Assessing the ASME Section III, Division 5, Class A Primary Load Design Rules Against Creep Notch Ef- fects

Applied Materials Division

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ABSTRACT

This report assesses the ASME Section III, Division 5, Subsection HB, Subpart B rules covering the design and construction of high temperature Class A nuclear reactor components for their robustness against creep notch effects. The creep notch effect combines the effect of multiaxial stresses on material creep deformation and damage. This report considers both effects, including the potential for creep mechanism shifts affecting the extrapolation of design data from high stress, short-time experimental data to low stress, long time operating component conditions. The report summarizes the history of and literature on multiaxial creep and surveys the current ASME design rules dealing with multiaxial effects. The report then describes the results of two dedicated numerical studies, one assessing the robustness of the ASME rules against uncertainty in extrapolating from uniaxial creep test data to multiaxial component conditions and a second study examining the potential effects of a mechanism shift from high stress, dislocation-mediated creep to diffusion-dominated creep at lower stresses. The final conclusion of the report is that the ASME rules adequately guard against multiaxial creep failure, though there are several aspects of the Code that could be optimized to provide less over-conservative design predictions or to provide a more consistent design margin as a function of temperature and stress. In a few areas, particularly the potential for mechanism shifts outside the currently-available experimental data, the Code rules should be reevaluated as additional experimental data and new modeling and simulation results become available.

EXECUTIVE SUMMARY

Operating high temperature nuclear components will be subject to general, multiaxial states of stress. The primary failure mechanism of concern, at least for steady operation, is creep damage caused by internal pressure and relaxing thermal and peak stresses. The vast majority of the creep deformation and failure data used to construct the design data used in the ASME Boiler & Pressure Vessel Code and other, international codes and standards comes from uniaxial creep tests. Moreover, available creep test data is for shorter times and therefore higher stresses than what will be experienced in operating components with realistic, economic design lives. Therefore, there is the potential for error and non-conservative designs when attempting to design multiaxial, long-lived components based on uniaxial, short-time data.

This report then assess the design rules for Class A components contained in the ASME Boiler & Pressure Vessel Code Section III, Division 5, Subsection HB, Subpart B against the effect of multiaxial stresses on creep rupture and deformation. This assessment includes a literature survey, a summary of the ASME rules dealing with multiaxial deformation and damage, and two detailed numerical studies. These studies focus on the robustness of the current ASME rules against uncertainty in extrapolating from uniaxial to multiaxial creep conditions and on the potential effects of mechanism shift impacting the extrapolation of creep data from short-term experimental to long-term design conditions.

The overall conclusion of the report is that the current ASME rules adequately account for multiaxial creep deformation and rupture both through explicit design rules dealing with multiaxiality and through the general conservatism of the design methods and design data used in the Code. The bases of this conclusion are:

- Experimental notched creep tests on the Class A materials have only ever demonstrated notch strengthening behavior.
- The Code primary load design rules use the Tresca stress which, somewhat surprisingly, seems to be a fairly conservative means to extend uniaxial creep rupture data to multiaxial failure.
- The creep-fatigue rules provide a backstop on the Code primary load design criteria by explicitly accounting for damage accumulated during stress relaxation from initially high stresses at notches and other stress concentrations. That is, the creep-fatigue rules account explicitly for the peak stresses at notches.
- The ASME rules are generally conservative, both in design analysis approximations and in the design data. Some of this conservatism can be allocated to guard against “unknown-unknown” effects outside the currently-available experimental database.

The report does make a few recommendations on future or ongoing actions, aimed at optimizing the Code design rules and reevaluating these conclusions as additional data becomes available:

- New design methods often use the von Mises stress instead of the Tresca stress for computational convenience. Simply swapping the two stress measures could eliminate some

of the conservative, bounding approximations identified here as helping guard against multiaxial deformation and rupture effects.

- Triaxial creep testing or modeling and simulations methods providing realistic triaxial creep failure data should be developed to directly assess the Code, including the effective stress measures, against direct test data. Component testing would also be useful in providing failure data for realistic component stress states.
- Similarly, methods for testing or exploring via modeling mechanism shifts in the Class A structural materials should be developed to assess the Code rules directly against realistic long-term creep data.
- For very severe notches the Code predictions of multiaxial creep failure may be inadequate as the effective stress measures used by the Code do not include the effect of the maximum principal tensile stress. This is not a design issue, as the other Code provisions will screen out geometries with severe stress concentrations. However, it may be an issue for future fitness-for-service evaluations, which will require evaluating cracked or otherwise defective components.

ACRONYMS AND ABBREVIATIONS

The following is a list of acronyms and abbreviations used throughout the document:

<i>Acronym</i>	<i>Expression</i>
ANL	Argonne National Laboratory
CPFE	Crystal plasticity finite element
HBBDATA	HBB data package program
INL	Idaho National Laboratory
ORNL	Oak Ridge National Laboratory
SCL	Stress Classification Line

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1 Introduction

1.1 What are creep notch effects?

The vast majority of creep data collected to support engineering design is from uniaxial, bar-type specimens. However, actual engineering structures have complex, multiaxial 3D stress states. The difference between uniaxial creep deformation and failure and deformation and failure under multiaxial loadings can be critical in the design of high temperature engineering components.

Researchers often test multiaxial creep effects using notched creep experiments [1], [2]. Standard creep frames, delivering uniaxial load, can run notched samples, simplifying data collection. Based on the results of these types of tests materials can be categorized as “notch-strengthening” or “notch-weakening.” Figure 1.1 illustrates how notched-bar results can sort materials into these categories. The figure shows two creep samples under the same applied dead load. The first sample is a standard, smooth-bar test where the stress in the specimen gauge is uniaxial. The second test has a notch, which means the stress in the gauge is multiaxial. However, both specimens have the same gauge diameter, which means the average stress on the critical plane is the same for both.

Figure 1.1. Illustration of notch weakening versus strengthening behavior.

If the notched sample fails before the smooth sample then the material is notch weakening. If the notched sample fails after the smooth sample then the material is notch strengthening. This characterization is useful for engineering design. However, it is a simplistic description of complex material behavior, which is discussed further below.

Notch effects, or multiaxial creep behavior in general, can be critical for engineering design. All realistic components have penetrations, openings, nozzles, supports, and other geometric features

that act as stress concentrations. Even shell/header transitions can act as geometric discontinuities and effect the state of stress. Future microreactor designs [3] or new types of components like compact heat exchangers [4] have very complex geometries and generally complicated, 3D stress states.

1.2 What causes notch effects?

The combination of non-uniform load across the critical section, multiaxial creep deformation, leading to stress relaxation, and multiaxial creep damage effects lead to experimentally-observed notch effects. Figure 1.2 illustrates how the often-complex interaction of these three factors contribute to notched creep behavior.

Figure 1.2. Illustration of the interaction of stress redistribution, multiaxial damage accumulation, and crack propagation leading to creep notch effects.

The figure shows qualitative results from a simulation of a notched creep sample. The simulation tracks the development of creep damage in the specimen, along with the redistribution of strain and stress. The figure shows two snapshots of the results as slices through the center of the specimen. The slices show results from the start of the creep test, right after the simulation applies the load to the specimen, and then at the end of the simulation when damage has developed, in this case at the root of the notch.

The top set of images show the von Mises stress. These results demonstrate that the stress across the critical section of the specimen is not constant, either across the section or as a function of time. The stresses are not constant across the section because of the stress concentration induced by the notch. The stresses are not constant in time because of stress redistribution caused by creep.

As creep damage is generally proportional to the stress state, the non-constant stress distribution causes the specimen to accumulate creep damage non-uniformly in space and time during the test.

The bottom two images show snapshots of the creep damage accumulated in the specimen at the start and end of the creep test. Again, the results here are qualitative. As discussed below, experiments demonstrate that material accumulate damage differently depending on the stress state, even for constant-stress conditions. This means that creep damage (and time to rupture) varies with the stress state, even if the stress is constant and non-varying. This means there is a second, independent factor contributing to creep notch effects: how materials accumulate damage under multiaxial load.

The final snapshot of the damage state at the end of the simulation illustrates the third contribution to creep notch effects. The simulation stopped when one location accumulated enough creep damage to remove the load carrying capability of the material in that region. However, because the stress state and damage accumulation are non-uniform across the section gauge, this sample still retains load carrying capability. That is, the specimen would only fail after a creep crack propagates across the entire critical section, separating the specimen in half. This is in contrast to a standard, uniaxial smooth-bar creep test where the onset of damage and specimen failure occur nearly at the same time.

This example demonstrates that the complicated interaction of three factors – stress relaxation, multiaxial damage accumulation, and non-uniform stress and damage distributions – causes creep notch effects. Of these three, the first two are relevant to the design of high temperature components using Section III, Division 5 of the ASME Code. The Code does not currently represent creep crack propagation by conservatively setting the design limits as the initiation of a crack (i.e. the design method is supposed to prevent crack initiation). The remainder of the report is then an evaluation of the techniques used by the Code to represent stress relaxation and multiaxial damage accumulation.

1.3 Effective stress measures

The concept of an effective stress is often used to understand and model multiaxial creep deformation and failure. Consider models suitable to represent the creep deformation and failure of a standard, uniaxial, smooth-bar creep specimen. These models often relate the uniaxial stress, temperature, and, sometimes, a set of internal variables to the uniaxial creep strain rate

$$\dot{\epsilon}_{cr} = f(\sigma, T, \mathbf{h}) \quad (1.1)$$

and similarly represent creep damage accumulation in the material with another ordinary differential equation

$$\dot{D} = g(\sigma, T, \mathbf{h}). \quad (1.2)$$

In these equations σ is the uniaxial stress, T is the temperature, $\mathbf{h}$ is some set of internal variables, $\dot{\epsilon}_{cr}$ is the uniaxial creep rate, and $\dot{D}$ is the creep damage over the test cross section. The key idea of the damage model is that the integrated damage reaches a value of one, $D = 1$, when the creep specimen fails. The simplest models of this type are simple history-independent power laws:

$$\dot{\epsilon}_{cr} = A\sigma^n \quad (1.3)$$

$$\dot{D} = B\sigma^m. \quad (1.4)$$

For uniaxial specimens this type of 1D model is sufficient to describe the experimental data. Integrating the creep rate equation gives the creep curve and integrating the damage equation gives the time to rupture.

However, these simple models are not sufficient for multiaxial creep deformation and rupture. *Effective stress measures* extend these 1D models to general multiaxial states of stress. An effective stress measure is a map between a general state of stress, represented by a stress tensor $\boldsymbol{\sigma}$, and a scalar stress measure σ_e . A proper effective stress meets two conditions:

1. It maps any general state of stress to a scalar greater than or equal to zero: $\sigma_e(\boldsymbol{\sigma}) \geq 0$
2. It maps a uniaxial stress tensor $\boldsymbol{\sigma}_u = \sigma(\mathbf{n} \otimes \mathbf{n})$ to the uniaxial stress: $\sigma_e(\boldsymbol{\sigma}_u) = \sigma$.

These two conditions ensure that the 3D models remain mathematically sound and that the predictions line up with the uniaxial data used to calibrate the models. Given two of these effective stress measures, one for creep deformation $\sigma_{e,\varepsilon}$ and one for creep damage, $\sigma_{e,D}$ the simple 1D equations can be extended to three dimensions:

$$\dot{\boldsymbol{\varepsilon}}_{cr} = A\sigma_{e,\varepsilon}^n \mathbf{g} \quad (1.5)$$

$$\dot{D} = B\sigma_{e,D}^m. \quad (1.6)$$

The creep damage equation retains the same structure and the same meaning – the material fails under multiaxial load when $D = 1$. The creep strain in 3D is a tensor and so the model also needs to specify the flow direction $\mathbf{g}$. However, the most common assumption for the flow direction is an associative flow model so that

$$\mathbf{g} = \frac{d\sigma_{e,\varepsilon}/d\boldsymbol{\sigma}}{\|d\sigma_{e,\varepsilon}/d\boldsymbol{\sigma}\|}. \quad (1.7)$$

For this type of model the effective stress fully-determines the creep rate.

As described in further detail in Chapter 2, nearly all models use the von Mises stress as the deformation effective stress:

$$\sigma_{vm} = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}} \quad (1.8)$$

where σ_1 , σ_2 , and σ_3 are the principal stresses in order so that $\sigma_1 \geq \sigma_2 \geq \sigma_3$. Because all the models discussed in this paper use the von Mises stress to describe material deformation this report often refers to the damage effective stress simply as the “effective stress.”

