / 3S_m$ in our cases are due to increases in temperature ranges only (Figure 4-4). The pressure cycling range was not changed. The formula of $K_e^{'} = K_v \times \frac{S_{p,t}}{S_{n,m} + S_{p,t}} + K_e \times \frac{S_{n,m}}{S_n}$ in Step 6 shows that the mechanical contribution $S_{n,m}$ will not change while the total S_n increases due to changes in thermal loads so that the weighting of K_e decreases. Note that K_e is large as compared to K_v and changes in K_e term would affect the K_e' more. Thus, the resulting K_e' drops, even though the K_v increases.

Figures 4-15 and 4-16 are for the transition pipe at the thin section and the thick section, respectively. Figures 4-17 and 4-18 are for the nozzle at the thin section and the thick section, respectively. The results from the unified method showed their conservatism as compared to the finite element method results, but were not as over-conservative as the K_e in the current ASME code.

All the calculations of S_n and other stress terms in this study are based on NB-3200 rules of the ASME code. NB-3600 differs from NB-3200 in the definition of S_n by removing the through thickness thermal bending stress in piping as shown in Equation 10 of NB-3653. If Equation 10 is used, the load cases investigated in this study will show that the $S_n / 3S_m$ is always below one, even when the structure is plastically deformed.

Table 4-1
Stress-Strain Relations for Stainless Steel

Points	Strain	Stress, *ksi*
1	0.106%	30
2	0.2%	36.294
3	0.6%	50.462
4	1%	58.819
5	2%	72.415

Table 4-2
Load Cases to Calculate K_e by Definition

Load Cases	Analysis Type	Straight Pipe	Transition Pipe	Nozzle
$\Delta T(200°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta T(325°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta T(450°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta T(600°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta P + \Delta T(200°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta P + \Delta T(325°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta P + \Delta T(450°F)$	Elastic	√	√	√
	Plastic	√	√	√
$\Delta P + \Delta T(600°F)$	Elastic	√	√	√
	Plastic	√	√	√

