Carleton Supercritical Water Loop: Design-driven Structural Analysis

by

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Abstract

A Supercritical Water (SCW) loop can be considered a complex structure designed to operate under extreme thermal-mechanical loading. The operation of the loop involves pressurization and heating. Some of its components operate at supercritical pressure and are subjected to a high temperature gradient. Thermoelastoplastic stresses may be developed. From the design point of view the magnitude of the stresses is an important factor to be taken into consideration, as plastic deformations may lead to premature failure.

The primary objective of this thesis is to conceive a tool to assist the design phase of the Carleton Supercritical Water Loop. More specifically, this thesis focuses on developing the structural simulations of the loop through the initial pressurization and heating during a start-up process. Finite element analyses were carried out employing representative design transients to ensure that the SCW loop will have sufficient strength to operate safely under the required service condition.

The finite element analyses were performed using the VrSuite software from GOLDAK technologies. The software has been extensively used for structural analyses at Carleton University, and has been benchmarked against a broad range of test cases for which analytical and high-fidelity experimental data is available, including geometric configurations similar to the various components of the present SCW loop.

To my parents

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Nomenclature

E	modulus of elasticity of the pipe material (Pa)
k	thermal conductivity of the solid (W/m/K)
L	length of the pipe (m)
r	radius (m)
r_i	inside radius of the pipe (m)
r_o	outside radius of the pipe (m)
T	temperature function (K)
T_r	radius-dependent temperature (K)

Greek letters

σ_r	radial stress (Pa)
σ_v	effective stress (Pa)
σ_z	axial stress (Pa)
σ_θ	tangential stress (Pa)
α	thermal expansion coefficient (/°C)
ν	Poisson's ratio

Chapter 1: Introduction

Although nuclear power reactors generate 17 % of the total electricity around the world, they operate only in 31 countries [1]. As the world develops under growing concerns about energy resource availability and climate change, the nuclear power is expected to play an important role in the future energy supply.

Even though the current nuclear power plant designs are considered technically acceptable, and economically satisfactory, a new generation of nuclear reactors is being designed. The “Generation IV” of nuclear reactors, which are to be deployed commercially by 2030, is being designed to be advantageous in the areas of economics, safety, sustainability, proliferation resistance and reliability.

The CANDU-Supercritical Water Reactor (CANDU-SCWR) is one of the concepts considered as one of the Generation IV innovation nuclear reactors. The CANDU-SCWR uses water as coolant that exits the fuel channel at supercritical conditions (25 MPa and 625°C). The Supercritical water (SCW) heat transfer properties are still a topic under investigation. SCW loops are being designed and installed in order to address this issue. Carleton University has designed a facility to study uncertainties associated with SCW heat transfer properties [2].

The supercritical water loop to be housed at Carleton University can be considered a structure of unique design features. It has been designed to operate under extreme thermal-mechanical loading. As its components operate at supercritical conditions,

thermoelastoplastic stresses may be developed. From the design point of view the magnitude of the stresses is an important factor to be taken into consideration.

The rest of this chapter is organized as follows. In Section 1.1, the objectives of the research are presented. In Section 1.2, the structure of the rest of the thesis is outlined.

1.1 Research Objectives

The main objective of this thesis is to perform a design-driven analysis aimed to support the design phase of the Carleton SCW loop. More specifically, this thesis aims to accomplish the following:

- To present a comprehensive survey of stress analysis for high-pressure vessels with emphasis in thermal stresses.
- To perform thermal and stress simulations using the VrSuite software from GOLDAK technologies and to benchmark it against classical problems with exact analytical solutions.
- To import the time-dependent fluid temperature histories resulting from the transient thermal analysis of the loop as input to structural simulation.
- To evaluate the severity of thermal stresses for a small number of potential worst-case transients.
- To perform a structural simulation with displacement constraints that are compatible with the loop platforms mounted at several points along the length of the SCW loop.

- To evaluate the results of the simulations.
- To evaluate the overall structural behaviour and functionality of the SCW loop for the intended series of experiments to study the heat transfer characteristics of water at supercritical conditions.

1.2 Idealizations, Assumptions and Approximations

1. The initial state of the SCW loop was assumed to have zero residual stress. This means that residual stresses and distortion introduced by manufacturing, by welding, by bolting flanges, Grayloc connections and misfit in assembly were ignored.
2. The detailed geometry of more complex parts such as the pump, heat exchanger and valves was approximated.
3. The geometry of simpler parts, such as tee, elbows and pipes was assumed to be accurate but no variations in the geometry were assumed.
4. The published temperature dependent material properties were assumed to be accurate.
5. Fluctuations in pressure and vibrations were ignored.
6. The time dependent fluid flow analysis of fluid pressure, density temperature provided by Mason *et al* (2010) [3] was assumed to be accurate.
7. The sensitivity of the design parameters was not evaluated.

1.3 Thesis Outline

The rest of the thesis is organized as follows:

Chapter 2: Presents results of literature review.

Chapter 3: Presents the benchmarking of VrSuite. Two classical problems have been analyzed; a thick-walled pipe subjected to internal pressure and constrained to have zero axial elongation and an internally heated thick-walled pipe.

Chapter 4: Discusses the approach followed to obtain the temperature-versus-time information for all points along the length of the loop and use it as thermal constraints in the structural simulations.

Chapter 5: Presents the finite element structural analysis for five simulation scenarios for steady-state and transient conditions.

Chapter 6: Discusses the main results of this thesis. Thermal and stress simulations results are commented, emphasis is given to the results that most influence the predicted displacement, stress and strain.

Chapter 7: A summary of the work performed is discussed. An overview of the simulation results is presented. Possible ways to enhance the research presented in this thesis are suggested.

Chapter 2: Background

2.1 Supercritical Fluids and Heat Transfer Correlations.

Although water is the primary coolant under investigation for SCW reactors, the use of a simulant fluid such as CO₂ or refrigerants plays an important role for research purposes [4]. When a simulant fluid is used, the experimental results should be scaled to the equivalent property of the water.

The use of phase diagrams is a convenient mode to represent states and transitions of a single simple substance, such as water. The phase boundary between liquid and gas (blue line) shown in Figure 2-1 ends at point called critical point. This point represents the highest temperature and pressure at which vapour and liquid can coexist in equilibrium.

