HEAT TRANSFER AND TURBULENCE MEASUREMENTS IN SUPERCRITICAL PRESSURE WATER

J. Licht[1], M. Anderson[2] and M. Corradini[2]
[1] Atomic Energy of Canada Limited, Chalk River, Ontario, Canada
[2] University of Wisconsin-Madison, Wisconsin, USA

Abstract

A series of integral heat transfer measurements in a square annular flow passage were performed for bulk water temperatures of 175-400°C with an upward mass velocity of 315 and 1000 kg/m^{2}s and heat fluxes of 0, 220, and 440 kW/m^2, all at a pressure of 25 MPa. Measured mean and turbulent velocities in conjunction with simulations with the CFD code FLUENT show that buoyancy effects cause a significant reduction in turbulent quantities at a radial location similar to what is called the law of the wall region for isothermal flow.

1. Introduction

The supercritical water reactor (SCWR) has been selected as one of the next steps in nuclear reactor designs [1]. The SCWR is essentially a light water reactor (LWR) operating at higher pressure (25 MPa) and higher exit temperature (510°C) with the goal of increasing the thermal efficiency from 33% to 44% while building upon the well established LWR's and supercritical fossil plant designs. The coolant enthalpy passes above and beyond the two-phase dome, remaining single phase, but undergoing large changes in its thermophysical properties. A significant amount of research on heat transfer to supercritical fluids has been carried out over the past 50 years and has been summarized by Pioro et al. [2]. Experiments have shown that upward flowing, variable property fluids can cause deterioration in heat transfer. Hall [3] suggested that deterioration was caused by changes in the shear stress and derived an expression for the wall shear stress for a variable property fluid. From this, a criterion (Jackson's criterion) was developed to identify significant changes in the shear stress due to radial gradients in density (buoyancy effects). Buoyancy effected heat transfer is not unique to supercritical fluids and can occur at subcritical pressures. A significant amount of subcritical mixed convection heat transfer research was done in Russia and summarized by Petukhov et al. [4]. However, there has recently been an increasing amount of work in this area as summarized by Jackson [5], where Jackson used the laser Doppler velocimetry (LDV) technique to measure local, instantaneous velocities under conditions of deteriorated heat transfer. Additional velocity measurements were done by Kang et al. [6] using R-113 as a coolant, and by Wardana et al. [7] using air. In an effort to further study heat transfer at supercritical pressures, a Supercritical Water (SCW) heat transfer facility was built at the University of Wisconsin-Madison with optical access for local measurements of turbulent velocities and density [8]. Additionally, heat transfer simulations were carried out with the Computational Fluid Dynamic (CFD) code FLUENT, which, along with the experimental data, are used to explain the fluid flow characteristics resulting in the deterioration in heat transfer.

2. Experimental facility

A SCW heat transfer facility has been built at the UW-Madison to allow for a detailed study of heat transfer to SCW in a circular and square annular geometry. The loop (Figure 1) has dimensions of approximately 2 m wide by 3 m tall and is made of 4.29 cm inner diameter Inconel 625 piping. A 3.3 m long Fuel Element Simulator (FES) with a diameter of 1.07 cm spans the entire right leg of the loop and protrudes out both ends. This design permits the use of 16 thermocouples, evenly spaced, along the inner cladding of the 1.01 m heated length. The center portion of the right leg of the loop serves as the test section, allowing a 76 cm entrance length for both upward and downward flow studies. The circular annular test section geometry is a 1.07 cm diameter FES within a 4.29 cm diameter flow channel. The square annular test section geometry is a 1.07 cm diameter FES within a 2.88 cm wide flow channel. The FES is centered within the flow channel with six spacers; four of which are located on either side of the tees and two that are 5 cm from either end of the heated section. The FES can generate up to 50 kW, giving a maximum, uniform heat flux (Q") of 1.5 MW/m^2. A pump capable of operating at supercritical conditions generates mass velocities (G) in the range of 200 to 2000 kg/m^{2}s. The current configuration is upward flow; however the facility was designed for flows in either direction with only minor modification. The facility is capable of operating at any steady state heat flux condition by using a variable heat removal system made up of copper cooling coils. Eight copper coils of various contact area are tightly wrapped to the Inconel piping. Heat removal by the cooling coils can be set to match that supplied by the FES by simply controlling the number of coils receiving cooling water and controlling their respective flow rates. A description of the mean velocity and turbulence measurement system and technique is given elsewhere [8].