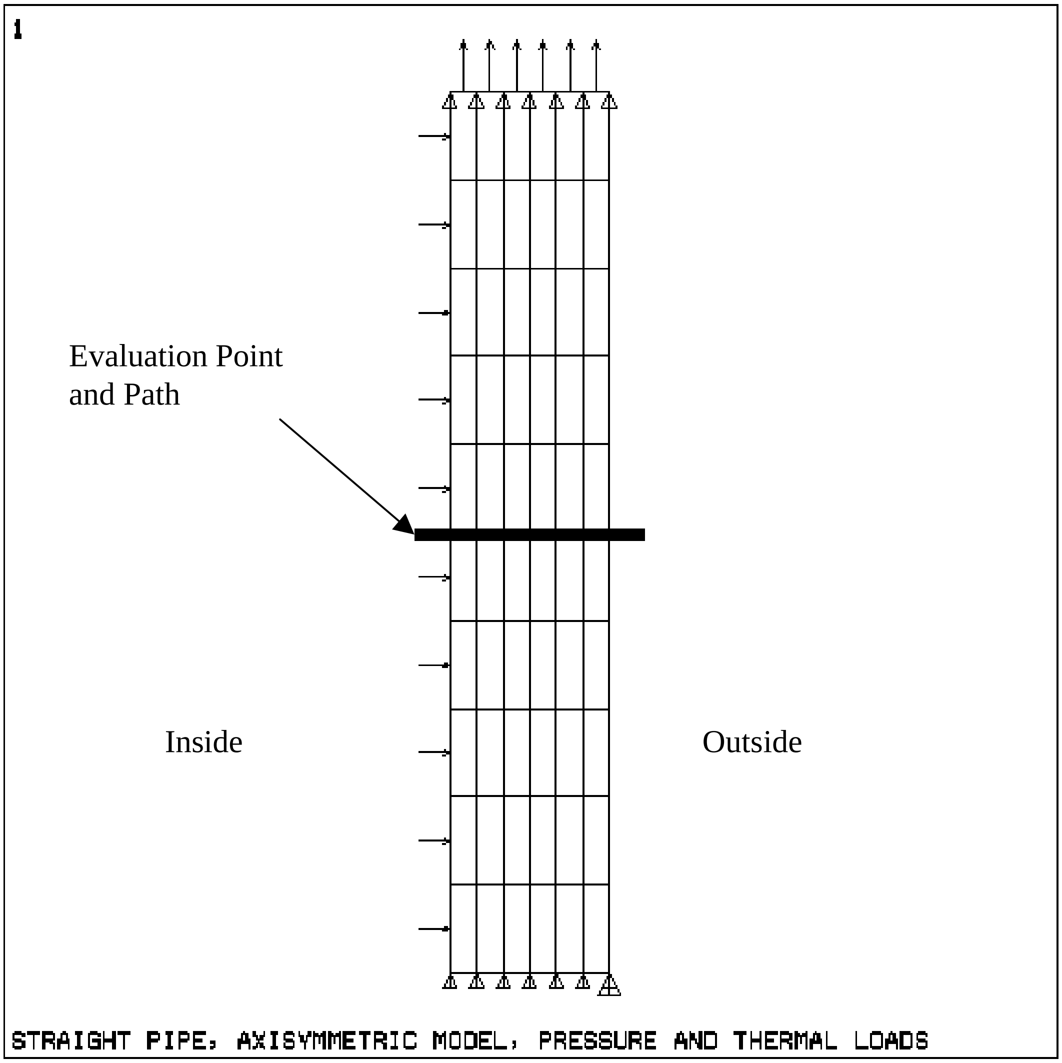

Figure 4-1
Axisymmetric Model of Straight Pipe

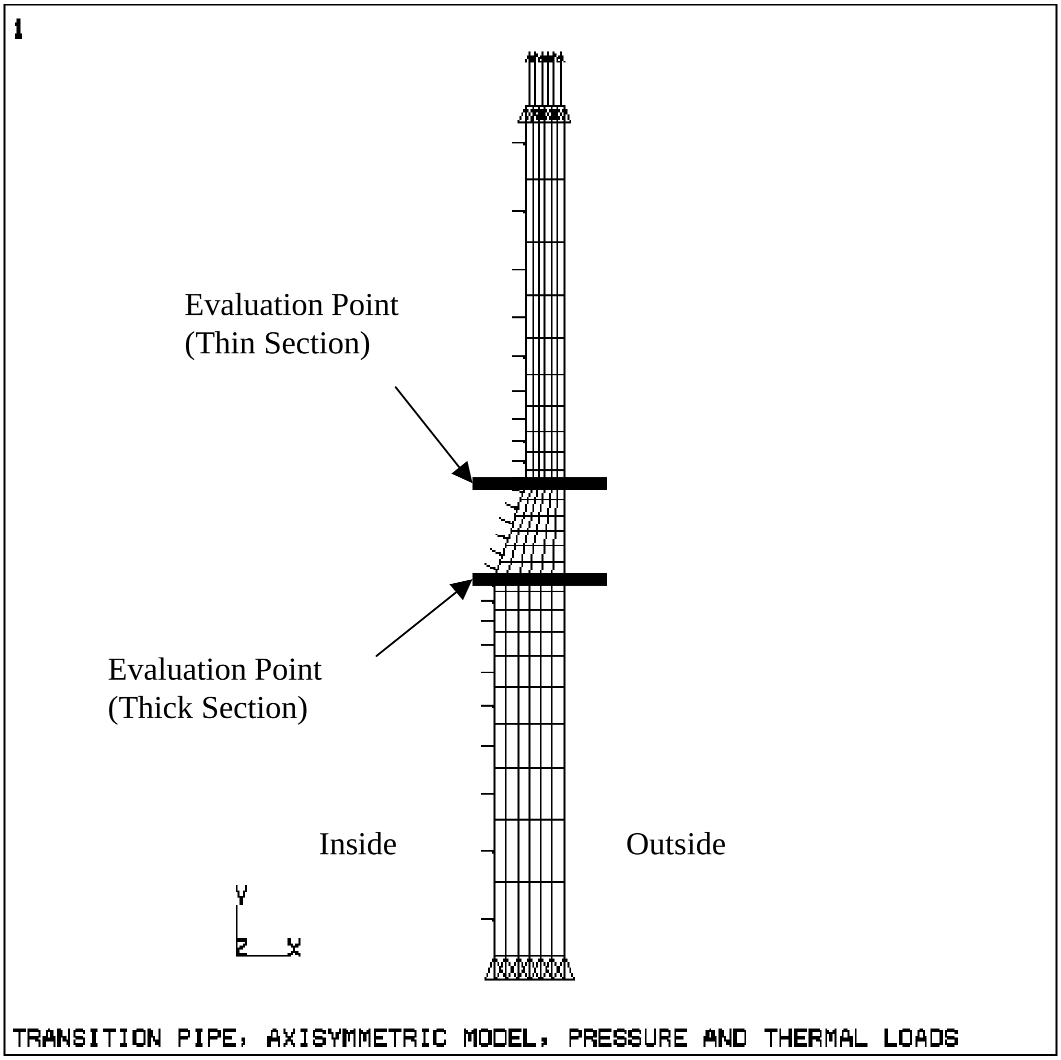

Figure 4-2
Axisymmetric Model of Transition Pipe

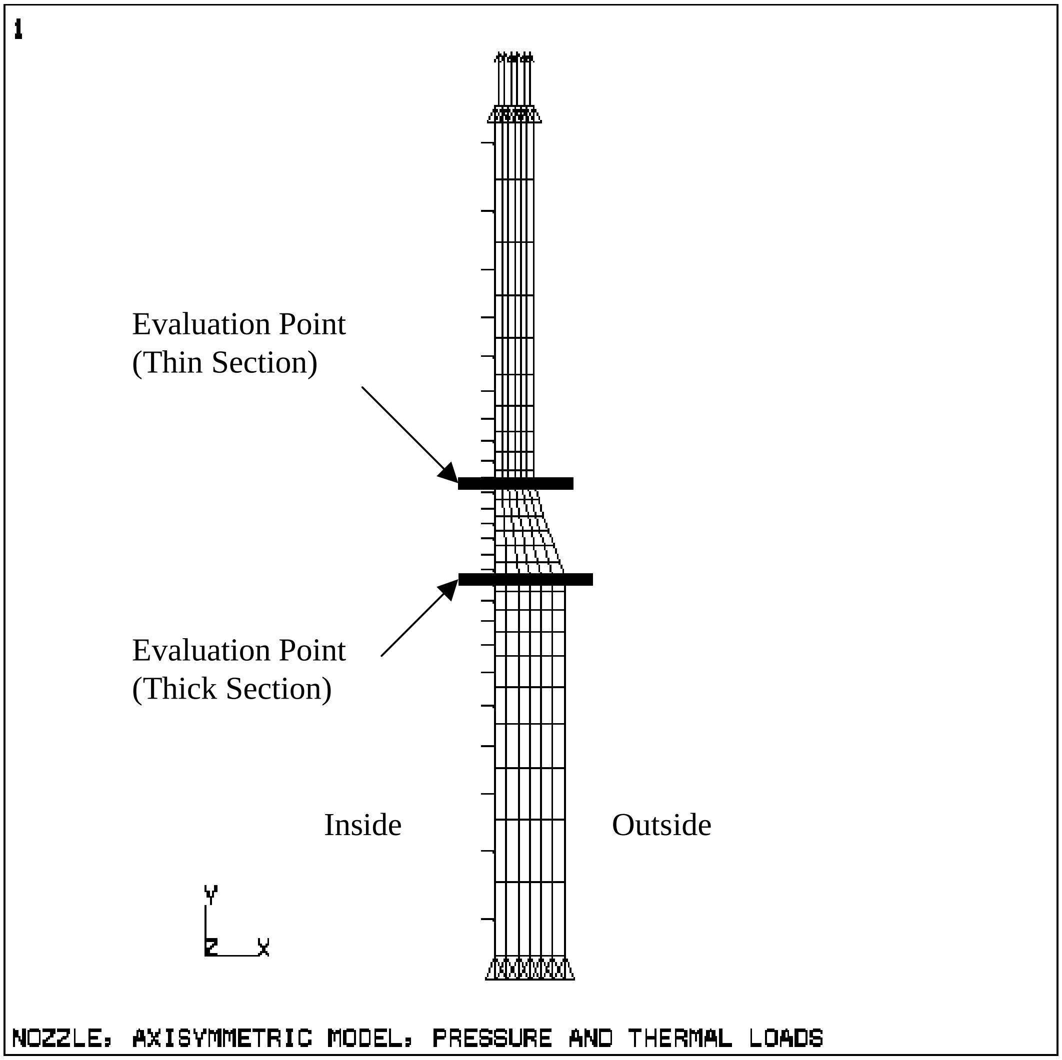

Figure 4-3
Axisymmetric Model of Nozzle