The P-V phase diagram illustrated in Figure 2-2 depicts how primary phases of water (liquid and vapour) change as pressure and specific volume are changed at constant temperature. For a decrease of volume at constant pressure T_1 , below the critical temperature, the vapour begins to condensate into liquid. As the pressure and temperature remain constant and volume decreases in the two phase region; more vapour is converted

into liquid until a point is reached where all is liquid. The critical point is located where the saturated liquid and saturated vapour curves meet.

Figure 2-1 Phase diagram of water in pressure-temperature space. [5]

Figure 2-2 P-V Diagram for water [6].

The significant variation of the thermodynamic properties is the special characteristic of the coolant near the critical point [7]. This is illustrated in Figure 2-3. Fluids at supercritical conditions behave similarly. For a given pressure there is a certain value for temperature, known as the pseudo-critical point for that pressure, at which the specific heat reaches its peak value.

Although considerable research studies have been carried out in the past to explore the heat transfer properties of SCW there are several issues still under investigation, Balouch *et al.* (2010) argue that there is no one definitive correlation to predict the complete heat transfer coefficient spectrum of SCW. Cheng *et al.* [8] performed one of the most extensive literature reviews on heat transfer in SCW. The authors claim that most of the empirical correlations have the general form of a modified Bittus-Boelter equation. Dyonisius *et al.* [9] refer to Pioro *et al.* for an updated review of heat transfer correlations.

Research topics like degradation of heat transfer [10], new heat transfer correlation for supercritical fluids [11], numerical simulation of fluid flow [4], [12], corrosion and water chemistry tests under irradiation [13], make SCW one of the most investigated objects of research in the nuclear industry nowadays.

Figure 2-3 Thermophysical property variations according to temperatures for supercritical pressure water at 25.0 MPa. [7]

2.2 SCW Loops

SCW loops are complex structures designed with the sole objective of investigating heat transfer properties of water under supercritical conditions. Balouch *et al* [2] surveyed SCW test facilities and summarized the most important features of the current designs. Although dissimilar models have been reported, in general terms the primary components of a SCW loop are the heat exchanger, the fluid-circulation pump and the test section. In order to acquire data, control the system and ensure safety, additional components are required.

There are some design requirements that determine the design of a SCW loop. The operating conditions is the most important one, as the test-sections are required to operate at supercritical temperatures (260-600°C) and pressures (23-27 MPa). The geometry of the test section is another factor to take into consideration. For the CANDU-SCWR cross-sections of the test section are shown in Figure 2-4.

Figure 2-4 Cross-sections of the CANDU-SCWR's test-section geometries [2].

2.3 Carleton SCW Loop

This section summarizes the most important features of the Carleton University SCW loop which is illustrated in Figure 2-5. Details of the design of the supercritical water loop and its components are discussed in Section 5.2 of the technical report [2].

Depending on the temperature, pipes are made of stainless steel 316 or 316H. Components that are not subjected to severe temperatures, are constructed from Type 316

and components that come into contact with the high temperature supercritical water are constructed of Type 316H.

The pipes' outer diameter and inner diameter are 48.26 mm (1.9 in) and 27.94 mm (1.1 in), respectively. Grayloc™ fittings or standard Class 2500 ASME flanges are used to connect the pipes and the components of the loop. The approximate dimensions of the loop are 2x3x5 m (width x length x height) for the test section positioned vertically. The loop design allows the test section to be positioned horizontally; in this case the length is 5.5 m.

A high-current DC power supply can transfer up to 300 kW of heat to the water in the 2.3 m long test section. A heat exchanger, which is rated to operate at 30 MPa and 630 °C, lowers the temperature of the water to 260 °C.

Figure 2-5 Layout and major components of the Carleton SCW loop [2].

Flow rates in the range of 0.4 to 1.6 kg/s are generated by a centrifugal pump that can operate at pressures as high as of 28 MPa. The water flow rate regulation is achieved by the pump itself, and two globe valves. A high-pressure bladder accumulator is used to pressurize the loop

2.4 Stresses in High-Pressure Vessels

Cylindrical pressure vessels are frequently used in industry for both liquid and gases flow under pressure [14]. The magnitudes of the applied pressure, as well as the radius of the cylinder and its shape determine the resulting normal stresses. For thick-walled pipes exact analytical solutions derived from elasticity are presented in Equations (1.1) through (1.4) of Section 3.3, these equations can be used for stress analysis in a thick wall pipe

subjected to internal pressure. Figure 2-6 shows the stress distributions through the wall thickness for the hoop and radial stresses.

When a pressure vessel is subjected to temperature gradient through its thickness expansion and contraction is produced, which develops thermal stresses. Equations (1.6) through (1.9) from Section 3.4.1 give the stresses for an unrestrained hollow cylinder with radially symmetric temperature [15]. Thick-walled high-pressure vessels are considered unique and due to their high metal content, preference is given to Finite Element Analysis (FEA) to study problems related to the strength calculation of such vessels [16].

Figure 2-6 Stress distributions of hoop and radial stresses [14].

2.5 Related Works

Considerable research works have been carried out in the past to explore the stresses in thick-walled pipes and high pressure vessels. The most relevant ones to this thesis are discussed as follows.

C. O. Smith (1960) [15] discussed the effect of temperature and temperature gradients in developing of thermal stresses and strains. Equations for stresses in circular cylinder subjected to a temperature gradient are presented in this paper.

D. Mohr *et al* (1982) [17] presented an integrated thermal-hydraulic-structural analyses conducted for intermediate heat exchanger (IHX). The results revealed the locations of the stress limiting areas. The FEA analyses evaluated the capability of the heat exchanger to withstand the thermal stress and accumulated fatigue damage. From the point of view of the author of this thesis, this paper can be considered as a classic paper for high-pressure vessels FEA.

Krishchuk (1984) [16] presented a practical application of the FEM to study the thermal and stress-strain state of thick-walled high-pressure vessels. The main advantages of the FEM for thermal stress analysis over other numerical and analytical methods are clearly stated.

High temperature gradients in high-pressure vessels may lead to thermoelastoplastic stresses. S. Jahanian (1996) [18] performed a nonlinear strain hardening thermoelastoplastic analysis for a thick-walled pipe. The fact that the properties of the

material are assumed to be temperature dependent is an important parameter which was considered in this numerical analysis.

