COMPARATIVE STUDY AND ADVANCEMENT ON A SUPERCRITICAL-WATER HEAT-TRANSFER CORRELATION FOR VERTICAL BARE TUBES

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Abstract

This paper presents an analysis of convective heat-transfer in water flowing in vertical bare tubes at supercritical conditions. A large dataset within conditions similar to those of SuperCritical Water-cooled Nuclear Reactors (SCWRs) was obtained from the Institute for Physics and Power Engineering (IPPE, Obninsk, Russia). This dataset was compared with existing heat-transfer correlations from the open literature, and a new more comprehensive heat-transfer correlation for predicting Heat Transfer Coefficient (HTC) values is proposed. A dimensional analysis was conducted to obtain a general form of empirical correlation using a combination of various dimensionless terms. This empirical correlation was verified using the experimental dataset obtained at the Normal Heat-Transfer (NHT) regime using statistical analysis. The final correlation showed the best fit for the experimental dataset within a wide range of flow conditions.

1. Introduction

SuperCritical Water-cooled nuclear Reactors (SCWRs) are high-pressure (~25 MPa) and high-temperature (outlet temperatures up to 625°C) reactors that will operate above the thermodynamic critical point of water (22 MPa and 374°C) (see Figure 1) [1], [2]. As part of the Generation-IV International Forum (GIF), SCWR concepts are currently under development worldwide. Figure 2 outlines the difference in the operating conditions (pressures, temperatures and entropies) of current generation reactor systems in comparison to SCWRs. Compared to existing Pressurized Water Reactors (PWRs), SCWRs would involve increasing the coolant pressure from 10 – 16 MPa to about 25 MPa, the inlet temperature to about 350ºC, and the outlet temperature to about 625ºC. The coolant would pass through the pseudocritical region before reaching the channel outlet [1].

1.1 SCWR Concepts

SCWRs can be divided into two subcategories: 1) Pressure-Vessel (PV) reactors, and 2) Pressure-Tube (PT) reactors. Currently, both Canada and Russia are working on the development of PT-reactor concepts. One of the main objectives for developing and utilizing SCWRs is that SuperCritical Water (SCW) Nuclear Power Plants (NPPs) offer an increased thermal efficiency, approximately 45 – 50%, compared to that of current generation NPPs (30 – 35%). Additionally, they allow for a decrease in capital and operational costs.

Generation-IV reactor concepts (see Table 1) under development at AECL [3] and RDIPE [4] have a main design objective of achieving major reductions in unit energy cost relative to existing PWR designs [5]. This approach builds on using existing SCW experience in operating fossil-fired thermal power plants. A major contribution to this energy cost reduction would result from boosting the outlet coolant temperature, thereby increasing the thermal efficiency of the NPP.

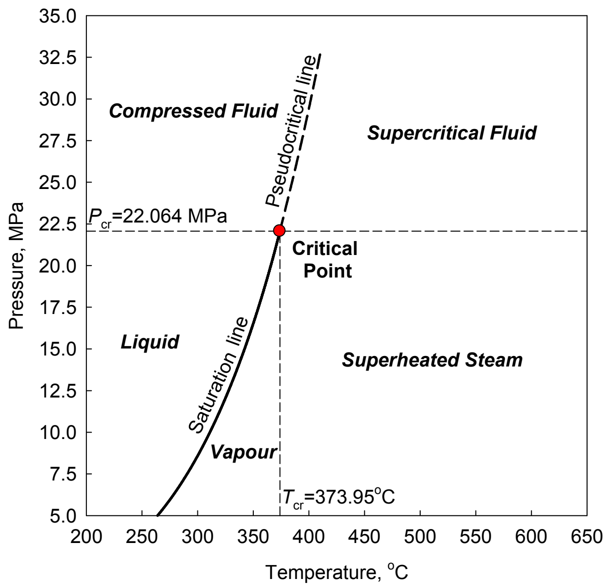


Figure 1 Pressure-Temperature Diagram for Water in Critical Region [2].

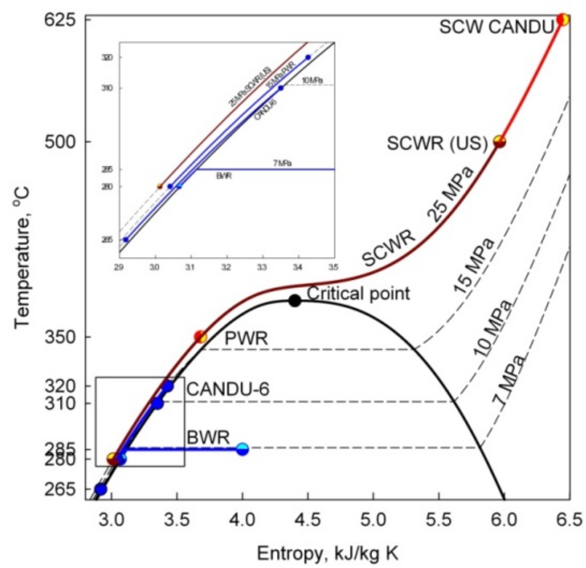


Figure 2 Temperature-Entropy Diagram Comparison of Current Generation Nuclear Reactors and SCWRs [1].

