FINITE ELEMENT ANALYSIS OF STRESS RUPTURE IN PRESSURE VESSELS EXPOSED TO ACCIDENTAL FIRE LOADING

by

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Abstract

A numerical model that predicts high temperature pressure vessel rupture was developed. The finite element method of analysis was used to determine the effects that various parameters had on pressure vessel failure. The work was concerned with 500, 1000 and 33000 US gallon pressure vessels made of SA 455 steel. Experimental pressure vessel fire tests have shown that vessel rupture in a fully engulfing fire can occur in less than 30 minutes. This experimental work was used both to validate the numerical results as well as to provide important vessel temperature distribution information.

Due to the fact that SA 455 steel is not meant for high temperature applications, there was little published high temperature material data. Therefore, elevated temperature tensile tests and creep rupture tests were performed to measure needed material properties. Creep and creep damage constants were calculated from SA 455 steel’s creep rupture data.

The Kachanov One-State Variable technique and the MPC Omega method were the creep damage techniques chosen to predict SA 455 steel’s high temperature time-dependent behaviour. The specimens used in the mechanical testing were modeled to numerically predict the creep rupture behaviour measured in the lab. An extensive comparison between the experimental and numerical uniaxial creep rupture results revealed that both techniques could adequately predict failure times at all tested conditions; however, the MPC Omega method was generally more accurate at predicting creep failure strains. The comparison also showed that the MPC Omega method was more numerically stable than the One-State Variable technique when analyzing SA 455 steel’s creep rupture.

The creep models were modified to account for multiaxial states of stress and were used to analyze the high temperature failure of pressure vessels. The various parameters considered included pressure vessel dimensions, fire type (fully engulfing or local impingement), peak wall
temperature and internal pressure. The objective of these analyses was to gain a better understanding of the structural failure of pressure vessels exposed to various accidental fire conditions.

The numerical results of rupture time and geometry of failure region were shown to agree with experimental fire tests. From the fully engulfing fire numerical analyses, it was shown that pressure vessels with a smaller length to diameter ratio and a larger thickness to diameter ratio were inherently safer. It was also shown that as the heated area was reduced, the failure time increased for the same internal pressure and peak wall temperature. Therefore, fully engulfing fires produced more structurally unstable conditions in pressure vessels than local fire impingements.
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**Nomenclature**

- \(a,b,c,d\): Theta Projection Concept creep constants
- \(A,n\): Creep constants shared by Norton and Kachanov
- \(A', \chi, B'\): One-State Variable equation creep constants
- \(b\): Body force vector
- \(C\): System damping matrix
- \(C_{L,M}\): Larson-Miller constant
- \(d\): Symmetric component of gradient of the velocity vector
- \(e\): Euler-Almansi strain tensor
- \(\epsilon\): Base of natural logarithm
- \(F\): Global force matrix
- \(K\): Global stiffness matrix
- \(m\): Creep constant shared by Norton, Kachanov and MPC Omega
- \(M\): System mass matrix
- \(P_{L,M}\): Larson-Miller parameter
- \(p, c\): MPC Omega creep constants
- \(r\): Kachanov creep constant
- \(s\): Surface in spatial description
- \(S\): Engineering stress
- \(S_{ij}\): Deviatoric stress component
- \(T\): Temperature
- \(t\): Time
- \(t_s\): Time in service
- \(t_r\): Rupture time
- \(T\): Traction vector
- \(u\): Displacement vector
- \(v\): Volume in spatial description
- \(\dot{v}\): Velocity vector
Greek

$\alpha$ One-State Variable equation creep constant related to multiaxial stress-state behaviour

$\alpha_i, \beta_i, \gamma_i, \delta_i$ Theta Projection Concept creep constants

$d\lambda$ Plastic compliance

$\varepsilon$ True strain

$\varepsilon_n$ Engineering strain

$\bar{\varepsilon}$ Effective strain

$\varepsilon^c$ Uniaxial creep strain

$\dot{\varepsilon}^c$ Uniaxial creep strain rate

$\bar{\varepsilon}^c$ Effective creep strain rate

$\dot{\varepsilon}_y^c$ Multiaxial creep strain rate component

$\dot{\varepsilon}_0$ MPC Omega creep constant, initial creep strain rate

$\varepsilon^p$ True plastic strain

$\Omega$ Surface

$\Omega_p$ MPC Omega creep constant, omega

$\sigma$ True stress

$\sigma_n$ Engineering stress

$\sigma_m$ Hydrostatic stress

$\bar{\sigma}$ Effective stress

$\sigma_r$ One-State Variable equation rupture stress

$\sigma_1$ Maximum principal true stress

$\sigma$ Cauchy stress tensor

$\theta_1, \theta_2, \theta_3, \theta_4$ Theta Projection Concept creep constants

$\omega$ Creep damage

$\dot{\omega}$ Creep damage rate

$\rho$ Density at time t
Chapter 1 Introduction

1.1 Motivation

Pressure vessels have been experimentally shown to fail when exposed to accidental fire loads. The failure can be highly destructive if the pressure vessel contains both a saturated liquid and vapour at the time of failure as a boiling liquid expanding vapour explosion (BLEVE) could occur (Birk et al., 2006). Failure occurs primarily due to high wall temperatures in the vapour wetted region of the vessel. With a properly working and sized pressure relief valve (PRV), the vessel can still fail due to wall strength degradation at high temperatures even though the vessel pressure is within design limits (Birk et al., 2006). Experimental pressure vessel fire tests have shown that thermally unprotected pressure vessels could fail in less than 30 minutes (Birk et al., 2006). This is a topic of engineering interest since some pressure vessel design standards, such as that for certain North American Rail Tank-Cars (CFR 49 Part 179, 2008; CGSB-43.147, 2005; ASME, 2001), require that pressure vessels resist failure for at least 100 minutes in defined fully engulfing fires. Therefore, if a pressure vessel is exposed to an accidental fire, there exists the possibility for a highly destructive explosion to occur in a relatively short amount of time. The motivation for this work was to employ finite element analysis (FEA) in order to better understand the structural failure of pressure vessels exposed to a variety of accidental fire loading conditions.

1.2 Pressure Vessel Failure

This work deals with pressure vessels containing pressure liquefied gases such as propane, and any pressure vessel discussed here should be assumed to be filled with propane. Also, the work deals only with pressure vessels made of ASME SA 455 steel and all pressure vessels discussed here should be assumed to be made of SA 455 steel. However, pressure vessels
made of other steels could also be analyzed using the same techniques presented in this work; new material constants would need to be investigated.

When a pressure vessel is heated due to an accidental fire, its wall material weakens and its internal pressure rises. These two effects push the tank towards failure. However, most pressure vessels are equipped with a pressure relief valve (PRV) that is programmed to open in order to release pressure from the vessel once the internal pressure has risen above a predetermined value. Therefore, in the event of a vessel being exposed to an external heat source, a race between the fire weakening the vessel material and the PRV emptying the vessel occurs. Figure 1-1 shows a pressure vessel failure that occurred during an experimental fire test.

![Experimental pressure vessel fire test](image)

**Figure 1-1:** Experimental pressure vessel fire test (Birk private communication, 2008).

Depending on the fill level of a pressure vessel, a certain percentage of the vessel volume will be occupied with saturated liquid and the remainder will be occupied with vapour. The compressed liquid will occupy the bottom portion of the pressure vessel as it has a higher density than the vapour. Data from fire tests (see for example Birk et al. (1994)) show that the liquid wetted wall stays below about 130°C during a fully engulfing fire, while temperatures of the
vapour wetted wall can rise to anywhere between 550°C to 750°C depending on the intensity of the fire as well as other external factors. The liquid wetted wall remains much cooler because the liquid propane is much more efficient at removing heat from the vessel wall than the vapour is. This is because liquid propane has a higher thermal conductivity, density and specific heat than vapourous propane; i.e. liquid propane has much higher convective heat transfer coefficient than vapourous propane. Phase change due to boiling of the liquid also enhances heat transfer from the vessel wall to the liquid. When pressure vessels are exposed to engulfing fires, they consistently fail at the location of peak wall temperature, somewhere in the vapour space wall region.

Short-term, high-temperature failures are termed stress-rupture failures. Stress-rupture includes two significant subcategories: short-term overheating and high-temperature creep (Viswanathan, 1989).

Short-term overheating failures are driven by plastic deformation. In short-term overheating, a component is exposed to excessively high temperatures to the point that material yielding begins. Considerable deformation of the given component is observed during short-term overheating in the form of reduced wall thickness and the failure surface resembles a knife edge fracture. Due to the excessively high temperatures, material phase changes are common during short-term overheating (Viswanathan, 1989).

High-temperature creep failures are driven by tertiary creep deformation. Temperatures are high enough to result in relatively fast failure times but not high enough to cause material yielding. Little or no reduction in wall thickness is observed during high-temperature creep; however, significant and measurable creep deformation does occur. The fracture surface will have thick edges since creep damage creates link ups of individual voids (Viswanathan, 1989).
Kraus (1980) experimentally observed that the quicker a tertiary creep dominated creep failure occurred, the more ductile the rupture was. This indicates that there exists a cross over region between high-temperature creep and short-term overheating in which both phenomena are acting to deform the material.

It is important to note that a combination of these two failure modes can occur. For example, temperature and stresses can be high enough to cause material yielding but not structural failure by plastic deformation. In this case, after the material yields, high-temperature creep would continue to occur until failure. In order to accurately predict pressure vessel failure behaviour, both stress-rupture failure modes need to be incorporated into the numerical analysis model. Therefore, the analysis model will compute high-temperature creep deformation while checking for and if necessary, computing localized plastic deformation. This is an elastic-plastic-creep deformation model.

From pressure vessel fire tests conducted by Birk et al. (2005) both stress rupture failure modes were observed. Very rapid failures (<10 mins) in which peak wall temperatures were very high (above 720°C) had knife edge fractures, indicating short-term overheating. Failures that took longer and had lower peak wall temperatures (<680°C) were characterized by some necking and rough failure edges, indicating a combination of short-term overheating and high-temperature creep.

1.3 Objectives

In order to develop a pressure vessel finite element analysis failure model, three main objectives were completed: SA 455 steel’s elevated temperature tensile and high-temperature creep properties were measured, creep properties were extracted from experimental data and
validated for uniaxial finite element creep analysis, and a pressure vessel finite element analysis model was developed.

In order to model the phenomena that govern the tank’s failure, SA 455 steel high temperature material properties were needed. ASME Boiler and Pressure Vessel Code (ASME, 2004) provided some limited data on the temperature dependency of SA 455 steel’s mechanical properties. However, it was impossible to get detailed true stress-true strain tensile data from this ASME data. Moreover, there was no published creep rupture data for SA 455 steel. This lack of high temperature data was due to the fact that SA 455 steel is not intended for high temperature applications. However, for the accidental loading being investigated, these properties were needed to conduct the analyses. Therefore, a series of mechanical property evaluation tests were performed at Queen’s University.

Creep of steels is usually studied on the time scale of hundreds or thousands of hours. Since stress rupture of pressure vessels occurring on the order of minutes was investigated, it was not clear which creep damage model(s) would best suit this application. Thus, an extensive literature review was conducted to choose creep damage models that showed potential to produce accurate results for this work. To ensure that the chosen creep damage models were performing adequately, finite element analyses were carried out using the creep constants extracted from the experimental material data to predict the uniaxial creep behaviour observed in the lab. For these analyses, the creep models were implemented into commercial FEA computer packages ANSYS and ABAQUS via creep user subroutines. An in-depth comparison between experimental and numerical analysis of the uniaxial creep behaviour of SA 455 steel was performed to validate all derived creep constants.
The scope of this work was to model the behaviour of pressure vessels exposed to a range of accidental loading conditions using concepts of solid mechanics. Therefore, the work was conducted with constitutive equations that were macroscopic in nature (dealt with stresses and strains averaged over representative volumes). Nevertheless, these macroscopic representative equations were developed to reflect the microstructural mechanisms that govern the material behaviour as best as possible.

Several finite element analysis models were developed using ANSYS and ABAQUS to examine the stress rupture behaviour of various pressure vessels exposed to various loadings. The effects that wall thickness, wall temperature, vessel internal pressure, local hot spots and vessel dimensions have on pressure vessel failure were investigated.
Chapter 2 Literature Review

The literature used to gain background information on high temperature structural failure of pressure vessel steel is split into the following three areas: mechanical testing, numerical creep failure prediction and pressure vessel failure analysis.

2.1 Mechanical Testing

2.1.1 Tension Tests of Metallic Materials

The American Standard of Testing Materials (ASTM) standard E 8M-98 (ASTM, 1998) presents the needed information to conduct a uniaxial tensile test on a metallic material. The following is a brief summary of key points presented in the standard that apply to the mechanical testing that was performed. For a plate-type specimen with pin ends, the standard specifies that the gage length must be at least 4 times the thickness of the sample. During the tensile tests, a strain rate between 0.05min⁻¹ and 0.5min⁻¹ must be used. An extensometer should be used to get accurate strain measurements during the room temperature tensile test. The yield strength is determined by the offset method; finding the intersection of the tensile test curve with the 0.2% strain offset of the elastic portion of the tensile test curve. The ultimate tensile strength is found by dividing the maximum load achieved in the tensile test by the original sample cross sectional area. Elongation is the percentage change in gage length achieved during the testing. The gage length is to be measured to the nearest 0.05mm before and after the tensile test in order to calculate the elongation.
2.1.2 Elevated Temperature Tension Tests of Metallic Materials

The American Standard of Testing Materials (ASTM) standard E 21-92 (ASTM, 1998) presents the needed information to conduct an elevated temperature uniaxial tensile test on a metallic material. The following is a brief summary of key points presented in the standard that apply to the mechanical testing that was performed. All of the points mentioned above in the summary of standard E 8M-98 apply to elevated temperature tests as well. In addition, the following criterion must also be met. For a gage length of 2” or less, at least two thermocouples must be attached to the specimen, one near each end of the gage length. For a test temperature up to 1000°C a difference of ± 3°C is allowed between the indicated test temperature and the nominal test temperature, above 1000°C, ± 6°C is allowed. Temperature overshoots during initial sample heating should not exceed these limits. Unless otherwise stated, the time of holding at test temperature prior to the start of a test should not be less than 20 minutes. This time is given to ensure that the specimen has reached equilibrium and that the temperature can be held between the limits mentioned above.

2.1.3 Creep Rupture Tests of Metallic Materials

The American Standard of Testing Materials (ASTM) standard E 139 – 96 (ASTM, 1998) presents the needed information to conduct creep rupture tests on a metallic material. The following is a brief summary of key points presented in the standard that apply to the mechanical testing that was performed. A creep rupture test is defined as “a test in which progressive specimen deformation and the time for rupture are measured”. These measured values depend on test load and temperature. The specimen specifications mentioned in standard E 8M-98 apply to creep rupture testing as well. For a gage length of 2” or less, at least two thermocouples must be
attached to the specimen, one near each end of the gage length. For a test temperature up to 1000°C a difference of ±2°C is allowed between the indicated test temperature and the nominal test temperature, above 1000°C, ±5°C is allowed. Temperature overshoots during initial sample heating should not exceed these limits. Unless otherwise stated, the time of holding at the test temperature prior to the start of the test should not be less than 1 hour. This time is given to ensure that the specimen has reached equilibrium and that temperature can be held between the limits mentioned above. During testing, temperature overshoots are much more of a concern than a temperature drop and any test that has a temperature rise above the limits specified above should be rejected. Over temperature may considerably accelerate creep and alter test results whereas, low temperatures usually will not damage the material in this fashion and if exposure time at low temperatures is not significant the test results may be kept. Strain readings must be made at least every 24 hours or 1% of the estimated duration of the test. The load should be applied such that shock loads or overloading due to inertia is avoided. The following data should be included in the report from a creep rupture test: time to rupture, test temperature, nominal test stress, elongation, minimum creep rate and time to various (0.1%, 0.2%, 0.5%, 1.0%, 2.0%, 5.0%) total strains; for a full list of data to be reported from a creep rupture test, see ASTM standard E 139 – 96 (ASTM, 1998).

2.2 Numerical Creep Failure Prediction

Creep failure can be numerically predicted using a creep damage model. The idea of quantifying a creep damage parameter was originally introduced by Kachanov (1960). The main purpose of a creep damage model is to predict failure in a component that is exposed to creep loading conditions. Most of the models work similarly when coupled with finite element analysis. A damage parameter is integrated over time at each node or integration point of the
finite element analysis model. That node or integration point is considered to have failed when the damage parameter reaches a certain critical value (usually 1). Creep deformation and creep damage models commonly used in research and industry are reviewed in the following subsections.

2.2.1 Creep

The important dates in the development of the mathematical modeling of creep were (Lemaitre and Chaboche, 1985):

- 1910, Andrade’s law of primary creep.
- 1929, Norton’s law which linked the rate of secondary creep to stress.
- 1934, Odqvist’s generalization of Norton’s law to the multiaxial case.

Collins (1993), among others, provides comprehensive information on creep phenomena, creep behaviour and design for creep.

Creep deformations are permanent, time-dependent and are considered to develop and accumulate in accordance with the concepts of incompressible material deformation. Time dependent implies that the rate of deformation is significant enough that it must be considered in the engineering assessments pertaining to the viability of a structure. Continued creep deformation occurs under constant load and generally, as a rule of thumb for metals and metal alloys, it must be accounted for if temperatures are above 40%-50% of the homologous material melting temperature; that is temperature in degrees Kelvin. Extended time at elevated temperatures can act as a tempering process, reducing material strength.

The melting point of a generic medium carbon steel is about 1450°C (Reed-Hill and Abbaschian, 1992). Therefore, creep of medium carbon steels should be taken into account at temperatures above about 420°C – 590°C. Note, that there exists no published data on the
melting temperature of SA 455 steel but that its carbon content classifies it as a medium carbon steel (0.33%). However, SA 455 steel also has a significant amount of Manganese (1.14%) which might affect its melting temperature. From this information and from the known temperature distributions of pressure vessels exposed to accidental fires, the creep rupture properties of SA 455 steel were measured in the range of 550°C – 720°C.

Although creep is a plastic flow phenomenon, creep rupture typically occurs without necking and can be catastrophic. However, for the case of relatively fast creep ruptures, Kraus (1980) has experimentally observed ductile failure surface; indicating a combination of creep and plastic deformation.

Good creep resistance features for materials include:

• Metallurgical stability under long-time exposure to elevated temperatures;
• Resistance to oxidation and corrosive media; and
• Larger grain size since this reduces the area of grain boundaries, where much of the creep damage is perceived to take place.

The rate at which a material creeps depends on numerous aspects, which include:

• Material microstructure
• Temperature
• Stress
• Mode of loading (uniaxial, multiaxial, etc.)
• Neutron flux (nuclear applications)

The relationship between temperature, stress and failure time is plotted in Figure 2-1, where it is seen that higher temperatures will cause reduced failure time and increased stress will also cause reduced failure time. Kraus observed that failures on the left portion of the graph are
ductile, whereas failures on the right portion are brittle, using reduction in area as a measure of ductility.

![Creep failure time dependence on temperature and stress](image)

**Figure 2-1: Creep failure time dependence on temperature and stress (Kraus, 1980).**

Note that this work is interested in failures that occur in less than 100 minutes, i.e. on the extreme left of Figure 2-1.

The most common way to obtain creep data is from uniaxial creep rupture tests in which a tensile specimen is loaded under constant load to failure. Due to the nature of this test, the time required can be long. A less common creep test is the stress relaxation test under which the specimen is loaded to a constant strain and the decrease of the load is monitored. This test is somewhat faster. Displacement (converted to strain), load (converted to stress) and time to failure are all recorded from a creep rupture test. Figure 2-2 shows the typical creep curve of a metallic material. The affect that temperature and stress have on a typical creep curve was plotted in Figure 2-3 and 2-4.
Figure 2-2: Creep strain vs. time; typical plot generated from a creep rupture test (Kraus, 1980).
Figure 2-3: Temperature dependence of a typical creep curve (Kraus, 1980).

Figure 2-4: Stress dependence of a typical creep curve (Kraus, 1980).
Examination of Figure 2-2 indicates that there are 3 distinct phases of creep: primary, secondary (or steady state) and tertiary. It is important to note that the shape of this graph is strongly affected by test temperature and stress, see Figure 2-3 and 2-4. In the primary phase of creep, creep deformation decelerates until the minimum creep rate is reached. The amount of creep deformation and thus creep damage accumulated during this phase is often neglected as it is considered small compared to the total creep strain at failure. The secondary phase of creep is often called “steady state creep” since during this phase the creep rate is constant. During the tertiary phase of creep, deformation accelerates. Depending on the exposure temperature, applied stress and the material microstructure, this phase will vary significantly.

Stress relaxation is another interesting creep deformation phenomena. Stress relaxation occurs when strain is held fixed during creep loading conditions and the typical behaviour is shown in Figure 2-5.

![Figure 2-5: Effect of stress level on relaxation at constant temperature (Kraus, 1980).](image)
2.2.2 Kachanov/Rabotnov (Empirical based Continuum Damage Mechanics)

Kachanov was the first to develop the idea of continuum damage mechanics. His work was revised by Rabotnov so that, among other things, the damage parameter varied from 0 in an undamaged state to 1 in a failed state. The equations developed by Kachanov and revised by Rabotnov have been widely used. Some modifications to the original equations have been proposed by a number of authors who were attempting to increase analysis accuracy for their specific applications. Hyde, Becker and Sun (1993-2006) (among others) have had success predicting creep failure in a variety of applications using the Kachanov type equations in forms they refer to as the One-State Variable equations and Three-State Variable equations.