Alzaharnah et al (2001) [19] studied the effect of a fully developed laminar flow on the thermal stresses in thick wall pipes. The results presented in this paper can be applied to heat exchanger, pipes and other components, which are parts of a SCW loop.

David (2010) [20] presented three practical examples of the use of Computational Fluid Dynamics (CFD) and FEA to solve difficult industry thermal stress problems. The resulting temperature histories are used as Dirichlet temperature constraints for the structural analysis.

Although the results presented by the above-mentioned researchers have been crucial for the development of this thesis, no reference could be found which presents a detailed FEA structural analysis for a SCW loop.

Chapter 3: FEA validation

3.1 Introduction

The water flowing inside a SCW loop is subjected to changes in temperature. Heat is added in the test section and then extracted in the heat exchanger. The high temperature gradient causes some parts to expand and some others to contract, leading to thermal stresses. As stated earlier, the operation of the SCW loop will involve pressurization followed by heating processes. Depending on the magnitude of the combination of the thermal stresses and stresses due to high pressure, the physical integrity of the entire structure could be at risk.

There is no an all-embracing analytical method to study the thermoelastoplastic stresses of different designs of high-pressure vessels currently available. However, numerical methods, particularly FEM, have numerous advantages for the analysis of thermal stress and stress-strain state of high-pressure vessel [16].

Any FEA validation aims to determine whether or not the actual behavior of an object is accurately described by the analysis results. In order to verify the accuracy of an FEA model it is typically benchmarked against analytical or high-fidelity experimental data.

The SCW loop finite element analyses were performed using the VrSuite software [21]. This software has been used extensively for several years for stress analyses at Carleton University. Two classical problems constitute the benchmark for VrSuite. First, the stress analysis of a thick-walled pipe subjected to internal pressure. Second, the thermal and stress analysis of a thick-walled pipe subjected to a temperature gradient.

3.2 Benchmarks Test Setup

Benchmark geometry, material and loading are chosen to correspond closely to those of the SCW loop. Tests were performed with the pipe segment set to 500 mm length. The inner and outer radius was set to 13.97 mm and 24.13 mm, respectively. The pipe material was selected as AISI 316 steel. The mechanical and thermal properties of the material at 300 K are shown in Table 3-1. Figure 3-1 shows the temperature-dependent properties taken from the material library of VrSuite.

Table 3-1 Mechanical and Thermal properties of the material at 300 K

Steel	Thermal Conductivity (k)	Thermal Expansion Coefficient (α)	Yield Strength at 300 K	Modulus of Elasticity (E)	Poisson's Ratio (ν)
AISI Type 316	16.2 W/m/K	15.9 $10^{-6}/^{\circ}\text{C}$	205 MPa	193 GPa	0.30

Figure 3-1 Temperature-dependent properties of AISI 316 steel.

The pipe was discretized using an 8-node brick mesh. For each benchmark two different refinements were carried out. The first refinement consists of a fine mesh, with 10 elements through the thickness and 25 elements along the length. The second refinement consists of a coarse mesh, with 2 elements through the thickness and 25 elements along the length. Table 3-2 shows characteristics of these meshes. The fine and the coarse mesh are presented in Figure 3-2 and Figure 3-3, respectively. Figure 3-4 illustrates the line along which temperature and stresses have been calculated. The analysis was run on one core of an Intel Quad Core Q9650 3.0 GHz processor.

The spatial discretization of the components of the SCW loop, to be presented later, was performed with a resolution that is similar to that of the coarse mesh.

Figure 3-2 Fine mesh with 10 elements through the thickness and 25 elements along the length.

Table 3-2 Mesh characteristics

Refinement	Elements along thickness	Total # Elements	Total # Nodes
1	10	6000	6864
2	2	1200	1872

Figure 3-3 Coarse mesh with 2 elements through the thickness and 25 elements along the length.

Figure 3-4 Line for temperature and stresses measurement.

3.3 Thick-Walled Pipe Subjected to Internal Pressure

The classical problem of a thick-walled pipe subjected to internal pressure and constrained to have zero axial elongation constituted the first benchmarks for VrSuite. The geometry of this test case is given in Figure 3-5. The pipe was pressurized to 25 MPa. Equations (1.1) through (1.4) are used to calculate the analytical solution for stresses. For the fine and coarse meshes the predicted stresses are compared to the analytical results in Figure 3-6 and Figure 3-7, respectively. Table 0-1 and Table 0-2 in Appendix B show a tabular comparison.

Figure 3-5 Thick-walled pipe under internal pressure.

$$\sigma_r = \frac{P_i r_i^2}{r_o^2 - r_i^2} \left[1 - \frac{r_o^2}{r^2} \right] \quad (1.1)$$

$$\sigma_\theta = \frac{P_i r_i^2}{r_o^2 - r_i^2} \left[1 + \frac{r_o^2}{r^2} \right] \quad (1.2)$$

$$\sigma_z = \frac{P_i r_i^2}{r_o^2 - r_i^2} \quad (1.3)$$

$$\sigma_v = [\sigma_\theta^2 + \sigma_r^2 + \sigma_z^2 - (\sigma_\theta \sigma_r + \sigma_\theta \sigma_z + \sigma_r \sigma_z)]^{\frac{1}{2}} \quad (1.4)$$

Figure 3-6 Comparison of numerical and analytic results for stresses (fine mesh).

Figure 3-7 Comparison of numerical and analytic results for stresses (coarse mesh).

Figure 3-8 illustrates the effective stresses through the thickness of a pipe subjected to an internal pressure of 25 MPa.

Figure 3-8 Effective stresses through the thickness of a pipe subjected to an internal pressure of 25 MPa.

Plasticity Due to Pressure Increase.

If the internal pressure of the pipe is increased to a magnitude sufficiently high, the inner layers of the metal might not be able to recover the initial shape once the pressure is released, that is the pipe have been deformed plastically. The resistance of the pipe to stress corrosion cracking could be improved depending on magnitude of the resulting

residual compressive stresses. Figure 3-9 shows the stress distribution in the pipe once the internal pressure is increased to 150 MPa.

Figure 3-9 Stress distribution of a pipe subjected to an internal pressure of 150 MPa.

3.4 Thermal and Stress Analysis of a Thick-Walled Pipe Subjected to Temperature Gradient

The main objective of this section is to create a benchmark example for thermal-stress analysis. Analytical and numerical solutions are found, as well as the comparison of

results. The effect of temperature gradient on the magnitude of thermal stresses is also presented.