These reactors might use the direct cycle with the coolant from the reactor flowing directly to turbines. This feature allows for a simplified flow circuit in which steam generators, steam dryers, steam separators, etc. can be eliminated. A further benefit of using SCWRs is their ability to facilitate hydrogen co-generation, on an economical scale, through either thermochemical cycles or direct high-temperature electrolysis.

The current Canadian SCWR concept includes a fuel channel comprised only of a pressure tube insulated internally, which would enable the pressure tube to operate at temperatures close to that of the moderator. This fuel-channel design would be used for supercritical water heating from 350 to 625°C. A re-entrant fuel-channel design, allowing the pressure tube to operate at the supercritical water inlet temperature, might be used for a nuclear steam re-heat at subcritical pressures. The current heat-transfer evaluation has shown that PT SCWRs are feasible. A further study on conceptual thermal-design options for pressure-tube SCWRs can be found in [6].

Supercritical fluids have unique properties [7], [8]. It is well established that thermophysical properties of any fluid, including water, experience significant changes within critical and pseudocritical regions. Figure 3 illustrates these variations for water passing through the pseudocritical point at 25 MPa, the proposed operating pressure of SCWRs.

The most significant changes in properties occur within ±25°C from the pseudocritical temperature (384.9°C at 25 MPa). The National Institute of Standards and Technology (NIST) [9] Reference Fluid Properties (REFPROP) software was used to calculate these thermophysical properties. Crossing from high-density fluid to low-density fluid does not involve a distinct phase change. Phenomena such as dry-out (critical heat flux) are therefore not applicable. However, at supercritical conditions, a Deteriorated Heat-Transfer (DHT) regime may exist [1].

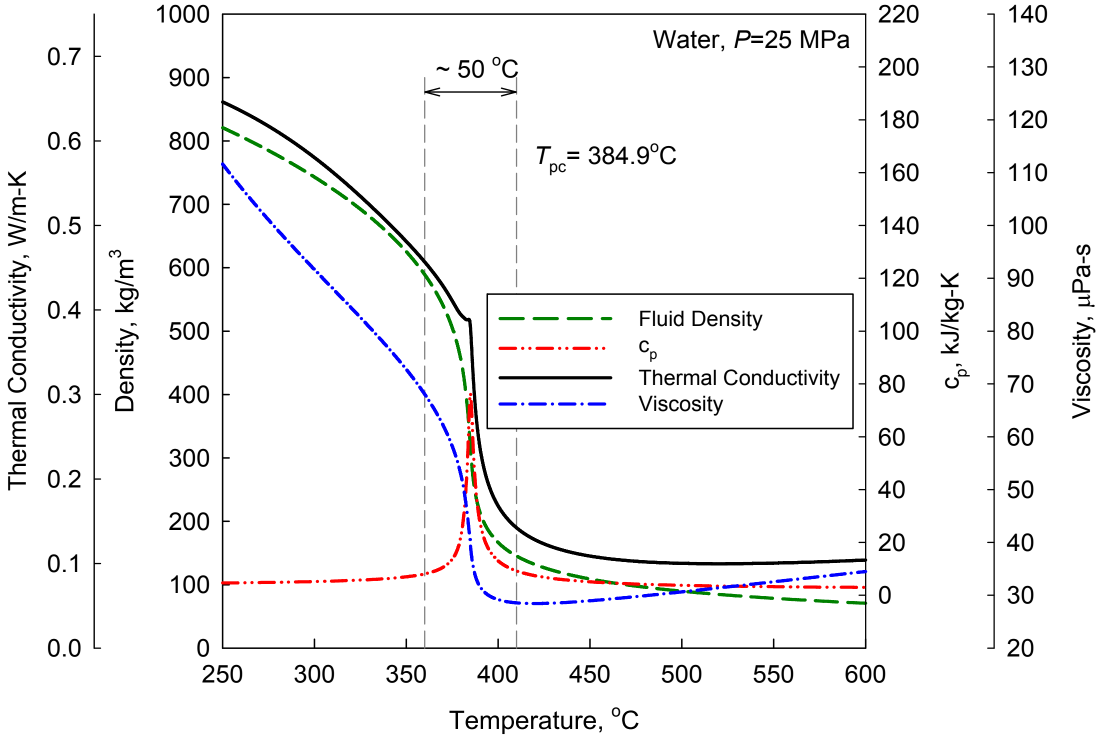


Figure 3 Selected Properties of Supercritical Water at Pseudocritical Point [2].

Table 1 lists parameters of current PT-SCWR concepts being developed by AECL (Canada) and RDIPE (Russia).