The One-State Variable equations were derived empirically to match experimental tertiary creep data results and are presented in Section 3.0.

Hyde (1998) presents in detail the steps required to extract the needed One-State Variable creep constants from experimental creep rupture data. Becker et al. (2002) have shown the One-State Variable creep equations to be accurate using numerous benchmarks. Hyde et al. (2006), Sun et. al (2002) and Hyde et. al (1999) have shown that the One-State Variable equations can accurately predict creep behaviour of various steel pipes exposed to creep loading conditions. Figure 2-6 shows the uniaxial validation that Hyde et al. (2006) performed to ensure that the One-State Variable equations and Three-State Variable equations could predict creep behaviour of Bar 257 steel.
Due to its simplicity and reliability, the One-State Variable form of empirical based Continuum Damage Mechanics Technique was one of the creep damage models chosen to numerically model the creep behaviour of SA 455 steel in this work.

2.2.3 MPC Omega Method

The MPC Omega method was developed by the Materials Properties Council and was presented by Prager (1995). The MPC Omega method allows one to predict remaining life and creep strain accumulation of the component being assessed.

The method is based on the idea that the current creep strain rate, along with a brief history of creep strain rates, provides enough information to be able to predict creep behaviour of a component both for past creep exposure and future creep exposure. The sole parameter of the method, which is derived experimentally, is $\Omega_p$ and it can be found by plotting the natural
logarithm of true creep strain rate vs. true creep strain from creep rupture test data (Figure 2-7); the slope of this straight line gives the parameter $\Omega_p$. If a straight line for this plot is not obtained from the data, then another creep damage model should be sought. It is important to note that $\Omega_p$ is dependent upon temperature, stress and mode of loading. Generally, $\Omega_p$ (Prager, 2000):

- Increases with decreasing stress
- Increases with decreasing temperature
- Increases with increasing multiaxiality
- Is far less sensitive to stress and temperature than strain rate

A large $\Omega_p$, 30, 50 or even 200 or more, indicates that most of the service life was spent at very low strains. In the final stages of life, strain rate accelerates rapidly to failure. High Omega behaviour may be due to creep softening or brittleness (Prager, 2000).

A low $\Omega_p$, would indicate that the service life was spent mostly in a tertiary state of creep strain. However the acceleration of the creep strain is not as prominent as that in the final stages of life for a material with high Omega values (Prager, 2000).
From creep rupture data one can create a table of the stress and temperature dependent $\Omega_p$ values as shown in Table 2-1.

Table 2-1: Experimentally derived Omega values for carbon steel (Prager, 1995).

<table>
<thead>
<tr>
<th>STRESS (KSI)</th>
<th>950</th>
<th>975</th>
<th>1000</th>
<th>1025</th>
<th>1050</th>
<th>1100</th>
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<td>8.87</td>
<td>8.24</td>
<td>7.67</td>
<td>7.16</td>
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<td>4.00</td>
<td>11.77</td>
<td>10.93</td>
<td>10.00</td>
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<td>7.96</td>
<td>7.45</td>
<td>6.95</td>
</tr>
<tr>
<td>3.50</td>
<td>12.33</td>
<td>11.54</td>
<td>10.46</td>
<td>9.67</td>
<td>8.96</td>
<td>8.33</td>
<td>7.70</td>
<td>7.20</td>
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<tr>
<td>3.00</td>
<td>12.95</td>
<td>11.90</td>
<td>10.86</td>
<td>10.13</td>
<td>9.38</td>
<td>8.71</td>
<td>8.11</td>
<td>7.67</td>
</tr>
<tr>
<td>2.50</td>
<td>13.61</td>
<td>12.50</td>
<td>11.51</td>
<td>10.52</td>
<td>9.63</td>
<td>9.12</td>
<td>8.49</td>
<td>7.91</td>
</tr>
<tr>
<td>2.00</td>
<td>14.29</td>
<td>13.10</td>
<td>12.06</td>
<td>11.12</td>
<td>10.29</td>
<td>9.54</td>
<td>8.87</td>
<td>8.26</td>
</tr>
</tbody>
</table>
The MPC Omega method is included in the American Petroleum Institute Recommended Practice 579 on Fitness-for-Service as a reliable method of predicting creep failure in high temperature applications with commonly used metals (API, 2000). One advantage of this method is that the Fitness-for-Service report includes all the needed material constants for several common high temperature application metal alloys. The inclusion of the MPC Omega method into the API Recommended Practice shows that it is a creep failure prediction method that is trusted by many practicing engineers.

Prager (1996) has shown that the MPC Omega method can accurately predict high temperature tube life and creep crack growth curves. Kwon et. al (2007) have shown that the MPC Omega method can accurately predict creep life of heater tubes. Evans (2004) has compared MPC Omega method’s creep property prediction accuracy against other creep damage techniques. Evans concluded that the MPC Omega method produced relatively accurate results compared with other creep life prediction techniques (Theta Projection Concept, CRISPEN and Kachanov) and that the MPC Omega method excelled at predicting minimum creep rates.

The MPC Omega method has been proven to be a reliable technique for predicting creep rupture by several researchers and by the American Petroleum Institute. Therefore, the MPC Omega Method was one of the creep damage models chosen to numerically model the creep behaviour of SA 455 steel in this work.

The empirically derived equations of the MPC Omega Method are presented in Section 3.0.

2.2.4 Theta Projection Concept

The Theta Projection Concept was developed by Evans and Wilshire (1984) to accurately extrapolate creep rupture data. In order to accomplish this, a large number of constants
(minimum 17) need to be derived from experimental creep rupture data. It is important to note that the Theta Projection Concept does not model secondary state creep, it only models primary and tertiary creep and views secondary creep as simply a transition between the other two.

The Theta Projection Concept attempts to predict total true creep strain using the following empirically derived equation:

\[ \varepsilon^c = \theta_1 \left( 1 - e^{-\theta_2 t} \right) + \theta_3 \left( e^{\theta_4 t} - 1 \right) \]  
(Equation 2-1)

where, \( \varepsilon^c \) is creep strain, \( t \) is time, \( \theta_1 \) and \( \theta_3 \) define the extent of creep strain and \( \theta_2 \) and \( \theta_4 \) quantify the curvatures of the primary and tertiary stages of creep respectively. An advantage of this method is that once the stress and temperature dependencies of the four \( \theta \) functions are determined, any creep rupture parameter can be easily computed (Brown et al. 1986).

Empirically, the following function was developed to quantify the stress and temperature dependencies of \( \theta_i \):

\[ \log \theta_i = \alpha_i + \beta_i \sigma + \gamma_i T + \delta_i \sigma T \]  
(Equation 2-2)

where, \( \alpha_i, \beta_i, \gamma_i \) and \( \delta_i \) are material constants used to define the stress and temperature dependency of \( \theta_i \).

Failure is determined by exhaustion of ductility, a failure strain, which is dependent upon stress and temperature and extrapolated using the Theta Projection Method. The temperature and stress dependency of the failure strains is given by the following equation:

\[ \varepsilon_f = a + bT + c\sigma + d\sigma T \]  
(Equation 2-3)

where a new set of \( a, b, c \) and \( d \) material constants need to be determined, bringing the total number of constants required to 20.

A drawback of the Theta Projection Method is that failure is based on a creep failure strain. Hyde (1998) suggests that in most cases, experimental rupture times are much more
reliable and less prone to variability than rupture strains. Depending on the material and testing conditions, the creep failure strain can show significant variance. However, for some materials under certain testing conditions the creep failure strain can be almost constant (Brown et al. 1986). Therefore, when using the Theta Projection Method an investigation of failure strain variability is important.

Researchers have applied the Theta Projection Concept to successfully predict creep behaviour of various components. Evans and Whilshire (1986) and Brown et al. (1986) successfully modeled creep of turbine blades using the Theta Projection Concept. Loghman and Wahab (1996) modeled creep of thick tubes using the Theta Projection Concept. Law et al. (2002) modeled creep of thick-walled pressure vessels with a through-thickness thermal gradient using the Theta Projection Concept. Figure 2-8 shows the stress redistribution that occurred in the pressure vessel due to creep deformation. Law et al. were not concerned with analyzing pressure vessel failure. Rather, their main objective was to investigate the creep deformation behaviour of a pressure vessel with a through-thickness temperature gradient. Figure 2-9 shows that the surface of the pressure vessel with the higher temperature will, as expected, creep faster.

Figure 2-8: Theta projection modelling of +5°C thermal case, von-Mises stresses (Law et al., 2002).
When modeling creep deformation, i.e. when failure is not being predicted, the Theta Projection method has been shown by many researchers to perform reliably. However, when attempting to numerically predict creep failure, the failure time is much more important than failure strain. Therefore, it was decided that a creep damage model based on failure times or creep strain rates was more appropriate for this work, so the Theta Projection Concept was not used.

2.2.5 Physically Based Continuum Damage Mechanics

There are several mechanisms by which a microstructure can degrade in service. Therefore, it is unlikely that a single equation, like those used in empirically based continuum damage mechanics, would be able to accurately predict creep behaviour. Thus, physically based continuum damage mechanics was introduced by Ashby and Dyson (1984). Physically Based CDM is much more complex than empirical CDM and has many more parameters that need to be quantified. It is beyond the scope of this literature review to introduce all of the physically based
continuum damage mechanics equations. Due to its complexity this creep damage model was not used in this work. However, this is a powerful creep failure prediction tool utilized by many researchers and thus a brief overview is provided.

According to Dyson (2000) the following four steps need to be completed to implement a physically based CDM model:

- identification of each applicable damage mechanism;
- definition of a dimensionless (damage) variable for each mechanism;
- incorporation of each variable within a kinetic equation for creep;
- development of an evolution equation for each variable.

Several microstructural parameters are incorporated into the physically based continuum damage mechanics equations. For example, the following damage variables could be incorporated into the model: (i) stress redistribution around hard regions; (ii) a progressive multiplication of the mobile dislocation density; (iii) coarsening of the strengthening particles; and (iv) nucleation-controlled, constrained grain boundary cavitation.

Dyson (2000) obtains the values of these parameters by initially choosing a set of values based on theoretical expectations. Creep curves are then computed by integrating the creep strain rate equations at different initial stresses and temperatures using a fifth-order Runge-Kutta numerical method with adaptive step-size control. Using the software CREEPSIM-2, convergence towards the “best” parameter set was observed by repeated systematic manual changes of parameters within the imposed physical bounds.

Physically based CDM has been successfully applied to predict creep behaviour in a variety of components made of various engineering alloys by Ashby et al. (1984), Dyson et al. (1989) and Hayhurst et al. (1994) among others. Yang et al. (2004) performed creep failure
analyses on variously notched tubes using Physically Based Continuum Damage Mechanics; Table 2-2 summarizes their findings.

**Table 2-2: Predicted failure times for plain and notched tubes with experimental data for comparison. Test conditions: 565°C, 36.6 MPa. (Yang et al., 2004)**

<table>
<thead>
<tr>
<th>Testpiece</th>
<th>ASME Subsection NH, h</th>
<th>Robinson’s Rule, h</th>
<th>CDM Predicted Life, h</th>
<th>Actual Life, h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plain Tube (10% eccentricity)</td>
<td>50</td>
<td>369</td>
<td>4612</td>
<td>4842</td>
</tr>
<tr>
<td>Internally Notched Tube</td>
<td>33</td>
<td>128</td>
<td>2535</td>
<td>2907</td>
</tr>
<tr>
<td>Externally Notched Tube</td>
<td>33</td>
<td>325</td>
<td>3520</td>
<td>3844</td>
</tr>
</tbody>
</table>

Yang et al.’s results illustrate the superior predictive capability of Physically Based CDM methodology compared with more conventional methods. It should be noted that for creep failure analyses, predictions within about 20% of measured failure are considered to be excellent. They attribute this significant difference in accuracy to the fact that:

- Conventional methods do not allow for stress redistribution as a result of creep deformation.
- Physically based CDM defines failure as occurring when the component can no longer carry equilibrium loads as a result of damaged elements.
- The ASME approach defines failure based on surface flaws.

For the numerical creep failure analysis of pressure vessels, the concern is not with modeling the individual mechanisms of creep deformation, but rather with developing a model that would reliably predict the global material behaviour associated with high-temperature failure of steel. Also, Physically Based Continuum Damage Mechanics would make computation and material property testing times much higher than needed for this application. Thus, it was decided that the Physically Based Continuum Damage Mechanics would over complicate the analysis and was therefore not practical for use in this work.
2.2.6 Robinson’s Life Fraction Rule

Robinson’s Life Fraction Rule was originally proposed to estimate creep lifetimes of components operating under variable temperatures during service (Robinson, 1938). The key advantage of this method is that, due to its simplicity, it can be applied to pretty much any creep model analysis to predict component lifetime. However, according to Dyson (2000), a major drawback of the method is that under conditions of varying stresses it will generally not produce accurate results. This can be explained with the same argument that is used to show why creep strain equations based on time hardening are not as accurate as those based on strain hardening when dealing with varying stresses (Kraus, 1980); see Section 3.1.2 for more details.

The creep damage of the component is given by Equation 2-4, and is integrated over its lifetime with respect to stress and temperature.

\[
\omega = \frac{t_s}{t_r(T, \sigma)}
\]  

(Equation 2-4)

where, \( \omega \) is creep damage, \( t_s \) is time in service, \( t_r \) is rupture time, \( T \) is temperature and \( \sigma \) is stress.

Since the damage values over each time period are assumed to be independent of each other, failure will occur when \( \sum_t \omega \) reaches a value of 1.

Due to the fact that Robinson’s Life Fraction Rule has been shown to produce unreliable results under varying stress conditions, it has not been used much recently by researchers. For this same reason it was not used for this work.
2.2.7 Larson-Miller

The Larson-Miller parameter presented by Larson and Miller (1952) is a quick way to organize and extrapolate experimental creep rupture data so that it can be plotted as a single function on a single graph. The Larson-Miller parameter combines test temperature and rupture time in a single parameter, the Larson-Miller parameter, so that it can be plotted against stress. It should be mentioned that this method, on its own, cannot be coupled with finite element analysis to evaluate creep rupture in components.

The function used to combine test temperature and rupture time is:

$$P_{L-M} = T(C_{L-M} + \log t_r)$$

(Equation 2-5)

where, $P_{L-M}$ is the Larson-Miller parameter, $T$ is absolute temperature ($^\circ$R), $C_{L-M}$ is the Larson-Miller constant and $t_r$ is the rupture time (hrs).

For example, stress is plotted versus the Larson-Miller Parameter in Figure 2-10 below.

A polynomial of first or second order is generally found to best describe the $P_{L-M}$ function. This function can be extrapolated past the experimental data in order to obtain creep information outside of test condition boundaries. Because the Larson-Miller technique depends on a “common” constant ($C_{L-M} \cong 20$), it can easily be used for relative assessments of creep behaviour of various materials. Note that choosing a value of $C_{L-M}=20$ is standard practice, when no other information indicating otherwise is present.
It is possible to calculate the optimal value of the Larson-Miller constant for a specific set of creep rupture data. This process was outlined by Kraus (1980). It is important to note that in order to calculate the Larson-Miller constant, the following units must be used: degrees Rankine for temperature and hours for time. Also, at least two sets of data are needed that each have at least two creep rupture tests that were conducted at different temperatures but with the same stress.

The Larson-Miller technique on its own cannot be integrated over time to account for varying operating temperatures and stress conditions. Since most applications will see varying temperature and stress histories, the Larson-Miller technique cannot be coupled with a finite element analysis on its own in order to predict creep rupture. However, one could combine Robinson’s life fraction rule with the Larson-Miller technique and implement this combined model into a finite element analysis to predict creep failure. The Larson-Miller technique would provide Robinson’s rule with the stress and temperature dependent life times needed to calculate life fractions consumed at each node, element or integration point of a finite element model.
These fractions would then be summed at each node, element or integration point over the life of the component to determine both failure times and failure locations.

The Larson-Miller technique was used to organize SA 455 steel’s stress rupture data. However, as discussed, it was not used to numerically predict creep behaviour.

2.3 Pressure Vessel Failure Analysis

Fire tests of pressure vessels have been conducted by many researchers (Birk, Cunningham, Ostic & Hiscoke 1997, 2004; Droste & Schoen 1988; Moodie, Cowley, Denny, Small & Williams 1988, among others) in order to achieve a more detailed understanding of high temperature pressure vessel failure mechanics. These tests collected vast amounts of data including, vessel wall temperatures, internal pressure and liquid and vapour temperatures in the vessel. The data was analyzed to make conclusions on how various test parameters affected failure times of the pressure vessels as well as whether or not a boiling liquid expanding vapour explosion (BLEVE) occurred. Results from experimental pressure vessel fire tests provide valuable information regarding vessel temperature distribution, internal pressure history and failure time.

All the reviewed work was similar in that great detail was given to modeling the thermal response of the vessels but the stress analysis conducted was quite simplistic. Thus, the current work built upon past work conducted by others by utilizing the thermal response data collected and analyzed to input thermal loads as temperatures at nodes into finite element analyses that conducted detailed stress analyses of the vessels.
2.3.1 Thermal Response of a Pressure Vessel in an Accidental Fire

Droste and Schoen (1988) fire tested unprotected and thermally protected pressure vessels containing liquid propane gas in fully engulfing fire conditions. The goal was to examine the safety margins of unprotected pressure vessels. The pressure vessels were equipped with several NiCr/Ni thermocouples and pressure measurement devices. They observed that the temperature values of the tank wall in the liquid space of the vessel were similar to those of the compressed liquid propane's temperature. Also, they observed that the vessel temperatures in the vapour space were much higher and that the maximum tank wall temperature was always measured at the top of the horizontal cylindrical vessel. A typical temperature distribution history of a pressure vessel thermally protected with an insulation jacket is shown by Droste and Schoen (1988) in Figure 2-11.

![Figure 2-11: Thermally protected pressure vessel shell temperatures (Droste and Schoen, 1988).]
Moodie et al. (1988) carried out five fire tests on commercial propane pressure vessels with initial fill levels of 22% to 72%. The main objective of this work was to gain a better understanding of how the pressure vessels heated up in a fully engulfing fire. To avoid failure, the fires were allowed to burn until peak vessel wall temperature reached about 600°C. They observed that the outer liquid-wetted wall temperatures ranged from 50°C to 140°C and that a through thickness temperature gradient of about 20°C was present in the liquid-wetted wall. They also observed that all the vapour-wetted wall temperatures behaved similarly, increasing rapidly once the fire had become established, but they leveled off once the pressure relief valve started venting the vessel; see Figure 2-12. Moodie et al. also measured a “frothing” region, a region in which wall temperatures sharply changed from hot vapor-wetted wall temperatures to much cooler liquid-wetted wall temperatures. They reported that temperature measurements just above the initial liquid level indicated that some upwelling of liquid and frothing had occurred once the fire was established; this was expected since the liquid will expand when heated and the liquid surface will contain vapour bubbles as it boils.

![Figure 2-12: Thermally unprotected peak vessel wall temperatures; various initial fill levels](Moodie et al., 1988).
The thermal response behaviour of pressure vessels in accidental fires as observed by Droste et al. and Moodie et al. was also observed by Birk et al. (1994, 2006) during many of their experimental vessel fire tests, see for example Figures 2-13 and 2-14. The “04-3” in figure captions refers to testing conducted in 2004 and the test number.

Figure 2-13: Wall thermocouple layout for test 04-3 (Birk et al., 2006).
2.3.2 Pressure Vessel Failure due to Accidental Fire Loading

The pressure vessels tested by Droste and Schoen had a fill level of 50%. They postulated that there were two reasons for thermally unprotected pressure vessels to rupture in a fire accident:

- Increase of internal pressure caused by the temperature rise of the compressed propane and insufficient pressure relief of the safety valve.
- Decrease of the vessel’s structural strength caused by an increasing vessel wall temperature and a corresponding drop of the yield and tensile strength.
Droste and Schoen calculated the bursting pressure of the vessel under the assumption that rupture will occur when the material’s temperature dependent yield strength is exceeded. The comparison of calculated values to test results in Figure 2-15 shows that the vessels have a reasonable margin of safety and that the assumption that rupture occurs when the material’s yield strength is exceeded is conservative.

Figure 2-15: Comparison of calculated tank bursting pressures and test results

(Droste and Schoen, 1988).
Droste and Schoen observed that internal pressure rise was not stopped or significantly delayed by the start of discharge of the pressure relief valve. From this work it was concluded that the thermal insulation design tested was able to prevent vessel failure even in a 90 minute full fire engulfment.

Moodie et al. (1988) concluded from their work, among other things, that an increase in wall temperatures in the vapour space and the long term behaviour of the vessel internal pressure during PRV operation would ultimately determine the integrity of the vessel structure.

Birk (2006) also found that pressure vessel rupture by fire impingement is due primarily to pressure induced stress and high wall temperatures in the vapour space wall region.