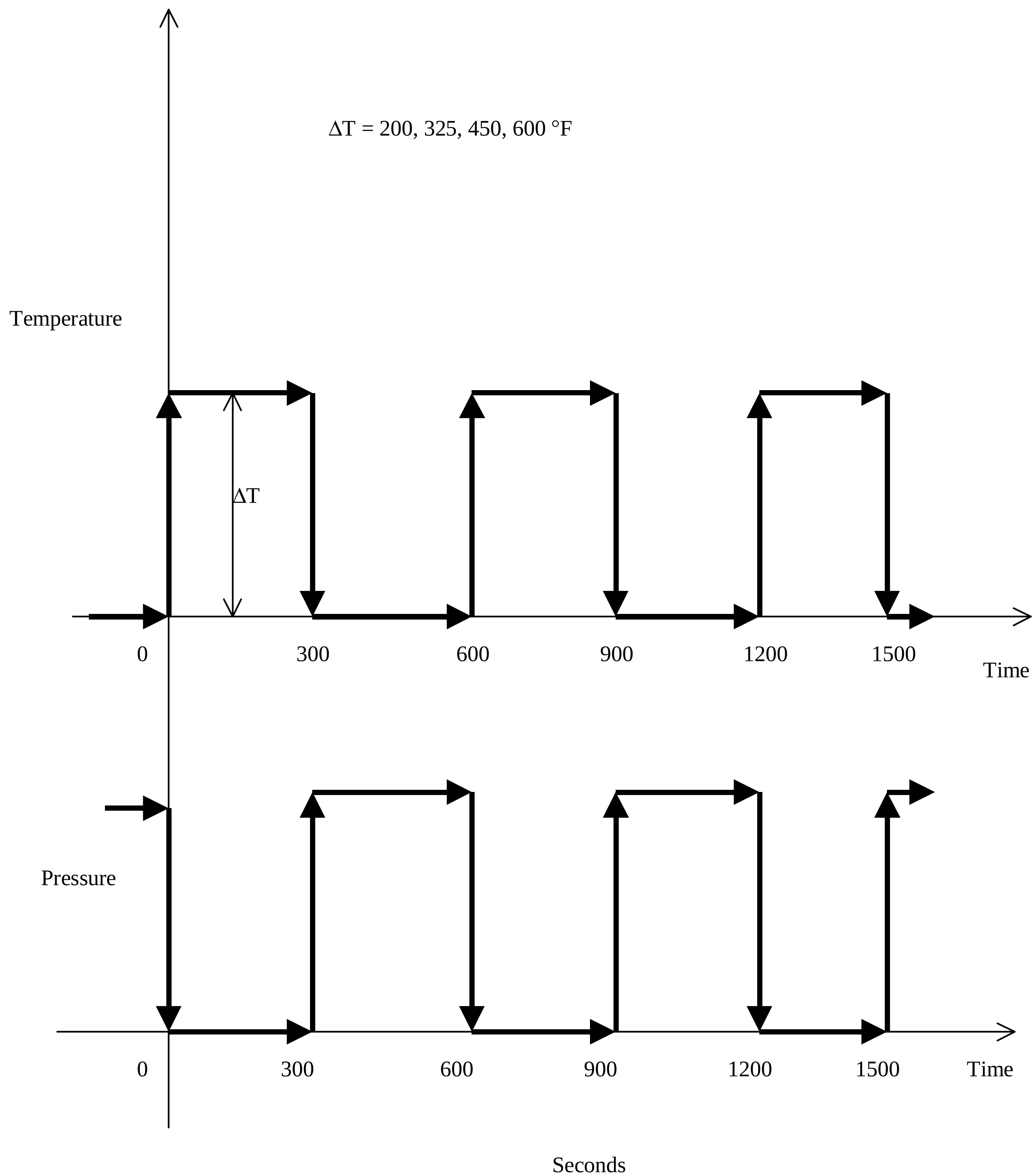

Figure 4-4
Internal Temperature and Pressure Histories

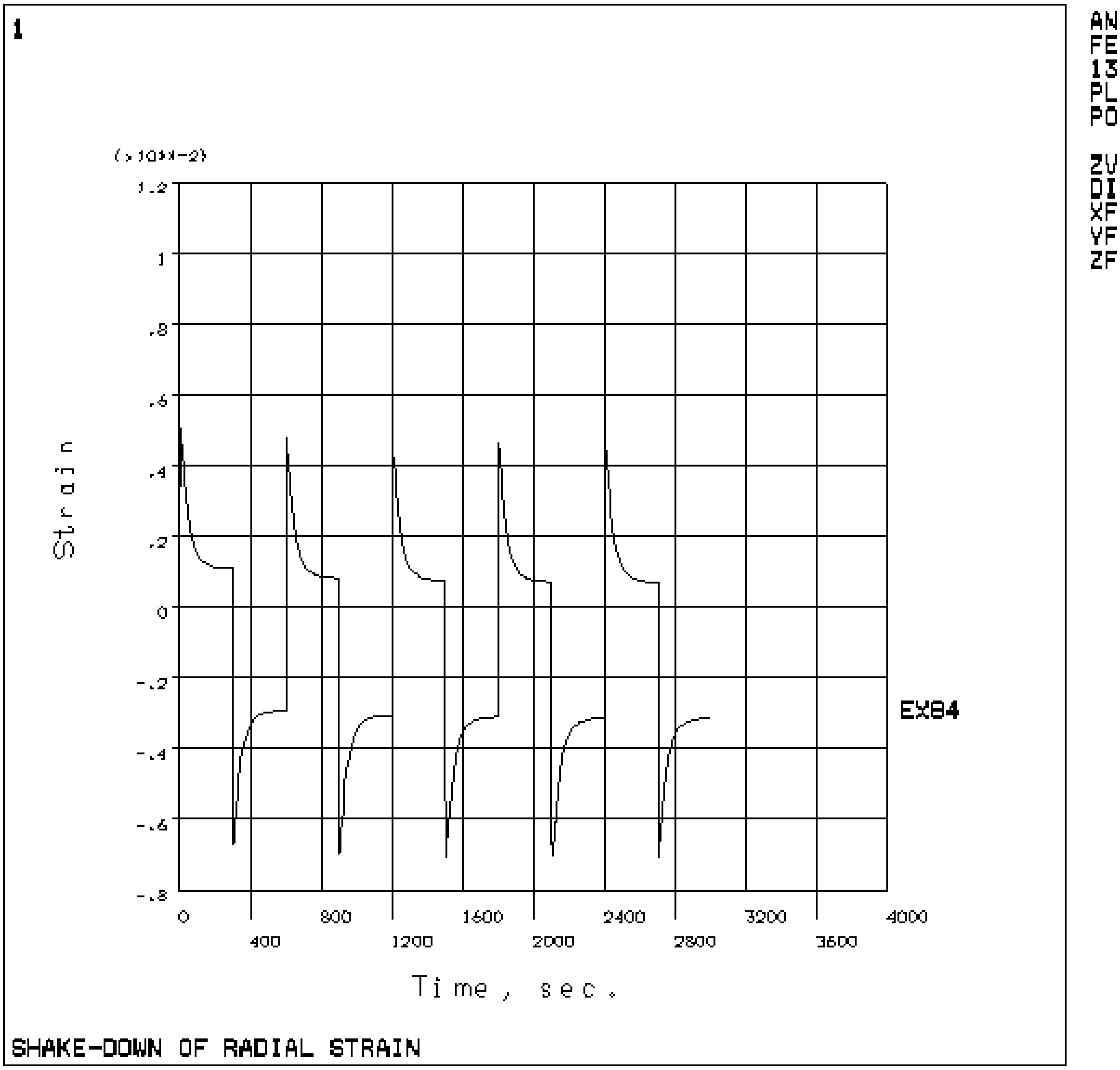

Figure 4-5
Shake-Down Of Total Radial Strain

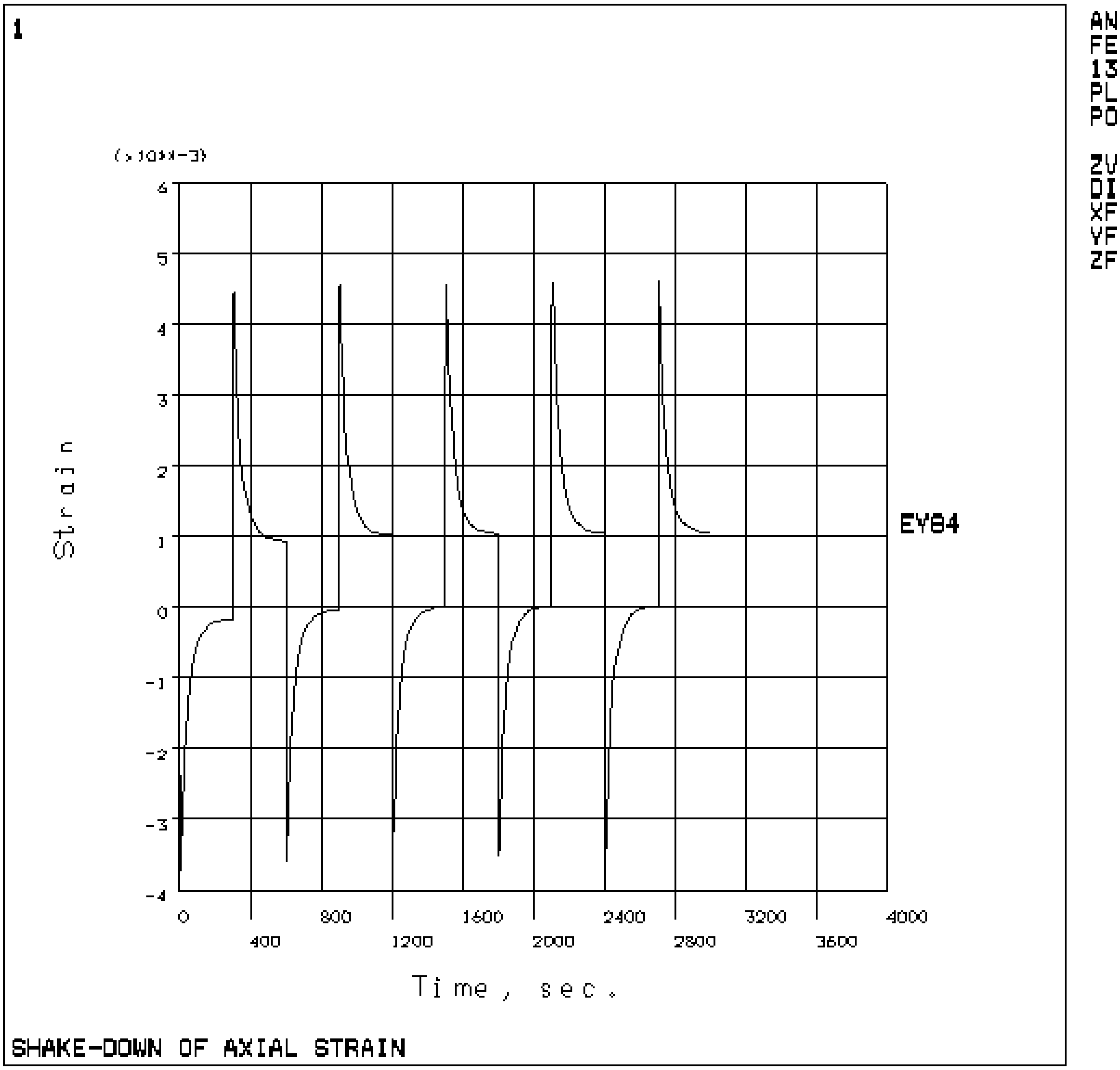

Figure 4-6
Shake-Down Of Total Axial Strain

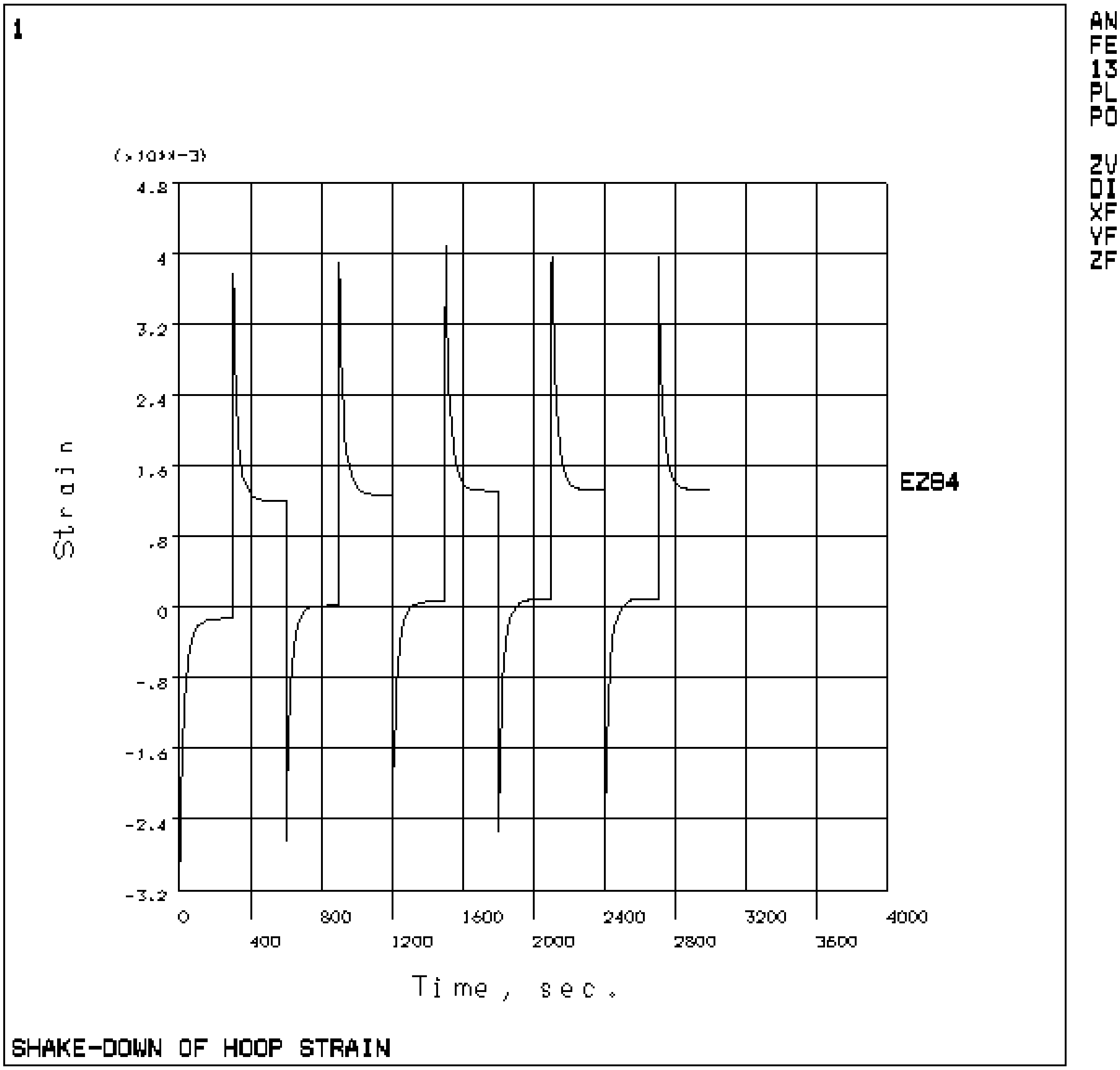