3.4.1 FEA Validation for Thermal-Stress Analysis

A benchmark to evaluate the thermal stresses due to temperature gradient is analyzed analytically and numerically. In this section a uniformly heated pipe is considered. The inside and outside temperature of the pipe are prescribed to 400 K and 300 K respectively. The axial ends of the pipe are set to adiabatic condition. The comparison of temperature through thickness for analytical and numerical solution is shown in Figure 3-10. The geometry and temperature distribution of this test case are given in Figure 3-11.

In order to evaluate the effect of mesh density on the intended stress analysis of the SCW loop, the thermal-stress analysis of this benchmark was performed by using the same two meshes defined in the previous section, identified as fine and coarse mesh earlier.

The analytical result for the steady-state radial temperature distribution in the pipe wall is given by the following the equation (1.5). It was found after solving the corresponding heat equation for steady-state condition with no heat generation [2].

$$T_r = \frac{T_i - T_o}{\ln \frac{r_i}{r_o}} \ln \frac{r}{r_o} + T_o \quad (1.5)$$

Figure 3-10 Temperature through thickness for analytical and numerical solution.

Figure 3-11 Geometry and temperature distribution of the thick-walled pipe.

Table 3-3 Information about thermal stress simulation.

Simulation	Elements along thickness	Total # Elements	Total # Nodes	Analysis Time (sec)	CPU Time (sec)
1	10	6000	6864	3600	636
2	2	1200	1872	3600	62

Equations (1.6) through (1.9) are used to calculate the exact analytical solution for the thermal stresses. The predicted hoop, radial, axial and effective stresses for the fine and coarse mesh are compared to the analytical results in Figure 3-12 and Figure 3-13, respectively.

$$\sigma_{\theta} = \frac{E\alpha}{(1-\nu)r^2} \left[\frac{r^2 + r_i^2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} Trdr + \int_{r_i}^r Trdr - Tr^2 \right] \quad (1.6)$$

$$\sigma_r = \frac{E\alpha}{(1-\nu)r^2} \left[\frac{r^2 - r_i^2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} Trdr - \int_{r_i}^r Trdr \right] \quad (1.7)$$

$$\sigma_z = \frac{E\alpha}{1-\nu} \left[\frac{2}{r_o^2 - r_i^2} \int_{r_i}^{r_o} Trdr - T \right] \quad (1.8)$$

$$\sigma_v = [\sigma_{\theta}^2 + \sigma_r^2 + \sigma_z^2 - (\sigma_{\theta}\sigma_r + \sigma_{\theta}\sigma_z + \sigma_r\sigma_z)]^{\frac{1}{2}} \quad (1.9)$$

Figure 3-12 Thermal stresses for analytic and numerical solution (fine mesh).

Figure 3-13 Thermal stresses for analytic and numerical solution (coarse mesh).

Plasticity Caused Due to Thermal Stresses

The magnitude of thermal stresses depends on the temperature gradient through the thickness of the pipe. If the prescribed temperature of the inner surface of the pipe is increased to 800 K as the outer surface temperature is 300 K, the thermal stresses will be high enough to cause plasticity. Therefore, the distribution of the temperature is changed. Hoop and axial stresses are shifted towards the outer radius and the maximum value of

the effective stresses will be on the outer surface of the pipe. Figure 3-14 shows the stress distribution of a pipe with thermally induced plasticity.

Figure 3-14 Stress distribution of a pipe with thermally induced plasticity.

Transient Thermal Stresses.

Another simulation was run for a pipe subjected to a temperature transient. The pipe is analyzed for 300 kW of heat input in 3600 seconds with a linear ramp-up rate starting at 0 kW, and without a pressure loading.

The magnitude of the thermal stresses is usually lower for a pipe with variable temperature, basically due to a uniform thermal expansion. After an initial increase of the effective stresses caused by the thermal shock, a decrease is observed. For this test the thermal stresses become negligible at steady state. This can be seen clearly in Figure 3-15.

Figure 3-15 Minimum –Maximum time-dependent temperature and Minimum –Maximum time-dependent effective stresses for a thick-walled pipe under transient temperature.

Combined Effect of Pressure and Time-Dependent Temperature on the Stresses of a Pipe

In order to evaluate the combined effect of pressure and temperature transient on the stress distribution a last test was prepared. Finally, the pipe was analyzed for 300 kW of heat input in 3600 seconds with the pipe pressurized at 25 MPa. Figure 3-16 illustrates that after the initial thermal shock the effective stresses keep increasing until the steady-state is reached. The comparison of Figure 3-6 with Figure 3-17, and Figure 3-8 with Figure 3-18 confirms that for this test the effective stresses are mainly ruled by the high pressure inside the pipe.

The results of the stress analyses performed for the above-noted benchmark examples are qualitatively comparable to those of the research studies carried out in the past by others authors to explore the stresses distribution in a single straight pipe.

Figure 3-16 Minimum –Maximum time-dependent temperature and Minimum –Maximum time-dependent effective stresses for a thick-walled pipe under internal pressure and transient temperature.

Figure 3-17 Stresses at steady-state for a thick-walled pipe under internal pressure and time-dependent temperature.

Figure 3-18 Effective stresses through the thickness of a pipe subjected to an internal pressure of 25 MPa and time-dependent temperature.

Chapter 4: Thermal Simulation of the SCW loop

4.1 Introduction

The importance of a proper prediction of temperature field in a structure is said to be critical in a thermal stress analysis [20]. In order to keep the thermal stresses in a reliable range the heating and cooling rates should be studied. There are certain conditions where estimating fluid flow and temperatures cannot be performed by following standard handbook procedures. Numerical thermal simulation can be used to overcome this issue where standard heat transfer correlations are not relevant anymore and direct measurement of the temperature is difficult or simply not possible.

The time-dependent temperature histories resulting from the transient thermal analysis of the loop [2] are intended as input to the stress analysis. The temperatures versus time profiles at a total of 16 locations along the length of the loop are used as thermal constraints in the stress analysis. To obtain the temperature-versus-time information for all points along the length of the loop the traditional approach of finding the fitting equation with the lowest sum of squared absolute error was followed. These equations are used to interpolate by time and position.