2.3.3 Computer Modeling of Pressure Vessel Failure in Accidental Fire Loading

In order to better understand the mechanisms involved in high temperature pressure vessel failure there have been numerous computer codes developed to model the response of pressure vessels exposed to accidental fires. Birk (1988) presented a computer model, the Tank-Car Thermal Computer Model (TCTCM), which could simulate the thermal response of a pressure vessel and its lading to external fire impingement and also predict vessel failure. The computer model was developed as a tool to study the effectiveness of various thermal protection systems of pressure vessels. The computer code could model a fully engulfing fire or a local hot spot. It could account for various protection systems such as pressure relief valves, thermal insulation, radiation shielding, temperature sensing relief valves and internal dissipating matrices. TCTCM models the pressure vessel as a two-dimensional representation of a circular cylindrical pressure vessel (axial gradients and end effects are not accounted for). The model was made up of a series of submodels that simulated the following processes:

- Flame to vessel heat transfer,
• Heat transfer through the vessel wall and associated coverings,
• Interior-surface to lading heat transfer,
• Thermodynamic process within the vessel,
• Pressure relief device operating characteristics,
• Wall stresses and material property degradation, and
• Tank failure.

Rupture of the pressure vessels was predicted by TCTCM as a function of vessel geometry, internal pressure, vessel wall temperature and temperature dependent material strength. It was observed that if very high wall temperatures were achieved, then the vessel ruptured even if the relief valve maintained the vessel internal pressure well below the normal burst pressure.

TCTCM is capable of modeling the physical phenomena which govern heat transfer between the fire and the pressure vessel and its contents, see Figure 2-16. Despite a highly simplified calculation of vessel stress, the computer model was capable of accurately predicting failure times of various experimental pressure vessel fire tests, see Figures 2-17 and 2-18.
Figure 2-16: Predicted and measured tank wall temperatures vs time from fire ignition for upright uninsulated fifth-scale tank-car exposed to an engulfing fire (Birk, 1988).

Figure 2-17: Predicted tank wall stresses and material strength vs time from fire ignition for upright uninsulated fifth-scale tank-car exposed to an engulfing fire (Birk, 1988).
Figure 2-17 shows that for some analyses the Tank-Car Thermal Computer Model was not always in perfect agreement with experimental observations. Birk (1988) indicated that the disagreement between predictions and the test could be caused by a number of factors such as flaws in the tank wall, poor welds, or unmeasured hot spots in the tank wall (due to limited thermocouple locations). Figure 2-18 shows excellent agreement between predicted failure time and experimentally measured pressure vessel failure time.

The computer model, TCTCM, developed by Birk (1983) was used to assess the fire survivability of aluminum 33.5 lb propane cylinders vs steel propane cylinders. The study showed that the aluminum cylinders would fail in about eight minutes in an engulfing fire and

![Figure 2-18: Predicted tank wall stresses and material strength vs time from fire ignition for upright uninsulated full-scale tank-car exposed to an engulfing fire (Birk, 1988).](image-url)
that the steel cylinders would empty before failure occurred. Although no direct comparisons were able to be made to experimental findings, accident data supports this finding. This work by Birk (2003) adds to the reliability of the Tank-Car Thermal Computer Model.

Birk (2005) updated the Tank-Car Thermal Computer Model to accomplish the following:

- Make the code quasi 3D so that it can account for a wide range of possible thermal protection defect locations and geometries.
- Include graphical output features to confirm defect geometries.
- Refine lading thermal model to account for liquid temperature stratification and its effect on tank pressure.
- Refine lading thermal model to include effects of defect position on tank pressurization.
- Refine PRV model to include real-world PRV operating characteristics (such as spring softening).
- Refine vapour space and PRV model to account for liquid entrainment into PRV at very high fill levels.
- Refine tank failure model to include high-temperature stress-rupture.
- Ensure that the code runs in a reasonable length of time (<2 hours for the simulation of a 100 minute fire scenario).

This updated computer model, Tank2004, was validated by Birk et al. (2006) against experimental fully engulfing fire tests of pressure vessels with thermal protection defects, see Table 2-3. The structural failure analysis was conducted using Robinson’s Life Fraction Rule with the assumption that the effective failure stress is dominated by von Mises stress. The von Mises stress used for these analyses is simplified as being a function of vessel dimensions and
internal pressure. Therefore, creep stress redistribution is ignored by the Tank2004 computer
model when calculating stress. The updated computer model produced good agreement with
experimental results and it was concluded that the assumption that von Mises stress drives the
tank rupture is reasonable.

However, as the size of the hot spot decreased, the Tank2004 computer model was found
to be less and less accurate. This was due to the fact that, as the hot spot size decreases creep
stress redistribution becomes increasingly prominent. Therefore, in order to accurately predict
high temperature failure of pressure vessels with hot spots, the stress state of the pressure vessel
needs to be completely modeled. The finite element analysis method is commonly used to
accurately compute the stress state of components numerically (among many other things).
Therefore, to improve upon Tank2004’s calculation of structural failure, the use of FEA is the
obvious choice to completely model the stress state and high temperature structural behaviour of
pressure vessels exposed to hot spots and fully engulfing fires.

Table 2-3: Summary of Tank2004 stress rupture predictions compared to actual failure
times; 500 gallon SA 455 steel pressure vessels (Birk et al., 2006).

<table>
<thead>
<tr>
<th>Test</th>
<th>Actual failure time</th>
<th>Predicted failure time from SR analysis from nominal von Mises stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>04-3</td>
<td>1440 s</td>
<td>1409 s</td>
</tr>
<tr>
<td>04-4</td>
<td>3370</td>
<td>3929</td>
</tr>
<tr>
<td>04-5</td>
<td>3550</td>
<td>2258</td>
</tr>
<tr>
<td></td>
<td></td>
<td>3608</td>
</tr>
</tbody>
</table>
Chapter 3  Theory

3.1  High Temperature Steel Failure

Short-term, high-temperature failures are termed stress-rupture failures. Stress-rupture includes two significant subcategories: short-term overheating and high-temperature creep (Viswanathan, 1989).

During short-term overheating, temperatures and stresses are high enough to cause plastic deformation. Since short-term overheating failures are driven by plastic deformation they are considered to be time-independent. Section 3.1.1 provides a review of the basic theoretical concepts of plasticity.

During high-temperature creep, temperatures and stresses are high enough to cause tertiary creep deformation that will lead to relatively fast failure times. Note that the temperatures and stresses are not high enough to cause material yielding. Since high-temperature creep failures are driven by tertiary creep deformation they are considered to occur as time-dependent phenomena. Section 3.1.2 provides a review of the basic theoretical concepts of creep.

It is important to note that short-term overheating and high-temperature creep can occur simultaneously to cause failure. For example, stresses and temperatures can be high enough to cause material yielding, but not failure by plastic deformation. In this case high-temperature creep would continue to occur until failure. Another example is that if a material was deforming due to high-temperature creep, it may become unstable as it nears failure. This instability can lead to enough wall reduction to cause local plastic deformations as failure occurs. This type of “combined” failure is often observed in experimentation (Kraus, 1980). Short-term overheating produces a sharp knifed edge fracture face, while high-temperature creep produces a rough, thick
fracture face. However, in experimental fire tests of pressure vessels, fracture faces with wall reductions of about 50% are common (Birk et al., 2006). This fracture face cannot be categorized as either short-term overheating or high-temperature creep and thus it is concluded that it reflects a combination of the two failure modes.

3.1.1 Plasticity

Plasticity can be defined as “that property that enables a material to be deformed continuously and permanently without rupture during the application of stresses exceeding those necessary to cause yielding of the material” (Mielnik, 1991).

The engineering stress-strain curve diverges from the true stress-strain curve as plastic deformation commences. Therefore, when analyzing plastic deformation, we must use true stress-stain data. The relationships between engineering and true stress and strain are shown in Equations 3-1 and 3-2. Engineering stress and strain are calculated using the assumption that original cross sectional area and original length are almost identical to current cross sectional area and length. For metals, this is found to be a valid assumption while strains are less than about 0.2%. Therefore, for large strain problems, true stress and true strain must be used to accurately model material behaviour.

\[
\sigma = \sigma_n (1 + \varepsilon_n) \quad \text{(Equation 3-1)}
\]

\[
\varepsilon = \ln(1 + \varepsilon_n) \quad \text{(Equation 3-2)}
\]

where, \(\sigma\) and \(\varepsilon\) are true stress and strain respectively and \(\sigma_n\) and \(\varepsilon_n\) are engineering stress and strain.
The theory of plasticity is used to relate stresses and strains once the material has been stressed past its yielding point. The yield point of a material is often defined by the 0.2% offset stress, see Figure 3-1.

![Figure 3-1: 0.2% offset used to define tensile yield strength (Hibbeler, 2003).](image)

It is evident that a major difficulty in relating stress and strain during plastic deformation is that the relationship between the two is no longer linear. Therefore, some assumptions are made to model plastic behaviour. The von Mises theory of plasticity is most often used to model plasticity in ductile metals. The key assumptions made for von Mises plasticity are (Mielnik, 1991):

- The material is homogeneous
- The volume remains constant, i.e, $\Delta V/V$ and the sum of the plastic strain increments is zero, or

$$d\varepsilon_1^p + d\varepsilon_2^p + d\varepsilon_3^p = 0 \quad \text{(Equation 3-3)}$$

for plastic deformations where $\varepsilon^p$ represents true plastic strain.
- A hydrostatic state of stress does not influence yielding.
Plasticity theory is composed of three parts that are needed to relate stress and strain during plastic deformation: a yield criterion, a flow rule, and a hardening rule. The general theory has many unique forms that were created to fit various sets of experimental data for various materials and loading conditions.

The Yield Criterion

For general plasticity of metals, yielding is said to have occurred when the effective stress reaches the tensile yield stress of the material obtained from a uniaxial tensile test. To perform plasticity analyses when dealing with three-dimensional states of stress, von-Mises effective stress is most often used in the yield criterion to calculate the effective stress when dealing with ductile metals, see Equation 3-4.

$$\bar{\sigma} = \frac{1}{\sqrt{2}} \left[ (\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2 \right]^{\frac{1}{2}}$$  \hspace{1cm} (Equation 3-4)

where, $\bar{\sigma}$ is the effective von Mises stress, $\sigma_1, \sigma_2, \sigma_3$ are the principal stresses.

The Flow Rule

Constitutive stress-strain relationships that describe the path of plastic deformation of a material are called flow rules. Flow rules for any yield criterion have the following form (Mielnik, 1991):

$$d\varepsilon_{ij}^p = \frac{\partial F(\sigma_{ij})}{\partial \sigma_{ij}} (d\lambda)$$  \hspace{1cm} (Equation 3-5)

where, $F(\sigma_{ij})$ is the isotropic yield function if the flow rule is considered associative, $\varepsilon_{ij}^p$ is the true plastic strain component, $\sigma_{ij}$ is the true stress component and $d\lambda$ is the plastic compliance. The flow rule is termed an associated flow rule when the plastic potential surface
coincides with the yield surface. Metals conform to this rule quite well. For the von-Mises yield criterion the flow rule obtained is:

\[
\frac{d\varepsilon_1}{S_1} = \frac{d\varepsilon_2}{S_2} = \frac{d\varepsilon_3}{S_3} = d\lambda
\]  

(Equation 3-6)

where, \(d\lambda\) is the plastic compliance and \(S_i\) are the deviatoric stress components. The deviatoric stress components are calculated as shown in Equations 3-7 and 3-8.

\[
S_1 = \sigma_1 - \sigma_m
\]
\[
S_2 = \sigma_2 - \sigma_m
\]
\[
S_3 = \sigma_3 - \sigma_m
\]

(Equation 3-7)

where,

\[
\sigma_m = \frac{1}{3}(\sigma_1 + \sigma_2 + \sigma_3)
\]

(Equation 3-8)

Deviatoric stress components are stresses that have the hydrostatic state of stress subtracted out; this is done since plastic deformation is assumed to be a constant volume process and thus hydrostatic stresses are assumed not to cause plastic deformation in the von Mises yield criterion.

With,

\[
d\lambda = \frac{3d\bar{\varepsilon}}{2\bar{\sigma}}
\]

(Equation 3-9)

where, \(\bar{\varepsilon}\) is the effective strain and \(\bar{\sigma}\) is the effective stress.

Equation 3-5 can now be rewritten as follows:
\[ d\varepsilon_1 = \frac{d\varepsilon}{\sigma} \left[ \sigma_1 - \frac{1}{2} (\sigma_2 + \sigma_3) \right] \]
\[ d\varepsilon_2 = \frac{d\varepsilon}{\sigma} \left[ \sigma_2 - \frac{1}{2} (\sigma_1 + \sigma_3) \right] \]
\[ d\varepsilon_3 = \frac{d\varepsilon}{\sigma} \left[ \sigma_3 - \frac{1}{2} (\sigma_1 + \sigma_2) \right] \]  
(Equation 3-10)

Note that since plastic deformation is assumed to be incompressible, Poisson’s ratio (\( \nu \)), \( \nu = 0.5 \).

The Hardening Rules

Hardening rules are used to describe the changing of the yield surface during progressive yielding. These rules are implemented in finite element analysis codes to allow stress states for subsequent yielding to be calculated. Generally, two hardening rules are available for metals, isotropic hardening and kinematic hardening. When dealing with cyclic loads involving load reversals, the Bauschinger effect should be observed. The Bauschinger effect states that the yield strength of a metal will decrease when the direction of loading is changed.

In general, isotropic hardening works well for large deformations that occur without cyclical loading. Figure 3-2 below shows how the isotropic hardening rule works.

![Diagram of isotropic hardening rule](image)

**Figure 3-2: Loading path generated by the isotropic hardening rule (ANSYS, 2004).**
Note that the Bauschinger effect is not taken into account, so a material will have the same yield strength in compression and tension during cyclic loading. This is unacceptable if the analysis is concerned with cyclic loading, so isotropic hardening should only be used for monotonic loadings.

Kinematic hardening is used when plastic deformations are small and loading involves one or more reversals because it models the Bauschinger effect. As seen in Figure 3-3 below, the yield stress is path dependent, that is the Bauschinger effect is taken into account. The yield surface will not change in size, instead it translates as yielding progresses.

![Figure 3-3: Loading path generated by the kinematic hardening rule (ANSYS, 2004).](image)

It should be noted that in reality, metals exhibit a combination of isotropic and kinematic hardening behaviour, but for simplicity one of the models is often used based on loading conditions.

### 3.1.2 Creep

Creep deformation was already briefly introduced in Section 2.2.1. However, the mathematical formulation of creep is presented in this section as part of the theory used to complete this work. Figure 3-4 shows a typical creep curve measured during a creep rupture test.
Examining Figure 3-4 it was initially determined that in order to model the creep process in one dimension an expression of the following form was needed:

$$\varepsilon^c = F(\sigma, T, t)$$  \hspace{1cm} (Equation 3-11)

where $\varepsilon^c$ is creep strain, $\sigma$ is stress, $T$ is temperature and $t$ is time.

When developing mathematical creep relations it is customary to exclude tertiary creep since it is often ignored during engineering creep analyses. This is because most engineering analyses are not concerned with predicting failure. However, the mathematical development of
primary and secondary creep equations presented here is also applicable to tertiary creep. To proceed with the development of mathematical creep relations the following creep expression is assumed:

\[ \varepsilon^c = A\sigma^m t^n \]  

(Equation 3-12)

where, A, m and n are temperature dependent material constants.

The creep expression given by Equation 3-12 is known as the Bailey-Norton creep law. A, m and n are material constants that are functions of temperature. This creep law models primary and secondary creep. For variable stress problems, the creep strain rate is of interest. Whether the creep strain rate is a function of time or of creep strain is an important factor and will determine whether the creep law is termed time hardening or strain hardening. The time derivative of the Bailey-Norton creep law is shown in Equation 3-13.

\[ \dot{\varepsilon}^c = \frac{\partial \varepsilon^c}{\partial t} = A\sigma^m n \dot{t}^{n-1} \]  

(Equation 3-13)

where \( \dot{\varepsilon}^c \) is the creep strain rate.

Equation 3-13 shows the time hardening form of the Bailey-Norton creep law. In order to derive the strain hardening form, first time needs to be solved for as a function of creep strain (from Equation 3-12), see Equation 3-14.

\[ t = \left(\frac{\varepsilon^c}{A\sigma^m}\right)^{\frac{1}{n}} \]  

(Equation 3-14)

Substituting Equation 3-14 into Equation 3-13, the strain hardening form of the Bailey-Norton creep law can be derived as shown in Equation 3-15.
\[ \dot{\varepsilon}^c = A^{\frac{m}{n}} \sigma^{m/n} (\varepsilon^c)^{(n-1)/n} \]  

(Equation 3-15)

Figures 3-5 and 3-6 illustrate the difference between the time hardening and strain hardening creep formulations.

Figure 3-5: Creep strain history prediction from time hardening theory (Kraus, 1980).
Under conditions of varying stress the time hardening and strain hardening solutions for creep strain will differ. The time hardening law uses the creep strain rate at time, \( t \), when the stress changes to calculate a new creep strain rate. The strain hardening law uses the creep strain rate at creep strain, \( \dot{\varepsilon}^c \), to calculate a new creep strain rate when the stress changes. From
experimental work, it has been shown, as expected, that the strain hardening law produces more accurate results than the time hardening law under conditions of varying stress. However, creep equations of the strain hardening form are typically more computationally intense to solve.

Creep deformation is most often analyzed in components under multiaxial stress conditions. Therefore, it is important to derive multiaxial creep strain relations. Kraus (1980) observed that the resulting multiaxial creep relations (assuming von Mises yielding behaviour for ductile metals) must satisfy the following requirements:

- The multiaxial formulation must reduce to the uniaxial formulation when appropriate.
- The model must express the constancy of volume experimentally observed during creep deformation.
- As observed experimentally, the model’s creep strain should not be influenced by the hydrostatic state of stress.
- For an isotropic material, the directions of principal stress and strain should coincide.

Therefore, initial formulation of multiaxial creep strain was given by:

\[ \dot{\varepsilon}_i^c = \lambda S_{ij} \]  

(Equation 3-16)

where, \( \lambda \) is the creep compliance and \( S_{ij} \) is the deviatoric stress component.

The effective creep strain rate is given by Equation 3-16. Note that it is of the same form as the von Mises effective strain which is used to combine the different strain components into a single effective strain value.
\[
\dot{\varepsilon}^c = \left( \frac{\sqrt{2}}{3} \right) \left( \dot{\varepsilon}_{11}^c - \dot{\varepsilon}_{22}^c \right)^2 + \left( \dot{\varepsilon}_{22}^c - \dot{\varepsilon}_{33}^c \right)^2 + \left( \dot{\varepsilon}_{33}^c - \dot{\varepsilon}_{11}^c \right)^2 + 6 \left[ \dot{\varepsilon}_{12}^c \right]^2 + \left( \dot{\varepsilon}_{13}^c \right)^2 + \left( \dot{\varepsilon}_{23}^c \right)^2 \right]^{1/2}
\]

(Equation 3-17)

where \( \dot{\varepsilon}^c \) is the effective creep strain rate.

Now, if Equation 3-16 is substituted into Equation 3-17 and the equation of effective stress is observed, Equation 3-4, it is found that:

\[
\lambda = \frac{3}{2\bar{\sigma}} \dot{\varepsilon}^c
\]

(Equation 3-18)

where, \( \bar{\sigma} \) is the effective stress.

Substituting Equation 3-18 into 3-16, it is that found that:

\[
\dot{\varepsilon}_{ij}^c = \frac{3}{2\bar{\sigma}} S_{ij} \dot{\varepsilon}^c
\]

(Equation 3-19)

Therefore, to mathematically model the multiaxial creep strain rate, the effective creep strain rate must be derived from uniaxial creep tests as follows. First, a uniaxial creep model is developed to model the experimental creep curves. For example, an experiment that yields primary and secondary creep curves could be modeled with the Bailey-Norton law, Equation 3-20.

\[
\varepsilon^c = A \sigma^n t^n
\]

(Equation 3-20)

Converting strain and stress values to effective strain and stresses knowing that effective quantities have been defined to reduce to the uniaxial forms when appropriate:

\[
\bar{\varepsilon}^c = A \bar{\sigma}^n t^n
\]

(Equation 3-21)
And the time derivative of Equation 3-21 is:

$$\dot{\varepsilon}^c = A n \sigma^m t^{n-1}$$  \hspace{1cm} (Equation 3-22)

Therefore, for an assumed Bailey-Norton creep law, the time hardening multiaxial creep equation is given by Equation 3-23.

$$\dot{\varepsilon}^c = \frac{3}{2 \sigma_y} A n \sigma^m t^{n-1}$$  \hspace{1cm} (Equation 3-23)

Which can be further simplified to Equation 3-24.

$$\dot{\varepsilon}^c = \frac{3}{2} S_y A n \sigma^m t^{n-1}$$  \hspace{1cm} (Equation 3-24)

The strain hardening multiaxial creep equation (Equation 3-25) is obtained by substituting the value of $t$ (from Equation 3-21) into Equation 3-24.

$$\dot{\varepsilon}^c = \frac{3}{2} S_y A n \sigma^m t^{n-1}$$  \hspace{1cm} (Equation 3-25)

This is how one would mathematically model multiaxial creep using uniaxial creep rupture data assuming von Mises yielding behaviour.