The report references several history (damage) effective stress measures, outlined briefly here.

1.3.1 The von Mises stress

The von Mises stress, defined above, can also be used as a damage effective stress.

1.3.2 The stress intensity or Tresca stress

The ASME Code uses the stress intensity or Tresca stress

$$\sigma_{tresca} = \max\{\sigma_1 - \sigma_2, \sigma_2 - \sigma_3, \sigma_1 - \sigma_3\} = \sigma_1 - \sigma_3 \quad (1.9)$$

For $\sigma_1 \geq \sigma_2 \geq \sigma_3$ as kind of damage effective stress.

1.3.3 The Huddleston stress

The Huddleston effective stress, in its more common one parameter form [5], is defined as

$$\sigma_h = \sigma_{vm} \exp \left[b \left(\frac{J_1}{S_s} - 1 \right) \right] \quad (1.10)$$

with $J_1 = \sigma_1 + \sigma_2 + \sigma_3$, $S_s = \sqrt{\sigma_1^2 + \sigma_2^2 + \sigma_3^2}$, and b is a material-specific parameter that must be calibrated against experimental data.

1.3.4 Hayhurst-Leckie stress

The Hayhurst-Leckie stress [6] is a linear combination of the von Mises stress and the maximum tensile (positive) principal stress:

$$\sigma_{mix} = (1 - \alpha)\sigma_{vm} + \alpha\sigma_1. \quad (1.11)$$

The parameter α is material specific and must be calibrated to experimental data.

1.4 Historical background

The earliest references to notched creep effects in the open literature date to 1953 from an ASTM “Symposium on Strength and Ductility of Metals at Elevated Temperatures: With Particular References to the Effects of Notches and Metallurgical Changes.” Already in this collection of papers the key issues are clear: notched creep behavior varies by material and materials can transition from notch strengthening to notch weakening behavior based on the loading conditions [7]–[9].

Later experiments confirmed many of these early observations [10], [11]. Hayhurst, Leckie, and coworkers extended these studies, using notched creep samples, to other materials and provided a mathematical framework to simulate multiaxial creep deformation and damage [6], [12], [13]. Their studies clearly demonstrated that notch dependence depends on the material and the type of

multiaxial loading and provided several influence models for representing notch effects in structural analysis. Many subsequent studies further confirmed these results and greatly expanded the array of modeling options.

1.5 Current ASME Section III, Division 5 Class A materials

At least simplistically, if a material is notch strengthening then simple engineering design methods relying on uniaxial test data will conservatively bound the actual, multiaxial deformation and failure of the components under steady load. As described extensively in Chapter 2, Section III, Division 5, Subsection HB, Subpart B of ASME Boiler and Pressure Code, covering the design and construction of Class A components, at least partially accounts for detailed multiaxial creep effects in the design process. However, all the currently-available experimental data for the Class A materials in the 2019 edition of the Code (and associated Code Cases) indicates that the materials are notch strengthening. Fundamentally then, this provides a final backstop against any inadequacy in the Code's multiaxial design approach.

Several studies examine notch effects in Grade 91 steel for several different notch geometries and covering the ASME Class A temperature range [14]–[16]. These studies all indicate notch strengthening behavior within the range of conditions covered by the tests. A similar study on 2.25Cr-1Mo also reports notch strengthening behavior [17]. One study on 0.5Cr-0.5Mo-0.25V steel (not a Class A material but a similar ferritic material to the Code-qualified 2.25Cr-1Mo) does suggest a tendency to notch weakening as the severity in the tested notch increases [18]. However, all the tests performed in the study did show notch strengthening, i.e., the notched creep life was greater than the unnotched life. The reported tendency was that the ratio of the notched to unnotched life decreased as the notch severity increased. If the results of this study could be reproduced for either Grade 91 or 2.25Cr-1Mo it would suggest a tendency towards weakening behavior for severe notches.

Many of the studies of notch effects on 316H stainless steel come from UK research programs and tend to apply creep ductility/ductility exhaustion models rather than ASME stress-based models for failure. These studies are then concerned with multiaxial ductility models [19], including for post-service material [20] and do not align the tests along constant net-stress conditions, like classical notch sensitivity studies. However, where comparisons at constant stress can be derived from the test results the material is notch strengthening [21]. A similar study on 316FR (not Code qualified) also shows notch strengthening behavior [22].

The historical data on 304H is more aligned with classical notch strengthening/weakening studies. Ramteke reports a similar trend as [18] for 304H. In his study the material remained notch strengthening in the absolute sense, relative to smooth bar standard geometries, but the degree of strengthening decreased as the notch severity increased [23]. Historical data, ultimately collected at Oak Ridge National Laboratory and reported in [24] suggests 304H was strengthening for all

conditions tested. Similar studies on Cu-modified 304H agree with this result: the material was notch strengthening for all conditions tested[25].

There are a fair number of notch effect studies on Alloy 800H dating from around 1990. However, most of them are not currently publicly accessible. Based on public-information summaries of the tests the material seems to be generally notch strengthening.

A long-term notched creep study on Alloy 617 is ongoing at Idaho National Laboratory. Based on the short-term data used in the recent Code qualification effort [26] and the ongoing, longer-term test data [27] this material is notch strengthening for all tested conditions.

Given this summary, all the Class A materials are notch strengthening for the actual conditions tested to date (see Table 1.1). For some of the Class A materials there is an indication that for some loading conditions the material *could* transition to notch weakening. This would have to occur for very low stresses or for very severe notches. Material simulation studies suggest this transition could have to do with a mechanism shift from dislocation-mediated to diffusion-mediated bulk creep mechanisms [28], [29]. Low stress/long-life loading is of a concern for component design, as this is the regime actual plant components will operate in and is a regime where it is difficult or impossible to collect experimental data.

There is some indication that the creep ductility in all the Class A materials decreases as the notch severity increases [19], [20], [30]. However, deconvolving creep deformation from damage and time to failure becomes difficult of multiaxial ductility studies. Moreover, the ASME Code does not directly reference the ductility of the materials in the design methods.

Table 1.1. Summary of notch test results in the literature for the Class A materials.

Material	References	Behavior	Comments
Grade 91	[14]–[16]	Notch strengthening for all tests	For other ferritics transition towards weakening for sharp notches
2.25Cr-1Mo	[17]	Notch strengthening for all tests	For other ferritics transition towards weakening for sharp notches
316H	[21]	Notch strengthening for all tests	Supported by ductility studies
304H	[23], [24]	Notch strengthening for all tests	Transition towards weakening for sharp notches
800H	None publicly available	Notch strengthening for all tests	
A617	[26], [27]	Notch strengthening for all tests	

1.6 Outline of this report

This report assesses the adequacy of the Section III, Division 5, Subsection HB, Subpart B load controlled (primary) stress limits rules for accounting for notch effects. These rules cover Class

A construction. While the focus of the report is on primary load design, as that deal directly with steady creep conditions and classical notch effects, the report also covers some aspects of the creep-fatigue design rules, particularly those contained in Nonmandatory Appendix HBB-T.

Chapter 2 describes the current Code rules dealing with multiaxial creep and assess their adequacy in capturing the available experimental notch creep data for the Class A materials. This analysis brings two issues to light: the adequacy of the Code effective stress measures and the potential for mechanism shifts altering the material deformation behavior at low stresses when compared to the generally higher-stress creep test data. Chapters 3 and 4 then deal with both of these issues with dedicated analysis studies. Finally, Chapter 5 summarizes the key conclusions of this report and describes a set of recommendations related to the current Code rules. The basic conclusion of the report is that the current Code rules adequately account for material notch effects. However, some optimization of the rules is possible, particularly as it relates to multiaxial creep deformation and effective stress measures aimed at capturing multiaxial creep rupture.

2 Current Rules Dealing with Creep Notch Effects

As described in Chapter 1, the notch effect is caused by a combination of factors including the material-specific evolution of the peak stress via stress relaxation and the material-specific resistance to multiaxial creep. This report focuses on primary load design which, in effect, is the design of the structure in steady creep conditions, after the relaxation of the initial peak stress. This chapter then focuses on rules in Section III, Division 5, Subsection HB, Subpart B of the ASME Code [31] covering multiaxial primary load design. However, for reasons outlined below, the chapter does discuss creep-fatigue design rules in Nonmandatory Appendix HBB-T and the peak stress as a backup to primary load design.

This analysis applies to the 2019 edition of the ASME Code. However, the provisions discussed here are basically unchanged in the 2021 edition. The discussion in this chapter assumes a basic familiarity with the structure and content of the Section III, Division 5 Class A rules, in particular with the basic process of the load controlled and deformation-controlled design processes.

2.1 Stress multiaxiality in the primary load design rules

Inherently, the ASME primary load design process is an allowable stress approach. The designer calculates a stress measure at each location (for the primary load rules each section, often implemented as a Stress Classification Line or SCL) and compares it to some allowable stress. One key provision of the primary load design rules in Section III, Division 5, Subsection HB, Subpart B and, indeed, in the rest of the ASME Code, is that the designer here only considers stresses caused by primary loads. The designer must determine this primary stress distribution from a linear elastic analysis of the component under the full load using the guidance contained in Table HBB-3217-1 and engineering judgement. For high temperature design this stress classification procedure approximates a steady state creep analysis. That is, the designer is approximating the stress distribution under steady creep conditions. While there is uncertainty and potential errors associated with the stress classification process, notably that two different designers could classify the stresses differently on the same component without violating any of the Code rules, the discussion here assumes that the designers made reasonable choices and are, in effect, working with the actual steady state stresses.

The high temperature rules in Subsection HB, Subpart B are unique in that they deal with two sets of loading conditions – the Design Loads and the Service Loads – and two corresponding sets of allowable stresses – S_o for the Design Loads which has no explicit time-dependence but is based on extrapolated 100,000 hour creep properties and S_{mt} for the Service Loads, which has explicit time dependence. Both sets of load-controlled design criteria approximate bending stress redistribution across a section. There are additional provisions unique to high temperatures for the Service Loads detailing how to account for multiple Service Load cases, by summing across load cases with a use-fraction approach. None of these details, however, deals explicitly with multiaxial load effects.

All of the allowable stress provisions, for both the Design and Service Loads, use the stress intensity to convert a multiaxial state of stress to a scalar stress measure to compare to the allowable

stresses. The stress intensity is essentially the Tresca stress or the maximum shear stress, defined as

$$S = \sigma_{tresca} = \max\{\sigma_1 - \sigma_2, \sigma_2 - \sigma_3, \sigma_1 - \sigma_3\} = \sigma_1 - \sigma_3 \quad (2.1)$$

where σ_1 , σ_2 , and σ_3 are the principal stresses $\sigma_1 \geq \sigma_2 \geq \sigma_3$ corresponding to the general multi-axial primary state of stress at the section. The stress intensity is therefore a type of *effective stress measure*, as defined in Chapter 1.

The allowable stress measures, when controlled by creep properties, are based on standard uniaxial tests. Specifically, in the creep regime the allowable stress measure S_o is the lesser of

3. 100% of the average stress to produce a creep strain of 0.01% over 1,000 hours.
4. 67% (below 815° C) of the average stress to cause rupture in 100,000 hours.
5. 80% of the minimum stress to cause rupture in 100,000 hours.

See Section II, Part D, Mandatory Appendix 2 for additional details. S_{mt} , when controlled by the time-dependent allowable stress S_t , is defined in HBB-3221 as the lesser of

1. 100% of the average stress to produce a total strain of 1% in a given time.
2. 80% of the minimum stress to initiate tertiary creep in the given time.
3. 67% of the minimum stress to cause rupture in the given time.

In the common case where the allowable stress is controlled by creep rupture, the Class A primary load design rules can be thought of as a check against creep rupture using the steady state creep stress distribution and a Tresca effective stress measure. This provides us a quantitative means to assess the Code rules for notch effects. The primary load rules do not consider peak stress relaxation and so the only multi-axiality effect is how the material ruptures under multi-axial load. We can then assess the effectiveness of the Code rules by assessing the effectiveness of the Tresca effective stress against other stress measures or actual data. Chapter 3 makes this assessment.