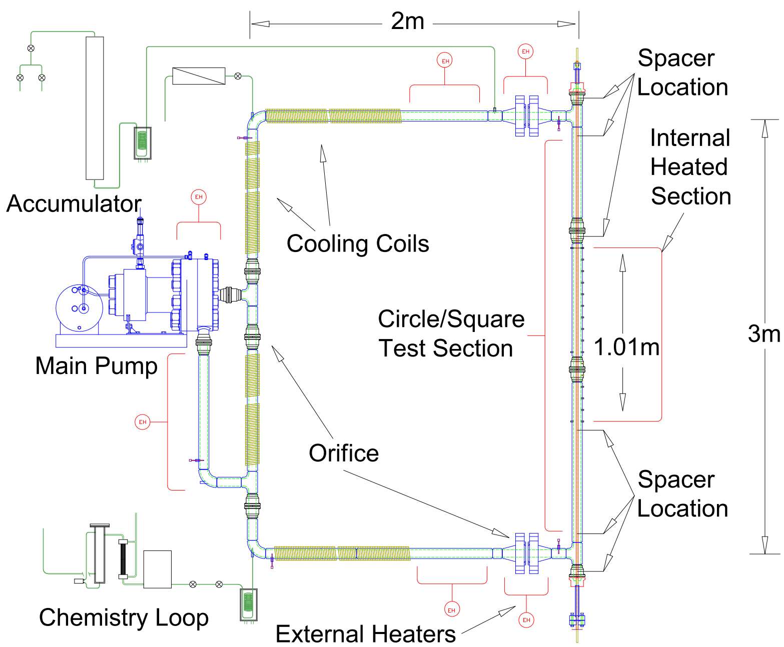

Figure 1 Heat transfer loop with a circle or square annular geometry test section.

3. Results and discussion

3.1 Integral heat transfer measurements

Figure 2 shows a subset of the experimentally measured wall temperatures versus bulk fluid enthalpy for heat transfer experiments performed over a wide range of boundary conditions along with wall temperature predictions from Jackson's Nusselt correlation [9]. Each group of data points represents a separate set of experimental conditions. It should be noted that, for each experiment, the axial bulk temperature (T_b) change is relatively small (<5°C) due to the test conditions and large hydraulic diameter of the test section. The first graph shows that the high mass velocity (1000 kg/m^{2}s) data matches well with Jackson's correlation for high and low heat flux for bulk inlet temperature spanning the pseudocritical temperature (T_{pc} = 385°C at 25 MPa). The second graph shows that at lower mass velocities (315 kg/m^{2}s), deterioration in heat transfer can occur resulting in wall temperatures significantly increasing above that predicted by Jackson's correlation. For example, at a heat flux of 220 kW/m^2, the wall temperatures agree well with that predicted by Jackson's correlation for bulk temperatures above and well below the pseudocritical temperature. At a bulk temperature just below the pseudocritical temperature, the data shows a rise and recovery spanning wall temperatures from ~400°C to 450°C. When increasing the heat flux to 440 kW/m^2, the range in bulk temperature in which deterioration in heat transfer occurs increases, especially for bulk temperatures less than the pseudocritical temperature. The reason for this is detailed in sections 3.2 and 3.3. The next section discusses the mean and turbulent velocity measurements for a single experiment that exhibits deterioration in heat transfer.