### 3.1.3 Creep Damage Equations

From the literature review conducted in Section 2.2, it was determined that the Kachanov One-State Variable technique and the MPC Omega method were most suitable to predict stress rupture of pressure vessels exposed to accidental fires. The equations needed to model creep and creep damage using these techniques are presented Sections 3.1.3.1 and 3.1.3.2.
3.1.3.1 Kachanov One-State Variable Technique

The original Kachanov/Rabotnov creep damage law, derived empirically, is:

\[
\dot{\omega} = \left[ \frac{\sigma}{A(1 - \omega)} \right]^r \quad \text{(Equation 3-26)}
\]

where, \( \dot{\omega} \) is creep damage rate, \( \sigma \) is true stress, \( A \) and \( r \) are material constants and \( \omega \) is creep damage.

Equation 3-27 shows how creep damage was coupled with creep strain to model tertiary creep.

\[
\dot{\varepsilon} = A' \left( \frac{\sigma}{1 - \omega} \right)^n \quad \text{(Equation 3-27)}
\]

where, \( \dot{\varepsilon} \) is the creep strain rate, \( A' \) and \( n \) are material constants, \( \sigma \) is stress and \( \omega \) is creep damage.

Both equations were derived empirically to mathematically model experimentally measured tertiary creep behaviour. Equation 3-26 describes how creep damage will increase and Equation 3-27 describes how creep strains will increase.

The One-State Variable equations as applied by Hyde et al. (2006) have more constants added to them in an effort to more accurately predict tertiary creep behaviour.

The uniaxial forms of the One-State Variable equations are:

\[
\dot{\varepsilon} = A' \left( \frac{\sigma}{1 - \omega} \right)^n t^m \quad \text{(Equation 3-28)}
\]

\[
\dot{\omega} = B' \left( \frac{\sigma^\chi}{(1 - \omega)^\phi} \right)^m t^n \quad \text{(Equation 3-29)}
\]

where, \( t \) is time and \( B' \), \( \chi \), \( \phi \) and \( m \) are material constants.
The multiaxial forms of the One-State Variable equations, Equations 3-30, 3-31 and 3-32, are derived using the creep theory presented in Section 3.1.2.

\[
\dot{\varepsilon}_{ij}^c = \frac{3}{2} A \left( \frac{\bar{\sigma}}{1 - \omega} \right)^n S_{ij} \tau^m \quad \text{(Equation 3-30)}
\]

\[
\dot{\omega} = B \left( \frac{\sigma_r^{z}}{1 - \omega} \right)^\varphi \tau^m \quad \text{(Equation 3-31)}
\]

\[
\sigma_r = \alpha \sigma_1 + (1 - \alpha) \bar{\sigma} \quad \text{(Equation 3-32)}
\]

where, \( \dot{\varepsilon}_{ij}^c \) is the multiaxial creep strain rate component, \( \bar{\sigma} \) is the effective stress, \( S_{ij} \) is the deviatoric stress component, \( \sigma_r \) is the rupture stress, \( \sigma_1 \) is the maximum principal stress and \( \alpha \) is a material constant related to multiaxial stress-state behaviour.

Note that creep constant \( n \), in Equations 3-28 and 3-30, does not necessarily have to be equal to the Norton creep constant because the values of these creep constants are often optimized to best fit experimental uniaxial creep rupture data.

### 3.1.3.2 MPC Omega Method

The MPC Omega Method equations were developed empirically to model the microstructural mechanisms that govern tertiary creep in order to match experimentally measured tertiary creep behaviour.

The exponential approximation shown in Equation 3-33 was found to hold for levels of 20% true creep strain or more on the strain-time curve (Prager, 1995).

\[
\dot{\varepsilon}^c = \dot{\varepsilon}_0 e^{m e^c} e^{p e^c} e^{c e^c} \quad \text{(Equation 3-33)}
\]
where, $\dot{\varepsilon}^c$ is the creep strain rate, $\dot{\varepsilon}_0$ is the initial creep strain rate (material constant), $\varepsilon^c$ is the creep strain, $m$ is Norton’s exponent to account for the rate increase due to cross section reduction (stress increase), $p$ corresponds to micro-structural damage, and $c$ is used to account for deficiencies in Norton’s exponent and other micro-structural factors associated with the stress change.

Note that Equation 3-33 can be re-written as:

$$\dot{\varepsilon}^c = \dot{\varepsilon}_0 e^{(m+p+c)\varepsilon^c}$$  \hspace{1cm} (Equation 3-34)

The exponents in Equation 3-34 are then gathered as follows:

$$m + p + c = \Omega_p$$  \hspace{1cm} (Equation 3-35)

$$\dot{\varepsilon}^c = \dot{\varepsilon}_0 e^{\Omega_p\varepsilon^c}$$  \hspace{1cm} (Equation 3-36)

where, $\Omega_p$ is Omega (material constant).

The multiaxial creep strain rate, Equation 3-37, was derived using the creep theory presented in Section 3.1.2.

$$\dot{\varepsilon}_{ij}^c = \frac{3}{2} \frac{S_{ij}}{\bar{\sigma}} \dot{\varepsilon}_0 e^{\Omega_p\bar{\varepsilon}}$$  \hspace{1cm} (Equation 3-37)

where, $\dot{\varepsilon}_{ij}^c$ is the multiaxial creep strain component, $S_{ij}$ is the deviatoric stress component, $\bar{\sigma}$ is the effective stress and $\bar{\varepsilon}$ is the effective creep strain.

And $\Omega_p$ can be found from experimental data in the following way (see Figure 2-7):

$$\frac{d \ln \dot{\varepsilon}^c}{d\varepsilon^c} = \Omega_p$$  \hspace{1cm} (Equation 3-38)

A useful equation, that was derived empirically, for determining the life fraction consumed is:
\[
\frac{t_s}{t_r} = \text{Life Fraction Consumed} = \frac{\dot{\varepsilon}^c t_s \Omega_p}{\dot{\varepsilon}^c t_r\Omega_p + 1}
\]  
(Equation 3-39)

where, \(t_s\) is the time in service, \(t_r\) is the rupture time, \(\dot{\varepsilon}^c\) is the effective creep strain rate.

To extend the MPC Omega method to multiaxial loading and tube loading the Omega creep constant needs to be modified. For example, for tubes \(\Omega_p\) becomes (Prager, 2000):

\[
\Omega_{\text{tube}} = \Omega_{\text{uniaxial}} + n
\]  
(Equation 3-40)

where, \(n\) is the Norton creep exponent.

### 3.2 FEA Governing Equations

The following section summarizes the main governing equations used by the finite element analysis solver.

The pressure vessel’s numerical finite element analyses will model the structure’s elastic, plastic and creep deformations where necessary.

FEA code is based on solving numerous finite elements (mesh) which discretize the domain to achieve a converged solution which is in equilibrium. To achieve the solution, several conservation equations (i.e. mass, momentum and energy) are solved. The governing equations listed below are used by FEA codes to numerically model the physical phenomena of structural deformation.

**Mass**

\[
\frac{\partial \rho}{\partial t} + \text{div}(\rho \mathbf{v}) = 0
\]  
(Equation 3-41)
Linear Momentum
\[ \text{div} \mathbf{\sigma} + b = \rho \mathbf{\ddot{v}} \]  
(Equation 3-42)

Angular Momentum
\[ \mathbf{\sigma} = \mathbf{\sigma}^T \]  
(Equation 3-43)

Mechanical Energy
\[ \frac{D}{Dt} \left[ \frac{1}{2} \rho \mathbf{v}^2 \right] dv + \int_{\Omega} \mathbf{\sigma} : d \mathbf{v} = \int_{\partial \Omega} t \cdot \mathbf{v} ds + \int_{\Omega} b \cdot \mathbf{v} dv \]  
(Equation 3-44)

Virtual Work
\[ \int_{\Omega} (\mathbf{\sigma} : \delta \mathbf{e} - b \cdot \delta \mathbf{u}) dv - \int_{\partial \Omega} t \cdot \delta \mathbf{u} ds = 0 \]  
(Equation 3-45)

These governing equations are used to solve the global system matrix, Equation 3-46
\[ M \{\ddot{u}\} + C\{\dot{u}\} + K\{u\} = \{F\} \]  
(Equation 3-46)

where, M is the mass matrix, C is the damping matrix, F is the load vector and u is the unknown displacement vector.

For the case of a quasi-static analysis, Equation 3-46 can be simplified to Equation 3-47.
\[ K\{u\} = \{F\} \]  
(Equation 3-47)

### 3.3 FEA User Subroutines

Commercial FEA computer codes allow the user to implement user subroutines in order to generate customized versions of the computer codes. The user subroutines are most frequently written in the Fortran computer programming language. Common applications of user
subroutines include creating custom material models, elements and loadings. Most commercial FEA computer codes offer the option of implementing about 40 different user subroutines. ABAQUS (2007) provides the flow chart shown in Figure 3-7 to show where common user subroutines fit into the analysis process.

CREEP, as shown in Figure 3-7, is a user subroutine that allows the user to implement custom creep equations. DLOAD allows the user to implement custom loadings. For more information on various subroutines offered by ABAQUS see (AB AQUS, 2007).

For this work, creep user subroutines were used to implement the Kachanov One-State Variable and MPC Omega creep and creep damage equations into ANSYS and ABAQUS. The UTEMP user subroutine was used in ABAQUS to define temperatures as loads at nodes.
Figure 3-7 Where various user subroutines fit into FEA (ABAQUS, 2007)
Chapter 4  SA 455 Mechanical Testing

Experimental creep rupture and elevated temperature uniaxial tensile tests were preformed to measure needed mechanical properties of SA 455 steel. These properties were previously unavailable because SA 455 steel is not meant for high temperature applications.

4.1 Experimental Specimens

Tensile specimens were cut with a band saw from a plate 7.1 mm thick, 2.11 m wide and 3 m long. The chemical composition and tensile properties reported by the supplier were compared to the ASTM requirements in Tables 4-1 and 4-2. The comparison showed that the SA 455 samples conformed to the ASTM requirements of SA 455 steel. Specimens conforming to ASTM E 21-92 and E 139-96 specifications were produced, as discussed in Section 2.1. Figure 4-1 shows the dimensions of the rectangular cross sectioned specimens. The reduced area in the centre of the specimen is the gage length. The errors presented in Table 4-2 are measurement errors.

<table>
<thead>
<tr>
<th>Chemical composition (wt.%)</th>
<th>Supplier analysis</th>
<th>ASTM SA 455 requirement [3]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Carbon</td>
<td>0.26</td>
<td>0.33 max</td>
</tr>
<tr>
<td>Manganese</td>
<td>1.14</td>
<td>0.79 – 1.3</td>
</tr>
<tr>
<td>Phosphorus</td>
<td>0.013</td>
<td>0.035 max</td>
</tr>
<tr>
<td>Sulfur</td>
<td>0.006</td>
<td>0.035 max</td>
</tr>
<tr>
<td>Silicon</td>
<td>0.3</td>
<td>0.45 max</td>
</tr>
<tr>
<td>Iron</td>
<td>rem.</td>
<td>rem.</td>
</tr>
</tbody>
</table>
Table 4-2: Tensile properties of SA 455

<table>
<thead>
<tr>
<th>Tensile properties</th>
<th>Supplier specified</th>
<th>ASTM SA 455 requirement</th>
<th>Present Study*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Yield Strength, MPa</td>
<td>420</td>
<td>260 min</td>
<td>423 ± 0.41%</td>
</tr>
<tr>
<td>Tensile Strength, MPa</td>
<td>610</td>
<td>515-655</td>
<td>628 ± 0.41%</td>
</tr>
<tr>
<td>Elongation, %</td>
<td>21.8^</td>
<td>22^</td>
<td>23.9^ ± 0.25%</td>
</tr>
</tbody>
</table>

*Performed at Queen’s University, Kingston, Ontario.

^Over a 1.66” gauge length.

^Over a 2” gauge length.

^Over a 1.5” gauge length.

4.2 Experimental Apparatus

The elevated temperature tensile and creep rupture tests were performed with an 8500 series Instron servo-hydraulic testing instrument in the Department of Mechanical and Materials Engineering at Queen’s University. The test rig is shown in Figure 4-2. A 100 kN dynamic load cell with an error of ±0.25% (Instron, 2008) was used. Since a high temperature extensometer
was not available, displacement, later converted to strain, was measured by the Instron’s linear variable differential transformer (LVDT) with an error of ±0.25%. The error of the LVDT was assumed to be the same as the error of the load cell since no data was found to indicate otherwise.

![Figure 4-2: Instron with furnace, custom grips and sample.](image)

Custom grips, shown in Figures 4-3 and 4-4, were designed and manufactured to hold the test specimens in the furnace. The grips were machined from super alloy A-286, solution treated (982°C) and aged. A-286 has excellent high temperature strength and corrosion resistance, and guaranteed that the deformation within the grips during the creep-rupture testing was negligible compared to that in the gauge length of the specimen. The upper custom grip was attached to a universal coupling that has 2 degrees of rotational freedom, thus minimizing bending forces applied to the samples. Insulation rings show in Figure 4-4 were used to minimize heat transfer from the specimen to the grips.
Figure 4-3: Custom grip’s design drawing.

Figure 4-4: Custom tensile test grip.
A custom made radiant heat transfer clamshell furnace, shown in Figure 4-5, that mounts onto the Instron machine was used to heat the samples.

Figure 4-5: Custom clamshell furnace.

The heater’s maximum operating temperature was 980°C and its power output was 1130W at 120 VAC. Special ceramic fibre insulation was used to plug the top and bottom of the furnace openings to minimize heat loss from the furnace to the surroundings. An Omega CN2100-R20 temperature controller, shown in Figure 4-6, was used to control the sample temperature.

Figure 4-6: Omega CN2100-R20 temperature controller.
4.3 Data Acquisition

Each sample had 3 type K thermocouples spot welded onto it; one at the centre of the gage length and the other 2 at the top and bottom ends of the gage length. The type K thermocouple has a standard error of ± 2.2°C (Omega, 2008). The thermocouple at the centre of the gage length was used by the Omega controller to control the furnace. The thermocouples were spot welded to the gage length with wire ends welded onto opposite sides of the specimen. This was done such that the sample physically became the thermocouple bead and thus error due to bead contact to the sample and surrounding air could be eliminated.

Thermocouple preparation included:

• Spot welding each thermocouple to the gage length.

• Ensuring that there was no wire contact behind the weld. This was done to prevent shorting of the thermocouple from taking place.

• Ceramic insulators were strung over the thermocouple wires in the hot zone and Teflon covering was used to insulate the remaining wire.

An IOtech Personal DaqView 55, personal data acquisition system, was used to collect and store temperature data from the thermocouples at the ends of the specimen’s gage length, see Figures 4-7, 4-8 and 4-9. The thermocouple placed at the centre of the gage length was used by the Omega temperature controller to control the furnace.
Figure 4-7: Data acquisition system used to collect specimen temperatures from the top and bottom of the gage length.

Figure 4-8: Temperature history of top and bottom of gauge length during a creep test at 630°C; start up heating, 20 minute hold time and creep test time are shown.
Displacement and load were read and stored by a computer connected to the Instron’s computer system. These measurements were made once every second during the creep rupture tests and four times a second during the tensile tests. A high temperature extensometer was not available during testing so the displacements were read by the Instron’s LVDT. The stiffness of our system at various temperatures was later determined (see Section 4.5) so that all elastic strains occurring inside and outside the specimen’s gage length could be removed from the data.

4.4 Experimental Details

The specimens took about 50-70 minutes to reach testing temperature and then were held at the test temperature for 20 minutes for the elevated temperature tensile and creep rupture tests. The ASTM standard calls for an hour of holding time for creep rupture tests; however, these creep rupture tests are performed at high enough temperatures and stresses to produce failure in under 100 minutes. Therefore, it was decided that the 20 minute holding time indicated for elevated temperature tensile tests would be more appropriate. Due to the unusually long tensile
samples much difficulty was experienced when attempting to hold the temperature of the gauge length to ±2°C during the tests. Due to this difficulty, the test temperature tolerance was relaxed to +2°C, -4°C, see Figure 4-9. Near the end of the tests, the temperatures may have risen as high +4°C as instability and necking caused the gage length to move significantly in the furnace. This is acceptable according to the creep rupture testing standard provided by ASME (ASME E139-96, 1998) since the samples were not overheated (except slightly at the end of the tests) and exposure time at low temperatures was not significant; therefore, the test results were considered accurate. Otherwise, the creep rupture tests conformed to ASTM E 139-96 standards.

The load was ramped such that it took 5-10% of the total test time to reach the desired test load. The creep rupture tests were conducted at 550°C, 600°C, 630°C, 660°C, 690°C and 720°C. The test loads were chosen such that the creep rupture tests varied in failure time from 3 to 100 minutes. These are extremely low failure times as far as creep-rupture is concerned. However, this is the failure window of the studied application, so these were the conditions tested.

The elevated temperature tensile tests conformed with ASTM E 21-92 standards. The elevated temperature tensile tests were performed at 400°C, 500°C, 600°C and 720°C. No failure was observed to occur at the location of any spot weld, thus it was assumed that the results were not affected by the thermocouple welds, see Figure 4-10.
4.5 Elevated Temperature Tensile Tests

Room and elevated temperature tensile tests were performed on SA 455 steel. An extensometer was only used for the room temperature testing. At high temperatures, the specimen elongation was measured from the Instron’s linear variable differential transformer (LVDT). The measured displacement included the elastic deformation in the non-gage length section of the sample, the grips, the load train and the elastic and plastic deformation in the gauge length of the specimens. The stiffness of the test rig and specimen was measured at each test temperature by loading-unloading experiments. Using these stiffness values, all the elastic deformations were subtracted out of the total deformation data. Consequently, a data set of only...
the specimen’s plastic deformation was obtained. Table 4-3 shows the temperature dependency of SA 455’s yield stress and ultimate tensile strength found in the elevated temperature tensile tests. The temperature dependency of Young’s modulus of elasticity as reported by ASME (2004) for medium carbon steels is also shown in Table 4-3 for reference.

To verify that the measured values of SA 455 are valid, they were compared to published values of TC 128 (Birk et al., 2006) and a plain medium carbon steel (ASME, 2004), see Figure 4-11. The comparison showed, as expected, that TC 128 was stronger at higher temperatures and that plain medium carbon steel was significantly weaker at all temperatures.

<table>
<thead>
<tr>
<th>Temperature ± 2.2°C [°C]</th>
<th>Yield Stress ± 0.41% [MPa]</th>
<th>UTS ± 0.41% [MPa]</th>
<th>E (ASME,2004) [GPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>22</td>
<td>423</td>
<td>628</td>
<td>201</td>
</tr>
<tr>
<td>400</td>
<td>415</td>
<td>617</td>
<td>169</td>
</tr>
<tr>
<td>500</td>
<td>285</td>
<td>420</td>
<td>148</td>
</tr>
<tr>
<td>600</td>
<td>195</td>
<td>225</td>
<td>121</td>
</tr>
<tr>
<td>720</td>
<td>92</td>
<td>92</td>
<td>83</td>
</tr>
</tbody>
</table>
Figure 4-11: Comparison of ultimate tensile strength of SA 455, TC 128 and medium carbon steel.

Figure 4-12 shows a plot of the temperature dependency of the true stress versus true plastic strain curves. At room temperature (22°C) there was observed a detectable yield point and Lüders strain, which disappears at higher temperatures. The Lüders strain is commonly observed in mild and medium carbon steels, it is the region of yielding that displays almost perfectly plastic behaviour as plastic deformation begins due to unpinning of dislocations from nitrogen and carbon atmospheres (Hall, 1970). Higher temperatures are characterized by lower yield strengths and reduced work hardening. This data is needed to conduct high temperature elastic-plastic finite element analyses of SA 455 steel.
From Figure 4-12 it is shown that at 720°C, SA 455 steel experiences little plastic hardening and it quite weak. This can be explained by the fact that at 727°C steel’s microstructure changes from ferrite to austenite, a much weaker configuration (Callister, 1997).

