HEAT TRANSFER ENHANCEMENT AT SUPER-CRITICAL PRESSURES

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ABSTRACT

A Supercritical Water Reactor (SCWR) design has been proposed by AECL as part of Canada's commitment to the GEN IV program. In this design, the outlet coolant temperature would be 625 °C, which is much higher than the corresponding value for the CANDU reactor and would result in a much higher thermal efficiency. To maintain fuel integrity, it is essential that the maximum fuel sheath temperature of the SCWR would not exceed 850 °C. Consequently, a central issue in the thermal-hydraulic design of the Canadian SCWR fuel bundle is the development of means to enhance heat transfer and the avoidance of "deteriorated heat transfer" under normal operating conditions as well as during different accident scenarios. These are the objectives of current work at the University of Ottawa, which, together with work at several other Canadian Universities, supports the development of the Canadian SCWR.

This paper will discuss the following options for heat transfer enhancement under supercritical pressures: (i) choosing the operating conditions such as to avoid operating in the deteriorated heat transfer regime; (ii) using turbulence promotion devices that delay or eliminate the occurrence of deteriorated heat transfer and enhance heat transfer with a minimal increase in channel hydraulic resistance; and (iii) optimizing the fuel geometry such as to minimize sheath temperatures while maintaining fuel integrity.

The paper will present a critical analysis of previously published work on the effect of various flow obstructions and rod bundle spacers on heat transfer at high subcritical and supercritical pressures. The work to be discussed is mostly experimental but it also includes some numerical studies. An important experimental observation has been that the installation of flow obstructions delays or eliminates the onset of heat transfer deterioration and results in heat transfer enhancement. Predictions of spacer effects on supercritical heat transfer using CFD codes have been in fair agreement with experimental observations for normal flow conditions, but were deemed to be of questionable accuracy for deteriorated heat transfer conditions.

A supercritical flow loop using carbon dioxide as a coolant has been in operation at the University of Ottawa. Heat transfer measurements in upward flow through a three-rod bundle with wire-wrap spacers have been scheduled for the near future. Depending on the progress, some results in this and other configurations may be presented at the Conference. .

NOMENCLATURE:

Ac^*	Acceleration parameter
Bo^*	Buoyancy parameter
d	Rib depth or height (mm)
D	Diameter (m)
f	Friction factor
G	Mass flux (kg/(m^2.s))
H	Specific enthalpy (kJ/kg)
k	Fluid thermal conductivity (kW/(m.°K))
L	Distance (m)
Nu	Nusselt number
P	Pressure (MPa)
q"	Heat flux (kW/m^2)
Q^*	Thermal loading/expansion group
Re	Reynolds number
S	Rib pitch (mm)
T	Temperature (°K or °C)
ΔP	Pressure drop (kPa)

Greek Letters:

v	Specific volume(m^3)
α	Heat transfer coefficient (kW/(m^2.°K))
μ	Dynamic viscosity (N.s/m^2)
ζ	Tube hydraulic resistance
ρ	Fluid density(kg/m^3)
ε	Blockage ratio(%)

Subscripts:

b	Bulk
h	Hydraulic diameter
PC	Pseudo-critical
w	Internal wall

Abbreviations:

CHF	Critical heat flux
HPLWR	High pressure light water reactor
HTC	Heat transfer coefficient
HTD	Heat transfer deterioration
ID	Inner diameter
OD	Outer diameter
SC	Super-critical
SCHT	Super-critical heat transfer

1. INTRODUCTION

The Super-Critical Water-Cooled Reactor (SCWR) is one of the candidate designs considered by the Generation IV International Forum as an innovative nuclear energy system with increased safety, more compact size, lower cost of energy production and reduced volume of nuclear waste, compared to existing systems. The present research is in support of the Canadian National Program for the development of the SCWR.

Heat transfer in a fluid at near-critical and supercritical pressures has distinct characteristics, which are not present at subcritical pressures. When the fluid pressure and temperature are near their critical values, or when the pressure is supercritical and the temperature is near its pseudo-critical value, small changes in temperature cause large changes in the thermo-physical properties and the heat transfer may undergo enhancement or deterioration. Of particular importance for the safety of SCWR is the phenomenon of heat transfer deterioration (HTD) where the supercritical heat transfer (SCHT) falls below that expected during normal forced convective heat transfer, and where therefore higher wall temperatures can be encountered. As will be shown in this paper, super-critical heat transfer (SCHT) enhancement can be very effective in suppressing the heat transfer deterioration. Heat transfer deterioration most probably happens due to the existence of a less turbulent flow near the wall.