Figure 4-7
Shake-Down Of Total Hoop Strain

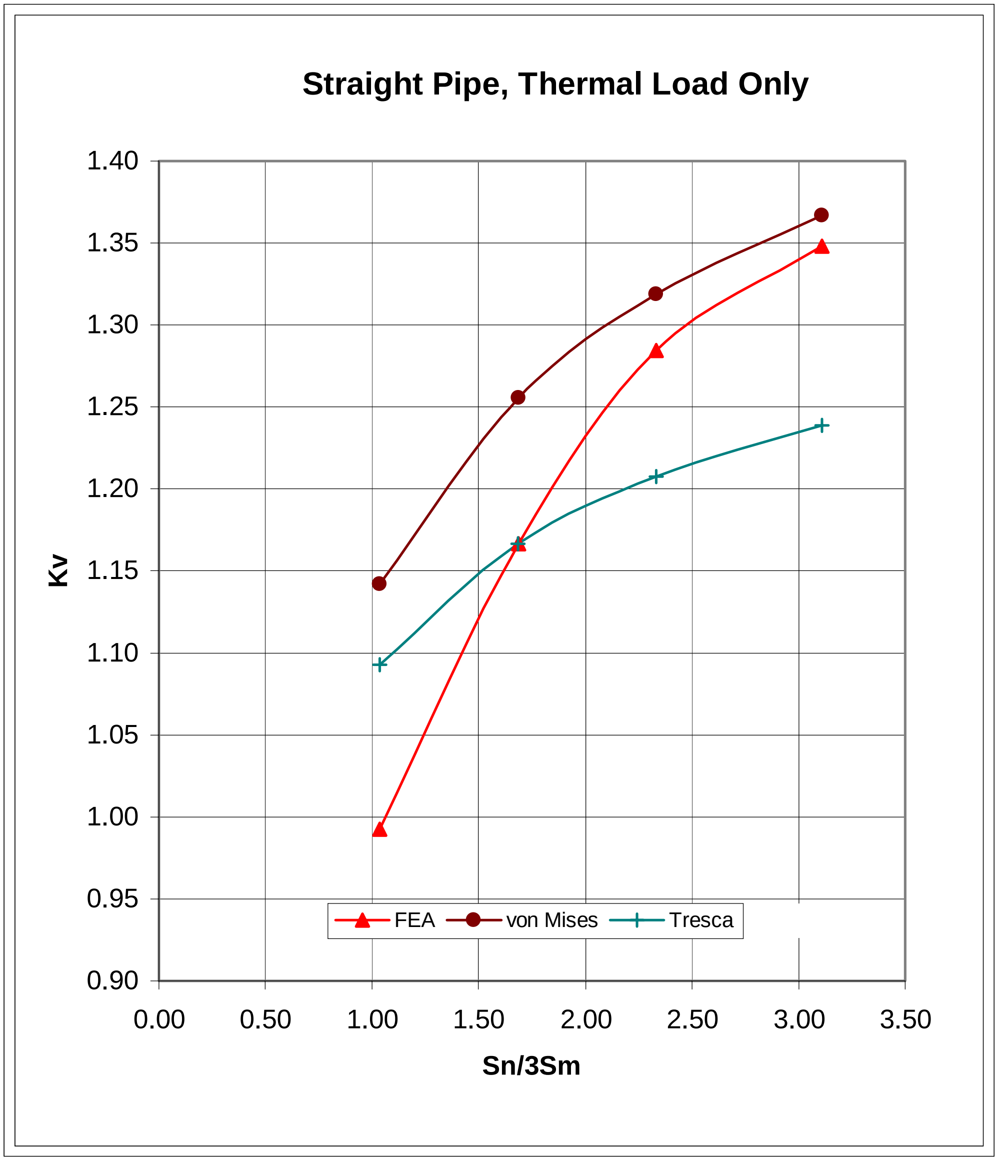

Figure 4-8
Poisson's Effect for Straight Pipe

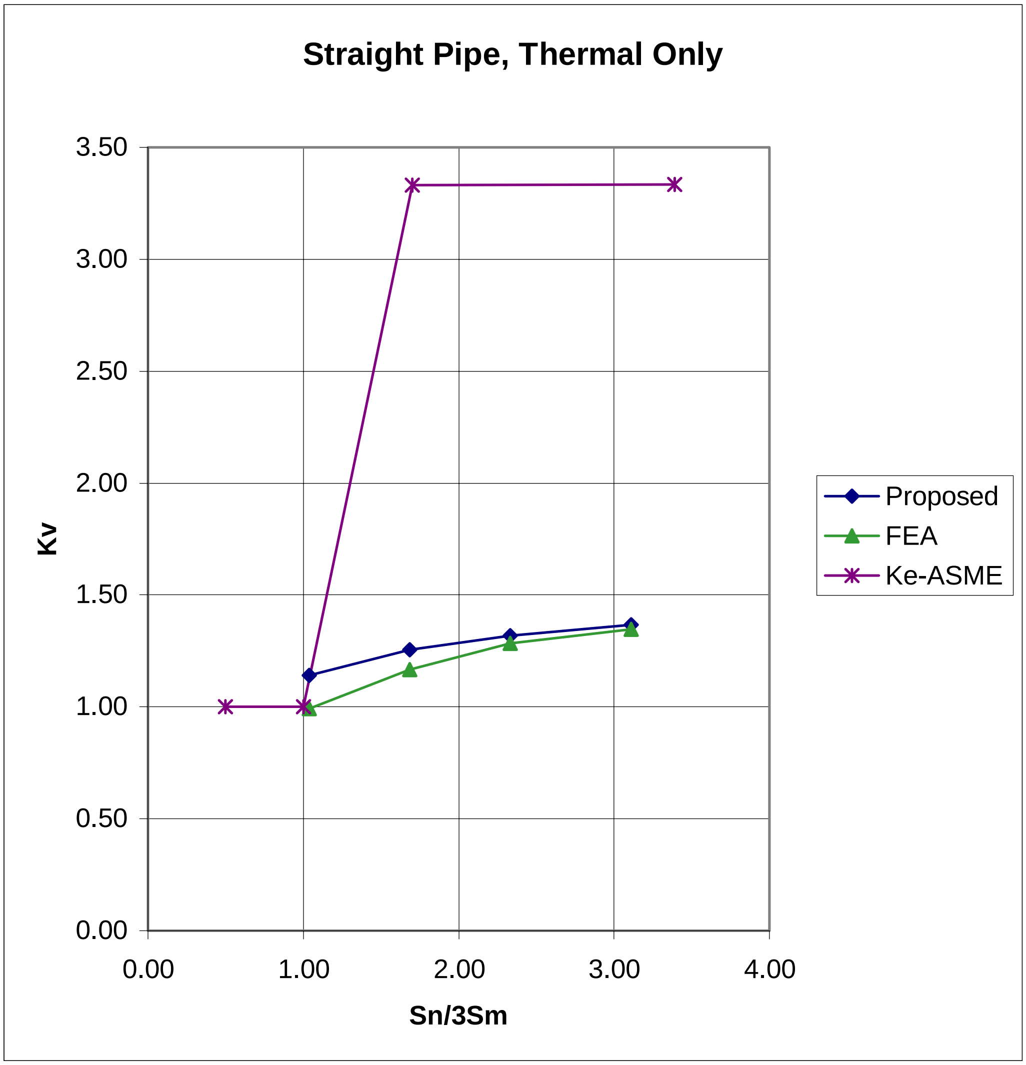

Figure 4-9
Comparing Poisson's Effect with Elastic Follow-Up Effect for Straight Pipe