### 4.6 Creep Rupture Tests

The creep rupture experimental results are summarized in Table 4-4. Note that since a high temperature extensometer was not used, the specimen extension was determined from the Instron’s actuator LVDT, which includes elastic deformation in the entire specimen, grips and load train. The LVDT displacement was converted to specimen extension by dividing it by the sample’s gauge length. Later on, all the elastic strains were subtracted out of the data using the system stiffness that was measured at all temperatures. The N/As in Table 4-4 represent tests for which computer problems caused no test data to be stored.
Table 4-4: Stress-rupture properties of SA 455 steel

<table>
<thead>
<tr>
<th>Mean temperature ± 2.2°C [°C]</th>
<th>Initial stress ± 0.41% [MPa]</th>
<th>Rupture time [min]</th>
<th>Elongation* at rupture ± 0.25% [%] LVDT sensor</th>
<th>Elongation** at rupture ± 0.05% [%] post measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>550</td>
<td>326.7</td>
<td>1.90</td>
<td>30</td>
<td>30.7</td>
</tr>
<tr>
<td>550</td>
<td>296.3</td>
<td>7.57</td>
<td>33.5</td>
<td>32.7</td>
</tr>
<tr>
<td>550</td>
<td>266</td>
<td>21.57</td>
<td>36.3</td>
<td>37.5</td>
</tr>
<tr>
<td>550</td>
<td>265.8</td>
<td>22.81</td>
<td>35.3</td>
<td>34.0</td>
</tr>
<tr>
<td>600</td>
<td>220.5</td>
<td>4.84</td>
<td>N/A</td>
<td>41.2</td>
</tr>
<tr>
<td>600</td>
<td>196.3</td>
<td>10.00</td>
<td>45.1</td>
<td>44.8</td>
</tr>
<tr>
<td>600</td>
<td>177.3</td>
<td>20.69</td>
<td>48.4</td>
<td>47.2</td>
</tr>
<tr>
<td>600</td>
<td>151.3</td>
<td>67.43</td>
<td>49.9</td>
<td>50.6</td>
</tr>
<tr>
<td>630</td>
<td>176.7</td>
<td>5.72</td>
<td>N/A</td>
<td>50.3</td>
</tr>
<tr>
<td>630</td>
<td>175</td>
<td>5.62</td>
<td>54.9</td>
<td>55.4</td>
</tr>
<tr>
<td>630</td>
<td>149.3</td>
<td>15.29</td>
<td>52.9</td>
<td>54.0</td>
</tr>
<tr>
<td>630</td>
<td>130</td>
<td>47.56</td>
<td>56.2</td>
<td>56.6</td>
</tr>
<tr>
<td>660</td>
<td>150</td>
<td>3.56</td>
<td>50.8</td>
<td>51.2</td>
</tr>
<tr>
<td>660</td>
<td>125</td>
<td>12.78</td>
<td>57.9</td>
<td>60.7</td>
</tr>
<tr>
<td>660</td>
<td>106.7</td>
<td>29.26</td>
<td>N/A</td>
<td>51.7</td>
</tr>
<tr>
<td>690</td>
<td>113</td>
<td>4.30</td>
<td>56.8</td>
<td>61.3</td>
</tr>
<tr>
<td>690</td>
<td>100</td>
<td>10.53</td>
<td>64.9</td>
<td>65.9</td>
</tr>
<tr>
<td>690</td>
<td>77</td>
<td>38.33</td>
<td>67.2</td>
<td>64.2</td>
</tr>
<tr>
<td>720</td>
<td>89.3</td>
<td>3.45</td>
<td>72.4</td>
<td>71.1</td>
</tr>
<tr>
<td>720</td>
<td>74</td>
<td>13.27</td>
<td>76.8</td>
<td>76.4</td>
</tr>
<tr>
<td>720</td>
<td>50</td>
<td>80.57</td>
<td>107.8</td>
<td>105.4</td>
</tr>
</tbody>
</table>

*Based on a 1.5” gauge length. Deflection measurements were made by the Instron, no extensometer used.

**Based on a 1.5” gauge length. Measured with vernier calipers.

The test specimen gauge length was measured after fracture with vernier calipers, and compared to the Instron reported value at rupture.

SA 455 steel’s stress rupture data presented in Table 4-4 is plotted in Figure 4-13 as stress versus time to rupture. For comparison purposes TC 128’s stress rupture is plotted in Figure 4-14; TC 128 is a pressure vessel steel that has similar composition to SA 455 steel. The failure times for SA 455 steel and TC 128 steel are similar as expected.

All SA 455 steel creep rupture tests produced conventional creep curves. The creep curves were all similar in that the tertiary stage of creep is the most prominent due to the relatively quick creep failure times. Examples of these plots from various testing conditions are shown in Figure 4-15; all the creep curves are shown in Appendix A. Note that the elastic strains in the sample, load train
and grips were subtracted out of the total strain to give creep strain. This was accomplished by measuring the stiffness of our test rig (specimen, load train and grips) at the various temperatures. Once this stiffness was found, all elastic strains were removed from the data.

![Figure 4-13: Plot of SA 455 steel’s stress rupture data.](image1)

![Figure 4-14: Plot of TC 128 steel’s stress rupture data (Birk et al., 2006).](image2)
Steady state creep of a material is most often modeled using Norton’s creep relationship given in Equation 3-20. Note that the stress and strain in Norton’s creep relationship are either true or engineering stress and strain, based on if the creep rupture data is from a constant true or engineering stress test. In order to derive Norton’s constants, one needs minimum creep rates under various conditions. Table 4-5 provides the minimum creep rates of SA 455 steel for our various testing conditions. In the creep rupture tests performed there was no significant primary or secondary creep, therefore, a minimum creep rate was based on the average of the creep rates during the first 30% of the test duration.

Due to the high loads at which the creep rupture tests were performed, creep failure times are relatively short and the shape of the creep curve is dominated by tertiary creep behaviour. However, even under these conditions, minimum creep rates as well as Norton’s creep constants can be useful.
Table 4-5: Minimum creep rates of SA 455 steel.

<table>
<thead>
<tr>
<th>Mean temperature ± 2.2°C</th>
<th>Initial stress ± 0.41% [MPa]</th>
<th>Minimum creep rate ± 0.5% [strain per second]</th>
</tr>
</thead>
<tbody>
<tr>
<td>550</td>
<td>326.7</td>
<td>1.10E-03</td>
</tr>
<tr>
<td>550</td>
<td>296.3</td>
<td>2.40E-04</td>
</tr>
<tr>
<td>550</td>
<td>266</td>
<td>7.08E-05</td>
</tr>
<tr>
<td>550</td>
<td>265.8</td>
<td>5.74E-05</td>
</tr>
<tr>
<td>600</td>
<td>196.3</td>
<td>1.95E-04</td>
</tr>
<tr>
<td>600</td>
<td>177.3</td>
<td>9.35E-05</td>
</tr>
<tr>
<td>600</td>
<td>151.3</td>
<td>1.55E-05</td>
</tr>
<tr>
<td>630</td>
<td>176.7</td>
<td>4.08E-04</td>
</tr>
<tr>
<td>630</td>
<td>149.3</td>
<td>1.23E-04</td>
</tr>
<tr>
<td>630</td>
<td>130</td>
<td>2.83E-05</td>
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<td>6.40E-04</td>
</tr>
<tr>
<td>660</td>
<td>125</td>
<td>1.59E-04</td>
</tr>
<tr>
<td>660</td>
<td>106.7</td>
<td>3.23E-05</td>
</tr>
<tr>
<td>690</td>
<td>113</td>
<td>6.37E-04</td>
</tr>
<tr>
<td>690</td>
<td>100</td>
<td>1.69E-04</td>
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<td>77</td>
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</tr>
<tr>
<td>720</td>
<td>89.3</td>
<td>9.30E-04</td>
</tr>
<tr>
<td>720</td>
<td>74</td>
<td>1.61E-04</td>
</tr>
<tr>
<td>720</td>
<td>50</td>
<td>1.85E-05</td>
</tr>
</tbody>
</table>

4.7 Creep Data Correlation

A correlation through all temperatures for the creep rupture data was desired. Given a temperature and stress the correlation accurately calculated failure times. When correlation failure times diverge from experimental failure times it was important that the correlation values remain conservative since a conservative analytical model was desired.

Larson-Miller (1952) is a widely used and accepted correlation of stress rupture data. The Larson-Miller model correlates stress rupture data using the Larson-Miller parameter (P), which is defined as follows:

\[ P_{L-M} = T(C_{L-M} + \log t) \]  

(Equation 4-1)
where, $P_{L-M}$ is the Larson-Miller parameter, $T$ is temperature measured in °R and $t_r$ is rupture time measured in hours. The Larson-Miller constant was originally indicated as taking on a value of 20 (Larson and Miller, 1952).

Since the creep rupture testing was conducted under conditions that generated unusually fast creep failures the validity of using $C_{L-M}=20$ was investigated. If one plots log of failure time versus $1/T$ and draws lines of constant stress, the y-intercept of the lines should be roughly the same value, $-C_{L-M}$. Such a plot for the measured SA 455 data is shown in Figure 4-16. Due to the testing conditions only 3 lines of constant stress could be drawn; the tests were conducted at constant temperature not constant stress conditions.

After examining Figure 4-16, it was apparent that a Larson-Miller constant between 17 and 20 would best suit the data set. It was found that a value of $C_{L-M}=19$ gave the most accurate correlation while being conservative. The Larson-Miller plot, with $C_{L-M}=19$, is presented in Figure 4-17. The values of the Larson-Miller parameter, $P_{L-M}$, are listed in Table 4-6.

![Figure 4-16: Investigating the suitability of $C_{L-M}=20$ for SA 455 steel.](image)
Figure 4-17: Larson-Miller correlation for SA 455 steel, $C_{LM}=19$.

Table 4-6: Larson-Miller parameter values

<table>
<thead>
<tr>
<th>Mean temperature ± 0.75% [°R]</th>
<th>Initial stress ± 0.41% [MPa]</th>
<th>Larson-Miller parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>1481.7</td>
<td>326.7</td>
<td>25930.1</td>
</tr>
<tr>
<td>1481.7</td>
<td>296.3</td>
<td>27493.3</td>
</tr>
<tr>
<td>1481.7</td>
<td>266</td>
<td>27529.5</td>
</tr>
<tr>
<td>1481.7</td>
<td>265.8</td>
<td>26819.6</td>
</tr>
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<td>1571.7</td>
<td>220.5</td>
<td>29134.9</td>
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<td>1571.7</td>
<td>196.3</td>
<td>29941.5</td>
</tr>
<tr>
<td>1571.7</td>
<td>177.3</td>
<td>28638.7</td>
</tr>
<tr>
<td>1571.7</td>
<td>151.3</td>
<td>28142.8</td>
</tr>
<tr>
<td>1625.7</td>
<td>176.7</td>
<td>29122.9</td>
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<td>1625.7</td>
<td>175</td>
<td>29215.4</td>
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<td>149.3</td>
<td>29922.3</td>
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<td>1625.7</td>
<td>130</td>
<td>30723.7</td>
</tr>
<tr>
<td>1679.7</td>
<td>150</td>
<td>29853.6</td>
</tr>
<tr>
<td>1679.7</td>
<td>125</td>
<td>30785.4</td>
</tr>
<tr>
<td>1679.7</td>
<td>106.7</td>
<td>31389.9</td>
</tr>
<tr>
<td>1733.7</td>
<td>113</td>
<td>31629.8</td>
</tr>
<tr>
<td>1733.7</td>
<td>100</td>
<td>32602.4</td>
</tr>
<tr>
<td>1733.7</td>
<td>77</td>
<td>30955.8</td>
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<td>1787.7</td>
<td>89.3</td>
<td>32794.1</td>
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<td>74</td>
<td>34194.6</td>
</tr>
<tr>
<td>1787.7</td>
<td>50</td>
<td>31749.4</td>
</tr>
</tbody>
</table>
To show the accuracy of this correlation, Figure 4-18 plots the correlation predicted failure times on the vertical axis and experimental failure times on the horizontal axis. The line y=x represents when the two times are exactly equal. Points below the y=x line are conservative predictions made by the correlation.

![Figure 4-18: Correlated vs measured failure times.](image)

The equation of the second order polynomial used to fit the data in Figure 4-17 is as follows:

\[
\sigma = (2.813 \times 10^{-6})P_{L-M}^2 - (2.042 \times 10^{-1})P_{L-M} + 3.742 \times 10^3 \quad \text{(Equation 4-2)}
\]

For many applications it is useful to have an equation that would generate failure times as a function of temperature and stress. The first step to such an equation would be to solve the above equation for \(P_{L-M}\), the Larson-Miller parameter.

\[
P_{L-M} = \frac{2.04 \times 10^{-1} - \sqrt{(2.04 \times 10^{-1})^2 - 4(2.81 \times 10^{-6})(3.74 \times 10^3 - \sigma)}}{2(2.81 \times 10^{-6})} \quad \text{(Equation 4-3)}
\]
Note that stress in Equation 4-3 was measured in MPa. It should also be noted that the ± is simply a subtraction, since the addition solution is physically meaningless. The solution for $P_{L-M}$ can be entered into Equation 4-4 to produce the failure time as follows.

$$t_r = 10^{(P_{L-M}/T)^{-19}}$$  \hspace{1cm} (Equation 4-4)

where, $t_r$ is rupture time in hours.

Note that generally the Larson Miller correlation has been shown to provide good predictions for extrapolated creep rupture data (Kraus, 1980). However, the creep rupture tests performed in this study were at relatively high stresses, so rupture times were low. Therefore, caution and engineering judgment is recommended when using this correlation to extrapolate creep failure times beyond 100 minutes.
Chapter 5  SA 455 Steel Creep and Creep Damage Constants

From the literature review summarized in Section 2.0 the Kachanov One-State Variable and
MPC Omega method were the creep damage models selected to numerically predict the tertiary creep
behaviour of SA 455 steel. The creep constants for these models were extracted from the experimental
creep rupture tests. This section presents the values of the creep constants and outlines the procedure
used to compute them from experimental data.

5.1 Kachanov One-State Variable Constants

The method used to determine the temperature dependent material constants $m$, $n$, $\phi$, $\chi$, $B'$
and $A'$ was described in great detail by Hyde, Sun and Tang (1998).

With some simple equation rearrangements, the creep strain was normalized by that of the
failure creep strain to give:

$$\frac{\varepsilon^c}{\varepsilon_f^c} = \left\{ 1 - \left[ 1 - \left( \frac{t}{t_r} \right)^{(1+m)} \right]^{1-n/(1+\phi)} \right\}$$

(Equation 5-1)

where $\varepsilon^c$ is creep strain, $\varepsilon_f^c$ is creep strain at rupture, $t$ is time, $t_r$ is rupture time and $m$, $n$, and $\phi$ are material constants.

In order to determine $m$, $n$ and $\phi$ material constants, a three variable least squares optimization
approach was conducted. The least squares optimization method is given by:

$$F = \sum_{i=1}^{n} \left( y_i - f(t_i, m, n, \phi) \right)^2$$

(Equation 5-2)
where, $F$ is to be minimized, $y_i$ is the normalized experimental data and $f(t_i, m, n, \phi)$ is the calculated normalized creep strain based on Equation 5-1.

This process gives the optimized values of $m$, $n$ and $\phi$ and will produce the best fit to the normalized experimental data plots. Once material constants $m$, $n$ and $\phi$ are found, $\chi$ and $B'$ can be found by plotting the natural logarithm of failure time versus the natural logarithm of nominal test stress. Figure 5-1 shows this plot for SA 455 steel.

![Figure 5-1: Relationship between natural logarithm of rupture time and natural logarithm of initial stress; plot needed to compute $\chi$ and $B'$ for SA 455.](image)

The material constant $\chi$ is given by the negative of the slope of the linear trend-line that fits the data and $B'(\phi+1)$ is given by $\exp(-y_{\text{intercept}})$. Therefore, the only material constant left to determine is $A'$. The value of $A'$ can be determined by using Equation 5-3. Equation 5-3 was derived
by integrating the Kachanov One-State Variable creep equation by time and isolating the strain at time
of failure. Details of this process are presented by Hyde et al. (1998).

\[
\varepsilon_f^c = \frac{A' \sigma^n t_f^{(1+m)}}{1 + m \left[1 - \left(n/(\phi + 1)\right)\right]} \quad \text{(Equation 5-3)}
\]

The material constants for SA 455, found as discussed above, are summarized in Table 5-1.

**Table 5-1: One-state variable continuum damage mechanics material constants for SA 455.**

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>A’</th>
<th>n</th>
<th>m</th>
<th>B’</th>
<th>(\phi)</th>
<th>(\chi)</th>
</tr>
</thead>
<tbody>
<tr>
<td>550</td>
<td>1.1x10^{-17}</td>
<td>5.3</td>
<td>0</td>
<td>3.8x10^{-15}</td>
<td>6.5</td>
<td>11.7</td>
</tr>
<tr>
<td>600</td>
<td>9.1x10^{-18}</td>
<td>5.7</td>
<td>0</td>
<td>1.6x10^{-20}</td>
<td>6.7</td>
<td>7.0</td>
</tr>
<tr>
<td>630</td>
<td>2.1x10^{-17}</td>
<td>5.8</td>
<td>0</td>
<td>1.2x10^{-19}</td>
<td>6.4</td>
<td>6.9</td>
</tr>
<tr>
<td>660</td>
<td>2.3x10^{-16}</td>
<td>5.6</td>
<td>0</td>
<td>1.9x10^{-17}</td>
<td>6.6</td>
<td>6.2</td>
</tr>
<tr>
<td>690</td>
<td>5.9x10^{-16}</td>
<td>5.8</td>
<td>0</td>
<td>1.7x10^{-15}</td>
<td>6.5</td>
<td>5.6</td>
</tr>
<tr>
<td>720</td>
<td>3.5x10^{-15}</td>
<td>5.8</td>
<td>0</td>
<td>2.5x10^{-14}</td>
<td>6.5</td>
<td>5.3</td>
</tr>
</tbody>
</table>

The temperature dependency of material constants \(n\), \(\phi\), \(\chi\), B’ and A’ was plotted in Figures 5-2 and 5-3. The zero values for m are not concerning since the value of m is often 0 or close to 0 for most steels (Hyde et al., 2006 and 1998). The values of the natural logarithms of A’ and B’ increased linearly with temperature. The exception to this was the value of \(\ln(B')\) at 550°C. The values of \(\phi\) showed little dependence on temperature. The values of n increased linearly with temperature and \(\chi\) decreased exponentially with temperature. In the literature review, no work was found that addressed the temperature dependencies of these creep constants. Therefore, the author cannot compare these trends to trends observed by others.
The two MPC Omega material constants, $\Omega_p$ and $\dot{\epsilon}_0$, are dependent on temperature and stress. Omega ($\Omega_p$) is the slope of the line that fits the plot of the natural logarithm of the true creep strain rate versus the true creep strain. This can be seen by the following equation:
\[ \frac{d \ln \dot{\varepsilon}^c}{d \varepsilon^c} = \Omega_p \]  

(Equation 5-4)

where, \( \dot{\varepsilon}^c \) is the creep strain rate, \( \varepsilon^c \) is the creep strain and \( \Omega_p \) is the material constant omega.

An example of such a plot used to determine \( \Omega_p \) is shown in Figure 5-4.

Once \( \Omega_p \) is known, the initial creep strain rate constant (\( \dot{\varepsilon}_0 \)) is found using Equation 2-4 and the experimental data. Norton’s creep exponent, \( n \), was found using the data from Table 4-5. Table 5-2 presents the Omega MPC material constants for SA 455. The stress dependency of \( \Omega_p \) and \( \dot{\varepsilon}_0 \) are plotted in Figures 5-5 and 5-6.

![Graph](image)

**Figure 5-4:** Determining SA 455 steel’s value of Omega for a test temperature of 600°C and stress of 177.3 MPa.
Table 5-2: SA 455 steel Omega MPC creep constants

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Stress (MPa)</th>
<th>Ωp</th>
<th>(\dot{\varepsilon}_0)</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
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<td>326.7</td>
<td>12</td>
<td>6.80x10^{-4}</td>
<td>13.66</td>
</tr>
<tr>
<td>550</td>
<td>296.3</td>
<td>15.8</td>
<td>1.36x10^{-4}</td>
<td>13.66</td>
</tr>
<tr>
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<td>266</td>
<td>20.5</td>
<td>3.77x10^{-3}</td>
<td>13.66</td>
</tr>
<tr>
<td>600</td>
<td>196.3</td>
<td>10</td>
<td>1.62x10^{-4}</td>
<td>9.87</td>
</tr>
<tr>
<td>600</td>
<td>177.3</td>
<td>11.9</td>
<td>6.72x10^{-5}</td>
<td>9.87</td>
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<td>11</td>
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<td>8.64</td>
</tr>
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<td>150</td>
<td>8</td>
<td>5.57x10^{-4}</td>
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<td>50</td>
<td>9.6</td>
<td>2.17x10^{-5}</td>
<td>6.57</td>
</tr>
</tbody>
</table>

Figure 5-5: Stress dependency of \(\Omega_p\).

The values of \(\Omega_p\) decreased linearly with stress and decreased with temperature. The values of \(\dot{\varepsilon}_0\) increased exponentially with stress and increased with temperature. These trends could not be
directly compared to the trends of the temperature and stress dependencies of $\Omega_p$ and $\dot{\varepsilon}_0$ as given by the API Fitness for Service because of the vast difference in failure time predictions. However, the general trends do compare well with those observed by Prager (2000).

![Figure 5-6: Stress dependency of $\dot{\varepsilon}_0$.](image-url)
Chapter 6  Numerical Analysis of Tensile Specimens

The creep and creep damage Equations 3-28, 3-29 and 3-34, 3-39 were implemented into both ANSYS and ABAQUS via usercreep subroutines using the values of the creep constants found in Section 5.0. Creep damage was calculated as a solution dependent variable in the user subroutines. The gage length of the test specimens used in the lab was modeled for the numerical analysis work. Failure was determined to have occurred when the damage parameter or life fraction consumed was equal to 1 in elements throughout the thickness of the samples or when the analyses became numerically unstable due to high creep strain rates. The objective of these finite element analyses was to validate the creep damage models for the case of uniaxial creep loading as well as to assess the non-linear creep analysis capabilities of both ANSYS and ABAQUS.

6.1  Tensile Specimen Model

In order to model the structural high temperature failure of pressure vessels, a material model capable of predicting the high-temperature creep behaviour of SA 455 steel was required.

Since creep of steels is usually studied on the time scale of hundreds or thousands of hours, it was not clear which creep damage model(s) would best suit this application.

The objective of Section 6 was to examine if the chosen creep damage models would accurately predict the high-temperature creep rupture behaviour of SA 455 steel and to validate the creep constants found in Section 5. From the literature review summarized in Section 2, it was determined that the Kachanov One-State Variable Technique and MPC Omega Method would be best suited to model the high-temperature creep behaviour of SA 455 pressure vessel steel.