The objective of this paper is to examine options of SCHT enhancement. In particular the following three options will be examined: (i) choosing operating conditions which would avoid the deteriorated heat transfer regime, (ii) using turbulence promotion devices to delay or suppress the occurrence of deteriorated heat transfer, and/or to enhance the heat transfer coefficient, and (iii) optimization of the fuel geometry such as to minimize sheath temperatures while maintaining fuel integrity. These three options are discussed in detail below.

2. OPTIMUM OPERATING CONDITIONS

Flow conditions strongly affect the SC fuel sheath temperatures. To ensure no overheating of the fuel elements, it is therefore important that flow conditions be chosen to minimize the possibility of HTD occurrence, and maximize the presence of heat transfer enhancement. The next section will describe how flow obstacles could delay or avoid the occurrence of HTD. This section will thus limit itself to a discussion of the conditions where HTD is more likely to occur.

Although the presence of heat transfer deterioration during upward flows in heated tubes has been documented by many investigators, there is no consensus in the literature on how to identify the onset of deterioration, and to separate their experimental SCHT data into those with and without HTD. Although many authors consider heat transfer to be deteriorated when the local heat transfer coefficient drops below the corresponding prediction of an empirical correlation developed for normal heat transfer (e.g. Dittus-Boelter Eqn.), they do not quantify the degree of heat transfer reduction required for reaching the deteriorated heat transfer region (**Zahlan *et al.* (2013)**).

Licht *et al.* (2008) sorted their SCHT results into three groups: a normal, an uncertain region, and a deteriorated region. This classification was based on the observation that some runs experienced steep localized wall temperature spikes (they were considered as runs with HTD), while other runs displayed small localized temperature increases (they were considered runs with possible HTD and were placed in the uncertain group).

Several criteria have been proposed to predict the onset of HTD. **Jackson *et al.* (2011)** proposed a model to predict HTD for a heated vertical tube based on the buoyancy (which has a partial laminarization effect on the turbulence) exceeding a certain value. **Jackson *et al.* (2011)** also developed a correlation to predict the acceleration effect on heat transfer; this correlation specifies that there is no a significant HTD effect if the acceleration parameter Ac_b^* were

$$Ac_b^* = \frac{Q_b^*}{Re_b^{1.625} Pr_b} < 2\times10^{-6}$$

,while **Zahlan *et al.* (2013)** reported that the Ac_b^* corresponding to their CO_2 data was 3.5×10^{-7}.

For a given mass flux, an onset of heat transfer deterioration correlation was developed by **Kim *et al.* (2005)**; where the HTD occurs when the heat flux exceeds some value of q_{onset}.

$$q_{onset} = 2\times10^{-4} G^2$$

In addition to the strong effect of heat flux and mass flux, flow geometry also affects the HTD boundary. Fig. 1 shows the effect of heated tube diameter on the HTD boundary for a given supercritical pressure for water (**Eter *et al.* 2013**).

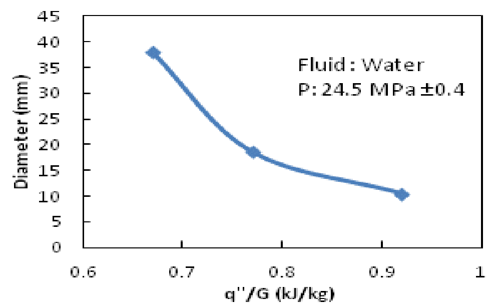

Figure 1. Dependence of the onset of deterioration condition on tube diameter (Eter et al. 2013).

3. EFFECT OF TURBULENCE PROMOTORS ON SCHT

This section examines the effectiveness of turbulence promotion devices to delay the occurrence of deteriorated heat transfer and /or to enhance the heat transfer coefficient with a minimal increase in channel pressure drop. This section is primarily based on a recent review of the literature of the impact of spacing devices in the near-critical and supercritical region (**Eter *et al.* 2013**) .

The objective of this section is to provide a better understanding of the flow obstruction effect on heat transfer and pressure drop at supercritical pressures.

3.1 Experimental Studies

Pioro and Duffey (2007), devoted Chapter 9 of their book to SCHT enhancement; they reviewed various investigations between 1968 to 2005. These and additional studies are discussed below The test conditions of these experiments are provided in Table 1 while the observed impact of the flow obstructions on SCHT is summarized in Table 2.

Shiralkar and Griffith (1970), observed that the swirl generated by a twisted tape inside a tube has a significant effect on heat transfer to upwardly flowing CO_2. However, the HTD was not completely eliminated, but its occurrence was switched to higher heat fluxes in both up-flow and down-flow.

Ackerman (1970), investigated heat transfer to water at supercritical pressures in vertical smooth and ribbed tubes over a wide range of pressures, mass fluxes, and heat fluxes. The author concluded that the usage of ribs result in the elimination of HTD at supercritical pressures as shown in Fig. 2.