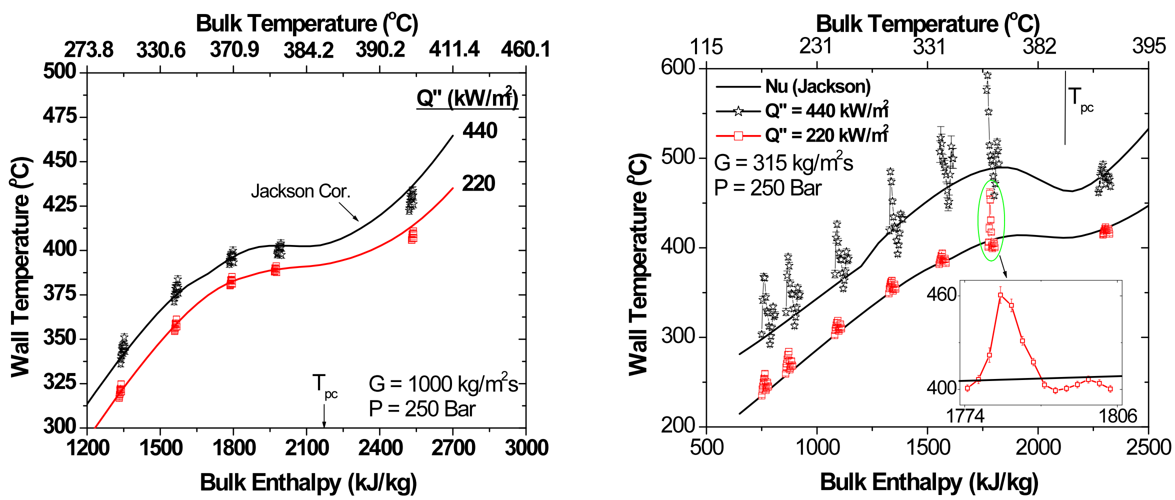

Figure 2 Heat transfer data compared with Jackson's Nusselt correlation for high (1000 kg/m^{2}s) and low (315 kg/m^{2}s) mass velocity.

3.2 Mean and turbulent velocity measurements

Figure 3 shows an example of what could be considered a mild deterioration in heat transfer. The experimental conditions are G = 300 kg/m^{2}s, T_b = 175°C and Q" = 220 kW/m^2. The wall temperature increases over the first 0.5 m and then decreases. At ~0.7 m the wall temperature undergoes a similar increase and then decrease in wall temperature.

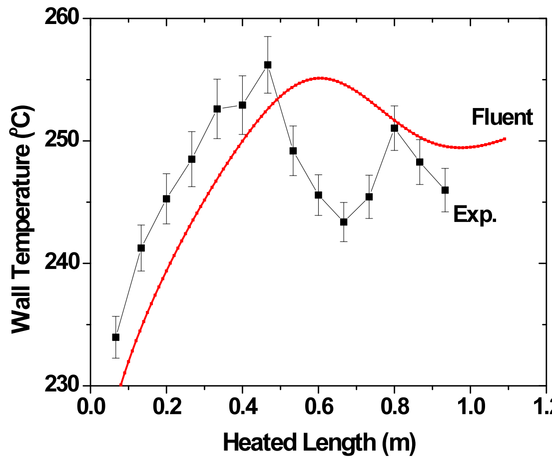

Figure 3 Heat transfer data and FLUENT simulation for an experiment that exhibits what can be considered a mild deterioration in heat transfer.

Figure 4 shows the mean axial velocity, axial and radial turbulence intensities, turbulent shear stress, turbulence production, and turbulent diffusivity profiles radially from the heater rod surface (R-R_i) at an axial location of 0.5 m of the heated section for the experiment shown in Figure 3. Figure 4 shows that the addition of a heat flux causes an increase in the mean axial velocity in the near wall region relative to the isothermal case. The increase in velocity is caused by buoyancy effects associated with the increase in near wall fluid temperature. Because water has constant properties for the isothermal case, the shape of the velocity profile in a log scale indicates the important regions described in non-dimensional scaling. The measurement nearest to the wall is found to be at non-dimensional distance of about $y^+ = 8$. The boundary between the law of the wall region and the buffer layer (y^+ ~30) is identified in the Figure 4a. These locations aid in the understanding of the profiles seen in the axial and radial turbulence for the isothermal case (Figure 4b, 4d). Both turbulent components behave similarly in the bulk of the flow, meaning that there are low turbulence levels in the bulk of the flow where the velocity gradients are small. As the wall is approached, the turbulence exhibits a peak and then begins to decline due to the viscous effects at the wall. When a heat flux is present, both turbulent components again behave similarly, however, non-dimensional locations for the case with a heat flux aren't as easily determined because of the variable properties. The turbulence increases in the bulk of the flow (R-Ri > 2), while the peak in turbulence is shifted closer to the wall. A similar effect is seen in the turbulent shear stress. The zero crossing of the turbulent shear stress shifts toward the heated wall indicating that the peak in axial velocity is also shifting toward the heated wall.