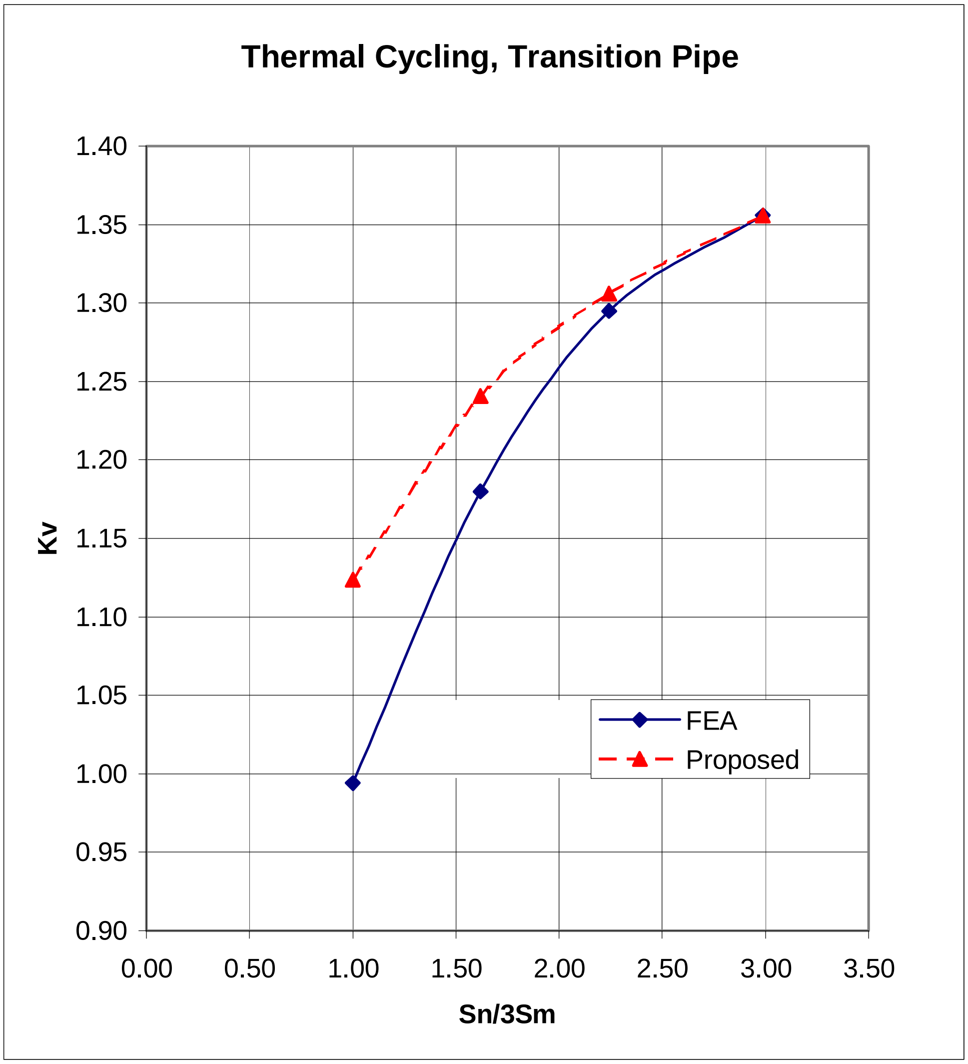

Figure 4-10
Poisson's Effect for Thin Section of the Transition Pipe

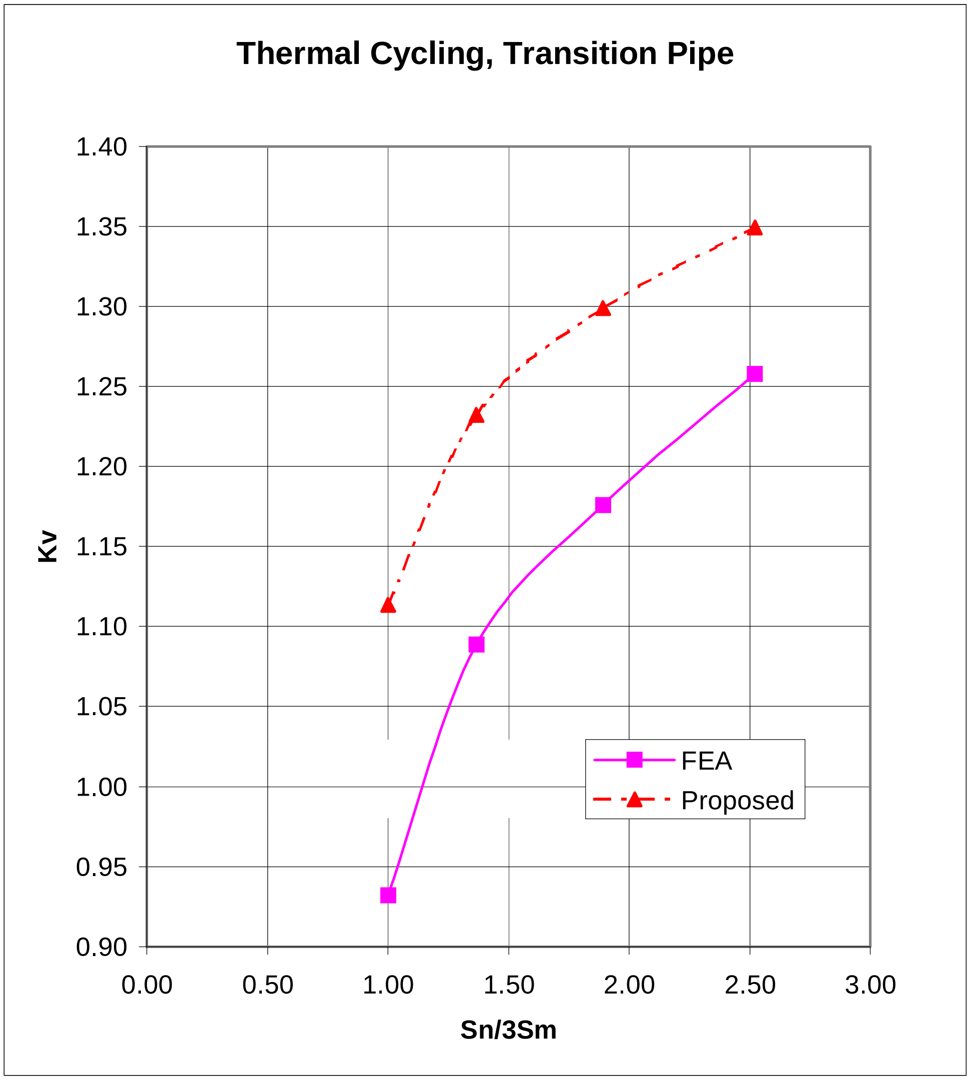

Figure 4-11
Poisson's Effect for Thick Section of the Transition Pipe

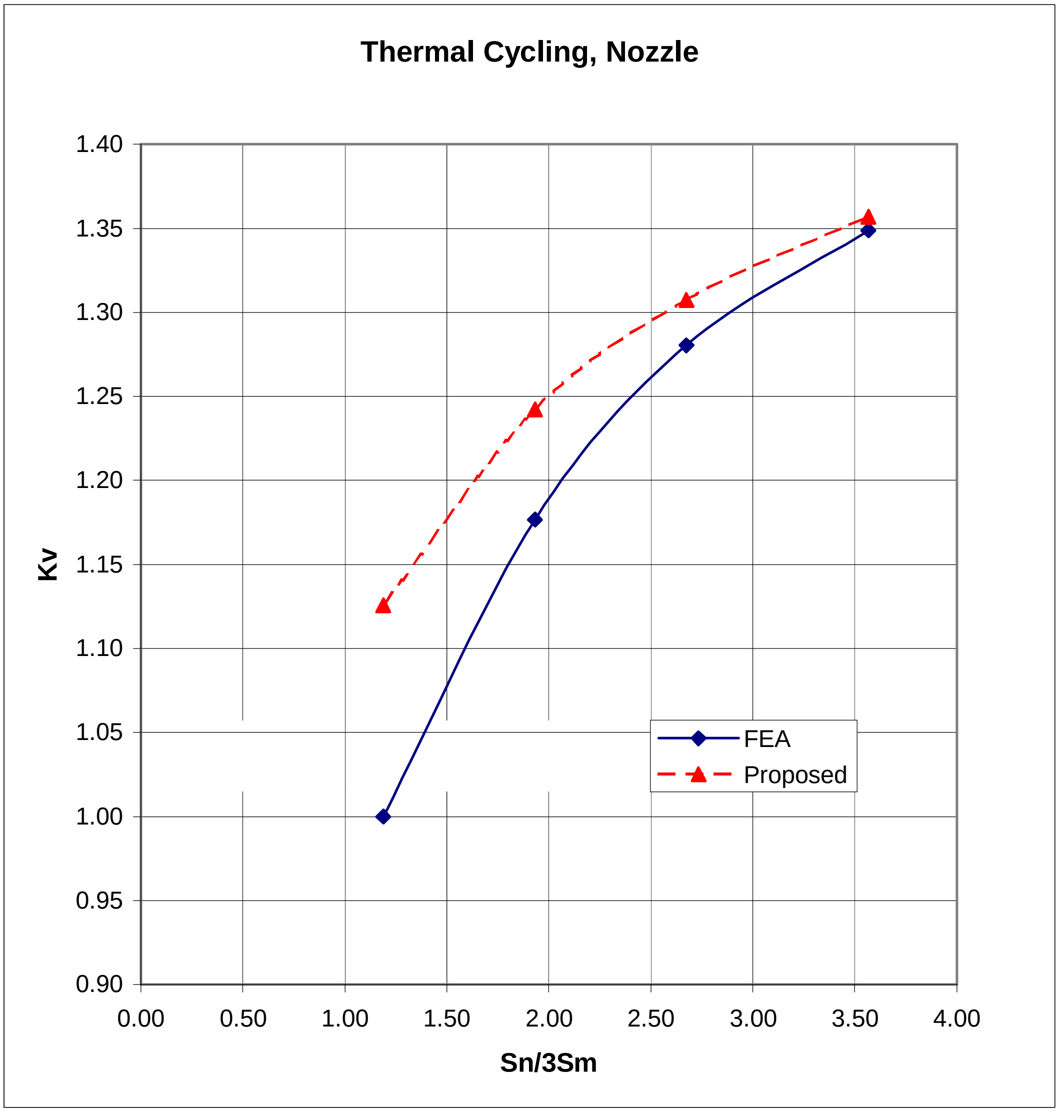

Figure 4-12