Commercial finite element analysis computer codes ANSYS and ABAQUS are both used in order to examine their performance when modeling relatively fast tertiary creep. An in-depth
comparison between experimental uniaxial creep rupture properties and those predicted by the finite element analyses was conducted.

6.1.1 Analysis Geometry and Mesh

The dimensions of the tensile specimen being modeled are shown in Figure 4-1. The objective of the numerical analyses was to predict the experimentally measured creep behaviour of the tensile specimen’s gage length. Therefore, only the gage length of the specimen was modeled. Figure 6-1 shows the part model used in the numerical analyses. From Figure 6-1, it is clear that the model includes a bit of specimen length outside of the gauge length at the top and bottom of the model. These lengths were included so that boundary conditions and loading do not have any local effects on the gage length; this was later confirmed by the analyses. The boundary conditions and loads are shown in Figure 6-2.

![Figure 6-1: Numerical model of the tensile specimen’s gage length.](image)
In ABAQUS, 448 C3D8R elements were used to mesh the gauge length model, see Figure 6-3. C3D8R is a three-dimensional solid hexahedral (brick) element with 8 nodes, reduced integration and hourglass control. Each node has three degrees of freedom, translation in the x, y and z directions. Reduced integration uses a lower-order integration to form the element stiffness. Reduced integration reduces running time, especially in three dimensions. For first-order elements, the accuracy achieved with full versus reduced integration is largely dependent on the nature of the problem (ABAQUS, 2007). The gage length model does not have any curved surfaces in which the stress state is of interest. Therefore, a first-order element was chosen to mesh the part since it is computationally less costly than a higher-order element. Analyses with both full and reduced integration were completed. Since run time was on the order of a couple of minutes on a PC with a 3.2 GHz Pentium 4 processor,
not much difference in run times was observed when using elements with full or reduced integration. Results were also not affected by the type of integration used. Therefore, it was decided to use elements with reduced integration.

Hourglassing can be a problem with first-order, reduced integration elements in stress/displacement analyses. Since the elements have only one integration point it is possible for them to distort in such a way that the strains calculated at the integration points are all zero, which in turn leads to uncontrolled distortion of the mesh, called hourglassing (ABAQUS, 2007). Therefore, ABAQUS implemented hourglass control algorithms into the formulation of these elements.

In ANSYS, 448 SOLID185 elements were used to mesh the gauge length model, see Figure 6-4. SOLID185 is a three-dimensional, first-order element with 8 nodes. Each node has three degrees
of freedom, translation in the x, y and z directions. Reduced integration with hourglass control was also used with the SOLID 185 element.

Figure 6-4: ANSYS meshed specimen gauge length.

Mesh convergence studies are important because they ensure that the results obtained from a numerical analyses were not dependent upon the model’s mesh. In an h-refinement mesh convergence study, the number of elements is increased while certain output variables such as maximum von Mises stress are monitored. In a p-refinement mesh convergence study, the order of the element is increased while certain output variables are monitored.

An h-refinement mesh convergence study was conducted to ensure that results were not dependent on mesh density. A p-refinement convergence study was not needed since higher-order elements were not expected to improve the results for these analyses as previously discussed. Results of the h-refinement mesh convergence study are shown in Figures 6-5 and 6-6.
Figure 6-5: Mesh convergence using maximum von Mises stress.

Figure 6-6: Mesh convergence using maximum displacement.

From the h-refinement mesh convergence study it was determined that 448 elements could accurately predict the structural behaviour while maintaining computational efficiency. This was
determined because when increasing from 448 elements to 850, the maximum von Mises stress only decreased by 1.3% and the maximum displacement only decreased by 2.4%. Since almost doubling the number of elements had such a small effect on the change of predicted displacement and stress, the solution is considered not to be dependent on the mesh.

6.1.2 Kachanov One-State Variable

The experimental creep rupture data presented in Section 4.6 showed that, under the conditions tested, SA 455 exhibits tertiary creep behaviour. As summarized in the literature review in Section 2.2, the Kachanov One-State Variable Technique is capable of modeling tertiary creep. The uniaxial form of the One-State Variable equations, as introduced in Section 3.1.3, is:

\[ \dot{\varepsilon}^c = A\left(\frac{\sigma}{1-\omega}\right)^n t^m \]  
(Equation 6-1)

\[ \dot{\omega} = B\frac{\sigma^p}{(1-\omega)^q} t^m \]  
(Equation 6-2)

where, \( \dot{\varepsilon}^c \) is the creep strain rate, \( A' \), \( B' \), \( n \), \( m \) and \( \Omega \) are material constants, \( \sigma \) is stress, \( \omega \) is creep damage and \( t \) is time.

These equations were implemented into ANSYS and ABAQUS via the creep user subroutine.

6.1.3 MPC Omega

As summarized in the literature review in Section 2.2, the MPC Omega Method is capable of modeling tertiary creep. The uniaxial form of the MPC Omega Method, as introduced in Section 3.1.3, is:
\[
\dot{\varepsilon}^c = \dot{\varepsilon}_o e^{\Omega_p \varepsilon^c}
\]  
\text{(Equation 6-3)}

\[
\frac{t_s}{t_r} = \text{Life Fraction Consumed} = \frac{\dot{\varepsilon}^c t_s \Omega_p}{\dot{\varepsilon}^c t_s \Omega_p + 1}
\]  
\text{(Equation 6-4)}

where, \(\dot{\varepsilon}^c\) is the creep strain rate, \(\dot{\varepsilon}_o\) is the initial creep strain rate (material constant), \(\varepsilon^c\) is the creep strain, \(\Omega_p\) is the material constant omega, \(t_s\) is time in service, \(t_r\) is rupture time and \(\dot{\varepsilon}^c\) is the effective creep strain rate.

These equations were implemented into ANSYS and ABAQUS via the creep user subroutine.

### 6.2 Results and Discussion

In order to validate the finite element analysis creep models, uniaxial creep runs were conducted under the same loads and temperatures as the creep rupture experiments. The purpose of this was to ensure that all derived creep material constants were correct and to verify if the MPC Omega and Kachanov methods could accurately predict the measured creep behaviour of SA 455 steel under uniaxial stress.

The results of the finite element analyses were compared to the experimental findings in Figures 6-7 through 6-12. By inspecting these plots it is apparent that each of the four types of analyses produced excellent failure time predictions. However, in terms of predicting true creep strain at rupture, the MPC Omega method generally outperformed the Kachanov technique. It was found that both ABAQUS and ANSYS commercial computer codes could model the high-temperature creep ruptures.
The numerical analysis creep strain plotted in Figures 6-7 through 6-12 is the average creep strain measured over the gage length calculated by dividing the total deformation along the gage length by the final length of the gage length. This is how creep strain was measured in the lab and thus to make an accurate comparison between numerical analysis results and experimental results, creep strain must be measured in the same way.

Figures 6-13 and 6-14 were plotted to enable quick examination of the overall accuracy with which the models predicted failure times and failure strains. Figures 6-15 and 6-16 show the typical FEA creep damage contour plots that were produced by the analyses.

![Graph showing creep strain vs. time for different stress levels and models.](image)

**Figure 6-7:** SA 455 steel uniaxial creep validation of finite element analyses, T=550°C.
Figure 6-8: SA 455 steel uniaxial creep validation of finite element analyses, T=600°C.
Figure 6-9: SA 455 steel uniaxial creep validation of finite element analyses, T=630°C
Figure 6-10: SA 455 steel uniaxial creep validation of finite element analyses, T=660°C.
Figure 6-11: SA 455 steel uniaxial creep validation of finite element analyses, T=690°C.
Figure 6-12: SA 455 steel uniaxial creep validation of finite element analyses, T=720°C.

Figure 6-13: Predicted failure time.
Figure 6-14: Predicted creep strain at failure.

Figure 6-15: Typical ABAQUS creep damage contour plot at failure.
It should be noted that the numerical analyses predicted two necks to occur in the samples due to the fact that the numerical analyses assumed the temperature over the gage to be constant and thus necks formed at the location of peak stresses. As previously mentioned, the comparison of creep strains presented in Figures 6-7 through 6-12 compared the averaged creep strain over the entire gage length. During the experiments the sample necked at only one location, the location of peak temperature. However, if gage length of the numerical analysis specimens was cut in half to account for model symmetry then the strain measured over the gage length would be the same and only capture one neck. Also, another solution to this discrepancy would be to analyze the gage lengths again with a slightly higher temperature occurring at some location along the gage length, perhaps the middle, in order to force only one neck to develop.

In order to further understand some of the formulation differences that exist between the Kachanov technique and the MPC Omega method, plots of damage accumulation versus time are shown in Figures 6-17 and 6-18. Examination of the plots revealed that when modeling high-temperature creep, the Kachanov type equations showed an exponential type creep damage increase while the MPC Omega method showed a linear creep damage increase. The Kachanov type equations are more accurate at predicting the realistic trend in creep damage accumulation, while the MPC Omega method damage parameter is more of a measure of life fraction consumed. Therefore, in most
cases, the MPC Omega method would be more numerically stable than the Kachanov technique when modeling stress rupture.

Numerical instability is often encountered when modeling creep rupture due to the increasing creep strain rates that lead to failure. The Kachanov creep strain rate is a function of creep damage, see Equations 6-1 and 6-2, since the creep damage increases rapidly near failure, as plotted in Figure 6-17, numerical instability is an issue as creep damage increases past 50%. From Figures 6-7 to 6-12, it is obvious that both the MPC Omega method and Kachanov technique can accurately predict the creep rupture behaviour of SA 455 steel. However, the MPC Omega creep strain rate is not a function of creep damage and so for high-temperature creep analysis, the MPC Omega method was found to be more numerically stable.

![Figure 6-17: Creep damage accumulation using the Kachanov technique; T=600°C S=177.3 MPa, solved with ABAQUS.](image)

![Figure 6-18: Creep damage accumulation using the MPC Omega method; T=600°C S=177.3 MPa, solved with ABAQUS.](image)
Chapter 7 Numerical Pressure Vessel Analysis

Numerical pressure vessel analyses were conducted to gain a better understanding of the vessel’s structural failure that occurred due to accidental fires. The effects of pressure vessel geometry, fire intensity and type of fire impingement were studied. This study is of engineering interest since it was shown experimentally that pressure vessels exposed to accidental fires can fail in less than 30 minutes (Birk et al., 2006) and some design standards (CFR 49 Part 179, 2008; CGSB-43.147, 2005; ASME, 2001) require that pressure vessels exposed to accidental fires must resist failure for at least 100 minutes. Both experimental and computational work will aid to improve the understanding of pressure vessel failure and will hopefully lead to safer pressure vessel designs.

The numerical analysis work conducted to validate uniaxial creep rupture was performed with both ANSYS and ABAQUS commercial FEA computer codes. However, when the problem was complicated by the multiaxial states of stress present in the pressure vessel analyses, ABAQUS clearly outperformed ANSYS. ANSYS was struggling with convergence issues when attempting to solve this non-linear problem. Therefore, only ABAQUS was used to conduct all of the pressure vessel analyses.

Implicit time integration was used for the creep analysis since it is unconditionally stable and can tolerate relatively large time steps (Becker et al., 1993). For an implicit integration creep analysis, the maximum allowable creep strain increment needs to be specified by the user. The smaller this value, the more accurate the solution. However, a smaller maximum creep strain increment will increase computational time as more increments will be needed. Therefore, the sensitivity of the solution with respect to the maximum creep strain increment should be investigated. In ABAQUS, the maximum creep strain increment is referred to as “CETOL”. From a sensitivity study, it was found that when CETOL was varied between the recommended values of $1 \times 10^{-3}$ and $1 \times 10^{-4}$, the failure time
and run time were essentially unchanged. Since the value of CETOL did not affect the results it indicated that the automatic time stepping was not being affected by the changing CETOL values. Therefore, in order to keep the analysis as accurate as possible, the smallest value of CETOL, $1 \times 10^{-4}$, was used for all of the analyses.

Non-linear FEA solution techniques were used to perform the numerical pressure vessel analyses. Sources of analysis non-linearities included the following:

- Geometric non-linearity due to large deformations
- Material non-linearity due to plastic and creep deformations.

Large deformations were present because this was a high temperature failure analysis of steel. Structural vessel deformations were large enough to have a significant effect on the load-deflection characteristics of the analysis. In general, to determine if large deformation behaviour should be accounted for, practitioners use Roark and Young’s (1975) rule of thumb, which indicates that large deformations should be accounted for if the transverse displacement exceeds about one-half the wall thickness. Material plasticity was present in the analysis when temperatures and stresses were high enough at one or more integration point(s) to cause material yielding. Plasticity was modeled as bilinear isotropic because loading was monotonic and thus cyclic loading effects were not present. Material creep was present in the analysis when temperatures at one or more integration point(s) were above a certain pre-determined temperature (550°C in the case of this work). For further information, Hinton (1992) provides a detailed discussion of the non-linearities associated with large deformation, plasticity and creep.

All ABAQUS numerical analyses were submitted using script files written in Python and user subroutines written in FORTRAN.
7.1 Geometry and Mesh

Numerical analyses were performed mainly on 500 US gallon pressure vessels. However, some analyses were also conducted on 1,000 and 33,000 US gallon pressure vessels.

ABAQUS’ S4 finite shell elements were used to mesh the pressure vessels. The S4 shell finite element has 4 nodes and six degrees of freedom at each node, three displacement components and three rotation components. The element allows transverse shear deformation. It uses thick shell theory as the shell thickness increases and becomes discrete Kirchhoff thin shell elements as the thickness decreases.

The theory of thin shell structures was first developed by Kirchhoff (1850), Aron (1874) and Love (1888). This theory was later refined by many researchers. The main condition that must be met in order to use thin shell theory is that the ratio of thickness to radius of curvature of the structure must be less than 0.1. The assumption of thin shell theory basically ignores the through thickness stresses as they are usually at least about 5 times lower than the normal stresses in the other two directions. This is similar to the assumption of plane stress, where one dimension of the structure is much smaller than the other two, only it is applied to shell structures. The 500 and 1000 US gallon pressure vessels both have thickness to radius of curvature ratios of 0.0148 and the 33,000 US gallon pressure vessel has a ratio of 0.0105. All three pressure vessels thickness ratios are about of an order of magnitude less than required to assume thin shell theory. Therefore, thin shell finite elements were used to mesh the pressure vessels.

For shell finite element S4 in ABAQUS, the number of integration points through thickness needs to be specified by the user. When the number of integration points through thickness was changed from 3 to 5, it made no difference to the final solution. ABAQUS documentation suggests that, for Gauss integration, 3 integration points be used through the thickness for conventional shell
elements. Therefore, the shell sections were defined to have 3 integration points through the thickness.

Checking mesh sensitivity is an important step in ensuring that an analysis is efficient and contains no significant mesh related errors. A p-refinement mesh convergence study was not possible because ABAQUS does not have a higher-order thin shell element with large strain capabilities. Therefore, an h-refinement mesh convergence study was conducted. The mesh in ABAQUS was generated using free meshing. Therefore, the sole parameter that will affect the mesh size will be the element size parameter. Decreasing the element size greatly increases run time since more elements will be needed to mesh the vessel. When the element size was increased 50% from 20mm to 30mm, for the 500 gallon pressure vessel, the predicted vessel failure time was only changed by 0.6%. Therefore, to keep run times low, an element size of 30mm was used. Note that a 40mm element size was attempted but the analysis became unstable since temperature gradients in elements became too large because not enough nodes were present to accurately map the thermal gradients.

When specifying a through-thickness temperature gradient in ABAQUS for shell elements, the user needs to decide on the number of temperature points through the thickness that will be used. Temperatures were defined through the shell thickness at all node locations in a piecewise fashion. This allowed for a through thickness temperature gradient to be established and the resulting loads from this gradient were incorporated into the analyses. Increasing the number of temperatures prescribed through the thickness increased the solution run time since more temperatures needed to be stored in memory and more calculations needed to be made. When the number of temperatures through the thickness was increased from 3 to 5, the solution did not change at all, however the solution run time increased by 22%. Therefore, 3 temperatures through the thickness were used.

The engineering drawings of the 500 and 1000 US gallon pressure vessels, Figures B-1 and B-2, indicate that the cylindrical part of the pressure vessel is made of SA 455 steel while the
hemispherical heads are made of SA 285-C. For this work it was assumed that the entire pressure vessel was made of SA 455 steel. Since failure was never predicted numerically or experimentally in the pressure vessel heads and since the materials have similar compositions, this assumption should not affect the results. The tensile strength of SA 285-C is 380-515 MPa and its minimum yield strength is 205 MPa (ASTM, 2007).

The engineering drawing of the 33,000 US gallon pressure vessel, Figure B-3, indicates that the pressure vessel is made of TC-128 steel. This work only measured SA 455 steel’s creep rupture data. Therefore, the 33,000 US gallon pressure vessel was modeled using SA 455 steel’s material properties.

The following subsections introduce the various pressure vessel dimensions and meshes.

7.1.1 500 US gallon Pressure Vessel

A detailed engineering drawing of the 500 US gallon pressure vessel is provided in Appendix B as Figure B-1. The pressure vessel was modeled as a shell structure with 5 faces, see Figure 7-1. Loads and boundary conditions are shown in Figures 7-2 and 7-3. The boundary conditions were set to mimic a pressure vessel resting on two stands; the pressure vessel was only fixed in the axial direction at one stand so as to prevent rigid body motion and to prevent the introduction of false axial stresses. The details of the shell mesh are shown in Figures 7-4 and 7-5.
Figure 7-2: 500 US gallon pressure vessel model with pressure loads shown as arrows.

Figure 7-3: 500 US gallon pressure vessel model, view of underside; boundary conditions on the left fixed all translational degrees of freedom and rotation about the 2 and 3 directions, boundary conditions on the right fixed translation in the 1 and 2 directions.
Figure 7-4: 500 US gallon pressure vessel mesh using 12,848 ABAQUS S4 finite shell elements.

Figure 7-5: 500 US gallon pressure vessel mesh, zoom in view of end detail.
7.1.2 1,000 US gallon Pressure Vessel

A detailed engineering drawing of the 1000 US gallon pressure vessel is provided in Appendix B as Figure B-2. It has the same dimensions as the 500 US gallon pressure vessel except that it is about twice as long. The pressure vessel was modeled as a shell structure with 3 faces, see Figure 7-6. Loads and boundary conditions are shown in Figure 7-7. Pressure loads on faces are represented by the pink arrows. The boundary conditions were set to mimic a pressure vessel resting on two stands; the pressure vessel was only fixed in the axial direction at one stand to prevent rigid body motion and to prevent the introduction of false axial stresses. The shell element mesh is shown in Figure 7-8. The detail of the mesh on the hemispherical head is similar to that shown in Figure 7-5.

Figure 7-6: 1000 US gallon pressure vessel model.
Figure 7-7: 1000 US gallon pressure vessel model, view of underside; boundary conditions on the left fixed all translational degrees of freedom and rotation about the 2 and 3 directions, boundary conditions on the right fixed translation in the 1 and 2 directions.

Figure 7-8: 1000 US gallon pressure vessel mesh using 20,048 ABAQUS finite shell elements.
7.1.3 33,000 US gallon Pressure Vessel

A detailed engineering drawing of the 33,000 US gallon pressure vessel is provided in Appendix B as Figure B-3. The 33,000 US gallon pressure vessel is manufactured with elliptical heads. However, for the sake of simplicity, in this analysis the heads were modeled as being hemispherical. This geometry change should not affect the results much since failure occurs far from the heads of the pressure vessels. The pressure vessel was modeled as a shell structure with 3 faces, see Figure 7-9. Loads and boundary conditions are shown in Figure 7-10. Pressure loads on faces are represented by the pink arrows. The boundary conditions were set to mimic a pressure vessel resting on two stands; the pressure vessel was only fixed in the axial direction at one stand to prevent rigid body motion and to prevent the introduction of false axial stresses. The shell element mesh is shown in Figure 7-11. The detail of the mesh on the hemispherical head is similar to that shown in Figure 7-5.

Figure 7-9: 33000 US gallon pressure vessel model.
Figure 7-10: 33000 US gallon pressure vessel model, view of underside; boundary conditions on the left fixed all translational degrees of freedom and rotation about the 2 and 3 directions, boundary conditions on the right fixed translation in the 1 and 2 directions.

Figure 7-11: 1000 US gallon pressure vessel mesh using 24,946 ABAQUS S4 finite shell elements.
7.2 Creep Damage Models

7.2.1 Kachanov One-State Variable Analyses

The Kachanov One-State Variable Technique was shown to accurately predict the uniaxial creep behaviour of SA 455 steel in Section 6. In order to use the technique to model the creep deformation of pressure vessels, the equations were modified, as discussed in Section 3.1.2, to capture the effects of multiaxial states of stress.

The multiaxial forms of the one-state variable equations are:

\[
\dot{\varepsilon}_{ij}^c = \frac{3}{2} A' \left( \frac{\bar{\sigma}}{1 - \omega} \right)^n S_{ij} t^m \\
\dot{\omega} = B' \left( \frac{\sigma_r^e}{(1 - \omega)^6} \right) t^m \\
\sigma_r = \alpha \sigma_1 + (1 - \alpha) \bar{\sigma}
\]

where, \( \dot{\varepsilon}_{ij}^c \) is the creep strain rate component, \( A', B', n, m \) and \( \Theta \) are material constants, \( \bar{\sigma} \) is the effective stress, \( \omega \) is creep damage, \( S_{ij} \) is the deviatoric stress component, \( t \) is time, \( \sigma_r \) is rupture stress, \( \sigma_1 \) is the maximum principal stress and \( \alpha \) is the multiaxial behaviour material constant.