Lee and Haller (1974), observed similar results to those of Ackerman when they tested flow inside a ribbed tube with no HTD at super critical pressures. They attributed that to the higher turbulence level of the flow, hence the heavier density bulk fluid from the core can move towards the wall much easier than in smooth tubes. They were able to test the ribbed tubes at 50% - 100% higher heat fluxes than smooth tubes without the occurrence of HTD.

Kamenetskii (1980) as reported by Pioro and Duffey (2007)**,** investigated the effect of the twisted tape on HTC in a horizontal tube at supercritical pressures. The results showed that the installation of the twisted tapes inside a bare tube reduced the flow stratification (the temperature difference between top and bottom of the tube was reduced compared to the bare tube), hence the heat transfer coefficient was increased. Although the twisted tapes are very efficient heat transfer enhancing devices in circular tubes, they cannot easily be employed in fuel bundle geometries.

Fedorov *et al.* (1986) as reported by Pioro and Duffey (2007) found a significant heat transfer enhancement in hydrocarbon flow at supercritical pressures by installing internal circumferential ribs inside a smooth tube; the ribbed tube resulted in a more uniform wall temperature profile along the heated length compared to the smooth tube and the enhancement increased with increasing heat flux.

Kalinin et al. (1998) as reported by Pioro and Duffey (2007), conducted experiments in supercritical kerosene flowing upward in a smooth and ribbed tube. The effect of different Reynolds numbers and geometrical parameters of the rib (depth and pitch) were studied. The results show that the Nusselt number, for a larger rib (d/D=0.85) is 35% - 90% higher than a smaller rib (d/D=0.95). As expected the pressure

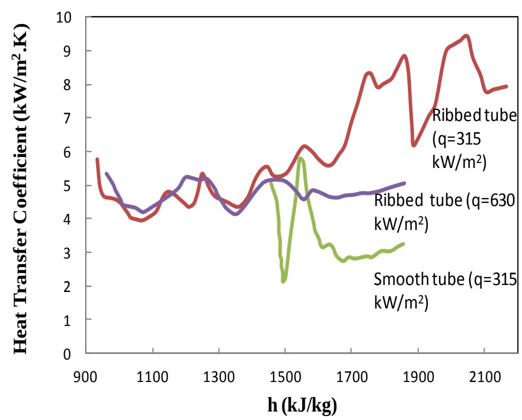

Figure 2. Heat transfer coefficients vs bulk enthalpy in ribbed and bare water-cooled tubes at G= 406.8 kg/(m^2.s) and P=24.8 MPa (Ackerman 1970).

drop increases with decreasing d/D, where the friction factor ratio (ζ / ζ_{Smooth}) is three times higher than that associated with d/D=0.95. Also, Nu/Nu_{Smooth} decreases as rib pitch increases.

Chen (2004) as reported by Pioro and Duffey (2007), investigated heat transfer to water flowing upward in a ribbed tube at supercritical pressures for a mass flux range of 400-1800 kg/(m^2.s), a heat flux range of 200-800 kW/m^2, a pressure range of 23-27 MPa, and a heated length of 1 m. He proposed correlations to calculate SCHT for upflow of water in a ribbed tube.

Bastron *et al*. (2005) investigated different methods to enhance heat transfer in HPLWRs. They found that using of an artificial surface roughness or a staircase type grid spacer can increase the HTC by 100%.

Dyadyakin and Popov (1977) as reported by Pioro and Duffey (2007), carried out experiments with water at supercritical pressures in a tight 7-rod bundle (rod OD = 5.2 mm) equipped with helical fins. They derived the following correlation:

$$Nu_x = 0.021 \mathrm{Re}_x^{0.8} \overline{\mathrm{Pr}}_x^{0.7} \left(\frac{\rho_w}{\rho_b}\right)_x^{0.45} \left(\frac{\mu_b}{\mu_{in}}\right)_x^{0.2} \left(\frac{\rho_b}{\rho_{in}}\right)_x^{0.1} \left(1 + 2.5\frac{D_{hy}}{x}\right)$$

where x is the distance along the heated length.

Bae *et al*. (2011), investigated the effect of inserting a helical wire (1.3 mm OD, 100 mm pitch) on heat transfer in an 6.32 mm ID tube during upflow of CO_2. They observed a 100% heat transfer enhancement compared to the plain tube data. The maximum heat transfer enhancement occurred in a region where the bulk temperature was near the pseudo-critical temperature. The enhancement effect of the wire gradually disappeared for bulk temperatures higher than the pseudo-critical temperature. For h > 400 kJ/kg, both the wall and the bulk temperatures were much higher than the pseudo-critical temperature, and the property variations across the channel no longer showed drastic changes. Here the heat transfer can be predicted by the simple Dittus–Boelter correlation. Also, the Dittus–Boelter correlation can be applied in the region of h < 250 kJ/kg where both the wall and the bulk temperatures were well below the pseudo-critical temperature.