To better understand the physical importance of these trends, the turbulent production and turbulent diffusivity of momentum are shown in Figures 4e and 4f. In the very near wall region (radially out to ~0.5 mm), the turbulent production and diffusivity are equal to or greater than the isothermal case. In non-dimensional units, this occurs out to $y^+ = 50$. Further out into the flow, the values are less than the isothermal case. Each figure is missing a data point at ~2 mm from the wall. This is due the fact that the turbulent shear stress and velocity gradient used in the calculation are essentially zero, so the associated uncertainties result in an erroneous data point. Progressing further into the bulk of the flow, the values are again larger than in the isothermal case. These results mean that during deterioration, there are actually increases in the production and diffusion of turbulence in the very near wall region. However, further out into the flow, starting at a location of about 0.5 mm, the buoyancy effect inhibits production and the diffusion of momentum into the bulk of the flow. The effects on the diffusion of momentum are also likely happening to the diffusion of heat. The turbulent Prandtl number is defined as the ratio of the diffusivity of momentum and heat and is typically assumed to be ~1. Recent measurements by Kang et al. [6] suggested that this assumption holds true for variable property heat transfer. This means that both the diffusivity of momentum and heat transfer, are reduced during deterioration at a radial position equivalent to what is called the law of the wall region for isothermal flow. While the difference between the deteriorated and isothermal case is not large, it must be remembered that the measurements are made at an axial location where the heat transfer is beginning to improve and wall temperature is decreasing. FLUENT simulations presented in the next section are used to gain further insight into how these fluid flow characteristics evolve over the length of the heat section.