Poisson's Effect for Thin Section of the Nozzle

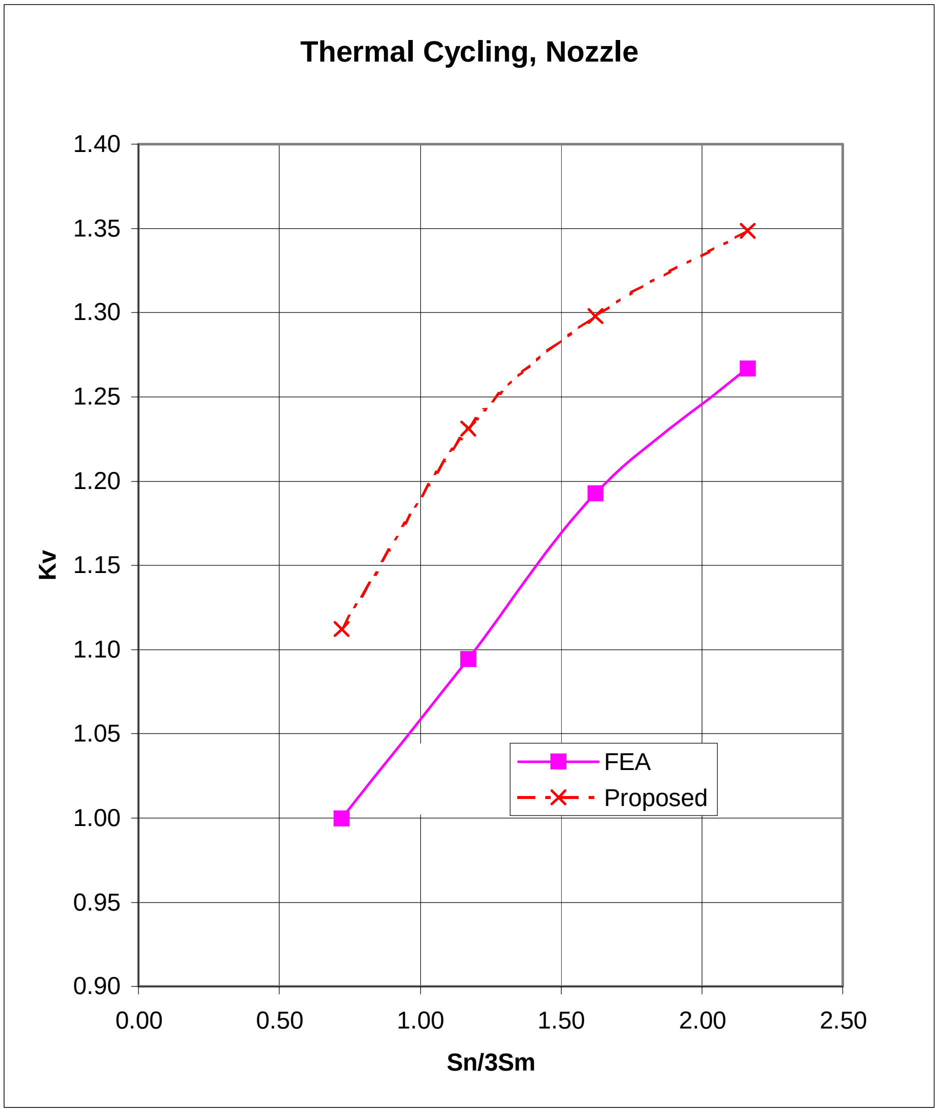

Figure 4-13
Poisson's Effect for Thick Section of the Nozzle

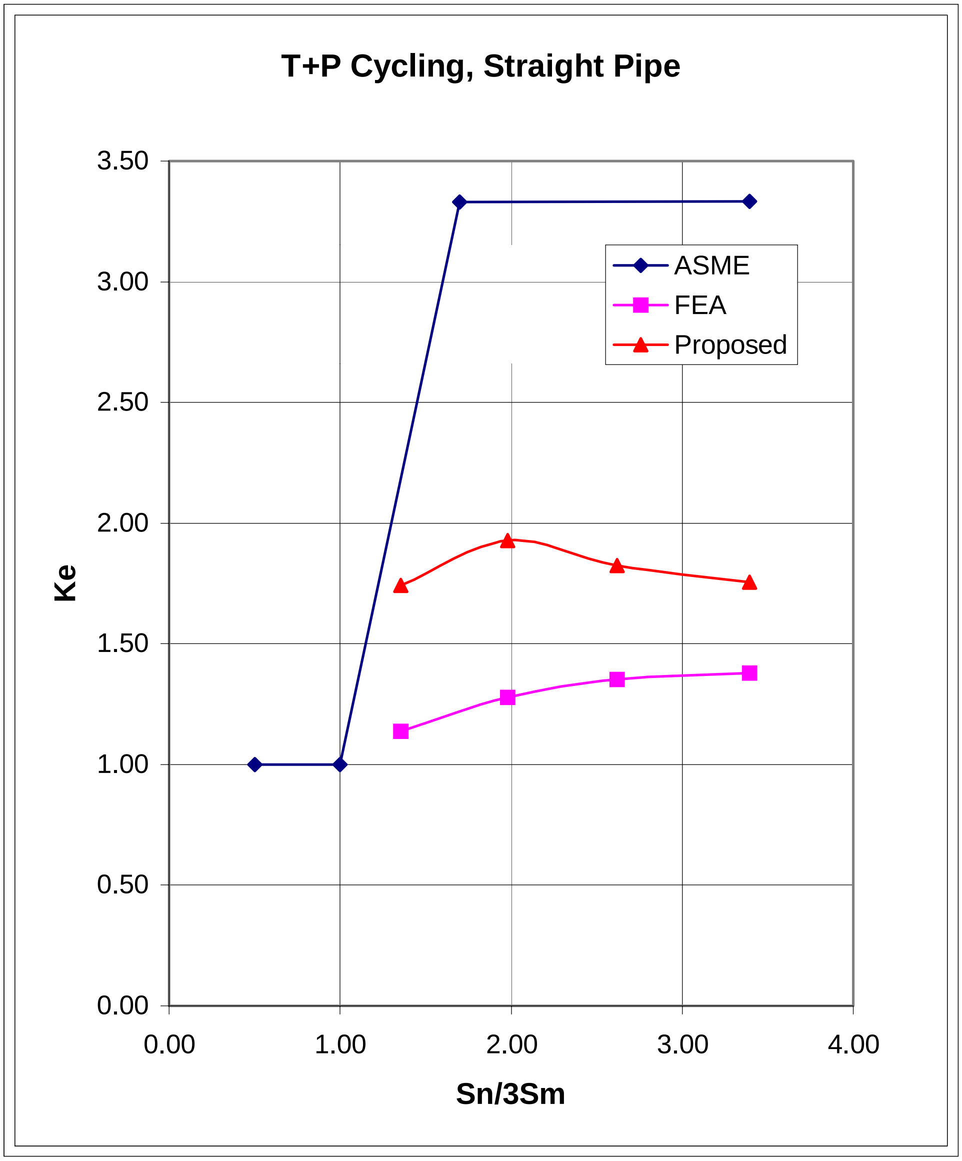

Figure 4-14
Strain Correction Factors for Straight Pipe Under Both Thermal and Mechanical Loads