The maximum heat transfer enhancement was 3.64 times heat transfer value of the plain tube, at h=300 kJ/kg which is close to the pseudo-critical point (h_{pc}=340 kJ/kg). Fig. 3 shows that the heat transfer coefficient decreases with increasing the heat flux at a given pressure and mass flux. However, in the region where the enthalpy is more than 450 kJ/kg, no significant effect of heat flux variation on the heat transfer coefficient was observed.

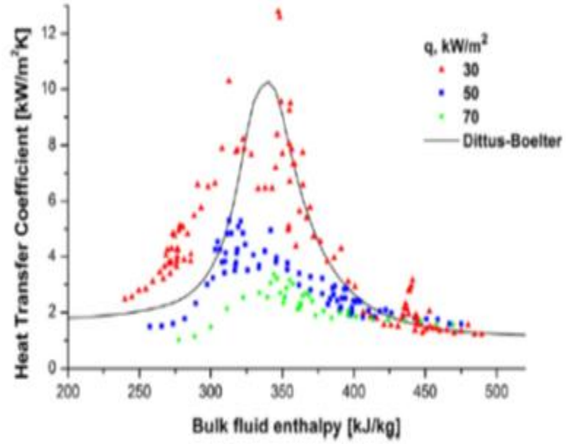

Figure 3. Effect of heat flux on CO_2 heat transfer in a tube with a helical wire insert at P=7.75 MPa and G=400 kg/(m^2.s), (Bae *et al*. ,2011).

Bae *et al.* (2011) also noted a significant increase in HTC with mass flux, especially near pseudo-critical region, which could not be explained by the expected increase in normal heat transfer with mass flux, the implication being that the increase in mass flux suppressed the HTD occurrence at the lower mass flux in the near critical region.

Wang *et al*. (2011, 2012), investigated experimentally and numerically the heat transfer characteristics of supercritical water flowing in an annular channel. A spiral spacer having 3 mm OD and 100 mm length was installed inside annular channels having a 6 mm and 4 mm gap; experiments were performed at various flow conditions. They observed that the axial distance, over which the spacer enhancement effect is effective, depends strongly on flow conditions. In addition, the spiral spacer suppressed HTD which occurred at high q"/G ratios.

Wang *et al*. (2011, 2012), observed that the HTCs for the 4 mm gap bare-annular channel were higher than those of the 6 mm gap bare-annular channel at q"/G= 0.86 and 0.82 for the 6 mm and 4 mm gap annular channels, respectively. This agrees with the reverse proportional relation between HTC and D as HTC ~ $D^{-0.2}$. However, the heat transfer coefficients for the spacer-equipped geometry are 25% - 80% higher than those of the geometry without spacer for the 6 mm gap annular channel with q"/G=0.86 as shown in Fig. 4 and 7% - 33% for the 4 mm gap annular channel q"/G=0.82 as shown in Fig. 5. As was also shown by **Bae *et al.* (2011)**, at temperatures away from the pseudo-critical temperature, the enhancement is the least. However, at a bulk temperature close to the pseudo-critical temperature, the heat transfer enhancement is most effective; here the spacer increases the HTC by about 80% .

Wang *et al*. (2012) found that the enhancement of heat transfer is relatively short-lived and decays with the increasing distance downstream from the spacer, which is consistent with the subcritical study of **Yao *et al*. (1982)**. However, the spiral spacer was able to suppress the deterioration in heat transfer at high ratios of heat flux to mass flux (q"/G=1.71).

Mori *et al*, (2012), investigated heat transfer characteristics of HCFC-22 flowing vertically upward and downward in a tube, a 3-rod bundle and a 7-rod bundle. The results showed that the heat transfer characteristics for upflow are similar in the tube and the 3-rod bundle where the heat transfer enhancement in the bundle is small at a low heat flux, while at a high heat flux, the heat transfer characteristics are different due to deterioration which happens in the tube but disappears in the bundle near the grid spacer. With down-flow, no heat transfer deterioration was observed in either the tube or the bundle. The HTC trends for the 7-rod bundle are similar to those of the 3-rod bundle.