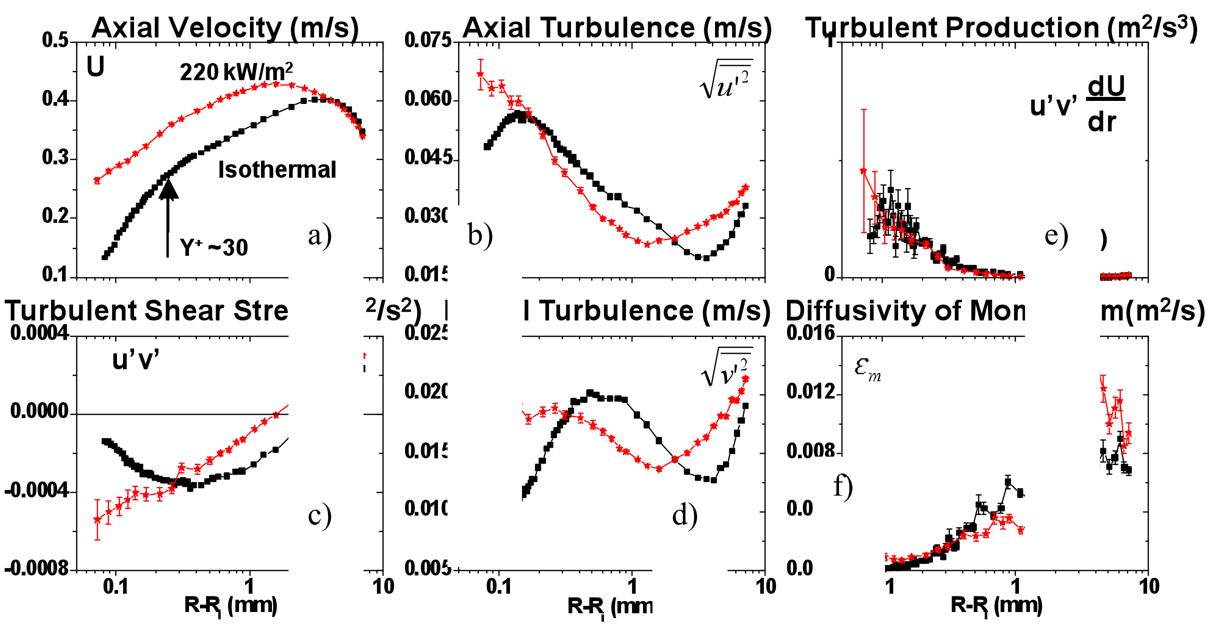

Figure 4 Mean and turbule asurements for an expe nt with 00 kg/m^{2}s, T$_b$ = 175°C and
Q" = 0 (isothermal) anc kW/m^2

3.3 FLUENT simulations

For the mild case of heat transfer deterioration presented above, a FLUENT simulation was performed to gain an understanding of the evolution of the fluid flow characteristics. Figure 3 shows the axial evolution of the axial wall temperature simulated with FLUENT. While computational fluid dynamics are not currently able to accurately simulate deterioration in heat transfer, FLUENT is able to reasonably simulate the wall temperature profile for this simple case.

The fact that deterioration occurs in mixed convection heat transfer is quite clear when considering simulations of the turbulence evolution along the heated wall in Figures 5 and 6. The vertical lines in the axial wall temperature graph represent the axial location where the radial profiles of the fluid temperature, mean velocity, turbulent shear stress, and diffusivity of momentum occur. The evolution of the fluid flow characteristics can be broken up into three parts.

First part (Figure 5, axial length x = 0 – 0.2 m). As the fluid enters the heated section, the near wall fluid begins to increase in temperature. This increase in temperature is associated with a decrease in density. Because the near wall density is lower than the density of the fluid in the bulk of the flow, the near wall velocity begins to increase due to buoyancy forces. This initially forms a flatter velocity profile, meaning that the velocity gradient is moved into the very near wall region where molecular viscosity effects dominate. The lack of velocity gradient in the typical law of the wall region drastically reduces the diffusion of momentum and heat, which prevents transport away from the near wall region, causing the wall and near wall fluid to further increase in temperature and further increases the density difference. The axial evolution of the fluid temperature profile (from 0.2 to 0.4 m) indicates that as the turbulent diffusion is reduced, the temperature within ~1.1 mm of the wall increases while the bulk fluid temperature remains unchanged. Initially there is a positive feedback between the evolution of the axial wall temperature and the deterioration in heat transfer.