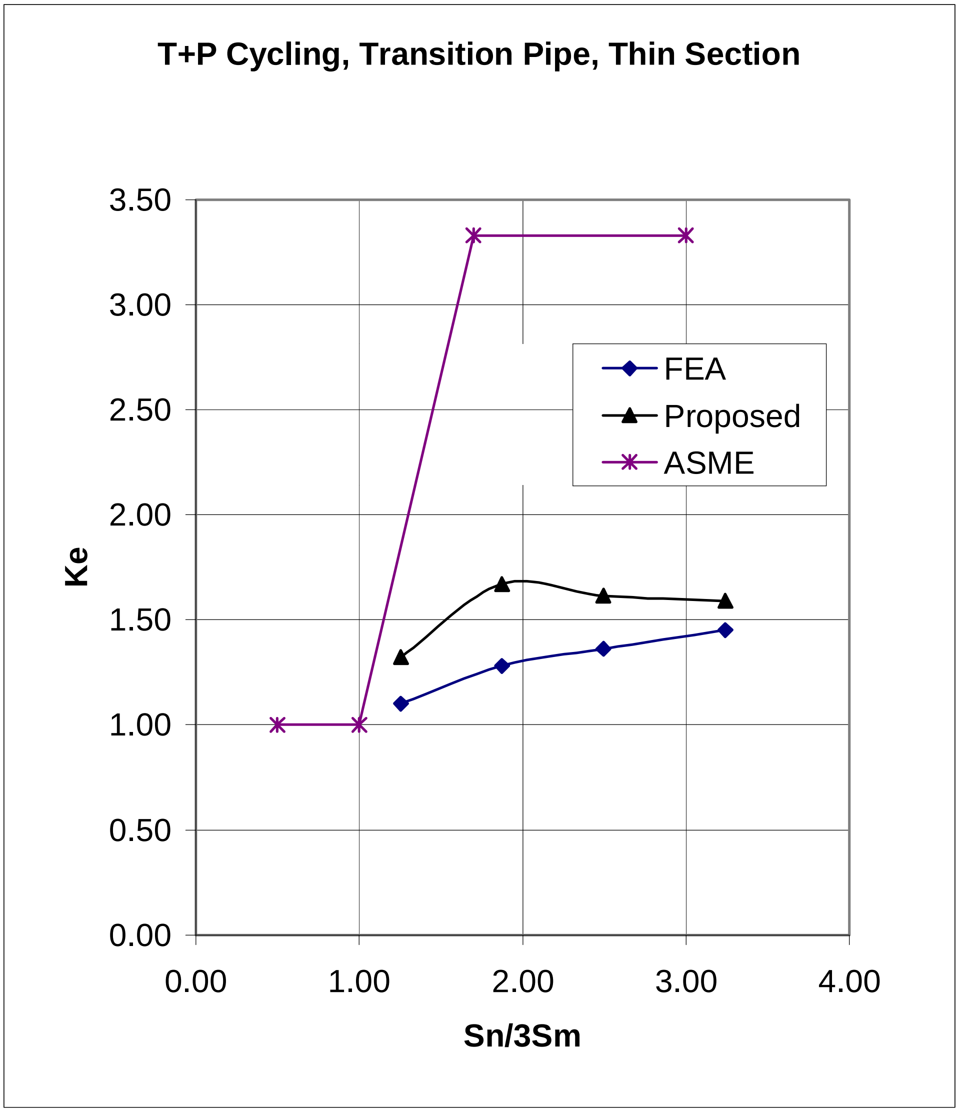

Figure 4-15
Strain Correction Factors for Transition Pipe Under Both Thermal and Mechanical Loads (Thin Section)

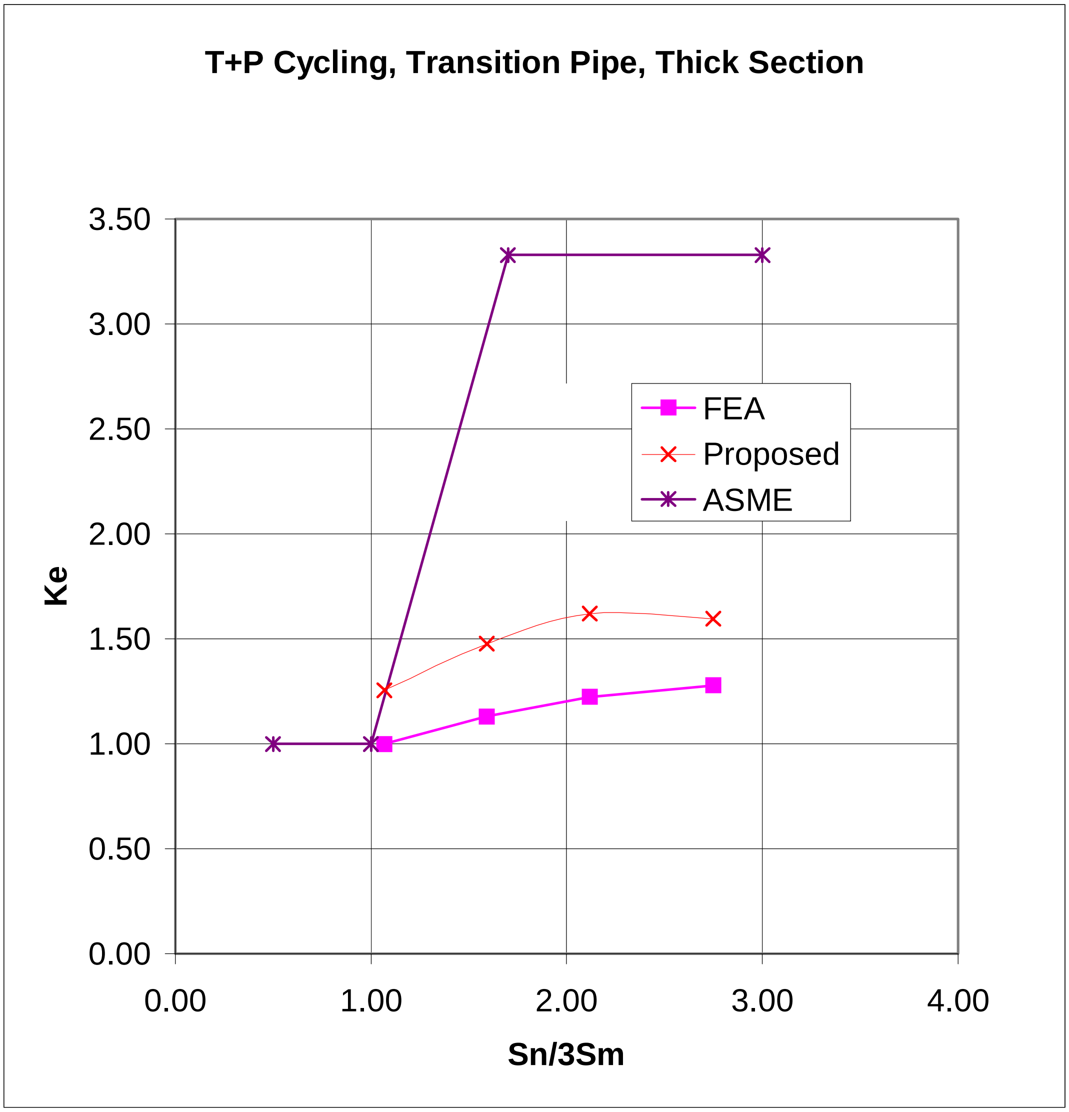

Figure 4-16
Strain Correction Factors for Transition Pipe Under Both Thermal and Mechanical Loads (Thick Section)