The following thermal simulations have been performed:

1. The temperature of the test section is ramped-up linearly in 3600 seconds until steady state condition is reached. The thermal simulation was performed completely in VrSuite.
2. Three different scenarios are considered, in which 300 kW of heat are introduced in the test section in 300, 1800 and 3600 seconds. In this case the temperature of the test section was restricted to 900 K.
3. The power is ramped-up in the test section to the maximum operating conditions. 300 kW of heat are introduced in the test section in 300 seconds until the steady state condition is reached. No restriction is imposed to the maximum temperature in the test section in order to simulate a potential worst-case transient.

4.2 Thermal Simulation Setup

In order to determine the temperature distribution in the structure of the SCW loop a thermal analysis was performed. The loop was modeled in Pro/Engineer, and then a discretization of the domain was performed in Abaqus FEA, by using a combination of 8-noded hexahedral and 10-noded tetrahedral elements. The generated mesh was imported in VrSuite. All thermal simulations used the same mesh. The total number of elements is 130444, and the number of nodes is 211752. One core of an Intel Quad Core Q9650 3.0 GHz processor was used for the analysis. Figure 4-1 illustrates the spatial discretization of the components of the SCW loop.

Figure 4-1 Spatial discretization of the components of the SCW loop.

4.2.1 Thermal Simulation for Steady-State

In order to determine the temperature distribution along the length of the loop a first thermal simulation was performed in VrSuite until the steady-state is reached. The test section was ramped-up linearly in 3600 seconds. The loop was design to operate at 533 K at the inlet of the test section and at 873 K at the outlet of the test section for steady state conditions. The results of this simulation were used later for the stress analysis of the SCW loop at steady state for maximum operating conditions. Figure 4-2 shows the temperature distribution in the SCW loop at steady-state.

Figure 4-2 Temperature distribution in the SCW loop at steady-state.

4.2.2 Transient Thermal Simulation

This section describes the transient thermal simulation of the SCW loop. The time-dependent temperature field derived from a thermal analysis of the SCW loop, performed by another member of the Carleton University's SCW loop Group, is used as input to the transient thermal analysis performed in VrSuite.

The thermal simulation discussed in [3] was run assuming that outside of the test section the wall temperature is slightly lower than the fluid temperature, while the loop is still

ramping up. At steady state, due to the assumption that the loop is perfectly insulated, the wall temperature equals the fluid temperature.

Three different scenarios are considered, in which 300 kW of heat are introduced in the test section in 300, 1800 and 3600 seconds, respectively. In this case the temperature of the test section was restricted to 900 K. As a result the fluid temperature through the test section only just passed the critical point at steady state.

A fourth crucial scenario for the power being ramped up in the test section to the maximum operating conditions was also analysed. This scenario simulates the highest time-dependent temperatures at which the loop is intended to be operated.

The thermal simulation was run for the fluid temperature data corresponding to sixteen locations along the length of the SCW loop. Figure 4-3 presents the locations of the control points.

The graph presented in Figure 4-4 suggests that the time-dependent temperature of the water computed at the control point in the test section inlet and outlet can be used to represent the temperature change for the rest of the points in the loop. Therefore, assuming a perfectly insulated loop, the points located from the outlet of the test section to the inlet of the heat exchanger (Point 10 to Point 13) are subjected to the same temperature transient as the test section outlet (Point 9). In the same manner, the points located from the outlet of the heat exchanger to the inlet of the test section (Point 14 to Point 7) have the same time-dependent temperature as the test section inlet (Point 8). This

simplification was considered for those simulations where the temperature of the test section was restricted to 900 K.

Figure 4-3 Control Points.

Figure 4-4 Temperature vs. Time for 300 kW applied in 3600 seconds.

The graph presented in Figure 4-5 shows the time-dependent temperature for the simulation where no restriction is imposed to the maximum temperature in the test section. For this specific case the thermal analysis in VrSuite was performed by considering the time-dependent temperature at points 8 (test section inlet), 9 (test section outlet), 10 (first elbow after the test section outlet), 13 (heat exchanger inlet) and 14 (heat exchanger outlet).

Functions that interpolate the wall temperature by time and position are used as Dirichlet thermal boundary conditions (BC) for the thermal simulation of the SCW loop.

Figure 4-5 Time dependent temperature of the wall computed at the control point.

4.2.3 Nonlinear Regression Equations for Thermal Simulation

For each of the selected control points the change of temperature over time was represented by a regression equation. Based on the traditional approach of finding an equation with the lowest sum of squared (SSQ) absolute error, the data was analysed using the function finder of a free online resource for curve fitting [22].

The function finder feature is used to search for the simplest and most reasonable equation that fits the data. The coefficients are determined by a genetic algorithm named

Differential Evolution, which makes an initial guess. A further refinement is performed by iteration, until continued iteration makes no significant improvement to the results. The best fit curve is determined among 11856 equations by presenting a ranking at the end of the analysis.

Figure 4-6 shows the temperature history at the inlet of the test section along with fitting curve for the scenarios where the maximum temperature is limited to 900 K.

Figure 4-6 Temperature history and fitting curve for the inlet of the test section.

Figure 4-7 shows the temperature history at the outlet of the test section along with fitting curve for the scenarios where the test section temperature is limited to 900 K.

Figure 4-7 Temperature history and fitting curve for the outlet of the test section.

Table 4-1 shows the equations that were found to fit well the temperature data generated with the thermal simulation. Appendix A contains the fitting results for each equation of the scenarios where the test section temperature is limited to 900 K. Table 0-3 and Table 0-4 in Appendix B show the data of temperature and its corresponding predicted values for the inlet and outlet of the test section. As an example for the resultant

temperature distribution along the length of the loop obtained through the above-noted interpolation process, Figure 4-8 illustrates the steady state conditions. Notice that the maximum temperature is lower compared to Figure 4-2, where the loop is operated at maximum conditions.

Figure 4-8 Temperature distribution in the loop at steady-state for the scenarios where the test section temperature is limited to 900 K.

Table 4-1 Best fitting equations.

Scenario	Ram-up time	Test Section Inlet	Test Section Outlet
Fluid temperature	300 sec	Gaussian Peak Modified	Morgan-Mercer- Flodin (MMF)
	1800 sec	NIST Hahn	Peters-Baskin Step- Stool
	3600 sec	Dose-Response C	NIST Hahn
Wall temperature	300 sec	NIST Hahn	NIST Hahn

Chapter 5: Stress Analysis of the SCW Loop

The supercritical water loop to be housed at Carleton University can be considered a unique structure, which has been designed to operate under extreme thermal-mechanical loading. Its components operate at supercritical pressure and are subjected to high temperature gradients. Thermoelastoplastic stresses may develop. From the design point of view the magnitude of the stresses is an important factor to be taken into consideration, as plastic deformations may lead to premature failure.