Of course, this only covers part of the notch behavior and part of the Code primary load design rules. Historically, ASME high temperature design rules assume the component spends a lot of time in the steady state and thus ignoring the higher stresses during the initial relaxation period is a reasonable approximation. For Class A, Division 5 components the creep-fatigue rules explicitly evaluate this transient damage, as described in 2.3 below. The other complication is that some of the allowable stress criteria are based on creep deformation, instead of rupture. For deformation the creep flow surface is an analogy to an effective stress measure for rupture. Interestingly, creep flow surfaces are seldom studied, at least when compared to creep rupture effective stresses [32]. The general consensus in the literature seems to be the standard J_2 flow theory, i.e. the von Mises effective stress, adequately represents multi-axial creep deformation. If this is the case then the Tresca stress will bound the deformation measure. Figure 2.1 illustrates why – the Tresca stress is always larger than or equal to the von Mises stress and approximates the von Mises surface. That is, for an arbitrary state of stress, the Tresca stress is greater than or equal to the von Mises stress. The surface plotted in the figure is a contour showing an arbitrary value of equivalent stress. Consider a point on the von Mises contour for a location where the two effective stresses are not

equal. The von Mises contour lies outside of the Tresca contour, meaning for that state of stress a larger value of the Tresca stress would be needed to match the fixed value of Mises stress.

This can also be seen directly from the mathematical definition of the two stresses in Eqs. 1.8 and 1.9 above. Define

$$\begin{aligned} l_1 &= \sigma_1 - \sigma_3 \\ l_2 &= \sigma_2 - \sigma_3 \\ l_3 &= \sigma_1 - \sigma_2. \end{aligned} \tag{2.2}$$

Because $\sigma_1 \geq \sigma_2 \geq \sigma_3$ we have $l_1 \geq 0$, $l_2 \geq 0$, and $l_3 \geq 0$. The proposition is that

$$\sigma_{tresca} \geq \sigma_{mises}. \tag{2.3}$$

Using the definitions above and squaring each side gives

$$l_1^2 \geq \frac{1}{2}(l_1^2 + l_2^2 + l_3^2) \tag{2.4}$$

or, after simplification

$$l_1^2 \geq l_2^2 + l_3^2 \tag{2.5}$$

notice that $l_1 = l_2 + l_3$, so

$$(l_2 + l_3)^2 \geq l_2^2 + l_3^2. \tag{2.6}$$

And finally

$$2l_2l_3 \geq 0 \tag{2.7}$$

thus proving that $\sigma_{tresca} \geq \sigma_{mises}$ for any state of stress.

Figure 2.1. Comparison between the von Mises and Tresca surfaces in the biaxial stress plane.

2.2 Rules directly dealing with multiaxial/triaxial stress states

The Code also contains provisions explicitly dealing with multiaxial loading and notch effects. HBB-3227.4 imposes a limit on the maximum primary load triaxiality, specifying that the sum of the three primary stresses must be less than four times the value of S_{mt} :

$$\sigma_1 + \sigma_2 + \sigma_3 \leq 4S_{mt}. \tag{2.8}$$

This provision is a primary load limit and so only applies to the approximate steady stress distribution, not to the initial peak stress.

For an efficient design the stress intensity will exactly equal the allowable stress S_{mt} . One standard definition of stress multi-axiality is

$$F = \frac{3\sigma_{mean}}{\sigma_{vm}} \quad (2.9)$$

where $\sigma_{mean} = (\sigma_1 + \sigma_2 + \sigma_3)/3$ and σ_{vm} is the von Mises stress. Approximating the von Mises stress with the Tresca stress, HBB-3227.4 imposes a triaxiality limit, for efficient designs, of

$$F \leq 4 \quad (2.10)$$

on structures meeting the Code criteria. This is a fairly severe triaxiality but does nevertheless rule out the application of the Code rules to extremely sharp, crack-like geometric stress concentrations.

Similarly, the examination requirements in HBB-5000 should prevent any unintentional (not designed) stress concentrations from cracks or other defects from existing in Code-qualified components. The ongoing inspection requirements in Section XI should likewise prevent, or at least detect, cracking and other defects during service.

2.3 The creep-fatigue rules as a backstop to primary load design

As described in Chapter 1, the notch effect arises from the combination of two factors: multi-axial creep deformation causing relaxation from the initial peak stress and multi-axial creep damage accumulation. The Section III, Division 5 primary load design rules by definition neglect the first factor, focusing on design against rupture (and other criteria) in the steady state.

Uniquely among the ASME high temperature design rules, Section III, Division 5, Subsection HB, Subpart B requires the designer to analyze and prevent strain accumulation, buckling, and creep-fatigue design limits which are controlled by deformation-controlled secondary and peak stresses.

The Code requirement resides in HBB-3250 and actual design approaches are in Nonmandatory Appendix HBB-T and two nuclear Code Cases N-861 and N-862.

Figure 2.2. Damage calculated at a notch using the primary load rules (blue) and the creep-fatigue rules (orange).

The creep-fatigue design limit is a backstop on the primary load rules, particularly for notches with significant peak stress concentrations. Figure 2.2 illustrates why. Nearly all nuclear components will see some type of cyclic service, but consider the limiting case where the component only undergoes a single load cycle – basically it starts up and runs until the end of its design life. In this case the fatigue damage is negligible as is creep-fatigue interaction. *However, the creep-fatigue design rules would still require the designer to calculate creep damage for the full, transient stress relaxation history, starting from the initial peak (and thermal, where applicable) stresses.*

Moreover, the creep-fatigue design rules explicitly or implicitly account for multiaxial rupture, depending on the method the designer selects:

- Design by elastic analysis: there are two options for calculating the stress relaxation history.
 - HBB-T-1433, Step 5, (a): This method alters a uniaxial relaxation history based on a stress relaxation analysis to explicitly account for the effect of stress multiaxiality.
 - HBB-T-1433, Step 5, (b): The designer may calculate an “algebraic” stress relaxation profile based on the isochronous stress-strain curves (ISSCs) in the Code. This option does not explicitly account for stress multiaxiality. However, the ISSC method is extremely conservative, resulting in a much slower relaxation from the initial stress which in turn translates to a greatly increased creep damage. This heightened damage covers the multiaxial rupture effect.

- Design by inelastic analysis uses the Huddleston effective stress measure (HBB-T-1411) [33] to convert from the general 3D state of stress to a scalar stress measure, index to calculated relaxation damage using the Code minimum stress to rupture table. The Huddleston stress has a well-developed technical background demonstrating it conservatively bounds biaxial rupture test data for a variety of materials [5], [34]. The creep damage calculation in the creep-fatigue rules tries to accurately capture the multiaxial effect on creep damage given a stress relaxation history. The Huddleston stress is one approach to accurately (best-estimate) capturing these effects. The creep damage calculation uses an explicit safety factor (K') to introduce conservatism.

Taken in combination, these the creep-fatigue backstop is a significant advantage for the Class A design rules, as it means they account for both the stress relaxation and multiaxial rupture effects leading to notch strengthening and/or weakening. This means that provided the underlying data or, more commonly, bounding design assumptions, remain valid then the creep-fatigue design should adequately address any material notch effects.

2.4 Summary

Based on this exposition, the Class A design rules should adequately account for notch effects provided at least *one* of the following is true:

1. The Class A materials are notch strengthening for all allowable operating conditions (i.e. for all conditions that pass the Code criteria).
2. The Tresca criteria reasonable approximates or bounds multiaxial rupture within the range of triaxialities allowed by the Code.
3. The bounding assumptions or multiaxial rupture data (depending on the method) in the creep-fatigue rules adequately account for any notch effects.
4. The general conservatism, including the explicit factors and statistical lower bounds used to generate the design data, are sufficient to cover any unaccounted-for notch weakening.
5. The main concern for #1 and #3 for this report are conditions outside the direct experimental database underlying the Code data, typically for low values of stress and long design lives. Assessing the Code rules against the underlying material database is out of the scope of this report.
6. The following two chapters provided analytical support demonstrating that the Code rules satisfy all of #2, #3, and #4. Chapter 3 examines multiaxial rupture and Chapter 4 multiaxial deformation.

3 Analytical assessment of the ASME rules using effective stress measures

3.1 Objectives of the analysis

The objectives of the analysis are to explore the effects of different effective creep stress definitions on creep damage calculation results obtained through inelastic analysis using FEA, and through comparing the results with those from elastic analysis (using standard ASME stress classification techniques) and the reference stress method. The purpose of these comparison is to:

- Assess the design approach in Section III, Division 5 of the ASME Boiler and Pressure Vessel Code against best-estimate and lower-bound predictions of the life of components subject to multiaxial stress.
- Compare the ASME design life predictions against other methods commonly used in creep design (i.e. the reference stress).

The focus of the analysis is on assessing the approaches for multiaxial loading using both realistic component designs and simpler, representative structures. The analysis starts from the assumption that the base ASME Section III, Division 5 rules produce conservative designs for uniaxial loads and focuses on the effect of stress multiaxiality. The analyses use Grade 91 material properties, as detailed multiaxial rupture data is available for this material.

3.2 Background

During the design process of high temperature pressure boundary components, there is a requirement for creep damage estimation to ensure safe operation during their service. The ASME BPVC Section III, Division 5 rules provide design checks against creep rupture in two locations for Class A components:

1. The primary load design rules include the minimum stress to rupture in the definition of the allowable stresses. The primary load design rules use the stress intensity (Tresca criteria) as the effective stress and rely on load classification and the general conservatism in the rules and allowable stresses to bound multiaxial failure.
2. In the limit of no fatigue loading, the creep-fatigue design rules in Nonmandatory Appendix HBB-T provide a check against creep failure by indexing creep damage against the Code minimum stress to rupture values. The design by elastic analysis rules include a multiaxiality correction to the stress relaxation profile used to determine the creep damage and the design by inelastic analysis rules apply the Huddleston effective stress measure to account for multiaxiality.

Appendix HBB-T [31] provides such tools as creep damage calculation equations with effective creep stress definition based on Huddleston's work [5]. Recent development in material constitutive equations and crystal plasticity calculation techniques show that this measure may not provide accurate results, and other definitions of effective creep stress are being developed to provide conservative creep damage calculation results for engineering design. Defining effective creep stress is a bridge to relate creep testing data, which is usually from uniaxial loading condition, to the

multiaxial loading condition of the components for engineering applications so the creep testing results can be apply to material under realistic, multiaxial conditions. Tresca and Mises stresses are two examples of effective stress related to material (time-independent) yielding for engineering design, while for creep damage calculation, there are varieties of theories [13], [35], [36] associated effective creep stress definitions. This report applies several different effective creep stress definitions, some of which were derived from our previous work, on several different models, both notched bar or typical pressure vessel component. These calculations provide a realistic range of *component* rupture times, without delving into the accuracy of each different effective stress measure. These predictions can then be compared to the ASME design life, according to the primary load design criteria, to assess the adequacy of the rules under multiaxial conditions.

While only two of the effective stress measures (Tresca and Huddleston) are used in the current ASME Code rules this report includes a wider variety of effective stress measures to provide a crude sensitivity analysis on the Code measures. The effective stress measures used here represent the best models assembled to represent multiaxial creep failure. The differences between the measures then represent the spread in predictions of multiaxial failure, given that, broadly speaking, the developers of the different effective stress models are working from the same dataset.

3.3 Scope of the analysis

Five effective creep stress definition were applied to model failure in finite element analysis on notched bars (semi-circular, elliptical, and V) and component models (tube plate, tube bend, dished head and tube T-joint). These measures are the Tresca stress, von Mises stress, Huddleston stress, maximum of Mises and maximum principal tensile stress, and the linear combination of the von Mises and maximum principal tensile stress with a weighting factor. The creep damage evaluation results by reference stress method and elastic analysis method are also presented for comparison.

3.4 Methodology

3.4.1 Software and hardware environment

The finite element analysis was accomplished using ANSYS Mechanical 2020 Version R2 on a computer cluster with Intel® Xeon® CPU E5-2698 CPU and 48 cores, the memory size was 500Gbytes. The ANSYS material model user defined subroutine was compiled by Intel Parallel Studio XE 2017.

3.4.2 Effective creep stress definitions

For the simplicity of the expression, the following effective creep stress definition will be presented in the principal stress space.