Mori *et al*. (2012) showed the grid spacer effect on the normalized HTC (α/α_o) vs. L/Dh plot for up-flow in Figure 6; the heat transfer coefficients are enhanced just downstream from the spacer but the enhancement decays gradually over a downstream distance of 60 Dh to 70 Dh. However, for down-flow, the impact of distance on α/α_o is seen to be minor. Although **Mori et al. (2012)** developed a correlation which represents the effect of the grid spacer on the normalized HTC for up-flow,

$$\frac{\alpha}{\alpha_o} = 1 + 5.55\varepsilon^2 e^{-0.05 L/D_h}$$

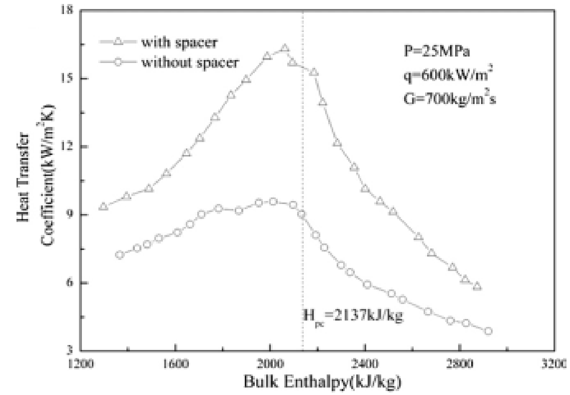

Figure 4. Comparison of the water heat transfer coefficients with and without spacers for 6 mm gap annular channel; (Wang *et al*. 2012).

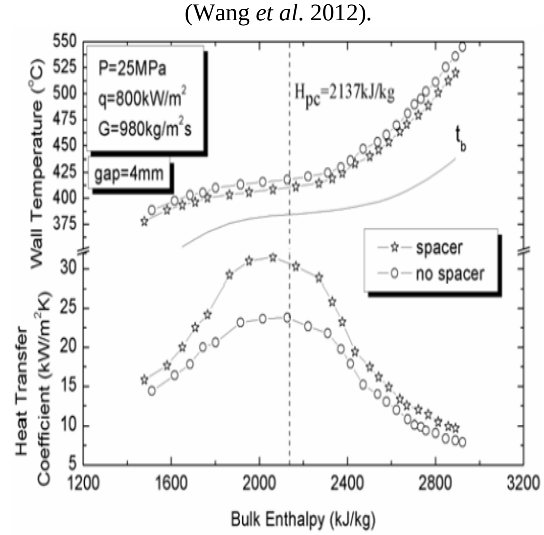

Figure 5. Comparison of the water heat transfer characteristics with and without spacers for the 4 mm gap annular channel (Wang *et al*, 2011).

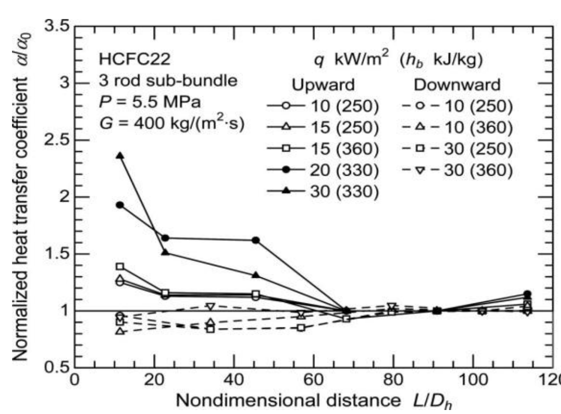

Figure 6. Effect of grid spacer on heat transfer. (Mori *et al*. (2012)).

this equation is expected to be an over simplification for SCHT as it is independent of heat flux and mass flux, and therefore is unlikely to be valid in the DHT region.

Hongzhi *et al*. (2009), performed experiments in a test section equipped with a helical wire wrap located inside a 15 mm wide square vertical channel. He found that at G=1200 kg/(m^2.s), the maximum enhanced heat transfer coefficient is 150% higher than that occurring away from the pseudo-critical temperature region. However, a significant deterioration in heat transfer occurred at G = 500 kg/(m^2.s), where the HTCs are less than 5 kW/(m^2.K) when $T_w>T_{pc}>T_b$, while the HTC is almost independent of heat flux when $T_w<T_{pc}$ or when $T_{pc}<T_b$ (i.e. at conditions away from the pseudo-critical region).

3.2 Numerical Studies:

The effect of flow obstacles on SCHT has also been investigated numerically using CFD tools.

Chandra *et al*. (2009), investigated numerically ,using SST k-ω turbulence model, the spacer effect on SCHT during up-flow of CO_2 inside an annulus equipped with a square ring (Height / Inner diameter = 0.5). The results show a significant effect of the square ring location on the length of the region over which heat transfer enhancement occurs: when the ring is located at 200 mm from the inlet, the heat transfer was enhanced up to L/D_{hy} = 50 downstream from the ring, while for a location of 50 mm from the inlet enhancement was felt up to L/D_{hy} = 25 downstream from the ring. **Yongliang *et al.* (2010)** who investigated the ring effect in a tube, found that the enhancement was up to L/D=62.5 downstream from the ring located midway of the tube.