The finite element analysis performed during the present research aimed to ensure that the SCW loop will have sufficient strength to operate safely under the required service condition. The use of finite element simulation was a valuable tool to determine the best way to constrain the loop. Multiple tests were run to decide the components and the directions to be constrained in order to minimize the maximum stresses.

5.1 Structural Simulation of the SCW Loop

5.1.1 Test Setup

In order to determine the distribution of the stresses in the structure of the SCW loop a structural analysis was performed. The loop was modeled by a combination of 8-noded

hexahedral and 10-noded tetrahedral elements. The total number of elements is 130444, and the number of nodes is 99223. The analyses time were 5400 and 3600 seconds. One core of an Intel Quad Core Q9650 3.0 GHz processor was used for the analysis. 53098 CPU seconds are required to complete the analysis for 600 seconds of time step size.

The structural simulation results presented herein are based on six different scenarios. Firstly an analysis was performed for at steady state condition with the loop pressurized at 25 MPa. This simulation, hereafter referred to as Scenario 1, aims to confirm that maximum displacement in the loop is within a range that will not interfere with the functionality of the test setup, and the stresses remain well within safe operating conditions at steady-state.

Secondly, the loop is pressurized to 25 MPa and analyzed for a 300 kW of heat input in the test section in 300, 1800 and 3600 seconds with a linear ramp-up rate starting at 0 kW. The temperature in the test section was limited to 900 K. The temperature of the fluid was used as Dirichlet temperature BC for the analysis. These three simulations hereafter referred to as Scenarios 2, 3 and 4, respectively, differ from the Scenario 1 because they are for transient conditions.

Finally the loop was analyzed for the most critical scenario where 300 kW of heat was introduced in the test section in 300 seconds with the loop pressurized at 25 MPa. No limit was imposed to the temperature in the test. This analysis aims to confirm the same as the Scenario 2 but for the most critical transient conditions.

The loop is constrained in all three directions at the points where the heat exchanger is mounted to the floor, and in two directions where the pump is mounted to the floor. Additionally, the structural simulation was set up with displacement constraints that are compatible with the loop platforms mounted at several points along its length as illustrated in Figure 5-1.

Figure 5-1 Platforms mounted at several points along of the loop.

5.1.2 Meshing Strategy

Meshing the SCW loop for structural analysis was the most time-consuming task of the entire process. Solids models were imported from Pro/Engineer and meshed in Femap with NX Nastran. The solid tetrahedral mesher was used for part with holes or other complicated intricacies. Hexahedral meshing was preferred whenever it was possible as hexahedral elements provide a better control over the number of elements through thickness and the CPU cost per element for a given accuracy is less.

The procedure in the hex meshing process involves dividing the part into simpler hex-meshable solids. Only solids that can be deformed smoothly into a cube can be meshed to hexahedral elements. Therefore one must break/slice complex parts it into multiple, simpler solids that can be hex-meshed. Figure 5-2 shows a tee joint subdivided into simpler hex-meshable solids. Figure 5-3 illustrates the resulting hexahedral mesh.

The CPU time can be reduced considerably when parts are hex meshed. Using tetrahedral implies a large number elements and nodes as illustrated in Figure 5-4.

Figure 5-2 Tee joint subdivided into simpler hex meshable solids.

Figure 5-3 Hexahedral mesh for a tee joint.

Nodes: 1221
Elements: 768

Nodes: 7158
Elements: 3965

Nodes: 5.86 x
Elements: 5.16 x

Figure 5-4 Comparison of the number of elements and nodes for a tee joint meshed with hexahedral and tetrahedral elements.

Chapter 6: Results and Discussions.

6.1 Thermal Analysis

The thermal analysis was performed to accurately predict the temperature distribution along the length of the SCW loop for steady-state and transient operations. The time-dependent temperature histories at a total of 16 locations along the length of the loop computed by Mason *et al* (2010) [3], were used to find the fitting equations, which were used as Dirichlet thermal BC conditions in the structural simulations. Figure 4-6 and Figure 4-7 show a good correspondence between the numerical data and the predicted results of the fitting equations. As examples for the resultant temperature distribution along the length of the loop Figure 4-2 and Figure 4-8 are illustrated. A large temperature gradient is observed along the test section. The locations with the highest temperatures are found from the outlet of the test section to the inlet of the heat exchanger. The temperature results from this analysis were used in subsequent stress analysis to predict the displacements and stresses in the SCW loop.

6.2 Stress Analysis

6.2.1 Results for Displacements

The resulting displacement distribution on the SCW loop for the simulation scenario 4 is shown in Figure 6-1. The displacements have been magnified by ten times for illustration purposes. In the figure the loop depicted with dashed lines corresponds to the undeformed original loop. The maximum displacement is found in the upper tee of the test section due to thermal expansion and the effect of high pressure. The maximum displacement location is the same for all simulations with Dirichlet temperature BC.

The high supercritical temperatures seem to have the greatest effect on the magnitude of the maximum displacement and on its the location as well. Figure 6-2 through Figure 6-4 show the effect of only pressure, only temperature, and pressure plus thermal on the resulting displacement distribution for simulation scenario 1. When the loop is subjected to pressure without any thermal effect, the maximum displacement is located in the lower tee of the test section and its magnitude is much lower. On the other hand, the magnitude and location for the maximum displacement when the loop is under pressure with thermal effect are roughly the same as the simulation with the temperature and zero pressure.

Figure 6-1 Predicted displacement at steady-state for simulation scenario 4.

Figure 6-2 Predicted displacement caused by the effect of 25 MPa of pressure at steady-state for simulation scenario 1.

Figure 6-3 Predicted displacement caused by the time-dependent temperature at steady state for simulation scenario 1.

Figure 6-4 Predicted displacement caused by combined effect of pressure plus thermal at steady-state for simulation scenario 1.