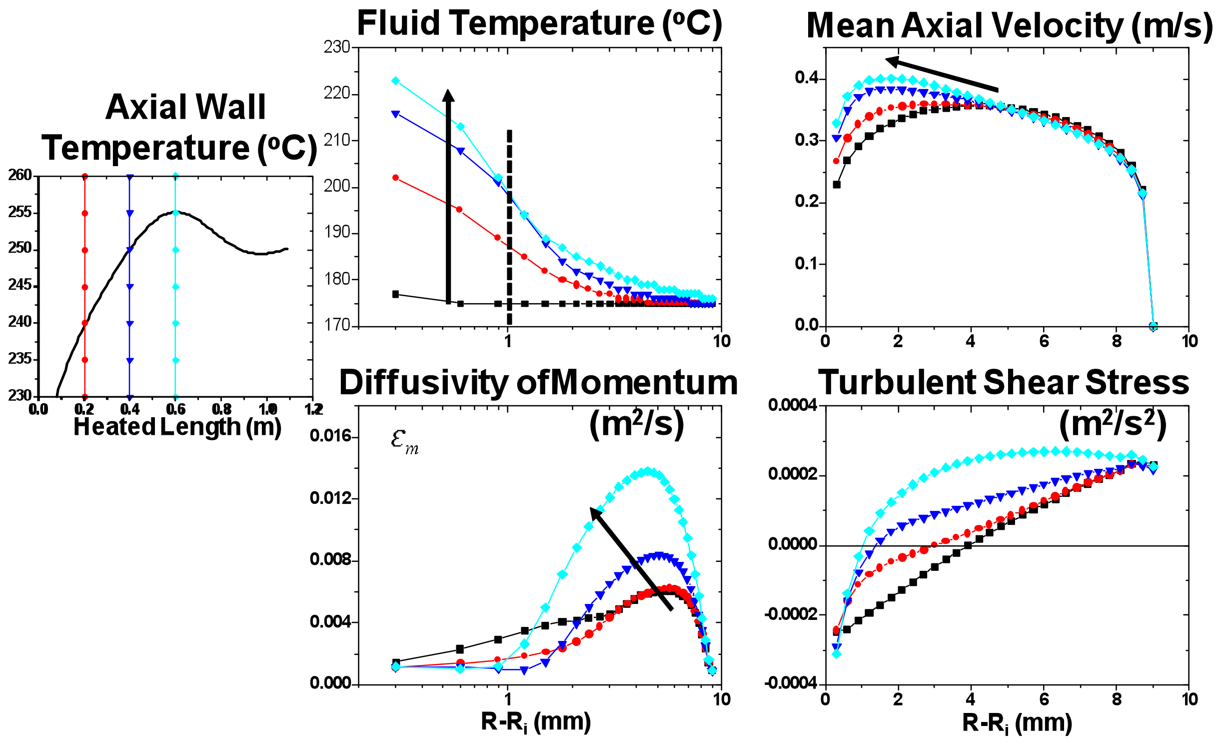

Figure 5 Simulated evolution of the wall temperature and turbulence parameters at different axial positions; spanning the evolution of the deterioration process (first half of the heated section).

Second part (Figures 5 and 6). As the wall and near wall fluid temperature increases and buoyant forces further act on the fluid, the peak velocity increases and shifts toward the heated wall. The velocity gradient between ~1 to 4 mm goes from a flattened profile at an axial position of 0.2 m to a profile with significant increase in velocity gradient at 0.6 m. This transition causes the diffusion of momentum in the bulk of the flow to increase and expand into the near wall region (Figure 5). The increased diffusion allows the energy to be removed from the near wall region causing the wall and near wall fluid temperature to decrease and the bulk fluid temperature to increase (temperature profile at 0.6 and 0.8 mm, Figure 6). As a consequence, the density difference between the inner and outer wall decreases, reducing the buoyancy force. As the peak velocity decreases and shifts away from the inner wall, the diffusivity will begin to decrease again, although its changes lag behind the changes in the velocity gradient. This decrease in diffusivity will again cause the wall temperature to increase, explaining the oscillatory shape of the wall temperature for both simulation and experiment. For large hydraulic diameter experiments, the bulk temperature isn't increasing significantly so these oscillatory changes in velocity gradient and diffusivity might continue in the axial direction. This type of wall temperature profile can be seen in experiments by Kenning et al. [10].