Assume three principal stresses are σ_1 , σ_2 and σ_3 , and $\sigma_1 > \sigma_2 > \sigma_3$, then Von Mises stress σ_v , the first of the five effective creep stresses to be defined can be expressed as

$$\sigma_v = \sqrt{\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2}{2}} \quad (3.1)$$

Tresca stress σ_t is the second definition of the creep effective stresses, and it can be calculated as

$$\sigma_t = \sigma_1 - \sigma_3 \quad (3.2)$$

The third definition is based on Huddleston's work [5], and as defined in ASME BPVC Section III Division 5 Appendix HBB-T [31], effective stress,

$$\sigma_e = \bar{\sigma} \exp \left[C \left(\frac{J_1}{S_s} - 1 \right) \right] \quad (3.3)$$

where $J_1 = \sigma_1 + \sigma_2 + \sigma_3$;

$$S_s = (\sigma_1^2 + \sigma_2^2 + \sigma_3^2)^{\frac{1}{2}};$$

$$\bar{\sigma} = \frac{1}{\sqrt{2}} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{\frac{1}{2}};$$

$\sigma_1, \sigma_2, \sigma_3$ are the three principal stresses, C is a constant for Gr.91 defined as follows:

(1) If $J_1/S_s > 1.0$, $C = 0.16$;

(2) If $J_1/S_s < 1.0$, $C = 0$.

The fourth effective creep stress definition σ_{max} is the maximum of the maximum tensile principal stress and Von Mises stress, Equation 3.4 below is applied to calculate its value

$$\sigma_{max} = \max \left(\frac{abs(\sigma_1) + \sigma_1}{2}, \sigma_v \right) \quad (3.4)$$

where $\max(a,b)$ is a function to get the maximum value out of two variables a and b ;

$abs(a)$ is a function to get the absolute value of the variable a ;

σ_v and σ_1 are as defined previously.

The fifth effective creep stress definition is similar to the reference stress definition in French nuclear code RCC-MRx [37] and related to the work of Sdobyrev [38] Hayhurst [13], it is a combination of maximum principal tensile stress and Von Mises stress with a weighting factor, in a form of

$$\sigma_{mix} = (1 - \alpha) * \sigma_v + \alpha * \frac{abs(\sigma_1) + \sigma_1}{2} \quad (3.5)$$

where $\alpha = 0.9233$, the value obtained through the regression of results from large scale crystalline plasticity calculations and σ_v , σ_1 and function $abs()$ are as defined previously.

3.4.3 Creep damage calculation in material model in ANSYS

Section HBB-T-1420 in Appendix HBB-T provides an equation based on time fraction for creep damage calculation:

$$D_c = \int_0^t \frac{dt}{T_d(\sigma_e, T)} \quad (3.6)$$

where t is the time, dt is the time increment, T_d is the rupture life with the effective stress σ_e and temperature T during that time increment, and its values is obtained from the HBBDATA function

TIME_RUPTURE(MATERIAL,TEMPERATURE,STRESS). The equation was combined into ANSYS USERMAT subroutine with five different effective stress definitions and creep damage values were calculated for each of them and the accumulated creep damage results were output as variables.

This “time-fraction” approach forms the basis of the ASME Section III, Division 5 creep damage approach, both for the primary load design rules and the creep-fatigue rules. While other approaches are used (for example, ductility exhaustion) applying this damage procedure to the different effective stress measures will produce estimated component lives consistent and comparable to the Section III, Division 5 rules.

3.4.4 Reference stress method for creep damage calculations

The reference stress method [39] is a creep rupture life assessment method suitable for multiaxial evaluation of pressure parts. The approach is based on limit load analysis to obtain reference stress σ_{ref} , which can be expressed as

$$\sigma_{ref} = \frac{P}{P_L} \sigma_y \quad (3.7)$$

where P is the load under which the reference stress is calculated, P_L is the collapse load and σ_y is the material yield strength. Before the σ_{ref} value is put into the HBBDATA function TIME_RUPTURE(MATERIAL, TEMPERATURE, STRESS) to get the allowable creep life $t_{allowed}$ it was first multiplied a factor of 1.25 to account for possible stress concentrations in the structure. The creep damage D_c for service life $t_{service}$ is expressed as

$$D_c = \frac{t_{service}}{t_{allowed}} \quad (3.8)$$

3.4.5 Elastic analysis through stress linearization

This method is from Section III, Division 5, Subsection HB, Subpart B and an Argonne report [40] gives details of its application. After stress linearization, the different stress intensities should satisfy the following

$$P_m < S_{mt} \quad (3.9)$$

$$P_L + P_b < K S_m \quad (3.10)$$

$$P_L + \frac{P_b}{K_t} < S_t \quad (3.11)$$

P_m , P_L , P_b , K_t , S_{mt} , S_m and S_t are defined in the ASME Code, and for Grade 91 material at 580 °C, the values for S_{mt} , S_m and S_t are 61.92, 111.13 and 61.92 in Mpa, respectively and they are from HBBDATA package.

The creep life under the load t_r can be calculated through HBBDATA function TIME_S_T(MATERIAL, STRESS, TEMPERATURE), the inputs are “Gr91” for material, $P_L + \frac{P_b}{K_t}$ value for stress in MPa and 580 for temperature in Celsius. The creep damage is calculated as the ratio of 300,000 hours to t_r .

3.4.6 Dimensional design for representative components

We based the geometric design of the representative components on design by rule formula found in the ASME Code.

3.4.6.1 Tube plate

Article A-8000 of the ASME Code gives a procedure for tube plate dimension determination, even though it is for the design of components with application temperature lower than those in creep range, we argue that it can still be approximately applied to the high temperature component design by substituting S_m with S_{mt} . After applying load and boundary conditions, the procedure can be simplified into two equations, which can be regarded as constraints for dimension determination. Eq. 3.3 is for the membrane stress, expressed as

$$\frac{P}{h} \frac{\Delta p R^*}{t} \leq S_{mt} \quad (3.12)$$

and Eq. 3.10 is for surface stress limit, and it is in the form of

$$\frac{P}{h} \Delta p \left(\frac{R^*}{t} \right)^2 \leq 1.79 S_m \quad (3.13)$$

where P is the nominal distance between hole centerlines, i.e., pitch;

h is the nominal width of ligament at the minimum cross section;

Δp is the differential pressure across the plate;

R^* is the effective radius of the perforated plate;

t is the plate thickness;

S_{mt} is the allowable stress for service load at 580° C for 300,000 hours, equal to 61.9Mpa

Figure 3.1 below shows a schematic of the tube plate and designation of some variables mentioned above.

Figure 3.1. Schematic of a tube plate geometry

In order to use Eq. 3.12 and Eq. 3.13, variables h , P and t should also satisfy the following two equations

$$\frac{h}{P} \geq 0.05 \quad (3.14)$$

$$\frac{t}{P} \geq 2 \quad (3.15)$$

The variable values applied in the tube plate model are list in the Table 3.1 below. Figure 3.2 shows the geometry for FEA, due to symmetry, only 1/12 of the model was included in the analysis.

Table 3.1. Tube sheet model geometry parameters

Variable	P (mm)	h (mm)	R* (mm)	t (mm)	ΔP (Barg)
Values	50	10	150	100	99

Figure 3.2. Tube plate model geometry

3.4.6.2 Dished head

Figure 3.3 below from the ASME Code shows the schematic of the tori-spherical head with designation of related geometrical variables. The head consists of two parts, one part is the cylindrical shell, the other is the head, and the schematic assumes equal wall thicknesses of the two parts.

Figure 3.3. Schematic of the dished head model

For the cylindrical part, the minimum wall thickness t_{min} is determined by circumferential stress through Eq. 3.16, and longitudinal stress through Eq. 3.17.

$$t_{min} = \frac{P \cdot R}{S_{mt} - 0.6P} \quad (3.16)$$

$$t_{min} = \frac{P \cdot R}{2S_{mt} + 0.4P} \quad (3.17)$$

where P is the applied internal pressure, R is the internal radius, and S_{mt} is the allowable for service load at 580° C for 300,000 hours and with a value of 61.9Mpa.

The minimum wall thickness for the head part is determined by Eq. 3.18 below

$$t_{min} = \frac{P*L*M}{2S-0.2P} \quad (3.18)$$

where parameter M can be expressed as

$$M = \frac{1}{4} \left(3 + \sqrt{\frac{L}{r}} \right) \quad (3.19)$$

Figure 3.3 defines the geometric parameter L .

Table 3.2 shows the parameter values and Fig. 3.4 presents the dished head model geometry.

Table 3.2. Dished head geometrical parameters

Parameter	P (Barg)	R (mm)	L (mm)	r (mm)	t (mm)
Value	200	200	333	42	81

Figure 3.4. Dished head model geometry

3.4.6.3 Tube bend

Paragraph 304.2.1 “Pipe Bends” of the ASME Code describes the procedure to determine minimum wall thickness for a tube bend and Fig. 3.5 below shows the nomenclature for the pipe bends.

Figure 3.5. Nomenclature for pipe bend

Based on these formula, the tube minimum wall thickness t_{min} can be expressed as

$$t_{min} = \frac{P \cdot D}{2 \left(\frac{SEW}{I} \right) + PY} \quad (3.20)$$

where P is the inside pressure load;

D is the tube outside diameter;

S is the allowable stress, S_{mt} was applied for our case for 300,000 hour life at 580° C;

E is the quality factor of a value of 1;

W is the weld joint strength reduction factor, equal to 1, as no weld joint in the bend model;

$Y=0.7$ for ferritic steel at 580° C.

I is a parameter related to locations on the tube bend, i.e., at the intrados, I is expressed as

$$I = \frac{4 \left(\frac{R_1}{D} \right) - 1}{4 \left(\frac{R_1}{D} \right) - 2} \quad (3.21)$$

while at extrados

$$I = \frac{4 \left(\frac{R_1}{D} \right) + 1}{4 \left(\frac{R_1}{D} \right) + 2} \quad (3.22)$$

In the tube bend model, a 90° 4" (OD 4.5") schedule 120 (0.437" wt) pipe was chosen, and the geometry is displayed in Fig. 3.6 below, due to the symmetry, only half of the geometry was included in the model.

Figure 3.6. Geometry of the tube bend model

3.4.6.4 T-joint

The T-joint configuration for the analysis is displayed in Figure 3.7 below based on Figure NB-3339.1(b)-1 of the ASME Code. The opening and related reinforcement dimension determination is the main focus of the following analysis.

Figure 3.7. Opening configurations for T-joint

Fig. 3.7 also demonstrates opening dimension constraints, with those in Table 3.3 and Table 3.4 based on Code equations, the T-joint dimensions were determined.

Table 3.3. Dimension constrains for the opening

Parameter	D/t	d/D	$d/\sqrt{Dt_n r_2/t}$
Range	10 to 100	0.5 max.	1.5 max

Table 3.4. Dimension constrains for reinforcement

Parameter $d/\sqrt{Rt_r}$	≤ 0.2	$\leq 0.4, > 0.2$	> 0.4
Required Minimum Reinforcing Area, A_r	None	$\left[4.05 \left(d/\sqrt{Rt_r} \right)^2 - 1.81 \right] dt_r$	$0.75 dt_r$

The model dimensions determined are shown in Table 3.5 and the nomenclatures in the table and the aforementioned in-equations are in Figure 3.7 and Figure 3.8. Figure 3.9 shows the model geometry and due to symmetry, only quarter of the geometry was included in the model.

Table 3.5. T-joint model dimension values

Parameters	R_0 (inch)	t_r (inch)	t (inch)	D (inch)	d (inch)	t_n (inch)	r_1 (inch)	r_2 (inch)	Theta (degree)	r_3 (inch)	L_c (inch)
Values	7	0.95	1.06	11.88	5.8	0.52	0.2	2.6	45	1.4	1.78

$$L_c = 0.75 \left(\frac{t}{D} \right)^{\frac{2}{3}} D$$

Figure 3.8. Nomenclature for opening reinforcement dimension constrains

Figure 3.9. T-joint model geometry

3.5 Assumptions

This analysis is based on the following assumptions:

1. The material for all the models is Grade 91, isotropic and homogeneous; material response is constant over time, i.e., effects of material degradation over time, are neglected. There is no coupling between the calculated creep damage and the material's deformation.
2. The material thermo-physical properties and constitutive equations applied in FEA are the same for both the welds and the base metal (in cases with welded construction).
3. Zero residual stress.
4. A viscoplastic material model, developed for use with the ASME Section III, Division 5 design by inelastic analysis method, represents the deformation of the material [41].