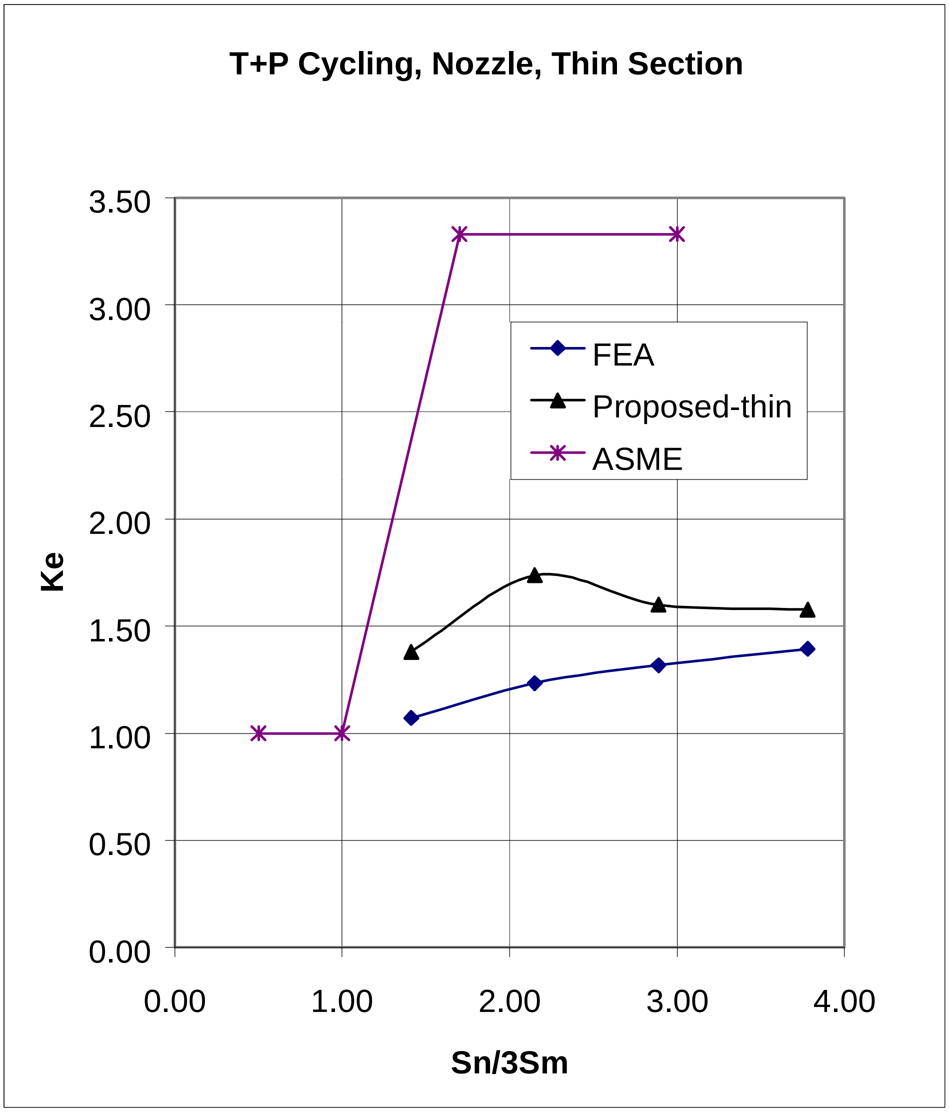

Figure 4-17
Strain Correction Factors for Nozzle Under Both Thermal and Mechanical Loads (Thin Section)

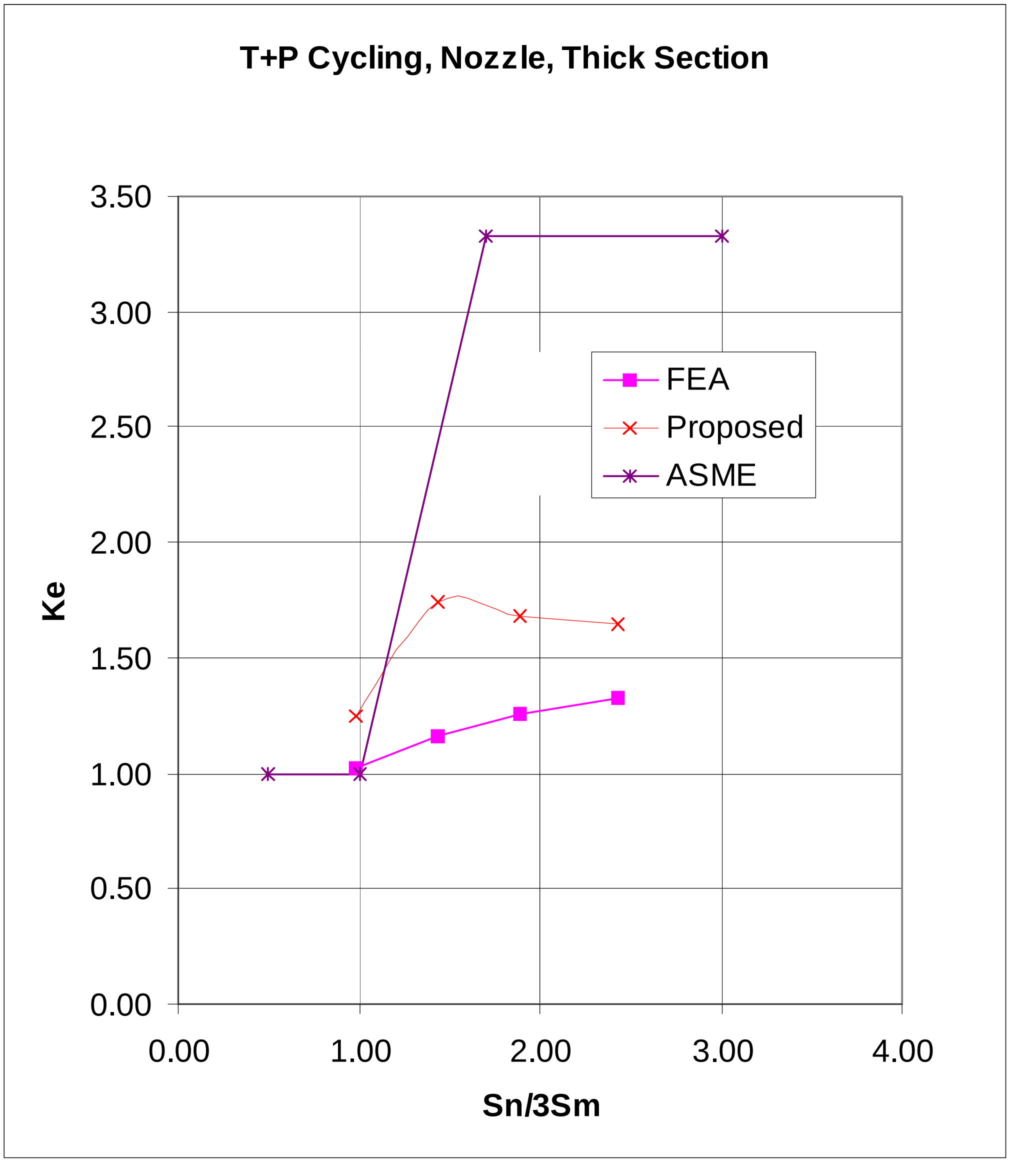

Figure 4-18
Strain Correction Factors for Nozzle Under Both Thermal and Mechanical Loads (Thick Section)

5

SUMMARY AND CONCLUSION

Three different configurations of pipe components were analyzed by the finite element method for comparison with a simplified approach in calculating the strain correction factors when plastic deformation occurs. The simplified approach is a variation of the procedures suggested in WRC Bulletin 361. The approach combines the effect of Poisson's ratio change in plasticity, the elastic follow-up effect, and the notch effect in one formulation, and is straightforward in implementation.

The results of the strain correction factor calculations showed that the proposed method used in this study is more realistic than the current ASME III method for large values of primary plus secondary stress range. The revised method is proposed as an alternative approach in a draft Code case to calculate K_e included in the Appendix of this report.

6

REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section III, Subsection NB, 1995 edition.

2. Grandemange, J. M., Heliot, J., Vagner, J., Morel, A., and Faidy, C., "Improvements on Fatigue Analysis Methods for the Design of Nuclear Components Subjected to the French RCC-M Code," Welding Research Council Bulletin 361, February, 1991.

3. ANSYS Program, Release 5.3, ANSYS, Inc., 1996.

A

DRAFT OF CODE CASE FOR ALTERNATIVE APPROACH

Case N-xxx

Alternative Approach to Calculate K_e in NB-3228.5 (b) and NB-3653.5 (c) Section III, Division 1, Class 1

Inquiry: What alternative methods might be used to compute K_e defined in NB-3228.5 (b) and NB-3653.5 (c)?

Reply: It is the opinion of the Committee that the strain correction factor, K_e, for $S_n > 3S_m$ may be replaced by the alternative K_{ep} using the expression provided as follows:

$$K_{ep} = \left(\frac{S_p}{S_{n,m} + S_{p,t}} \right)^{\left(\frac{1-n}{1+n}\right)} K_e^{'}$$

where:

$$K_e^{'} = K_v \text{ x } \frac{S_{p,t}}{S_{n,m} + S_{p,t}} + K_e \text{ x } \frac{S_{n,m}}{S_n}$$

$$K_v = \frac{(1-\nu)(1+\nu^*)}{(1+\nu)(1-\nu^*)}$$

$$\nu^* = 0.5 - 0.2\left(\frac{3S_m}{S_{n,m} + S_{p,t}}\right)^{1-n}$$

K_e = the multiplying factor in NB-3228.5

S_p = primary-plus-secondary-plus-peak stress intensity range

S_n = primary-plus-secondary stress intensity range

$S_{p,t}$ = secondary-plus-peak (excluding effects of stress concentration) stress intensity range due to local thermal transients

$S_{n,m}$ = stress intensity range due to mechanical loads

n = material parameter from Table NB-3228.5 (b)

The remainder of the requirements in NB-3228.5 or NB-3653.5 must be met.

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