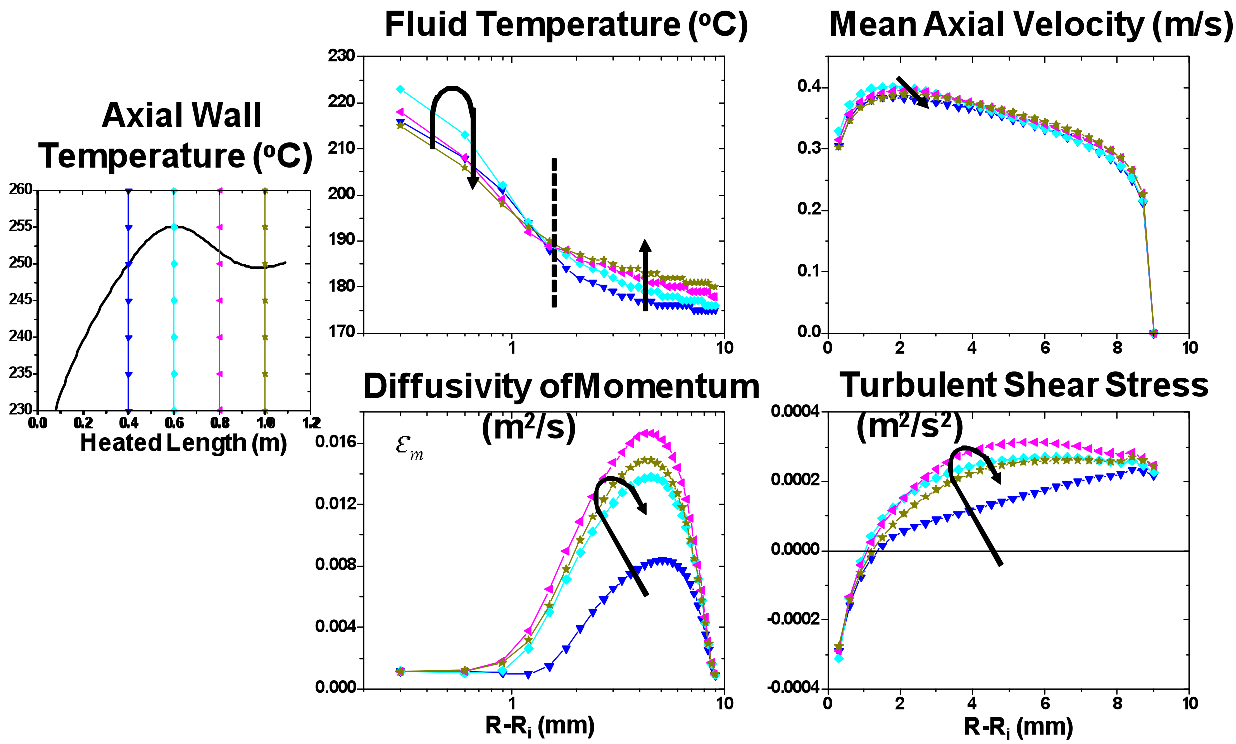

Figure 6 Simulated evolution of the wall temperature and turbulence parameters at different axial positions; spanning the evolution of the deterioration process (second half of the heated section).

Third part (not shown). The radial gradient in density is lost as the bulk fluid temperature passes through the pseudocritical temperature. The loss of density difference forces the velocity gradient to pass through a flattened profile to reach a velocity profile similar to that seen in isothermal flow. Thus, a wall temperature increase would be expected to be seen due to this effect. In an experiment performed by Hall et al. [11] using carbon dioxide as a surrogate fluid, a large localized spike in wall temperature occurred near the beginning of the heated section. Following this, as the bulk fluid temperature approached the pseudocritical temperature; a more broadly shaped wall temperature increase was seen. Burke et al. [12] demonstrated a similar wall temperature profile, but also measured radial mean axial velocity profiles at several axial locations. The initial increase in wall temperature was caused by a flattening in the mean axial velocity gradient (first part) followed by a recovery in the heat transfer as the peak in the mean axial velocity increased and shifted toward the heated wall (second part). As the bulk fluid temperature increased toward the pseudocritical temperature, a more gradual decrease in the heat transfer coefficient was observed and is associated with a transition of the mean axial velocity back to the typical profile seen in isothermal flow (third part).

The results of this work illustrate the complicated changes in fluid flow characteristics that cause deterioration in heat transfer. It also illustrates a deficiency in almost all integral heat transfer experiments aimed at measuring deteriorated heat transfer in supercritical pressure fluids. To understand this last statement, consider one channel in a SCWR with an inlet and outlet temperature (320°C to 625°C) spanning the pseudocritical temperature (385°C). The amount of power needed to heat water over this entire temperature range typically prohibits operating an experiment at these conditions. Instead, experiments are performed in short test sections with limited amounts of power such that they operate over a limited temperature range. This means a series of experiments must be performed with incremental increases in inlet temperature to cover the entire temperature range. This is acceptable for high mass velocity experiments where forced convection dominates the heat transfer. This is not true for low mass velocity experiments where buoyancy affects alter the fluid flow characteristics. Experiments performed over limited temperature ranges will not reproduce the local velocity profile and fluid flow characteristics that develop axially along the heated section and significantly affect the heat transfer. If it is important to know the heat transfer under conditions where deterioration may be present, then an experiment that spans the entire temperature range must be performed such that the evolution of the fluid flow characteristics are correctly captured.