3.6 Analysis on notched bars

The first set of analyses consider standardized (or quasi-standardized) notched creep samples. These tests are often used to evaluate notch/multiaxial effects using a standard creep frame. The frame delivers uniaxial load, but the notch geometry provides multiaxial stress in the gauge region. While these tests are easy to implement with standard creep testing equipment, the results are difficult to interpret as the stress state changes with time due to stress redistribution, even without accounting for softening due to creep cavitation. However, this limitation is not significant here as the comparison is entirely between FEA results.

Three notched samples, with semi-circular, semi-elliptical, or V-shape notches, are included in this report. The samples have outer cross sections with a 5mm radius, and inner (gauge) cross sections

with a 3mm radius. Due to different notch geometries, the stress states are different at the notches which results in different service lives due to creep damage.

3.6.1 Inputs

3.6.1.1 Geometries

Figures 3.10 to 3.12 below shows the geometries of the three notched bars.

Figure 3.10. Geometry of the semi-circular notched bar

Figure 3.11. Geometry of the semi-elliptical notched bar

Figure 3.12. Geometry of the V-notched bar

3.6.1.2 Boundary Conditions and Loads

Due to the symmetry, only a quarter of the bar was included in the bar model, three faces are symmetry planes the outer surface is free of load, the end surface was with a pressure of -22.3Mpa (under tension), that leads to the stress at the notch bar net section to be 61.9Mpa for a minimum creep rupture life of 300,000 hours at 580° C. Figure 3.13 shows the boundary conditions for the three notched bars, even though the figure shows a semi-circular notched bar, the boundary condition can be applied to other notched bars too.

Figure 3.13. Boundary conditions for the notched bars

3.6.2 Results

3.6.2.1 Stress states

The design for different notches is to introduce different stress states, and the parameters utilized to characterize different stress states in the reports are Tresca, Mises, maximum tensile principal stress, and J_I/S_s ratio used in Eq. 3.3.

Figure 3.14 shows the path for the stress state demonstration, label 1 shows the start location of the path, which is the center of the bar, and label 2 shows its end location, which is at the tip of the

notch. Figure 3.14 shows the path definition for semi-circular notched bar, the path definitions for semi-elliptical notched and V notched bars are similar.

Figure 3.14. Path definition for stress state demonstration

Figure 3.15. Stress state parameters (a) semi-circular notched, (b) semi-elliptical notched, (c) V-notched bars

Figure 3.15 shows the effective stress state parameter values along the path, the abscissa shows the normalized distance from the bar central line, “0” is the start of the path, the bar center, “1” is the end of the path, the notch tip. The J_1/S_s values of semi-circular notched bar are lower than those of semi-elliptical and V-notched bars, the latter two demonstrate similar values but the V-notched bar shows a higher J_1/S_s values near the tip of the notch. 300,000 hour creep lowers the J_1/S_s values in the region near the notch tip for semi-circular notched bar and there are only moderate changes for semi-elliptical and V-notched bars. At the tip of the notch, the magnitudes of maximum tensile principal stress, Tresca and Mises increase in the order of semi-circular notched bar, semi-elliptical notched bar and V-notched bar, and after 300,000 hour creep, the magnitudes for each of the three parameters are similar among the three bars with different notch geometries. For all the three bars, in the region towards the notch tip, there are humps for maximum tensile principal stresses with higher magnitude than those at the tip after 300,000 hour creep, and they increase with the same order as that for their magnitudes at the notch tip, i.e., semi-circular notched bar shows the lowest values, V-notched bar demonstrates the highest value and semi-elliptical notched bar with a value in between.

3.6.2.2 Creep damage for different effective creep stress definitions

In Subsection 3.4.2, different effective creep stresses for creep damage calculation are defined, they are Mises, Tresca, effective stress defined by Huddleston, maximum of maximum tensile principal stress and Mises and a combination of the above two with a weighting factor, and in the following paragraph, they are referred as Mises, Tresca, Huddleston, maximum stress and combined effective stress, respectively. Figure 3.16 to 3.18 display the creep damage contour plots at the end of 300,000 service life based on different effective creep stress definitions and Table 3.6 list the maximum creep damage at any point in each notched sample at 300,000 hours.

Figure 3.16. Creep damages at the end of 300,000 hour life for semi-circular notched bar with different effect creep stress definitions (a) Mises, (b) Tresca, (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

Figure 3.17. Creep damages at the end of 300,000 hour life for semi-elliptical notched bar with different effect creep stress definitions (a) Mises, (b) Tresca , (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

Figure 3.18. Creep damages at the end of 300,000 hour life for V notched bar with different effect creep stress definitions (a) Mises, (b) Tresca , (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

The above three figures demonstrate that the maximum creep damage for Mises, Tresca and Huddleston effective creep stress definitions with different notch geometries are always at the surface of the notch tips, while those for Maximum stress and combined effective creep stress definitions are inside the bars near the notch tips. The creep damage magnitudes for 300,000 hour life at 580° C are presented in Table 3.6 below.

Table 3.6. Creep damage at the end of 300,000 hour service with different creep effective stress definitions

Effective creep stress	Mises	Tresca	Huddleston $\sigma_e = \sigma \exp[C(I_1/S_s - 1)]$	Max.stress $\sigma_{max} = \max((\sigma_1 + \sigma_2)/2, \sigma_v)$	Combined effective stress $\sigma_{mix} = (1-\alpha)\sigma_v + \alpha((\sigma_1 + \sigma_2)/2)$
Semi-circular	0.03	0.06	0.04	0.07	0.06
Semi-elliptical	0.06	0.14	0.08	0.31	0.26
V-notched	0.09	0.26	0.14	1.13	0.86

The creep damage for 300,000 hour service increases in the order of: semi-circular notched, semi-elliptical notched, and V notched bars. The maximum stress definition, i.e., the larger of the von Mises stress and maximum principal tensile stress, always gives the highest creep damage. Within the three historical effective stress measures (von Mises, Tresca, and Huddleston), the Tresca stress bounds the other two.

3.7 Analysis on typical pressure vessel components

3.7.1 Inputs

3.7.1.1 Geometries

The geometries of the four typical pressure vessel components, i.e., tube plate, dished head, tube bend and T-joint are shown in Figure 3.2, Figure 3.4, Figure 3.6 and Figure 3.9, respectively in Section 3.4 of the report.

3.7.1.2 Boundary conditions and loads

Figure 3.19 displayed the boundary conditions and load for the tube plate model, three surfaces can only move within their planes, to restrain possible rigid body motions, the holes in the center is with cylindrical support and the tube sheet internal surface is under internal pressure of 240 Barg.

Figure 3.19. Boundary and load conditions for the tube plate model

The boundary and load conditions of dished head model is demonstrated in Figure 3.20, the internal pressure magnitude is 200 Barg.

Figure 3.20. Boundary and load conditions of the dished head model

Figure 3.21 shows the boundary and load conditions of a quarter model of the tube bend, similar to those for tube plate model, there surfaces can only move within their own planes to restrict rigid body motion, and the internal pressure is 110 Barg.

Figure 3.21. Boundary and load conditions for tube bend model

The boundary conditions and load for the T-joint are demonstrated in Figure 3.22 below. Due to symmetry, only $\frac{1}{4}$ of the geometry is included in the model. Three planes can only move inside

their own planes to prevent rigid body motion, one end of the run pipe is under end load due to internal pressure and the Y direction displacements were coupled on that place to ensure it moves as a plane, the internal pressure is 90 Barg.

Figure 3.22. Boundary and load conditions for T-joint model

3.7.2 Results

For the typical pressure vessel component analysis, the dimensions were determined through design by formula approach, then design checks were conducted using the ASME Section III, Division 5, Subsection HB elastic analysis method (stress classification approach) to obtain the design creep life usage, i.e., creep damage. Creep damage calculation results through reference stress approach were also calculated to compare those from inelastic analysis with different effective creep stress definitions, described in details in Section 3.4.2 and designated in Subsection 6.2.2.

3.7.2.1 Tube plate

3.7.2.1.1 ASME

Figure 3.23 (a) shows the stress intensity of the tube plate model, the ligament near the plate rim demonstrates a large area with stress intensity higher than 61.92Mpa, which is the allowable stress for 300,000 hour life at 580° C, a stress classification line was defined to pass through that region for elastic analysis and it is shown in Fig. 3.23(b).

Figure 3.23. Elastic analysis of tube plate, (a) stress intensity, (b) stress linearization path

Table 3.7. Stress linearization results

Unit: MPa

Path	P_m	P_L+P_b	P_L+P_b/K_t
Path 1	49.37	52.97	52

Assume $P_L=P_m$

Table 3.7 displays the stress linearization results for elastic analysis, they satisfy the requirement of Eq. 3.9, 3.10 and 3.11. Based on the procedure in Section 3.4.5. the creep damage of 300,000 hour service for the ligament based on P_m is 12.18%, and for the surface based on P_L+P_b/K_t , the value is 21.79%.

3.7.2.1.2 Reference stress

Figure 3.24 displayed the equivalent plastic strain contour plot of tube plate model before it collapses, the maximum plastic strain location is at the ligament near the rim, same locations as that for the stress intensity in Figure 3.23. The reference stress, obtained by following the procedure in Section 3.4.4, is 56.38Mpa, and the creep damage for 300,000 hour service at 580° C is 2.25%.

Figure 3.24. Equivalent plastic strain before tube plate collapses

3.7.2.1.3 Inelastic analysis with different effective creep stress definitions

Figure 3.25 shows the creep damage contour plots of the tube plate model, for the first four effective creep stress definitions, the maximum creep damage occurs at the ligament close to the plate rim, while for the combined effective creep stress definition, it appears at the hole wall next to the central hole. The exact maximum creep damage values are listed in the following section.

Figure 3.25. Tube plate creep damage after 300,000 hour service at 580° C with different effective creep stress definitions (a) Mises, (b) Tresca, (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

3.7.2.1.4 Summary

The calculation results of the creep damage after 300,000 hour service at 580° C are summarized in Table 3.8 below, the elastic method, i.e., stress classification, gives the highest value of 21.79%, reference stress method and inelastic method using Tresca as the effective creep stress shows creep damage around 2.5%, while the results provided by other effective creep stress definitions give results less than 1.5%. Even though the elastic method shows the highest creep damage, the highest creep damage location is different from that of other methodologies, and this will be analyzed in the following sections.

Table 3.8. Creep damage summary for tube plate model

Method	Effective stress definition	Highest damage location	Accumulated creep damage
Inelastic	Mises	ligament center near the rim	1.34%
	Tresca	ligament center near the rim	2.89%
	Huddleston	ligament center near the rim	1.34%
	max(max.tensile principal, Mises)	ligament center near the rim	1.34%
	0.9233Max.Tensile Principal +0.0767Mises	ligament bottom near the center	1.31%
Elastic	N/A	ligament top near the rim	21.79%
Reference stress	reference stress	ligament center near the rim	2.25%

3.7.2.2 Dished head

3.7.2.2.1 ASME

The stress intensity under normal operation load, i.e., internal pressure of 20Mpa, and the stress linearization paths are shown in Figure 3.26 (a) and (b) respectively. The ASME results are demonstrated in Table 3.9, they all satisfy Eqs. 3.9 to 3.11. Creep damage after 300,000 hour service is 61.89%. Elastic analysis shows that the critical location for creep damage is along path 1, this is different from that for inelastic analysis later.

Figure 3.26. Dished head elastic analysis (a) Stress intensity, (b) stress linearization path

Table 3.9. Stress linearization results for the dished head model

Unit: MPa

Path	Pm	PL+Pb	PL+Pb/Kt	
Path 1	59.43	59.43	59.43	Pb=0, PL=Pm
Path 2	31.94	31.94	31.94	Pb=0, Pm=PL
Path 3	35.73	54.96	51.11	PL=Pm

3.7.2.2.2 Reference stress

Figure 3.27 shows the equivalent plastic strain before the dished head collapse during limit load analysis, it is expected that the most severe creep damage will occur along Path 1 in Figure 3.26(b), as large deformation appears in that region in Figure 3.25, this location is the same as that from elastic analysis above. Following Section 3.4.4 procedures, the reference stress is 66.27MPa, and the creep damage due to 300,000 hour creep at 580° C under 20MPa internal pressure is 6.2%.