Bae *et al.* (2010), investigated numerically, using RNG k-ε turbulence model, the effect of inserting a helical wire (1.3 mm OD, and 105 mm pitch) in an 6.32 mm ID tube on SCHT during up-flow of CO_2 and compared the results with those obtained from KAERI's experiments (**Kim *et al.* 2006**). Fig. 7 shows that, while the experiment showed a strong enhancement effect of the wire (or a pronounced HTD without the wire), the numerical model showed a very weak effect of the wire.

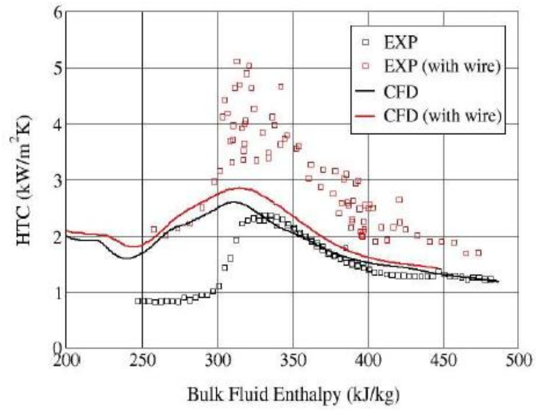

Figure 7. Comparison between the experimental results of CO_2 and the numerical results (lines). P=8.12 MPa, G = 400 kg/(m^2.s), and q''= 50 kW/m^2. (Bae *et al.*, 2010).

Shan and Chen (2010), investigated numerically, using ATHAS sub-channel analysis code, the effect of a wire wrap spacer and a grid spacer inside a SCWR fuel bundle having 40 pins in a square arrangement. The numerical results suggest that the wire wrap is much more effective than the grid spacer since the peak cladding temperature and pressure drop associated with the wire wrap spacer are lower than that associated with the grid spacer, and the coolant temperature is more uniform across the bundle. As expected, the most uniform temperature profile and the highest pressure drop occurred at the smallest wire pitch (H_{wire}/D_{rod}=20). **Wang *et al*. (2012)** suggested that the enhancement occurs when the fluid is subject to rotational forces due to the swirling effect of spacer. Therefore, the turbulence intensity near the spacer is expected to be higher than that without spacer.

Table 2 summarizes the experimental observed and numerical predicted effects of the flow obstacles on SCHT.

4. FUEL GEOMETRY OPTIMIZATION

The Canadian version of the SCWR is a pressure-tube type reactor design with light water as coolant and heavy water as moderator. As the coolant passes through the core region, the high-temperature and pressure keeps it in a

Table 1: Parameters range of experiments investigating the flow obstacle effect on SC and near-critical heat transfer.

Reference	Heated Length m	Test section geometry details	Conditions			P MPa	P/P_{Pc}*	Working Fluid
			Mass flux kg/m^2 s	Heat flux kW/m^2	Flow direction			
Shitsman 1967	1.5	Tube with a twisted tape, D=16 mm	600-690	90-370	Up	24.3 - 25.3	1.1-1.14	Water
Ackerman 1970	1.83	Ribbed tube (from rib valley to rib valley), Six helical ribs Pitch, D=21.8 and 18 mm	406.8	315.3-630.5	Up	24.8	1.13	Water
Kamenetsky Shitsman 1970	3	SS tube with a twisted tape δ_w =0.8 mm, spiral shape with S/D=15; D=22 mm	540	190-1300	Up	24.5	1.11	Water
Lee and Haller (1974)	4.57	SS tube with ribs, D=38.1 mm	542-2441	250-1570	Up	24.1	1.09	Water
Chen 2004	1	SS ribbed tube with ribs (Height =0.81mm , Pitch= 20.5 mm , Width = 9mm); D_{avg}=15.24 and D_{Ext}=28mm	400	300	Up	24	1.086	Water
Shiralkar and Griffith 1968	1.524	Tube with a twisted tape, Twist is one turn of 360° in four diameters, D=6.35 mm	Re $2.67x10^5$ $8.35x10^5$	50.24-452.16	Up	7.58	1.03	Carbon dioxide
Ankudirov and Kurgarov 1981	1.84	Helical wire inside a tube,D= 8 mm	2100-3200	Up to 1540	Up	7.7	1.04	Carbon dioxide
Bae *et al*.(2011)	2.65	1.3 mm OD Helical wire, 100 mm pitch, D=6.32 mm	400 to 1200	30-90	Up	7.75 and 8.12	1.05 and 1.1	Carbon dioxide
Wang *et al*. (2012)	1.4	Spiral spacer, 100 mm length, inside a 4 and 6 mm gap of annular channel, D_{hy}=8 and 12 mm.	350-1000	200-1000	Up	23-28	1.04-1.3	Water
Silin *et al*. 1993	-	Bundles, Drod=4 and 5.6mm, rod pitch=5.2 and 7mm.	350-5000	180-4500	Up	23.5 - 29.4	1.06-1.33	Water
Dyadyakin and Popov 1977	0.5	Tight bundle (7 rods), having Drod=5.2 mm and L=0.5 m, every rod has 4 helical fins thickness 1 mm, helical pitch= 400mm located in a hexagonal pressure tube.	500-4000	<4700	-	24.5	1.11	Water