Figure 6-5 shows the time dependent displacement for simulation scenarios 2 through 5. The magnitude of maximum displacement is determined by the maximum temperature at steady-state. This is illustrated in Figure 6-5. The maximum temperature at steady-state for simulation scenario 2 through 4 is the same for all of them; therefore the magnitude of the maximum displacement is also the same. Simulation scenario 5 presents the largest maximum displacement.

Figure 6-5 Minimum-maximum predicted displacement vs. time.

6.2.2 Results for Stresses

Figure 6-6 shows the predicted effective stresses for simulation scenario 4. The stress limiting area was found to be the area at the pipe connecting the pump and the flow-meter. Although this segment is located in the “cold” part of the loop, the high magnitude of stress seems to be caused by separate mechanisms. First, the bending stresses in the joining tee, and second, the mutual restraint resulting from the temperature differences

between the connecting pipe and the pump, influenced by the fact that the pump is free to translate in the z direction. The heat exchanger is constrained in all directions.

The high supercritical temperatures have the greatest effect on the total stresses of the SCW loop. This is illustrated in Figure 6-7 for simulation scenario 5, where a detailed view of the zone of maximum stresses is shown. Figure 6-8 shows that the effective stresses caused by the pressurization of the loop are very low compare with the stresses caused by the thermal expansion.

Figure 6-6 Predicted stresses for simulation scenario 4

Figure 6-7 Contour of the zone of highest effective stresses for a) pressure only, b) temperature only and c) pressure plus thermal at steady state for simulation scenario 5.

Figure 6-8 Predicted Von-Mises stresses vs. time for pressure only, time-dependent temperature only, and pressure plus thermal for simulation scenario 5.

Although the maximum effective stresses at steady-state are roughly of the same magnitude for simulation scenario 2 through 4, during transient heating these stresses may differ significantly. This is illustrated in Figure 6-9.

Figure 6-9 Effective stresses vs. time for simulation scenarios 1 through 4.

The results of the simulations show that the displacements will not interfere with the functionality of the test setup and the stresses remain well within safe operating conditions. Although some plasticity is observed, the maximum total strain was found to be 0.00132. A preliminary estimation, by comparing the maximum total strain with the results of [17], showed that the corresponding number of duty cycles is 50000.

The equation for the Coffin-Mason model published independently by L.F. Coffin (1954) and S. S. Mason (1953), is used to estimate the effect of low cycle fatigue induced by thermal stresses [23].

$$N_f \Delta \epsilon_p^2 = C_1 \quad (1.10)$$

Where:

N_f = Number of cycles to failure (cycles)

$\Delta \epsilon_p$ = Strain amplitude per load cycle

C_1 = Proportionality constant for a particular material

The results of these simulations, and many others that have been performed for different operational schedules, confirm the structural integrity and functionality of the SCW loop and its fitness-for-service for the intended series of experiments to study the heat transfer characteristics of water at supercritical conditions.

Chapter 7: Conclusions and Future Work

In this chapter the conclusions to the thesis will be presented. Additionally, suggestions for future expansion of the work described in this document will be provided.

7.1 Summary of Thesis

In this thesis, a summary of the design of the Carleton SCW loop is presented. Also, the most relevant related works have been discussed with emphasis in the use of FEA for stress analysis of thick-walled high-pressure vessels. The main purpose of this evaluation was to determine the requirements for design-driven analysis oriented to SCW loops.

The finite element analyses were performed using the VrSuite software from GOLDAK technologies. This software has been verified against two classical problems; a thick-walled pipe subjected to internal pressure and constrained to have zero axial elongation and an internally heated thick-walled pipe.

The time-dependent temperature histories resulting from the transient thermal analysis of the loop were used as input to the structural simulation. To obtain the temperature-versus-time information for all points along the length of the loop fitting

equations were found. The temperature distributions computed by Mason *et al* (2010) [19] were taken as input for structural analysis.

Finally, a finite element structural analysis was presented based on five simulation scenarios for steady-state and transient conditions. The structural simulation was set up with displacement constraints that are compatible with the loop platforms mounted at several points along its length.

7.2 Summary of Results

Given the very satisfactory results obtained in these two benchmarks discussed in Chapter 3, the spatial resolution utilized in these studies was adopted for the structural analysis of the Carleton SCW loop.

The fitting equations accurately predicted the time-dependent temperature histories at a total of 16 locations along the length of the loop.

Range of displacements, stresses and strains were obtained from the structural analysis. The combination of these results with the criteria from the ASME Design Code may be used to determine whether the Carleton SCW loop deployment could be conducted without causing unacceptable fatigue and creep damage.

The structural simulation confirmed that the predicted displacements are within a range that will not interfere with the functionality of the test setup.

Preliminary evaluation of the results of the simulations suggests the structural integrity and functionality of the SCW loop for the intended series of experiments to study the heat transfer characteristics of water at supercritical conditions.

7.3 Suggestions for Future Research

In order to complement the present study, the following additional work is suggested:

1. To use VrSuite to run fluid flow and advection diffusion analysis.
2. To study the effect of the pressure transient, i.e. water hammer, on the stresses of the SCW loops.
3. The detailed geometry of more complex parts such as the pump, heat exchanger and valves was approximated. Therefore a better approximation for future works should be considered
4. To run additional simulation by using finer meshes in the location of maximum effective stresses.
5. Further simulations should be performed in order to determine the best way to constrain the loop with the objective of further reducing the maximum stresses.
6. The results of the structural simulation show some plasticity. Additional investigation should be performed to evaluate the possible accumulated fatigue damage during the lifetime of the loop.

7. To run additional simulations for the initial state of the SCW loop in which residual stress is considered. For these simulations the residual stresses and distortion introduced by manufacturing, by welding, by bolting flanges, Grayloc connections and misfit in assembly should be taken into account.
8. Fluctuations in pressure and vibrations should be considered for future works.