Figure 3.27. Equivalent plastic strain before collapse during limit load analysis

3.7.2.2.3 Inelastic analysis with different effective creep stress definitions

The accumulated creep damages after 300,000 hour service at 580° C under 20MPa were calculated based on different effective creep stress definitions and the data are present in Figure 3.28 below. For the first four effective creep stress definitions, i.e., Mises, Tresca, Huddleston, maximum stress, and combined effective stress, the highest damage occurs at internal surface at the knuckle, but the for the last one, the combined stress definition, the highest creep damage location moves towards the wall inside. The exact creep damage values are presented in the next section with data from other creep damage calculation methodologies.

Figure 3.28. Dished head creep damages with different effective creep stress definitions. (a) Mises, (b) Tresca , (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

3.7.2.2.4 Summary

Table 3.10 displays the creep damages after 300,000 hour service at 580° C with 20MPa internal pressure, the elastic method gives the highest creep damage and using Tresca, i.e., stress density shows the second highest creep damage values. The inelastic method with five different effective creep stress definitions demonstrates the highest creep damage locations are near the dished head knuckle, while the elastic method and reference stress method gives the locations as the cylinder walls.

Table 3.10.Creep damage calculation results for the dished head model

Method	Effective stress definition	Highest damage location	Accumulated creep damage
Inelastic	Mises	Internal surface near knuckle	3.87%
	Tresca	Internal surface near knuckle	9.49%
	Huddleson	Internal surface near knuckle	3.88%
	max(max.tensile principal, Mises)	Internal surface near knuckle	3.87%
	0.9233Max.Tensile Principal +0.0767Mises	wall internal near knuckle	2.06%
Elastic	N/A	Cylinder wall	61.89%
Reference stress	reference stress	Cylinder wall	6.20%

3.7.2.3 Tube bend

3.7.2.3.1 ASME

Figure 3.29(a) shows the stress intensity distribution at the service load and it demonstrates that the maximum stress intensity occurs at the internal surface at the intrados. Figure 3.29(b) displays the stress linearization paths, one is at the bend and the other is away from the bend, both are at the intrados and across the wall.

Figure 3.29.Elastic analysis of tube bend (a) stress intensity, (b) stress linearization path

Table 3.11 shows the stress linearization results of the elastic analysis of tube bend, the results satisfy Eqs. 3.9 to 3.11, and the creep damage calculation based on P_m and $P_L P_b / K_t$ both demonstrate a value of 43.45% for 300,000 hours service at 580° C with internal pressure of 110 Barg.

Table 3.11.Stress linearization results for the tube bend model

Unit: MPa

Path	P_m	P_L+P_b	P_L+P_b/K_t	
Path 1	57.24	57.24	57.24	$P_b=0; P_m=P_L$
Path 2	51.08	51.08	51.08	$P_b=0; P_L=P_m$

3.7.2.3.2 Reference stress

Reference stress method gives a reference stress value of 69.97Mpa and a creep damage of 9.04% for 300,000 hours service at 580° C with internal pressure 110 Barg. Figure 3.30 demonstrates the equivalent plastic strain right before the bend collapses, the wall at the bend intrados is the critical location.

Figure 3.30. Equivalent plastic strain in the tube bend before it collapses during limit load analysis

3.7.2.3.3 Inelastic analysis with different effective creep stress definitions

The creep damage calculation results with different effective creep stress definition for 300,000 hour service at 580° C with 110 Barg internal pressure is shown in Figure 3.31 below. All the calculated high damage regions are at the inside wall surface near the bend at the intrados, and for

the combined stress definition, i.e., combination of maximum tensile principal stress and Von Mises stress, the high creep damage region occupies part of the wall inside.

Figure 3.31. Creep damage of tube bend with different effective creep stress definitions (a) Mises, (b) Tresca, (c) Huddleston, (d) Maximum stress, (e) Combined effective stress.

3.7.2.3.4 Summary

Table 3.12 summarizes the creep damage calculation with different methods applied, the stress classification method gives the highest creep damage of 43.45% while reference stress shows a value of 9.04%; out of the five effective creep stress definitions, Tresca stress gives the highest creep damage of 6.19%, and all other stress definitions demonstrate creep damages less than 3% for 300,000 hour service at 580° C with 110 Barg internal pressure.

Table 3.12. Summary of creep damage calculation results for tube bend

Method	Effective stress definition	Highest damage location	Accumulated creep damage
Inelastic	Mises	inside surface at the bend intrados	2.70%
	Tresca	inside surface at the bend intrados	6.19%
	Huddleston	inside surface at the bend intrados	2.79%
	max(max.tensile principal, Mises)	inside surface at the bend intrados	2.70%
	0.9233Max.Tensile Principal +0.0767Mises	inside surface at the bend intrados	2.21%
Elastic	N/A	tube wall at the bend intrados	43.45%
Reference stress	reference stress	inside surface at the bend intrados	9.04%

3.7.2.4 T-Joint

3.7.2.4.1 ASME

Figure 3.32 shows the stress intensity under 90Barg internal pressure service load and based on that information, four stress linearization paths are defined, they are across the walls at the joint crown (path A), at the run away from the joint (path B), at the branch away from the joint (path C) and at the joint saddle (path D). The stress linearization results are in Table 3.13, they demonstrate that Eqs. (9) to (11) are satisfied and the creep damage for 300,000 hour service life at 580° C are 48.33% for both P_m and P_L+P_b/K_t criteria.

Figure 3.32. Stress linearization for T-joint. (a) stress intensity, (b) stress linearization paths

Table 3.13. Stress linearization results of T-joint

Unit: MPa			
	P_m	P_b	P_t+P_b/K_t
Path A	55.02	0	55.02
Path B	54.59	0	54.59
Path C	54.79	0	54.79
Path D	57.9	0	57.9

$P_L = P_m$
 $P_m = P_L$
 $P_L = P_m$
 $P_m = P_L$

3.7.2.4.2 Reference stress

Figure 3.33 is the equivalent plastic strain contour plot from limit load analysis, it is a snap shot right before the T-joint collapses. The plot demonstrates the critical location for creep damage is on the run wall away from the joint, it means the joint design is good, and we can improve its performance against creep by increasing run wall thickness if we need to. The limit load analysis leads to a reference stress of 68.52MPa, and creep damage of 7.8% for 300,000 hour service at 580° C with 90 Barg internal pressure.

Figure 3.33. Equivalent plastic strain from limit load analysis of the T-joint model, (a) front view, (b) back view

3.7.2.4.3 Inelastic analysis with different effective creep stress definitions

The following Figure 3.34 to Figure 3.38 show the creep damage after 300,000 hours service with different effective creep stress definitions. As with all the five effective creep stress definitions, the maximum creep damage appears either at the opening internal corner at the joint crown or on the external wall at the joint saddle, each figure consists of two views, i.e., front view, showing the cross section of the model and back view, showing the saddle region of the joint.

Figure 3.34. Creep damage of the T-joint model after 300,000 hour service with the 1st effective creep stress definition (Von Mises), (a) front view, (b) back view

Figure 3.35. Creep damage of the T-joint model after 300,000 hour service with the 2nd effective creep stress definition (Tresca), (a) front view, (b) back view

Figure 3.36. Creep damage of the T-joint model after 300,000 hour service with the 3rd effective creep stress (Huddleston), (a) front view, (b) back view

Figure 3.37. Creep damage of the T-joint model after 300,000 hour service with the maximum stress (larger between the max tensile principal stress and Von Mises), (a) front view, (b) back view

Figure 3.38. Creep damage of the T-joint model after 300,000 hour service with the combined stress (weighted combination of the max tensile principal stress and Von Mises), (a) front view, (b) back view

3.7.2.4.4 Summary

Table 3.14 summarizes the creep damage calculation results with different methodologies, those results include maximum the creep damage locations and the magnitudes, it shows that the elastic

analysis gives the most conservative results and reference stress method is the next most conservative.

Table 3.14. Creep damage calculation result summary for the T-joint model

Method	Effective stress definition	Highest damage location	Accumulated creep damage
Inelastic	Mises	internal corner at the joint crown	5.31%
	Tresca	external surface at the joint crotch	6.63%
	Huddleson	internal corner at the joint crown	5.31%
	max(max.tensile principal, Mises)	external surface at the joint crotch	7.27%
	0.9233Max.Tensile Principal +0.0767Mises	external surface at the joint crotch	6.78%
Elastic	N/A	Path across the run pipe wall at the joint crown	48.33%
Reference stress	reference stress	inside surface along the run pipe axial direction	7.80%

3.7.3 Result Summary

Table 3.15 summarizes the creep damage results calculated by different methods for the four typical pressure vessel components.

Table 3.15. Creep damage results by different methods for four typical pressure vessel components

Componet	elastic	Creep damage calculation method					reference stress
		inelastic					
		Mises	Tresca	Huddleson	Max(max.tensile principal stress, Mises)	0.9233Max. Tensile principal stress + 0.0767Mises	
Tube plate	21.79%	1.34%	2.89%	1.34%	1.34%	1.31%	2.25%
Dished head	61.89%	3.87%	9.49%	3.88%	3.87%	2.06%	6.20%
Tube bend	43.45%	2.70%	6.19%	2.79%	2.70%	2.21%	9.04%
T-joint	48.33%	5.31%	6.63%	5.31%	7.27%	6.78%	7.80%

3.8 Discussion

3.8.1 Notched bars

The notched bar simulations compare the Code effective stress measures (Tresca and Huddleston) to a range of predictions via different effective stresses to ascertain which measures are most conservative for severely notched geometries. The notches in these samples likely bound the notch severity found in actual, well-design components and hence these simulations put a bound on potential notch effects.

Three notch geometries are modeled to demonstrate different stress states, Figure 3.15 shows that for Von Mises and Tresca, i.e., the 1st and the 2nd effective creep stress definitions, before creep starts (shown as dashed orange line and dashed blue line in the figure), their maximums are at the notch tips, and after 300,000 hour creep (shown as solid orange line and solid blue line in the figure), their magnitudes at the notch tips decrease and those at other locations away from the notch tips increase due to stress redistribution, but the trends are still the same, in other words, the

magnitudes are still the maximum at the notch tips and decrease towards the bar center. For all the three notch geometrical configurations, after 300,000 hour creep, the Tresca and Von Mises at the notch tips are with similar magnitudes, respectively, but the creep damage magnitudes are quite different there, as shown in Table 3.6. The accumulated creep damage values are more than doubled when the notch geometry changes from semi-circular to semi-elliptical shape, and increase 50% or more when the notch geometry changes from semi-elliptical to V-notched. The initial stress values are quite different between these two cases and the creep damage produced during the stress redistribution further contributes to the different creep damage values.

Similar magnitude changes and trend occur for the maximum tensile principal stress (the maximum stress definition) during 300,000 hour creep. After 300,000 hour creep, the maximum values for the three notch geometries are not at the notch tips, but rather located towards bar center. The magnitudes increase following the notch geometry order: semi-circular, semi-elliptical and V-notched.

Table 3.6 shows the creep damage magnitudes for bars with different notch geometries and with different effective creep stress definitions. All the calculation cases give results < 1 , except when maximum tensile principal stress is regarded as the effective creep stress and with V-notched bar, in that case, the creep damage is 1.13.

Overall, these results show that the Tresca stress used in the primary load design rules is quite reasonable, except potentially for very severe notches (i.e. the V notch). The other ASME rules likely screen out notches with this severity, but for future fitness-for-service uses where cracks may be present additional work on the effective stress measure may be required.

3.8.2 *Typical pressure vessel parts*

The goal of creep damage calculation for the pressure part design is to ensure their safe operation, i.e., no pressure burst, structural collapse, etc. occur during service. More accurate calculation results on the creep damage magnitudes and critical locations would improve component replacement decisions, maintenance schedules and inspection procedure, but for a design calculation a conservative bound on life is imperative. Creep damage calculation for four typical pressure vessel components has been carried out in the report with five effective creep stress definitions plus reference stress method and elastic method (stress classification). The calculation results in Tables 3.8, 3.10, 3.12 and 3.14 show that out of the seven approaches, the ASME elastic method with the Code allowable stress values gives the highest/ most conservative results for all the four components, as it applies the stress condition at full load condition in elastic state, no stress redistribution (or relaxation) is involved, so pressure vessel design using elastic analysis approach should provide conservative results against net section failure, compared with results from other approaches. Table 3.16 shows the first two highest creep damage calculation results, for all the four components, the highest calculated creep damage can be as higher as 650% more than the second highest creep damage.