Table 2: Experimentally observed and numerically predicted effects of the flow obstacles on SCHT.

Reference	Obstacle type and geometry	Effect of the flow obstacles
Shiralkar and Griffith (1970)	Twisted Tape in a tube	For q"= 146.59 kW/m^2, no HTD occurred for G > 1139 kg/(m^2.s), while HTD occurred for G $\leq$1356 kg/(m^2.s) in the bare tube.
Ackerman (1970)	Ribs in a tube	For P=24.8 MPa and G= 407 kg/(m^2.s), no HTD occurred at q"=630 kW/m^2, while HTD occurred at q" $\geq$315 kW/m^2 in the bare tube.
Lee and Haller (1974)	Ribs in a tube	Heat fluxes can be increased by 50%-100% over smooth tube values without occurrence of HTD.
Kamenetskii (1980)	Twisted Tape in a tube	Reduced flow stratification in horizontal tubes.
Fedorov *et al*. (1986)	Ribs in a tube	More uniform wall temperature profile along the heated length and circumferentially compared to the smooth tube; HT enhancement increased as heat flux increased.
Bae *et al*. (2011),	Helical wire in a tube	>100% increase in HTC compared to that in the plain tube when T_b is very close to T_{pc}. At enthalpies much higher than h_{PC} , HTC becomes independent of the heat flux.
Wang *et al*. (2012)	Spacer in annulus	25%-80% increase in HTC compared to that in the annulus without a spacer. The least enhancement occurs when T_b is away from the pseudo-critical temperature, i.e. $T_w<T_{pc}$ or when $T_{pc}<T_b$
Mori *et al*. (2012)	Grid spacer in a 3-rod and 7-rod bundles	Downstream from the grid spacer, significant HT enhancement was observed in the upward flow, but not in the downward flow.
Hongzhi *et al*. (2009)	Helical wire in a square annular channel	For P= 25 MPa and q"= 400 kW/m^2, the maximum heat transfer enhancement occurred at G=1200 kg/(m^2.s), while HTD occurred at G=500 kg/(m^2.s). At P= 23 MPa, the heat transfer enhanced by 100% compared to that at P=25 MPa, but the HTD occurred at low mass fluxes compared to P=25 MPa.
Chandra *et al*. (2009)	Ring in an annulus, and helical wire in a tube and annulus (numerical)	Effect of the ring was short-lived over a specific region downstream from the ring: when the ring is located at 200 mm distance from the inlet, the heat transfer was enhanced up to L/D_{hy} =50 downstream from the ring, while in the case of 50 mm distance from the inlet it was up to L/D_{hy} =25 downstream from the ring.
Bae *et al.* (2010)	Helical Wire in a tube (numerical)	HTD cannot be predicted by CFD at high heat fluxes compared to experiment. However, the predicted HTC in the wired tube increased about 10% compared to the bare tube.
Yongliang *et al.* (2010)	Ring in a tube (numerical)	Reduction in the wall temperature for a distance downstream from the ring.
Shan and Chen (2010)	Wire wrap and grid spacer in a SCWR (numerical)	Wire wrapped SCWR has a uniform coolant temperature, hence it has lower cladding temperature compared to a SCWR with grid spacers. Wire wrap also has lower pressure drop.

supercritical state. Because of the lack of phase change in the core at high operating temperature and pressure, design criteria based on the critical heat flux (CHF) concept are not applicable. The commonly accepted practice is to specify cladding temperature limits that must be met during various events. Therefore, the heat transfer coefficient between cladding and supercritical water coolant needs to be predicted accurately. On the other hand, thermo-physical properties vary strongly in the vicinity of the pseudo-critical point, which may result in deterioration of heat transfer. Therefore, it is important to analyze flow and enthalpy distributions within the Canadian SCWR bundle and to predict accurately the cladding temperatures.