In order to find the value of creep constant \( \alpha \), creep rupture experiments of notched tensile specimens need to be conducted as discussed by Hyde et al. (2006). These data were unavailable since no notched creep rupture specimens were tested. However, in the absence of these data, Equation 7-4 was used, as suggested by Lemaitre and Chaboche (1985).

\[
\sigma_r = \bar{\sigma} \left[ \frac{2}{3} (1 + \nu) + 3(1 - 2\nu) \left( \frac{\sigma_m}{\bar{\sigma}} \right)^2 \right]^{\frac{1}{2}}
\]

where, \( \nu \) is poisson’s ratio and \( \sigma_m \) is the hydrostatic stress.
These equations were implemented into ABAQUS via a creep user subroutine.

### 7.2.2 MPC Omega Analyses

The MPC Omega Method was shown to accurately predict the uniaxial creep behaviour of SA 455 steel in Section 6. In order to use the technique to model the creep deformation of pressure vessels, the equations were modified, as discussed in Section 3.1.2, to capture the effects of multiaxial states of stress.

The multiaxial forms of the MPC Omega equations are:

\[
\dot{\varepsilon}_ij^c = \frac{3}{2} \cdot \frac{S_{ij}}{\overline{\sigma}} \cdot \dot{\varepsilon}_0 \Omega_{tube} \varepsilon^c
\]  
\(\text{(Equation 7-5)}\)

\[
t_s = \frac{\varepsilon^c t_s \Omega_{tube}}{\varepsilon^c t_s \Omega_{tube} + 1}
\]  
\(\text{(Equation 7-6)}\)

where, \(\dot{\varepsilon}_ij^c\) is the creep strain rate component, \(\dot{\varepsilon}_0\) is the initial creep strain rate (material constant), \(\varepsilon^c\) is the creep strain, \(\Omega_{tube}\) is the material constant omega (see Equation 3-40), \(S_{ij}\) is the deviatoric stress component, \(\overline{\sigma}\) is effective stress, \(t_s\) is time in service, \(t_r\) is rupture time and \(\dot{\varepsilon}_0\) is the effective creep strain rate.

These equations were implemented into ABAQUS via a creep user subroutine.

### 7.3 Loading Conditions

From the literature review conducted in Section 2.3 it was shown that the thermal response of a pressure vessel exposed to an accidental fully engulfing or partially engulfing fire is well understood.
The main objective of this work was to develop a FEA model capable of numerically predicting the high temperature structural failure of a pressure vessel. Therefore, it was beyond the scope of this work to model the detailed heat transfer that occurs between the vessel contents, the vessel and the fire. Rather, the pressure vessel temperature distribution data experimentally gathered in the literature review was used to apply thermal loads at nodes as temperatures during a static structural analysis.

The relative reliability of the numerical model can be examined with experimental results and various loading conditions of interest can be investigated. First, to assess the general performance of the numerical analysis model, a time independent temperature and pressure distribution loading will be examined. Then, to investigate the affect of local hot spots on pressure vessel failure, time independent hot spots will be applied.

7.4 Time Independent Fully Engulfing Fire Loading

7.4.1 Description of Fully Engulfing Fire Loading

The loading discussed in this section was applied to 500, 1000 and 33000 US gallon pressure vessels.

There exists substantial variability in the exact temperature distribution of a pressure vessel exposed to a fully engulfing fire. Many factors such as wind, fire intensity and surroundings of the pressure vessel cause variations in temperature distribution. Variability in the internal pressure history exists since pressure is dependent on the heating distribution on the vessel and also it is controlled by a pressure relief valve (PRV). PRVs have been shown to have highly variable performance by Pierorazio and Birk (1998). Therefore, it was decided not to exactly model the complex loading history that a pressure vessel is exposed to during an accidental fire. Rather, the vessel loading was
simplified to consist of a time independent temperature distribution and internal pressure. This was achieved by ramping up the vessel temperature distribution and internal pressure at the beginning of the numerical analysis during a time independent load step. After that, temperatures, internal pressure and fill level were held constant during the remainder of the simulation.

Depending on the fill level of a pressure vessel, a certain percentage of the vessel volume will be occupied with liquid and the remainder will be occupied with vapour. The liquid, being more dense, will occupy the bottom portion of the vessel. In reality the fill level changes with time. Before the PRV opens the liquid level will rise due to liquid thermal expansion. The fill level will drop once the PRV is activated since vapour will leave the vessel as the lower internal pressure causes the liquid to boil. This complexity was removed from the present numerical analysis and the fill level was considered to stay constant at 50%. Constant fill levels of 30% and 70% were also analyzed and produced similar failure times to the 50% fill. This is because moving the large thermal gradient up and down between 70% and 30% fill does not change the stress state in the failure zone much.

Data from fire tests summarized in the literature review, Section 2.3, show that the liquid wetted wall stays below about 130°C during a fully engulfing fire, while temperatures of the vapour wetted wall will rise to anywhere between 550°C to 750°C depending on the intensity of the fire as well as other external factors. The liquid wetted wall remains much cooler because the liquid propane is much more efficient at removing heat from the vessel wall than the vapour is. This is because liquid propane has a much higher specific heat, thermal conductivity and density and also the boiling of liquid propane will absorb much heat.

When the PRV opens and vapour is vented, there will be boiling in the liquid and this will result in a frothing layer at the liquid-vapour interface consisting of a 2-phase liquid-vapour mixture as the liquid surface bubbles. This frothing layer acts to spread out the severe temperature gradient at
that location in the vessel wall. The frothing layer is also difficult to define precisely and has been simplified for this numerical analysis.

Figure 7-12 shows how various angles are defined and measured for this analysis in order to impose the discussed temperature distribution on the pressure vessels.

![Diagram showing angle measurement convention](image)

**Figure 7-12: Cross section representation of pressure vessel temperature distribution showing angle measurement convention.**

In this study, the size of the frothing region was defined by a constant angle \( \beta \), where \( \beta \) is the angle that spans the frothing region. A sensitivity study, see Figure 7-13, revealed that when \( \beta \) was increased from 10° to 30°, the predicted vessel failure time only increased by 4%. Therefore, using engineering judgment and from a desire to keep the analysis stable, \( \beta \) was set equal to 20°.
When pressure vessels are exposed to engulfing fires, they consistently fail at the location of peak wall temperature, somewhere in the vapour space wall region. In this study it was assumed that the peak wall temperature occurred on the top of the vessel. In order to achieve a peak wall temperature on the top of the vessel, temperature gradient, $\delta$, is defined. The physical interpretation of $\delta$ is that for every degree traveled from the top of the frothing region towards the top of the vessel, the temperature increases by the value of $\delta$. A sensitivity study, see Figure 7-14, on the value of $\delta$ revealed that increasing $\delta$ by a factor of 4 caused a 6% increase in vessel failure time. A $\delta$ value of 0.1 $^\circ$C/deg was found to be most appropriate since it predicted failure at the top of the vessel, yet was low enough to keep the temperature almost constant throughout the vapour wetted region of the vessel, behaviour typically observed in experimental pressure vessel fire tests.
In the finite element analysis model, temperatures were specified as temperature loads at
nodes via a piecewise function using the UTEMP user subroutine. The temperature in the vapour
wetted region was described by a linear function, see Equation 7-2. The temperature in the frothing
region was described by a cosine function that smooths out the large temperature gradient, see
Equation 7-3. The temperature in the liquid wetted region was constant, see Equation 7-4. The cross
sectional temperature distribution is plotted in Figure 7-15 and the 3D temperature distribution of the
vessel is shown in Figure 7-16.

\[
TEMP = TH - \theta \cdot \delta 
\]

\[\text{Figure 7-14: Sensitivity study of value of thermal gradient in the vapour wall space.}\]

\[\text{(Equation 7-2)}\]
\[ \text{TEMP} = \cos\left((\theta - \text{fill} + \beta)(180^\circ / \beta)\right) \left[ \frac{\text{TH} - \text{TC} - \left(\delta \ast (\text{fill} - \beta)\right)}{2} \right] + \frac{\text{TH} - \text{TC} - \left(\delta \ast (\text{fill} - \beta)\right)}{2} + \text{TC} \]  

(Equation 7-3)

\[ \text{TEMP} = \text{TC} \]  

(Equation 7-4)

\(\theta\) is the current angle

fill is the angle to the bottom of the frothing region (measured from top centre)

\(\beta\) is the angle of the frothing region (measured from bottom of frothing region)

\(\delta\) is the gradient of temperature rise in the vapour wetted wall.

\(\text{TH}\) is the maximum temperature in the vapour wetted wall.

\(\text{TC}\) is the temperature of the liquid wetted wall (130°C).

Also, during fire exposure, it is known that a through-thickness temperature gradient exists. This was analytically calculated to be about 14°C for the liquid wetted wall and 6°C for the vapour wetted wall and experimentally by Moodie et al. (1988). This through-thickness temperature gradient was incorporated into the finite element analyses.

Figure 7-15: Piecewise temperature distribution; theta is measured from top centre of vessel. \(\delta=0.1\), \(\beta=20^\circ\), \(\text{TH}=650^\circ\text{C}\), \(\text{TC}=130^\circ\text{C}\), \(\text{fill}=50\%\) or 90°.
Figure 7-16: Imposed vessel temperature distribution (°C) of 500 US gallon pressure vessel, with $\delta=0.1$, $\beta=20^\circ$, TH=650°C, TC=130°C, fill =50% or 90°.

7.4.2 Results and Discussion of the Fully Engulfing Fire Loadings

7.4.2.1 500 US gallon Pressure Vessel

The analyses were conducted with the temperature distribution discussed in Section 7.4.1, and parameter settings of $\delta=0.1$, $\beta=20^\circ$, TC=130°C and fill = 50% or 90°. The numerical analysis results of the 500 US gallon pressure vessels are summarized in Table 7-1 and Figure 7-17. It has been shown experimentally by Birk et al. (1994) that a pressure vessel in a fully engulfing fire will empty in about 15 minutes via the pressure relief valve. Therefore, the numerical analysis was focused on loading conditions that predict pressure vessel failure times of less than about 20 minutes. This is why an analysis was not conducted for a maximum temperature of 620°C and an internal pressure of 1.90MPa (275 psig). Pressure vessel failure, in all cases except one, was
determined to occur due to creep damage exhaustion; that is, creep damage values surpassed 0.99 at many locations along the top of the pressure vessels.

Table 7-1: MPC Omega numerically predicted failure times [min] of pressure vessels with respect to internal pressure and maximum temperature solved with ABAQUS.

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>P [MPa] (psig)</th>
<th>1.90MPa (275 psig)</th>
<th>2.07MPa (300 psig)</th>
<th>2.24MPa (325 psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td>680</td>
<td>4.4</td>
<td>2.6</td>
<td>1.1*</td>
<td></td>
</tr>
<tr>
<td>650</td>
<td>21.9</td>
<td>12.1</td>
<td>7.1</td>
<td></td>
</tr>
<tr>
<td>620</td>
<td>X</td>
<td>71.9</td>
<td>39.1</td>
<td></td>
</tr>
</tbody>
</table>

*Failure due to combined plastic and creep deformation.

Figure 7-17: 500 US gallon pressure vessel predicted failure times (MPC Omega, ABAQUS).

For the loading case of a maximum temperature of 680°C and an internal pressure of 2.24MPa (325 psig), there was plastic deformation in the failure zone. The combined plastic and creep failure was structurally unstable and thus computationally difficult to solve. This meant
that the predicted failure time for this loading condition was determined by ABAQUS’ inability to continue the analysis no matter how small the time step size.

All predicted failures, except $T=680^\circ C$ $p=2.24$MPa (325 psig), were similar to that shown in Figures 7-18, 7-19 and 7-20, occurring along an axial line on the top of the pressure vessel with no plastic deformation occurring in the failure zone. Therefore, these failures could be categorized as being mostly driven by stress rupture high-temperature creep. Plastic deformation only occurred in the vicinity of the frothing region. This was because the large temperature gradient through the frothing region produced large thermal stresses and thus plastic strains.

In the legend of the FEA contour plots, “fraction=0.775” indicates that the values at the integration points closest to the outer surface are being plotted. “SDV1” is creep damage, “CEEQ” is effective creep strain, “PEEQ” is effective plastic strain, “S” is stress and “NT13” is the nodal temperature on the outer surface of the pressure vessels.

Predicted MPC Omega creep damage and effective creep strain contour plots for all cases are shown in Appendix C.
Figure 7-18: Predicted MPC Omega creep damage at time of failure, $T=650^\circ C$ $p=2.07MPa$ (300 psig); solved with ABAQUS.

Figure 7-19: Predicted MPC Omega effective creep strain at time of failure, $T=650^\circ C$ $p=2.07MPa$ (300 psig); solved with ABAQUS.
Figure 7-20: Predicted MPC Omega effective plastic strain at time of failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.

The relatively fast failure times indicated that tertiary creep must be the dominant phenomena. Examination of Figure 7-21 confirmed that creep was indeed tertiary. Stress redistribution or relaxation is a much studied and widely known phenomenon that occurs with creep deformation (Hinton, 1992). Figure 7-22 shows how fast stress redistribution occurred with tertiary creep and how unstable it became with the onset of failure. In a fully engulfing fire, the stress in the failure zone quickly redistributed itself early on and then remained almost constant for the remainder of the vessel life until it dropped quickly with failure. Since the fire is fully engulfing, the wall temperature everywhere in the vapour space is near the peak wall temperature. This meant that a significant portion of the pressure vessel was softened and thus not much stress redistribution could occur during the course of the analysis since load equilibrium must be maintained. This is shown in Figure 7-22, as the stress level remained almost constant for most of the analysis. The extreme drop in stress near the end of the analysis showed that material on
the verge of failure will experience a vast reduction in its load bearing capacity. This phenomenon has been thoroughly studied and presented by other researchers as well (Becker, 2002).

![Figure 7-21: Time history of MPC Omega effective creep strain at point of eventual failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.](image1)

![Figure 7-22: Time history of effective stress at point of eventual failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.](image2)

The numerically predicted failure times are within expectations based on fire test results of Birk et al. (2006). Birk conducted partially engulfing fire tests of thermally protected 500 US gallon propane pressure vessels that were identical to those modeled in this study. In those tests
the tank wall temperature, pressure and fill level were changing with time and therefore direct comparisons cannot be made. However, they observed failure times in the range of 10 to 15 minutes of vessels averaging peak wall temperatures and internal pressures of about 650°C and 2.07MPa (300 psig). Also, from experimental fire test data, it is expected that for a maximum wall temperature of 650°C, the failure edge would be relatively thick and failure would be mostly due to high-temperature creep. Figure 7-23 shows the shape of a vessel failure that was fire tested with a 25% engulfing fire. The shape of the failure is very similar to that shown in Figure 7-19; although there is not as much vertical displacement predicted because the force of the exhausted fuel jet was not modeled. Therefore, it is believed that the results of this study are realistic since both failure time and failure mode were well predicted.

Figure 7-23: Failed fire tested 500 US gallon pressure vessel; 25% engulfing fire, peak wall temperature = 740°C, peak internal pressure = 2.59MPa (375 psig).
The same loading conditions summarized in Table 7-1 were also simulated using the Kachanov One-State Variable Technique. The One-State Variable Technique proved to be more unstable than the MPC Omega method for this application. This instability was expected as discussed in Section 6.2 due to the quick increase in creep damage that occurred near failure. These analyses consistently ended with convergence errors before creep damage values were at a critical level. For this reason not all loading cases were solved using the One-State Variable Technique. Table 7-2 summarizes the results found using the One-State Variable Technique. Failure for the One-State Variable Technique is predicted when the analyses became numerically unstable.

**Table 7-2: Kachanov One-State Variable Technique numerically predicted failure times [min] of pressure vessels with respect to internal pressure and maximum temperature solved with ABAQUS.**

<table>
<thead>
<tr>
<th>T [°C]</th>
<th>P [MPa] (psig)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2.07 MPa (300 psig)</td>
</tr>
<tr>
<td>650</td>
<td>10.8</td>
</tr>
<tr>
<td>620</td>
<td>X</td>
</tr>
</tbody>
</table>

The Contour plots of creep damage, effective creep strain and effective plastic strain at time of failure, found using the One-State Variable Technique, are plotted in Figures 7-24, 7-25 and 7-26. From Figure 7-17, it is seen that for the T=650, p=2.07MPa (300 psig) loading, the analysis becomes unstable when maximum creep damage is about 28%. Note that since the creep damage for the One-State Variable Technique does not increase linearly with time this does not mean that about 72% of the life time remains. From examining Figure 6-17, it is clear that for uniaxial creep loading, a Kachanov One-State Variable creep damage value of about 28% indicates that only about 10-15% of the component lifetime remains. Therefore, this shows that
the One-State Variable technique becomes unstable in these analyses when about 10-15% of pressure vessel life remains.

Figure 7-24: Predicted One-State Variable Technique creep damage at time of failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.

Figure 7-25: Predicted One-State Variable Technique effective creep strain at time of failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.
7.4.2.2 1,000 US gallon Pressure Vessel

The 1000 US gallon pressure vessel was analyzed to examine the effects of vessel dimensions on failure time. The 1000 US gallon pressure vessel was identical to the 500 US gallon pressure vessel except that it is about twice as long. The 500 US gallon pressure vessel has a length to diameter ratio of 3.26 while the 1000 US gallon pressure vessel has a length to diameter ratio of 5.53. The 1000 US gallon pressure vessel was solved exclusively with MPC Omega creep damage method since it was shown to be more robust than the One-State Variable Technique in Section 7.4.2.1. For a loading condition in which the maximum temperature was 650°C and the internal pressure was 2.07MPa (300 psig), the 1000 US gallon pressure vessel failed in 10.2 minutes, this is 1.9 minutes sooner then the 500 US gallon pressure vessel failed under the exact same loading conditions, see Table 7-3. The shorter 500 US gallon pressure vessel takes longer to fail then the longer 1000 US gallon pressure vessel because of end effects.
Since the 1000 US gallon pressure vessel is longer it is easier to bend and thus a higher state of stress is developed in the failure zone. This higher state of stress in the failure zone will cause quicker failures. Therefore, it was shown that pressure vessels with a lower length to diameter ratio are inherently safer when exposed to accidental fires.

The creep damage and creep strain contour plots are shown in Figures 7-27 and 7-28. The 1000 US gallon pressure vessel’s failure zone did not contain any plastic strain, see Figure 7-29. Therefore, the failure was driven by creep deformation and can be classified as a stress rupture high-temperature creep failure.

Figure 7-27: 1000 US gallon vessel predicted MPC Omega creep damage at time of failure, T=650°C p=2.07MPa (300 psig); solved with ABAQUS.
Figure 7-28: 1000 US gallon vessel predicted MPC Omega effective creep strain at time of failure, T=650°C  p=2.07MPa (300 psig); solved with ABAQUS.

Figure 7-29: 1000 US gallon vessel predicted MPC Omega effective plastic strain at time of failure, T=650°C  p=2.07MPa (300 psig); solved with ABAQUS.
Table 7-3: Comparison between 500 and 1000 US gallon analysis results; peak wall temp = 650°C, p=2.07MPa (300 psig).

<table>
<thead>
<tr>
<th>Pressure Vessel Size [US gallon]</th>
<th>Length / Diameter</th>
<th>Thickness / Diameter</th>
<th>Fail Time [min]</th>
<th>Length of Failure Zone [mm]</th>
<th>Length of Failure Zone / Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>500</td>
<td>3.26</td>
<td>0.0074</td>
<td>12.1</td>
<td>450</td>
<td>0.472</td>
</tr>
<tr>
<td>1000</td>
<td>5.53</td>
<td>0.0074</td>
<td>10.2</td>
<td>540</td>
<td>0.566</td>
</tr>
</tbody>
</table>

Table 7-3 shows that the length of the failure zone increased by 20% when the length of the pressure vessel was increased by about a factor of 2. The length of the failure zone is important since it is often used to examine whether or not a BLEVE occurred. Investigating if a BLEVE failure or jet release failure occurred is beyond the scope of this work since the internal fluid was not modeled. Predicting structural failure of the vessels was the current objective. However, the failure length data could be useful to other researchers in the field.

7.4.2.3 33,000 US gallon Pressure Vessel

The 33000 US gallon pressure vessel was analyzed to examine the effects of vessel dimensions on failure time. The 33000 US gallon pressure vessel has the same length to diameter ratio as the 1000 US gallon pressure vessel; however, it is about 3.5 times larger with respect to some of the major dimensions. Also, the 33000 US gallon pressure vessel has a smaller thickness to diameter ratio than the 500 and 1000 US gallon pressure vessels. The 33000 US gallon pressure vessel was solved exclusively with MPC Omega creep damage method since it was shown to be more robust than the One-State Variable Technique in Section 7.4.2.1. For a loading condition in which the maximum temperature was 620°C and the internal pressure was 2.07MPa (300 psig), the 33000 US gallon pressure vessel failed in 4.85 minutes, this is 67.1 minutes faster then the 500 US gallon pressure vessel failed under the exact same loading conditions. This huge
difference in failure time can be explained by two key geometrical differences, the larger length to diameter ratio and the smaller thickness to diameter ratio. As shown by the results of the 1000 US gallon pressure vessel, a longer pressure vessel will fail quicker than a shorter one with the same diameter. The thickness to diameter ratio of the 500 and 1000 US gallon pressure vessels was 0.0074, compared to 0.0052 for the 33000 US gallon pressure vessel. This 30% reduction in thickness to diameter ratio creates a 30% increase in vessel wall stress and thus has a great effect on the fire survivability of a pressure vessel exposed to a fully engulfing fire. The creep damage and creep strain contour plots are shown in Figures 7-30 and 7-31. The 33000 US gallon pressure vessel’s failure zone contained plastic strain, see Figure 7-32. Therefore, the failure was driven by both creep and plastic deformation and can be classified as a combination of high-temperature creep and short-term overheating stress rupture.