4. Conclusion

A series of integral heat transfer measurements in a square annular flow passage were performed for bulk water temperatures of 175-400°C with an upward mass velocity of 315 and 1000 kg/m^{2}s and heat fluxes of 0, 220, and 440 kW/m^2, all at a pressure of 25 MPa. Detailed mean and turbulent velocity measurements show that the turbulence, diffusivity of momentum, and likely the diffusivity of heat, are reduced during deterioration in heat transfer at a radial position equivalent to what is called the law of the wall region for isothermal flow. For the simple case of deterioration investigated in detail, FLUENT simulations offered qualitative insight into changes in fluid temperature and turbulent velocities responsible for the axial evolution of the wall temperature. Experiments investigating deterioration in heat transfer must be performed at conditions spanning the entire temperature range of a SCWR such that the evolution of the fluid flow characteristics is correctly captured.

5. References

[1] US Dept of Energy, Nuclear Energy Research Advisory Committee, Generation IV International Forum, "A Technology Roadmap for the Generation IV Nuclear Energy Systems", GIF-002-00, 2002, http://gif.inel.gov/roadmap/pdfs/gen_iv_roadmap.pdf.

[2] I.L. Pioro, "Heat Transfer and Hydraulic Resistance at Supercritical Pressure in Power-Engineering Applications", ASME, New York, 2006.

[3] W.B. Hall, "Forced convective heat transfer to supercritical pressure fluids", Arch Termodyn Spalania, Vol. 6, Iss. 3, 1975, pp. 341-352.

[4] B.S. Petukhov and A.F. Polyakov, "Heat Transfer in Turbulent Mixed Convection", Hemisphere, Washington, 1988.

[5] J.D. Jackson, "Studies of buoyancy-influenced turbulent flow and heat transfer in vertical passages", Proceeding of the Thirteenth International Heat Transfer Conference, Sydney, Australia, KN-24, 2006.

[6] S. Kang, B. Patil, J.A. Zarate and R.P. Roy, "Isothermal and heated turbulent upflow in a vertical annular channel-Part I. Experimental Measurements", International Journal Heat Mass Transfer, Vol. 44, Iss. 6, pp. 1171-1184, 2001.

[7] I.N.G. Wardana, T. Ueda, S. Kurihara and M. Mizomoto, "Turbulence structure in an annuli with strongly heated inner cylinder", Exp Fluids, Vol. 27, Iss. 2, pp. 137-144, 1999.

[8] J.R. Licht, "Heat transfer and fluid flow characteristics in supercritical water", PhD thesis, University of Wisconsin-Madison, Madison, WI, 2008.

[9] J.D. Jackson, "Consideration of the heat transfer properties of supercritical pressure water in connection with the cooling of advanced nuclear reactors", Proceedings of the 13th Pacific Basin Nuclear Conference, Shenzhen City, China, 2002.

[10] D.B.R. Kenning, R.A.W. Shock and J.Y.M Poon, "Local reduction in heat transfer due to buoyancy effects in upward turbulent flow", Proceedings of the 5th International Heat Transfer Conference, Tokyo, Japan, pp. 139-143, 1974.

[11] W.B. Hall and J.D. Jackson, "Laminarization of turbulent pipe flow by buoyancy forces", ASME (69-HT-55), pp. 1-8, 1969.

[12] P.J. Bourke and J.D. Pulling, "Experimental explanation of deterioration in heat transfer to supercritical carbon dioxide", ASME Paper no. 71-HT-24, 1971.