A variety of radial and axial power profiles in fuel channels exists and should be taken into account in the analysis while optimizing the fuel assembly, as changes in radial and axial power profiles would impact the heat transfer characteristics along the fuel. The ASSERT-PV subchannel code has been applied to generate data on fuel sheath temperatures for different geometries and for different axial and radial power profiles (the power profiles were established using physics codes). For the optimization of the fuel assembly, radial and axial power profiles that correspond to the beginning and end of cycles in the hot channel (i.e. channel where radial power peaking factor in the overall core is the largest) are applied.

Since the Canadian SCWR and fuel channel design underwent changes throughout the research period, the optimization of the fuel assembly was made based on *comparing* the solutions satisfying the requirements of fuel, thermalhydraulics and physics; such as low linear element ratings, low fuel sheath temperatures and negative coolant void reactivity (CVR).

Figure 8 shows the latest design of the fuel assembly and the fuel channel. The fuel assembly design consists of two rings, with 32 elements in each ring and a central flow tube. The solid central pin in the old Canadian SCWR designs has been replaced with a coolant flow tube through which the coolant enters and flows downwards and changes the direction when it enters to the fuel assembly section. **Pencer *et al.* (2013)** and **Dominguez *et al.* (2013)** have shown that the two-ring design with the central flow tube and 31 elements in each ring (see Figure 9 – the flow is in-plane in the flow tube while looking downward from the top of the reactor, and out-plane in the fuel assembly section, between the flow tube insulator and the liner tube) has improved thermalhydraulics, and physics characteristics of the fuel assembly compared to the previous designs of the Canadian SCWR fuel assemblies under steady-state conditions. This design has subsequently been further refined with 32 elements in each ring to provide better physics and thermalhydraulics characteristics. It should be noted that while optimizing the fuel geometry, spacing devices and heat losses from the fuel channel were not included. It is expected that the results are conservative without taking into account these effects: the sheath temperatures are expected to be lower when the spacer effects are included. It should also be noted that the calculated fuel sheath temperatures without incorporating these effects are below the design criteria of 850 °C.

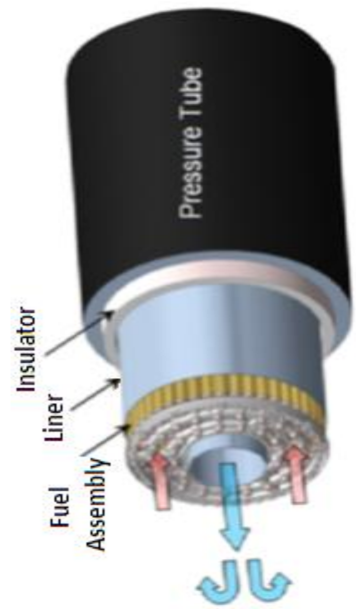

Figure 8. Cut-away view of the Canadian SCWR fuel bundle design in the High Efficiency Re-entrant Channel.

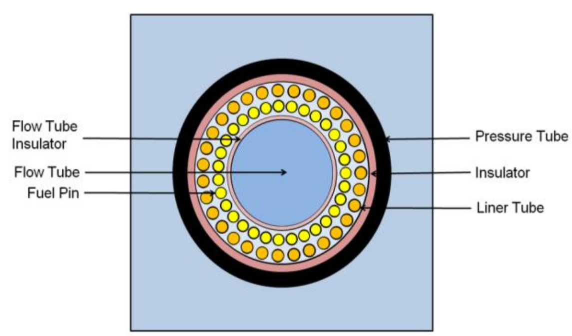

Figure 9. Cross sectional view of the Canadian SCWR fuel bundle and fuel channel design.

5. CONCLUSIONS AND FINAL REMARKS:

- From the limited bundle SCHT literature available one could tentatively conclude that heat transfer deterioration (HTD) in bundles is unlikely to occur. Whether HTD can possibly occur at low mass fluxes and high heat fluxes in up-flow remains to be determined based on future experiments.
- The highest heat transfer enhancement due to a flow obstruction occurred near the pseudo-critical temperature where the enhancement can exceed 100% compared to geometries without flow obstructions.
- Heat transfer enhancement downstream of a grid spacers decays with distance from the grid spacer in up-flow; in down-flow however, the effect of grid spacers on heat transfer was less pronounced or sometimes absent altogether.
- Predictions of spacer effects on supercritical heat transfer using CFD codes were in reasonable agreement with experimental observations for normal flow conditions (conditions away from the pseudo-critical temperature), but were deemed to be of questionable accuracy for deteriorated heat transfer conditions.
- Heat transfer at subcritical pressures was enhanced over a distance of L/D~30-35 downstream from the flow obstruction, while at supercritical pressures, the enhancement appears to be effective over a significantly longer distance (L/D= 60-70).
- The vast majority of experiments were in tubes and annular channels instead of bundles; relevant experiments in bundles are needed to get a better indication of the flow obstruction effect on SCHT during upflow.

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