![Figure 7-30: 33000 US gallon vessel predicted MPC Omega creep damage at time of failure, T=620°C p=2.07MPa (300 psig); solved with ABAQUS.](image)
Figure 7-31: 33000 US gallon vessel predicted MPC Omega effective creep strain at time of failure, $T=620^\circ C$ $p=2.07$MPa (300 psig); solved with ABAQUS.

Figure 7-32: 33000 US gallon vessel predicted MPC Omega effective plastic strain at time of failure, $T=620^\circ C$ $p=2.07$MPa (300 psig); solved with ABAQUS.
Table 7-4: Comparison between 500 and 33000 US gallon analysis results; peak wall temp = 620°C, p=2.07MPa (300 psig).

<table>
<thead>
<tr>
<th>Pressure Vessel Size [US gallon]</th>
<th>Length / Diameter</th>
<th>Thickness / Diameter</th>
<th>Fail Time [min]</th>
<th>Length of Failure Zone [mm]</th>
<th>Length of Failure Zone / Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>500</td>
<td>3.26</td>
<td>0.0074</td>
<td>67.1</td>
<td>360</td>
<td>0.377</td>
</tr>
<tr>
<td>33000</td>
<td>5.53</td>
<td>0.0052</td>
<td>4.85</td>
<td>3500</td>
<td>1.148</td>
</tr>
</tbody>
</table>

Therefore, the length of the failure zone normalized by the diameter of the pressure vessel increased by 205% when the length of the pressure vessel normalized by the diameter of the vessel was increased by about a factor of 2 and the thickness of the vessel normalized by the diameter of the vessel was decreased by 30%. The length of the failure zone is important since it is often used to examine whether or not a BLEVE occurred. Thus it was concluded from Table 7-3 and 7-4 that the length of the failure zone was more sensitive to a change in vessel thickness than it was to a change in vessel length.

7.5 Time Independent Hot Spots

It is worth examining the structural behaviour of pressure vessels exposed to local hot spots since not all accidental fire impingements are fully engulfing. For example, a nearby pipe could rupture and cause a jet fire to come in contact with a pressure vessel which would lead to a localized hot spot. Another example is that a thermally protected pressure vessel in a fully engulfing fire could have a defect in its thermal protection, which would also lead to a localized hot spot. The analyses were conducted as time independent because the heat up time will vary on many external factors such as fire intensity and wind. Therefore, the analyses considered steady state thermal conditions and predicted failure time as the time needed to achieve failure once
steady state conditions were reached. This is considered a valid assumption of loading since minimal creep or plastic deformation will occur during heat up due to low wall temperatures.

The hot spot analyses were only conducted on 500 US gallon pressure vessels because the bulk of the local fire impingement experimental fire testing done by Birk et al. was on 500 US gallon pressure vessels.

7.5.1 Description of Hot Spot Loading

The hot spot was placed on the top dead centre of the pressure vessel. The shape of the hot spot is a circle projected onto the top dead centre of the pressure vessel. This most closely simulated the case of a jet fire impingement. The size of the hot spot was increased until failure of the pressure vessel was observed within the time window of this work, about 20 minutes. It was found that a hot spot of radius 100mm generated such failure times at wall temperatures known to be achieved by a pressure vessel exposed to fire impingement, about 700°C.

The MPC Omega method was used to compute creep strain and creep damage for the hot spot loadings. The MPC Omega method was shown to be more robust than the One-State Variable technique when analyzing pressure vessel stress rupture in Section 7.4.2.

In order to apply the thermal loading as temperatures at nodes, the pressure vessel face needed to be partitioned into various areas, both for applying temperature loads and for ease of meshing, see Figures 7-33, 7-34 and 7-35. The partitioning of faces and meshing of the pressure vessel varied based on the size of the hot spot. The partitioned faces and meshes shown in Figures 7-33, 7-34 and 7-35 are for a 100mm radius hot spot. The partitioned faces and meshes for the 50mm and 75mm hot spots are not shown because they are simply smaller versions of what is shown in Figures 7-33, 7-34 and 7-35.
Figure 7-33: 500 US gallon pressure vessel part model for hot spot analysis (100mm radius).

Figure 7-34: 500 US gallon pressure vessel mesh for hot spot analysis (100mm radius) using 14,058 ABAQUS shell finite elements.
The temperature distributions were imposed based on experimentally measured pressure vessel hot spots (Birk et al., 2003). The experimentally measured hot spots did not produce pressure vessel failure. This was because the test was not meant to analyze pressure vessel failure but rather to map the temperature distribution that a local hot spot induced in the pressure vessel wall. Therefore, the measured temperature distribution, see Figure 7-36, was scaled up in order to impose temperatures high enough to cause failure.
Figure 7-36: Experimentally measured temperature distribution of a pressure vessel with a 7.6 cm defect (Birk and VanderSteen, 2003).

7.5.2 Results and Discussion of Time Independent Hot Spot Loading

A summary of the hot spot analysis results is presented in Table 7-5. Results of the hot spot analysis with a 100 mm radius and 720°C maximum wall temperature are shown in Figures 7-37 through 7-46. The results of the other hot spot analyses are presented in Appendix D.

Table 7-5: 500 US gallon pressure vessel hot spot predicted failure times [min].

<table>
<thead>
<tr>
<th>Hot spot radius [mm]</th>
<th>680</th>
<th>700</th>
<th>720</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>*</td>
<td>*</td>
<td>*</td>
</tr>
<tr>
<td>75</td>
<td>*</td>
<td>58.2</td>
<td>24.1</td>
</tr>
<tr>
<td>100</td>
<td>55.6</td>
<td>20.4</td>
<td>9.7</td>
</tr>
</tbody>
</table>

*Failure not observed in a 100 minute analysis.
Note that the hot spot radius referred to in these analyses refers to the radius of the inner, smallest circle, which contains the peak wall temperatures. However, Birk and VanderSteen (2003) found it more convenient to deal with the size of the defect in the thermal insulation during their experimental work. Therefore, to introduce the thermal defect size and bridge the gap between the numerical and experimental work terminology, Table 7-6 was created.

Table 7-6: Relationship between hot spot radius and thermal defect radius.

<table>
<thead>
<tr>
<th>Hot Spot Radius [mm]</th>
<th>Thermal Defect Radius [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>50</td>
<td>125</td>
</tr>
<tr>
<td>75</td>
<td>187.5</td>
</tr>
<tr>
<td>100</td>
<td>250</td>
</tr>
</tbody>
</table>

It was apparent from Table 7-6 that the thermal defect radius was always 2.5 times greater than the hot spot radius. This was the approximate trend observed experimentally by Birk and VanderSteen (2003) and thus it was used to relate the values.

Figure 7-37: Comparison between fully engulfing fire and hot spot failure times.
Figure 7-38: 500 US gallon pressure vessel, imposed temperature distribution [°C] for hot spot analysis (100mm radius); maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Note that since this is a hot spot analysis, the temperature of the liquid wetted wall was 50°C. This is lower than the 130°C liquid wetted wall temperature used in the fully engulfing fire analyses. The wall temperature was lower since the hot spot was applied on the vapour wetted region of the vessel, so there was no direct contact between the fire and the liquid wetted wall. The value of 50°C was obtained from experimental observations made by Birk throughout his pressure vessel fire testing.
Figure 7-39: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of MPC Omega creep damage at time of failure; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure 7-40: 500 US gallon pressure vessel hot spot analysis (100mm radius), time history of MPC Omega creep damage at location of eventual failure; red plot is inner surface, green plot is outer surface; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
All the calculated creep damage occurred near the hot spot. This was because creep damage can only occur in regions where creep deformation occurs. For SA 455 steel, creep deformation was determined as being significant at temperatures above 550°C. Therefore, the only regions that could contain creep damage and creep deformation were regions in which the temperature was above 550°C.

The creep damage plot in Figure 7-40 shows that for hot spot analyses the creep damage rate in the failure region decelerated throughout the analysis. This deceleration of creep damage was caused by the large and continuous stress redistribution that occurred during the hot spot analysis. This stress redistribution acted to redistribute load away from the hot spot region to stiffer, cooler, regions, Figure 7-47.

Figure 7-41: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of MPC Omega effective creep strain at time of failure; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure 7-42: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of MPC Omega effective creep strain at time of failure; zoomed in on failure zone; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure 7-43: 500 US gallon pressure vessel hot spot analysis (100mm radius), time history of MPC Omega effective creep strain at location of eventual failure; red plot is inner surface, green plot is outer surface; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
From examination of Figure 7-43, it was clear, as expected, that creep in the failure zone was predominantly characterized by tertiary creep.

Figure 7-44: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of effective plastic strain at time of failure; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

As shown in Figure 7-44, pressure vessels exposed to local fire impingements experienced plastic strains in the failure region. Therefore, failure was characterized by a combination of creep and plastic deformation.
Figure 7-45: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of hoop stress at time of failure; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure 7-46: 500 US gallon pressure vessel hot spot analysis (100mm radius), contour plot of hoop stress at time of failure; zoomed in on failure region; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Due to creep deformation induced stress redistribution, the stress in the failure region was continuously lowered as the failure region deformed.

**Figure 7-47:** 500 US Gallon pressure vessel hot spot analysis (100mm radius), time history of von Mises stress at location of eventual failure; red plot is inner surface, green plot is outer surface; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Direct comparisons with experimental fire tests were not possible because the analyzed local fire impingements were of different size and shape than those fire tested. Also, during the fire tests, there was a heat up time as well as fluctuations in temperature and pressure due to variations in wind and fire intensity which made a direct comparison not possible. However, the failure times were in the range expected as experimentally tested pressure vessels with 8% and 16% thermal defects failed in the range of 22 – 40 minutes (Birk et al., 2006) with peak wall temperatures comparable to values used in the analyses. The shape of an experimentally measured failure region is shown in Figure 7-48. Note that the failure region was smaller than those measured experimentally for fully engulfing fires (Figure 7-23). This same trend was
observed numerically. The failure region generated by a local fire impingement is smaller than that generated by a fully engulfing fire since less material is severely heated.

Figure 7-48: Experimentally observed rupture of 500 US gallon pressure vessel with small insulation defect (8% of tank surface); Peak wall temp=700°C, peak internal pressure = 2.59MPa (375 psig).
Chapter 8  Conclusions and Recommendations

A numerical study was performed to examine the structural failure of pressure vessels exposed to accidental fires. The MPC Omega method and the Kachanov One-State Variable technique were both employed to model SA 455 steel’s creep rupture behaviour. Creep rupture is usually analyzed on the time scale of hundreds or thousands of hours; therefore it was not clear which creep damage model would best model the creep rupture of SA 455 steel occurring on the scale of minutes. From the literature review, two techniques were chosen with the hope that either one or both of them would be able to numerically predict the relatively fast creep ruptures of pressure vessels. Also, since this analysis involved non-linear FEA, both ANSYS and ABAQUS were employed to ensure that a solution could be reached. The pressure vessel analyses modeled 500, 1000 and 33000 US gallon pressure vessels in fully engulfing fire conditions and a 500 US gallon pressure vessel in various local fire impingement conditions.

8.1 Conclusions

The current work developed numerical analyses that modeled the behaviour of pressure vessels exposed to accidental fires. From the current work, the following conclusions were made.

8.1.1 Creep Damage Methods

- Both the MPC Omega method and the Kachanov One-State Variable technique were shown to accurately predict the uniaxial creep failure times observed in the creep rupture experiments.
• The MPC Omega method was more accurate than the Kachanov One-State Variable technique at predicting the uniaxial creep failure strains observed in the creep rupture experiments.

• The MPC Omega method was more numerically stable than the Kachanov One-State Variable technique for analyzing the relatively fast creep rupture of SA 455 steel.

• Due to its numerical robustness, the MPC Omega method was used in all the various pressure vessel numerical analyses while the One-State Variable technique was only used in some of the analyses.

8.1.2 Commercial Finite Element Analysis Computer Codes

• Both ANSYS and ABAQUS performed adequately when analyzing the uniaxial tensile specimens to reproduce the experimental creep rupture results.

• When modeling the pressure vessels, ABAQUS clearly outperformed ANSYS. Numerous convergence issues were encountered with ANSYS, and in the end all the pressure vessel analyses were conducted with ABAQUS.

8.1.3 Fully Engulfing Fire

• Pressure vessels with a lower length to diameter ratio take longer to fail in a fully engulfing fire because they benefit from end effects which make it more difficult to bend a shorter pressure vessel.

• Pressure vessels with a higher thickness to diameter ratio are much safer in a fully engulfing fire because they have lower wall stresses.

• Pressure vessels with a higher length to diameter ratio are more likely to experience a BLEVE in a fully engulfing fire since their failure zone was predicted to be larger.
Pressure vessels with a lower thickness to diameter ratio are much more likely to experience a BLEVE in a fully engulfing fire since their failure zone was predicted to be much larger. These differences in potential BLEVE failure occurrence were predicted based on larger stresses that occurred in the failure zone due to the different vessel dimensions.

- A greater fire intensity will generate greater peak wall temperatures and internal pressures and will produce quicker pressure vessel failures. This is because the vessel wall material weakens with temperature increase and, hence, will creep faster. Also, the stresses in the vessel wall will increase with increasing pressure.

- Stress redistribution occurred only briefly at the beginning of the analyses and then stress levels in the area of peak wall temperatures remained almost constant for the duration of the analyses. This occurred because the entire vapour wetted region was at high temperatures and thus there was not much stress redistribution that could occur since load equilibrium must be maintained. Therefore, a fully engulfing fire makes the pressure vessel more structurally unstable than a local hot spot.

**8.1.4 Local Fire Impingement (Hot Spots)**

- The greater the size of the fire impingement, the quicker the pressure vessel will fail.

- Stresses in the area of the local fire impingement can be reduced drastically because there exists a large amount of cooler pressure vessel wall material that will carry load and maintain load equilibrium. This means that peak wall temperatures in hot spots can be larger than those in fully engulfing fire cases and still have longer failure times.
• Failure regions generated by local fire impingement are smaller than those generated by a fully engulfing fire. Therefore, a fully engulfing fire is more likely to cause a pressure vessel to BLEVE.

8.2 Recommendations

• To further simplify the process of separating high temperature creep and plastic deformation, creep rupture experiments should be performed with constant true stress loading instead of constant load or constant engineering stress loading. This would involve setting up a feedback loop that would decrease the load based on specimen deflection.

• To improve the accuracy of the creep and creep damage models, a high temperature extensometer should be used to measure specimen creep deflection in the lab instead of using the Instron’s linear variable differential transformer (LVDT).

• The mechanical testing could be done with round tensile specimens to minimize the stress concentration effects of edge imperfections.

• A full transient heat transfer analysis can be coupled with the structural analysis to fully model all of the occurring heat transfer phenomena as well as the changing fill level of the pressure vessel. This would be another project on its own to implement into the FEA but it would make the analysis more complete, as the resulting analysis would not rely on so many assumptions to be made about the loading conditions of the pressure vessel.
References


API Recommended Practice 579 “Fitness For Service”, American Petroleum Institute, Jan 2000.


ASME, ASME Pressure Vessel Code, Section VIII, Division I, Pressure Vessels, 2001, American Society of Mechanical Engineers.


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Appendix A
SA 455 Steel Measured Creep Curves
Figure A-1: SA 455 steel creep curve; $T=550^\circ$C.

Figure A-2: SA 455 steel creep curve; $T=600^\circ$C.
Figure A-3: SA 455 steel creep curve; T=630°C.

Figure A-4: SA 455 steel creep curve; T=660°C.
Figure A-5: SA 455 steel creep curve; T=690°C.

Figure A-6: SA 455 steel creep curve; T=720°C.
Appendix B

Detailed Drawings of 500, 1,000 and 33,000 US gallon Pressure Vessels
Figure B-1: Detailed engineering drawing of 500 US gallon pressure vessel.
Figure B-2: Detailed engineering drawing of 1,000 US gallon pressure vessel.
Figure B-3: Detailed engineering drawing of 33,000 US gallon pressure vessel.
Appendix C

Results of Numerical Analysis for Time-Independent Temperature and Pressure Loadings
Figure C-1: Predicted MPC Omega creep damage at time of failure, T=620°C p=2.07MPa (300 psig); solved with ABAQUS.

Figure C-2: Predicted MPC Omega effective creep strain at time of failure, T=620°C p=2.07MPa (300 psig); solved with ABAQUS.
Figure C-3: Predicted MPC Omega creep damage at time of failure, T=620°C p=2.24MPa (325 psig); solved with ABAQUS.

Figure C-4: Predicted MPC Omega effective creep strain at time of failure, T=620°C p=2.24 MPa (325 psig); solved with ABAQUS.
Figure C-5: Predicted MPC Omega creep damage at time of failure, T=650°C p=1.90MPa (275 psig); solved with ABAQUS.

Figure C-6: Predicted MPC Omega effective creep strain at time of failure, T=650°C p=1.90MPa (275 psig); solved with ABAQUS.
Figure C-7: Predicted MPC Omega creep damage at time of failure, $T=650^\circ C \ p=2.24 \text{MPa}$ (325 psig); solved with ABAQUS.

Figure C-8: Predicted MPC Omega effective creep strain at time of failure, $T=650^\circ C \ p=2.24 \text{MPa}$ (325 psig); solved with ABAQUS.
Figure C-9: Predicted MPC Omega creep damage at time of failure, $T=680^\circ C$ $p=1.90 MPa$ (275 psig); solved with ABAQUS.

Figure C-10: Predicted MPC Omega effective creep strain at time of failure, $T=680^\circ C$ $p=1.90 MPa$ (275 psig); solved with ABAQUS.
Figure C-11: Predicted MPC Omega creep damage at time of failure, T=680°C \( p=2.07\) MPa (300 psig); solved with ABAQUS.

Figure C-12: Predicted MPC Omega effective creep strain at time of failure, T=680°C \( p=2.07\) MPa (300 psig); solved with ABAQUS.
Figure C-13: Predicted MPC Omega creep damage at time of failure, $T=680^\circ C$, $p=2.24$MPa (325 psig); solved with ABAQUS.

Figure C-14: Predicted MPC Omega effective creep strain at time of failure, $T=680^\circ C$, $p=2.24$MPa (325 psig); solved with ABAQUS.
Figure C-15: Predicted effective plastic strain at time of failure, T=680°C, p=2.24MPa (325 psig); solved with ABAQUS.
Appendix D

Results of Numerical Analysis for Time-Independent Hot Spot Loadings
Figure D-1: 500 US Gallon pressure vessel, imposed temperature [°C] distribution for a 100 mm hot spot analysis; maximum temperature of 680°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-2: 500 US Gallon pressure vessel, contour plot of MPC Omega creep damage at time of failure; 100mm radius hot spot, maximum temperature of 680°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-3: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 100mm radius hot spot, maximum temperature of 680°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-4: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 100mm radius hot spot, maximum temperature of 680°C, pressure of 2.07MPa (300 psig), solved using ABAQUS; zoomed in view of failure region. Scale same as D-3.
Figure D-5: 500 US Gallon pressure vessel, contour plot of effective plastic strain at time of failure; 100mm radius hot spot, maximum temperature of 680°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-6: 500 US Gallon pressure vessel, imposed temperature [°C] distribution for a 100mm hot spot analysis; maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-7: 500 US Gallon pressure vessel, contour plot of MPC Omega creep damage at time of failure; 100mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-8: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 100mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-9: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 100mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS; zoomed in view of failure region.

Figure D-10: 500 US Gallon pressure vessel, contour plot of effective plastic strain at time of failure; 100mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-11: 500 US Gallon pressure vessel, imposed temperature [°C] distribution for a 75mm hot spot analysis; maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-12: 500 US Gallon pressure vessel, contour plot of MPC Omega creep damage at time of failure; 75mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-13: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 75mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-14: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 75mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS; zoomed in view of failure region.
Figure D-15: 500 US Gallon pressure vessel, contour plot of effective plastic strain at time of failure; 75mm radius hot spot, maximum temperature of 700°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-16: 500 US Gallon pressure vessel, imposed temperature [°C] distribution for a 75mm hot spot analysis; maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-17: 500 US Gallon pressure vessel, contour plot of MPC Omega creep damage at time of failure; 75mm radius hot spot, maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.

Figure D-18: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 75mm radius hot spot, maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.
Figure D-19: 500 US Gallon pressure vessel, contour plot of MPC Omega effective creep strain at time of failure; 75mm radius hot spot, maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS; zoomed in view of failure region.

Figure D-20: 500 US Gallon pressure vessel, contour plot of effective plastic strain at time of failure; 75mm radius hot spot, maximum temperature of 720°C, pressure of 2.07MPa (300 psig), solved using ABAQUS.