Appendix A

Location	Ramp-up time	Fitting Results
Test section Inlet	300 sec	Gaussian Peak Modified With Offset $y = a * \exp\left(-0.5 * \left(\frac{x - b}{c}\right)^d\right) + Offset$ <p>sum of squared absolute error = 1.0072916033775069E+04</p> <p>a = 1.6115325120993890E+02</p> <p>b = -5.3912988415542929E+02</p> <p>c = 1.1036111336111758E+03</p> <p>d = -4.0015808277043101E+00</p> <p>Offset = 3.0001979684335470E+02</p>
	1800 sec	NIST Hahn $y = \frac{a + b * x + c * x^2 + d * x^3}{1.0 + f * x + g * x^2 + h * x^3}$ <p>sum of squared absolute error = 3.0720033724500822E+03</p> <p>a = 3.0017420299184323E+02</p> <p>b = -2.3757638593902675E-01</p> <p>c = 1.8446025369876231E-04</p> <p>d = 2.6825420307528062E-08</p>

Test section Outlet		$f = -8.1037986023581028\text{E-}04$ $g = 4.8476544739427670\text{E-}07$ $h = 5.0284036638169406\text{E-}11$
	3600 sec	Dose-Response C $y = b + \frac{a - b}{1 + 10^{d*(x-c)}}$ Sum of squared absolute error = 1.1262582810762338E+04 $a = 4.6060755460098574\text{E+}02$ $b = 2.8604702228066765\text{E+}02$ $c = 1.5499449001666294\text{E+}03$ $d = -7.4037074801520746\text{E-}04$
	300 sec	Morgan-Mercer-Flodin (MMF) $y = \frac{a * b + c * x^d}{b + x^d}$ sum of squared absolute error = 3.9617001841783451E+04 $a = 3.0559989764164061\text{E+}02$ $b = 9.5071096310452500\text{E+}05$ $c = 6.6294150255492423\text{E+}02$ $d = 2.8178760426004676\text{E+}00$
	1800 sec	Peters-Baskin Step-Stool: yIII (6) With Offset $K = \ln(\exp(b2 * c1 * d1) + \exp(b2 * d1 * x))$ $yII = b1 * x + \frac{K}{d1}$ $L = \ln(\exp(b2 * c1 * d1) + \exp(b2 * c2 * d1))$

	$y_{III} = y_{II} - \ln\left(\exp\left(d2 * \left(b1 * c1 + \frac{L}{d1}\right)\right) + \frac{\exp(d2 * y_{II})}{d2} + Offset\right)$ <p>sum of squared absolute error = 7.8694104656548061E+03</p> <p>b2 = 3.9269854424455797E-01</p> <p>c1 = 1.0004698813880129E+03</p> <p>d1 = -2.5203810627209999E-02</p> <p>b1 = 9.9269591943331988E-04</p> <p>c2 = 6.5512757893388056E+01</p> <p>d2 = -3.7286931740020034E-02</p> <p>Offset = 2.9214570305071436E+02</p>
3600 sec	<p>NIST Hahn</p> $y = \frac{a + b * x + c * x^2 + d * x^3}{1.0 + f * x + g * x^2 + h * x^3}$ <p>sum of squared absolute error = 5.072003272460021E+03</p> <p>a = 2.9125731998552021E+02</p> <p>b = 6.1552302396745084E-02</p> <p>c = -1.1613243900990397E-04</p> <p>d = 9.0371468345991624E-08</p> <p>f = -4.2609389822521415E-04</p> <p>g = -1.7206140862157601E-08</p> <p>h = 1.2079720119033841E-10</p>

Appendix B

Table 0-1 Tabular comparison of numerical and analytic results for stresses (fine mesh).

r (m)	σ_r An (MPa)	σ_r FEA (MPa)	σ_θ An (MPa)	σ_θ FEA (MPa)	σ_z An (MPa)	σ_z FEA (MPa)	σ_v An (MPa)	σ_v FEA (MPa)
0.01397	-25.00	-19.80	50.21	45.92	12.60	11.69	65.13	57.16
0.01498	-20.07	-18.67	45.28	44.13	12.60	11.50	56.60	54.64
0.01600	-16.06	-15.88	41.26	41.12	12.60	11.43	49.64	49.64
0.01701	-12.74	-12.94	37.94	38.12	12.60	11.41	43.89	44.52
0.01803	-9.96	-10.27	35.17	35.43	12.60	11.41	39.08	39.91
0.01905	-7.62	-7.94	32.83	33.10	12.60	11.41	35.03	35.91
0.02006	-5.62	-5.94	30.83	31.10	12.60	11.41	31.57	32.49
0.02108	-3.91	-4.24	29.12	29.40	12.60	11.41	28.60	29.58
0.02209	-2.42	-2.83	27.63	27.98	12.60	11.41	26.03	27.17
0.02311	-1.13	-1.79	26.34	26.90	12.60	11.39	23.79	25.37
0.02413	0.00	-1.43	25.21	26.39	12.60	11.35	21.83	24.63

Table 0-2 Tabular comparison of numerical and analytic results for stresses (coarse mesh).

r (m)	σ_r An (MPa)	σ_r FEA (MPa)	σ_θ An (MPa)	σ_θ FEA (MPa)	σ_z An (MPa)	σ_z FEA (MPa)	σ_v An (MPa)	σ_v FEA (MPa)
0.01397	-25.00	-19.80	50.21	45.92	12.60	11.63	65.13	57.16
0.01905	-7.62	-9.16	32.83	30.88	12.60	11.40	35.03	34.96
0.02413	0.00	-1.43	25.21	26.39	12.60	11.35	21.83	24.63

Table 0-3 Temperature data and predicted values for the inlet of the test section where the maximum temperature is limited to 900 K.

Time	Temp for 3600 sec	Dose- Response	Temp for 1800 sec	NIST Hahn	Temp for 300 sec	Gaussian Peak
500	310	311	321	320	387	385
1000	337	335	377	378	442	441
1501	369	370	430	430	456	454
2001	405	405	453	452	458	458
2500	434	432	458	458	460	460
3001	445	447	459	460	460	460
3500	455	455	460	460	461	461
4000	458	458	460	460	461	461
4500	460	459	461	460	461	461
5000	460	460	461	461	461	461
5400	461	460	461	462	461	461

Table 0-4 Temperature data and predicted values for the outlet of the test section where the maximum temperature is limited to 900 K.

Time	Temp for 3600 sec	Dose- Response	Temp for 1800 sec	NIST Hahn	Temp for 300 sec	Gaussian Peak
500	381	382	463	462	652	655
1000	484	483	635	632	661	662
1501	582	586	658	660	663	663
2001	649	644	662	660	663	663
2500	657	660	663	661	663	663
3001	660	661	663	661	663	663
3500	662	661	663	662	663	663
4000	663	661	663	662	664	663
4500	663	662	664	663	664	663
5000	663	664	664	663	664	663
5400	663	666	664	664	664	663

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