Table 3.16. The first two highest creep damage calculation results

	highest creep damage		second highest creep damage		magnitude different (mag.1-mag.2)/mag.2)
	method	magnitude	method	magnitude	
Tube plate	elastic	21.79%	Tresca	2.89%	654%
Dished head	elastic	61.89%	Tresca	9.49%	552%
Tube bend	elastic	43.45%	reference stress	9.04%	381%
T-joint	elastic	48.33%	reference stress	7.80%	520%

There is a trend in which method produces the second-highest creep damage fraction. For thick components the Tresca effective stress produced the lowest service life estimate. For thinner components the reference stress approach produced the lowest best-estimate of life. Differences in stress relaxation over the 300,000 hours of creep explains this change.

For the three notched bar models, the maximum of the maximum principal tensile stress, i.e., the 4th effective creep definition and Von Mises, the first effective creep definition, give the highest creep damage out of the five effective creep stress definitions, but it is not the case for typical pressure vessel components, where out of the five aforementioned stress definitions, the former gives the highest creep damage in the T-joint model only and Tresca, the 2nd definition of the five effective creep stress definitions, gives the highest creep damages for the other three models. However, for all the four models, both effective stress definitions predict the highest creep damage location in the whole model at the same location, i.e., for T-joint model, the Tresca and 4th effective creep stress definition both predict the highest creep damage occurs at the external surface at the joint crotch. The above analysis also show that, to predict local creep damage for the typical pressure vessel components, Tresca is a better choice as an equivalent creep stress, except for T-joint model, where the effective creep stress definition defined as the maximum of the maximum tensile principal stress and Von Mises, determines creep life. In fact the T-joint, designed by using stress intensity (Tresca), showed premature cracking at the outside surface where there also used to have welds with pre-existing cracks and other stress concentration sites and maximum tensile principal stress plays critical roles under this circumstances [50].

3.9 Conclusions

Overall, the numerical results described here support the use of the Tresca and Huddleston effective stress measures used in the ASME Code, except for cases with very severe notches like the V-notch bar. The Tresca stress, used in the bounding, primary load design method, bounds damage accumulation in the components studied here and the Huddleston stress, used to accurately characterize the multiaxial effect in the creep-fatigue rules, falls within the range of best-estimate life predictions. More specifically, the Tresca stress provides conservative multiaxial rupture predictions for all the representative components and for the moderately notched U geometries. Moreover, regardless of which effective stress measure is the most accurate, the ASME Code is providing conservative designs – the calculated creep rupture damage at the end of the design life using inelastic analysis and any of the effective stress measures is much less than the use fraction for design-by-elastic analysis for all the components considered here.

4 Assessing the rules against potential material mechanism shifts

4.1 Objectives of the analysis

The objectives of this analysis are to compare the Section III, Division 5, design by elastic analysis results for the T-joint model described above to creep damage calculations using an alternative material model that includes a mechanism shift, causing a shift in the creep rate sensitivity at lower stresses. This model, described below, represents a material mechanism shift from dislocation to diffusion-based creep that likely exists in the actual material. The purpose of the analysis to evaluate the robustness of the Code rules against a material change that cannot be easily measured through short- or intermediate-term creep testing.

4.2 Background

Creep damage estimation is a key part of high temperature design. The creep-fatigue rules in Section III, Division 5, HBB-T provides a creep damage estimation procedure based on elastic analysis, using isochronous stress-strain curves encapsulating creep deformation behavior, or through inelastic analysis using the Huddleston effective stress [5]. Recent development in material constitutive equations and crystal plasticity calculation techniques suggest that the Gr.91 creep deformation mechanism may shift from a dislocation mechanism to a diffusion mechanism. This would manifest as a “bend” in the creep stress-strain rate curve. The potential danger would be extrapolating from the high-stress experimental data to low stress conditions without accounting for this mechanism shift. This could lead to non-conservative results. Figure 4.1 plots two sources of creep rate data: long-term creep deformation experiments currently underway in Japan (red line) and short-term, specialized extrapolated creep rate measurements (black line) [28], [42]–[46]. The extrapolated measurements, based on primary creep data, shows a dramatic shift from a highly rate sensitive, dislocation mechanism to a linear-viscous diffusion mechanism. The direct long-term measurements show a mechanism shift taking places at the same stress level. However, the effect on the minimum creep rate is much less dramatic than the extrapolated short-term measurements would suggest.

Figure 4.1. Experimental creep results from literature for minimum creep rate of Grade 91 loaded in uniaxial tension at 600° C (figure reproduced from [47]).

This section uses a typical pressure vessel component, a tube T-joint, as an example, to show the effect on creep damage when accounting for a mechanism shift of this type. The “mechanism-shift” model is based on Crystal Plasticity Finite Element (CPFE) results conducted at ANL for Grade 91. This CPFE model can predict creep curves and creep rates for the material at very low stress levels, typical of components but impossible to directly test because of the very long test durations [28], [29]. However, because CPFE is time consuming, the component simulations use a macroscale creep model calibrated to match the behavior of the detailed CPFE, rather than CPFE directly.

4.3 Scope of the analysis

The creep damage of a tube T-joint were analyzed through stress linearization and elastic analysis, and inelastic analysis with three different material models: the Grade 91 material model developed for the draft Appendix Z of ASME BPVC Section III, Division 5, a simplified model without a strain rate sensitivity mechanism shift, and a simplified model including the mechanism shift described above. The inelastic analysis uses Huddleston’s effective creep stress definition for the creep damage calculations.

4.4 Methodology

4.4.1 Software and hardware environment

The finite element analysis was accomplished using ANSYS Mechanical 2020 Version R2 on a computer cluster with Intel® Xeon® CPU E5-2698 CPU and 48 cores, the memory size was

500Gbytes. The ANSYS material model user defined subroutine was compiled by Intel Parallel Studio XE 2017.

4.4.2 Elastic analysis through stress linearization

This method is from Section III, Division 5 of the ASME Code and an ANL report [40] gives details of its application. After stress linearization, the stress intensities should satisfy the following

$$P_m < S_{mt} \quad (4.1)$$

$$P_L + P_b < S_m \quad (4.2)$$

$$P_L + \frac{P_b}{K_t} < S_t \quad (4.3)$$

P_m , P_L , P_b , K_t , S_{mt} , S_m and S_t are defined in Section III, Division 5, Subsection HB, Subpart B, and for Grade 91 material at 600° C, the values for S_{mt} , S_m and S_t are 48, 101 and 48 in Mpa, respectively. These data were taken from the HBBDATA package developed at ANL.

The creep life under the load t_r can be calculated through HBBDATA function TIME_S_T(MATERIAL, STRESS, TEMPERATURE), the inputs are “gr91” for the material, $P_L + \frac{P_b}{K_t}$ value for stress in MPa, and 600 for temperature in Celsius. The temperature varies here from the previous analyses and is slightly higher than likely nuclear use of Grade 91 to match the temperature available in the mechanism-shift model. The creep damage is calculated as the ratio of 300,000 hours to t_r .

4.4.3 Inelastic analysis

4.4.3.1 Thermal and mechanical load transients

Creep damage calculation for 300,000 hours service with inelastic analysis takes long time to accomplish, as we focus on low stress creep damage and fatigue damage is negligible, a 1000 hour load cycle was chosen for creep damage calculation, to get the results for 300,000 hours, the 1000 hour results will be multiplied by 300. The thermal and mechanical load transients were designed to minimize thermal stress peaks, and 1000 hours is sufficient to reach near-steady stress relaxation conditions for a Gr91 material components.

4.4.3.2 Effective creep stress definitions

An effective stress is used to map the stress relaxation history calculated as described above into a scalar to index to the ASME values of S_r .

Assume three principal stresses are σ_1 , σ_2 and σ_3 , and $\sigma_1 > \sigma_2 > \sigma_3$, and based on Huddleston's work [5], the effective creep stress applied in this report is defined as

$$\sigma_e = \bar{\sigma} \exp \left[c \left(\frac{J_1}{S_s} - 1 \right) \right] \quad (4.4)$$

where $J_1 = \sigma_1 + \sigma_2 + \sigma_3$;

$$S_s = (\sigma_1^2 + \sigma_2^2 + \sigma_3^2)^{\frac{1}{2}};$$

$$\bar{\sigma} = \frac{1}{\sqrt{2}} [(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2]^{\frac{1}{2}};$$

$\sigma_1, \sigma_2, \sigma_3$ are the three principal stresses, C is a constant for Gr.91 defined as follows:

(3) If $J_1/S_s \geq 1.0$, $C = 0.16$;

(4) If $J_1/S_s < 1.0$, $C = 0$.

We consider the Huddleston stress here as it is the effective stress measure used in the ASME creep-fatigue design rules.

4.4.3.3 Creep damage calculation in material model in ANSYS

Section HBB-T-1420 in Section III, Division 5 provides an equation based on time fraction for creep damage calculation:

$$D_c = \int_0^t \frac{dt}{T_d(\sigma_e, T)} \quad (4.5)$$

where t is the time, dt is the time increment, T_d is the rupture life with the effective stress σ_e and temperature T during that time increment, and its values are obtained from HBBDATA function TIME_RUPTURE(MATERIAL,TEMPERATURE,STRESS). The equation was combined into ANSYS USERMAT subroutine, the accumulated creep damage results at different time intervals were output as state variables.

4.5 Assumptions

This analysis is based on the following assumptions:

1. The material for all the models is Grade 91, isotropic and homogeneous; material response is constant over time, i.e., effects of material degradation over time, are neglected. There is no coupling between the calculated creep damage and the material's deformation.
2. The material thermo-physical properties and constitutive equations applied in FEA are the same for both the welds and the base metal (in cases with welded construction).
3. Zero residual stress.
4. A viscoplastic material model, developed for use with the ASME Section III, Division 5 design by inelastic analysis method, represents the deformation of the material [48].

4.6 Analysis on Tube T-joint

4.6.1 Inputs

4.6.1.1 Geometries

The tube T-joint geometry is shown in Figure 4.2 below. Due to symmetry, only quarter of the T-joint is included in the geometry for FEA.

Figure 4.2. T-joint model geometry.

4.6.1.2 Boundary conditions and loads

4.6.1.2.1 Thermal and mechanical load transients

In each load cycle the fluid flow temperatures in the main pipe and the branch ramp from 300° C to 600° C while the pressures change from 0 Barg to 40 Barg , and the flow keeps at 600° C /40 Barg for 976 hours and ramps down to 300° C /0 Barg in another 24 hours. The plots in Figure 4.3 and Figure 4.4 show the transients of temperature, pressure and heat transfer coefficients during one load cycle.

Figure 4.3. Temperature and pressure transients from one load cycle.

Figure 4.4. Heat transfer coefficient changes during one load cycle.

4.6.1.2.2 Boundary conditions

The internal walls of the T-joint face the fluid thermal transients described in Figure 4.3 and Figure 4.4, other surfaces are thermally insulated. The pressure boundary conditions and load for the T-joint are demonstrated in Figure 4.5 below. In the figure, three planes can only move inside their own planes to prevent rigid body motion, one end of the run pipe is under end load due to internal pressure and the Y direction displacements were coupled on that plane to ensure it moves as a plane, the internal pressure is 40 Barg at steady state and the pressure transient is displayed in Figure 4.3.

Figure 4.5. Pressure load boundary condition.

4.6.2 Results

4.6.2.1 ASME

Figure 4.6 demonstrates stress linearization paths for elastic analysis based on the procedure in the ASME Code and the results are presented in Table 4.1. Considering S_{mt} , S_m and S_t are 48, 101 and 48 in Mpa, respectively, the results in the table demonstrate that Eq. (1) to Eq.(3) are satisfied and the creep damage for 300,000 hour service life < 1 at 600° C. As HBBDATA provides the possible

lowest stress values for 300,000 hour life only, and the life for stress values lower than them are unknown, so is the life usage (damage).

Figure 4.6. Stress linearization paths for elastic analysis.

Table 4.1. Stress linearization results.

Unit:	MPa			
	P_m	P_b	$P_L + P_b/K_t$	
Path A	25.34	0	25.34	$P_L = P_m$
Path B	24.45	0	24.45	$P_m = P_L$
Path C	24.33	0	24.33	$P_L = P_m$
Path D	4.69	0	4.69	$P_m = P_L$
Path E	24.01	0	24.01	$P_L = P_m$
Path F	23.15	0	23.15	$P_m = P_L$

4.6.2.2 Inelastic analysis with different material models for Gr.91.

The following Figure 4.7 and Figure 4.8 show the creep damage after 300,000 hours service with different material model for Gr.91, the data are based on the inelastic FEA results for a 1000 hour load cycle by multiplying them by 300. For all these three material models, the maximum creep damage locations are at internal corner of the joint at the joint crown, and the model with “shift” shows a much large (3X more) maximum creep damage than those from the other two models, and the outside wall at the crotch also show a greater creep damage.

Figure 4.7 T-joint front view with 300,000 hour creep damage contour for different material models (a) normal, (b) non-shift, (c) shift.

Figure 4.8 T-joint back view with 300,000 hour creep damage contour for different material models (a) normal, (b) non-shift, (c) shift.

4.7 Discussion

The results in Figure 4.7 and Figure 4.8 demonstrate that for all the three material models applied, there are two regions with high creep damage, one is at the internal corner of the joint crown, the other is at the outside surface of the joint crotch. The creep damage in the higher damage region, using normal material mode results as reference, are higher when “shift” material model is applied, while the “non-shift” material model lowers the creep damage in those regions. One of these has a higher triaxiality factor, which is defined as

$$triaxiality\ factor = \frac{(\sigma_1 + \sigma_2 + \sigma_3)}{3\bar{\sigma}} \quad (4.6)$$

Figure 4.9 Triaxiality factor of T-joint.

Figure 4.9 shows a triaxiality factor contour plot, it demonstrates that one of the high creep damage regions, i.e., outside surface at the joint crotch, has high triaxiality, which prompts void formation [49] leading to higher creep deformation rate as included in the CPFEM model [28], [29].

4.8 Conclusions

These results suggest that a mechanism shift from dislocation-dominated creep at higher stresses to diffusion-dominated creep at lower stresses could reduce the margin against rupture in a properly-designed ASME Section III, Division 5 Class A component. However, the examples here demonstrate that the component would still retain an ample margin against rupture failure even in the low stress regime, it is only the relative margin that decreases.

Table 4.2. Creep damage fractions at 300,000 hours.

Model	Creep damage at 300,000 hours
Reference (Appendix Z)	0.070
Simple, no shift	0.038
Simple, with shift	0.236

The component passes the Section III, Division 5, Class A primary load design criteria for 300,000 hour life and passes the design-by-inelastic analysis creep-fatigue check (with negligible fatigue), using all three material models. However, as shown in Table 4.2, the three inelastic models all produce different final creep-damage fractions at the end of the component design life. The simple model, that does not include the shift, predicts a much lower damage fraction when compared to the shift model.

The unshifted model is analogous to the deformation model, via the isochronous stress-strain curves, currently contained in Section III, Division 5. These models are calibrated against exper-

imental data which (with the exception of the long-running Japanese tests) does not sample material deformation in the very low stress, diffusion-dominated regime. The “with shift” model developed for this report includes this mechanism shift. The difference between the creep damage fractions for the “with shift” and “without shift” models is indicative of the potential effect of this material phenomenon in the Code rules as a whole – it does decrease the relative margin against rupture, but Code designs still retain a substantial margin against rupture, even though they do not explicitly represent the effect.

While there is likely some reduction in margin in low stress low stress operating conditions, the final creep damage fractions at 300,000 hours are significantly less than 1.0, indicating a significant margin against rupture failure even for the mechanism-shift model. Moreover, this example likely overexaggerates the effect of the mechanism shift for several reasons:

1. The CPFE model, underlying the “with shift” model used here, was calibrated to match the Kloc [43] experimental data. This data was collected using specialized, sensitive samples and relies on extrapolating primary creep rates measured over comparatively short time periods to long-term deformation behavior. The long-term Japanese tests data [44] does suggest a mechanism shift occurs around the stress value predicted by the Kloc experiments. However, the Japanese tests suggest a much less dramatic mechanism shift. As such, the “with shift” model here may significantly overestimate the change in material mechanism. If all Grade 91 falls closer to the Japanese experimental data than the reduction in margin will be much less.
2. There may be, likely is, a corresponding shift in the material time-to-rupture data for low stresses, which is not captured in the Code rupture data. The rupture data in the ASME Code is extrapolated, like the deformation data, from higher stress experiments. As with the deformation data, this extrapolation may miss a mechanism shift under lower stress component operating stresses. These two shifts may be synergistic – that is the reduction in stress relaxation cause by a shift to diffusional creep may be offset by a corresponding shift in rupture life. For example, a creep-fatigue damage model base on ductility exhaustion, if the material ductility does not depend on stress, would not predict any intrinsic reduction in material rupture resistance corresponding to a reduction in the stress relaxation rate. This is, this apparent reduction in margin in the low-stress regime may be an artifact of the ASME damage approach and the fact these simulations account for a mechanism shift in material deformation but not in material rupture resistance.

5 Summary and Recommendations

5.1 Summary

Creep notch effects describe experimental results from notched versus smooth creep specimens with the same gauge cross section and hence same average principal stress. For notch strengthening materials the notched specimen will have a longer rupture life than the smooth specimen, for the same applied load. A notch weakening material has a shorter rupture life for the notched specimen than the smooth specimen. There is some limited experimental evidence that there is no definitive characterization of materials as notch strengthening and notch weakening. That is, a material may transition from a notch weakening to a notch strengthening behavior depending on the temperature, applied load, notch severity, and other factors.

Notch effects are caused through the interaction of two effects:

1. The stresses at a notch will relax over time due to creep deformation. Damage accumulates during this relaxation, contributing to failure. Relaxation is material dependent.
2. Materials fail differently under multiaxial loads than they fail under standard uniaxial creep conditions. Again, multiaxial rupture is a material-dependent phenomenon.

Notch effects are relevant to high temperature creep design. Realistic engineering structures contain stress concentrations caused by openings, nozzles and penetrations, thickness changes, supports, and even geometric discontinuities at shell/head transitions. All of these geometric features can be viewed as notches, which affect the creep deformation and failure of the material.

This report analyzes the rules in Section III, Division 5 of the ASME Code covering the design and construction of high temperature nuclear reactor structural components. The report focuses on the primary load design rules in the creep regime, guarding against creep failure under the approximate, steady creep conditions. The report analyzes the current rules and describes specific analyses aimed at quantifying the effect of notches and multiaxial stresses on the adequacy of the Code rules, focusing on multiaxial rupture and deformation mechanism shifts.

5.2 Recommendations

Generally, this report demonstrates that the primary load design rules in Section III, Division 5, Subsection HB, Subsection B for Class A components adequately account for notch effects. Several provisions of the Code combine to provide adequate coverage of notch effects, specifically guarding against notch weakening behavior causing premature failure:

1. Notch tests on the qualified Class A materials have only ever demonstrated notch strengthening behavior. The only indication of a shift in behavior towards notch weakening behavior for severe notches or lower stresses/lower rupture lives comes from extrapolation from relative notch weakening behavior or modeling/simulation results.
2. In a common limiting case the primary load design rules can be viewed as a direct check against rupture data using the steady state stress distribution and a Tresca effective stress.

The analyses described in Chapter 3 suggest, somewhat surprisingly, that the Tresca measure is generally conservative, meaning the primary load allowable stress checks using the stress intensity generally conservatively bound multiaxial rupture.

3. The creep-fatigue rules provide a backstop on notch effects by explicitly accounting for peak stress redistribution and using creep damage/relaxation stress measures explicitly designed to bound (design by elastic analysis) or represent (design by inelastic analysis using the Huddleston stress) multiaxial creep failure.
4. Generally, the ASME design rules are quite conservative versus best-estimate predictions of steady-state creep rupture regardless of the effective stress measure used to predict multiaxial rupture. This large conservatism induced by the factors on the allowable stress and conservative design approximations means that there is plenty of margin in designs produced according to the ASME rules to account for an unknown notch effects not quantified and assessed in this report. Given that many of the life estimation methods surveyed here are used to design non-nuclear components and represent the best current engineering practice, this suggests the ASME rules are substantially conservative against actual failure.
5. The ASME creep deformation and rupture data for the low stress, long-life component regime relies on extrapolating data from higher stress, shorter-term experiments. For some materials, like Grade 91, this extrapolation may miss a mechanism shift affecting the relative trend in creep deformation rate, rupture life, or both. This report demonstrates that missing such a mechanism shift may lead to a reduction in the Code margin against creep rupture. However, this study may overexaggerate the effect of the mechanism shift, as described in the main body of the report. Even for this exaggerated case the Code retains ample margin against creep rupture failure.

There are a few aspects of the current Code rules that warrant additional examination. In all cases this additional examination would be to optimize the rules and do not have significant implications for the adequacy of the current rules:

1. Recent design approaches under consideration at ASME replace the Tresca criteria with the von Mises criteria. Swapping out the Tresca stress for the von Mises stress without further consideration could remove some of the conservatism for multiaxial effects described in this report. This should be kept in mind as new design methods are developed.
2. The Code effective stress measures are only based on biaxial rupture test data, not truly triaxial stress conditions like those found in actual components. Triaxial creep tests are extremely difficult to perform and, in effect, not practically available. If a triaxial testing methodology becomes available or if physically-based modeling and simulation becomes accurate enough to predict triaxial rupture the Tresca and Huddleston measures should be reconsidered in light of the new data. However, the Code rules provide significant margin

against best-estimate predictions of multiaxial rupture and so this issue does not significantly impact the adequacy of the current rules.

3. Component testing would be invaluable in validating the ASME design rules. By definition component tests would assess the design rules under realistic states of stress. These types of tests would be the best validation of the Code multiaxial design rules.
4. The Code tacitly bases the deformation measures used to calculate stress relaxation in the creep fatigue rules (and, by extension, the backstop to the primary load rules accounting for peak stress redistribution) on comparatively short-term creep deformation data. Very long-term deformation or stress relaxation data is not available, especially for multiaxial stresses. One potential concern is a mechanism shift in the material, likely from dislocation-based creep deformation at higher, experimental stresses to diffusion-dominated mechanisms at lower stresses which would make the current deformation models, extrapolated from the higher stress regime, inaccurate. As with #2 above, if this data becomes available in the future, then the Code procedures should be reassessed. However, the Code provides significant design margin against rupture and so it is unlikely future data would demonstrate any inadequacy in the current rules. Material mechanism shifts are a potential concern, though the overall conservatism of the Code design approach would guard against any currently experimentally unknown mechanism shifts. The Code design approach should be continually assessed against new experimental data and modeling and simulation information examining the low stress regime, as it becomes available.
5. For very severe notches and stress concentrations, the Code effective stress measures (Tresca and Huddleston) may be insufficient as they do not include the role of the maximum principal stress. This should not affect the Code design rules, as the other provisions of the Code will screen out geometries with such severe notches. However, for future fitness-for-service use, which will require analyzing components with cracks or other flaws, a better effective stress measure may be required.

The scope of the report was limited to the classical notched creep effect, which occurs under steady loading conditions. Creep-fatigue will be a significant consideration for future high temperature reactor components. Section III, Division 5 does guard against the initiation of creep-fatigue damage through the rules in Appendix HBB-T. This report only considers the creep damage calculation part of these rules.

The Division 5 creep-fatigue design rules do consider the effects of multiaxial on the fatigue damage calculation and then the effect of fatigue multiaxiality on creep-fatigue life through the creep-fatigue interaction diagram. The Code calculates the fatigue damage fraction using Miner's rule, but with a multiaxial effective strain defined in HBB-T-1413. Moreover, the rules for multiaxial cycle counting refer to the well-established Section III, Division 1 rules. Additionally, except for special circumstances like thermal striping in certain locations in sodium-cooled reactors, the design of components with realistically long ASME design lives tends to be dominated by the creep damage calculation and not the fatigue damage.

To the extent that the fatigue damage calculation procedure generally agrees with current industry practice, including the Section III, Division 1 rules, specific validation of the Section III, Division 5 approach for multiaxial fatigue does not seem warranted. However, some additional verification

of the rules by detailed analysis or component testing might be prudent for the specialized circumstances, like thermal striping, where multiaxial fatigue might significantly contribute to the expected component failure modes.

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