Table 1 Major Parameters of SCW CANDU® and KP-SKD Nuclear-Reactor Concepts [1], [10].

Parameters	SCW CANDU	KP-SKD
Reactor type	PT	PT
Thermal power, MW	2540	1960
Electric power, MW	1220	850
Thermal efficiency, %	48	42
Pressure, MPa	25	25
Inlet temperature, °C	350	270
Outlet temperature, °C	625	545
Mass flow rate, kg/s	1300	922
Number of fuel channels	300	653
Number of fuel elements in bundle	43	18
Length of bundle string, m	6	–
Maximum cladding temperature, °C	850	700

Comparisons of selected thermophysical properties profiles for water along the fuel-channel heated length for a non-uniform Axial Heat Flux Distribution (AHFD) are shown in Figures 4 and 5 [6].

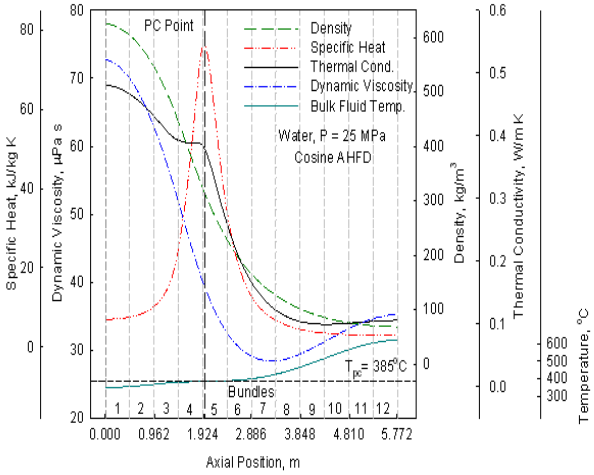


Figure 4 Bulk-Fluid Temperature and Thermophysical Properties Profiles for Water along Heated Length of SCWR Fuel Channel [2].

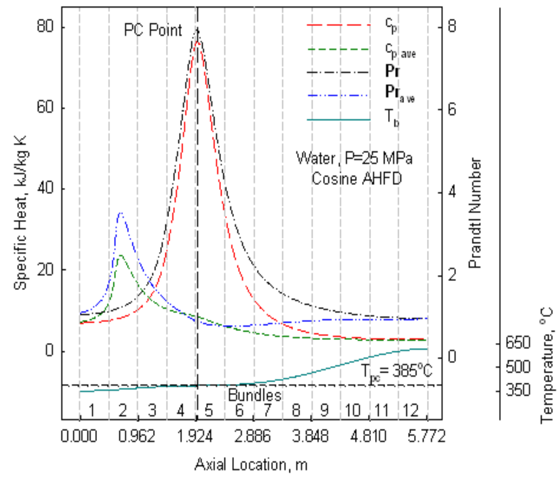


Figure 5 Prandtl Number and Specific Heat Profiles for Water along Heated Length of SCWR Fuel Channel [2].

The following bulk-fluid thermophysical properties were calculated: (a) density; (b) specific heat; (c) thermal conductivity; (d) dynamic viscosity; and Prandtl number. In addition to the regular bulk-fluid properties, the cross-sectional average specific heat and the corresponding average Prandtl number, which are used in various supercritical heat-transfer correlations (for details, see [1]), were shown in Figure 5 for reference purposes. The bulk-fluid temperature was calculated based on the heat-balance method.

This paper presents selected results on heat transfer to supercritical water flowing upward in a 4-m long vertical bare tube. Further results and analysis of this dataset can be found in [11] and [12]. The objective of this paper was to verify several well-known heat-transfer correlations for vertical bare tubes with a recent heat-transfer dataset. In addition, it was determined that an updated correlation for forced convective heat-transfer to supercritical water in a bare vertical tube could be developed and is presented in the following section.

2. Background

Currently, there is just one supercritical-water heat-transfer correlation for fuel bundles. This correlation was obtained for supercritical water flowing in a 7-element helically-finned bundle designed by Dyadyakin and Popov [1]. However, heat-transfer correlations for bundles are usually very sensitive to bundle design. Therefore, this correlation cannot be applied to other bundle geometries. To overcome this problem, a wide-range heat-transfer correlation based on bare-tube data can be developed as a conservative approach. This process is based on the fact that HTCs in bare tubes are generally lower than those in bundle flow geometries in which heat transfer is enhanced with appendages (endplates, bearing pads, spacers, buttons, etc.).

A number of empirical generalized correlations, based on experimentally obtained datasets, have been proposed to calculate the HTC in forced convection for various fluids including water at supercritical pressures. These bare-tube-based correlations are available in various literature sources, however, differences in HTC values can be up to several hundred percent [1].

2.1. Existing Correlations

The most widely used heat-transfer correlation at subcritical pressures for forced convection is the Dittus-Boelter correlation (1930) [13]. McAdams (1942) [14] proposed to use the Dittus-Boelter correlation in the following form for forced-convective heat transfer in turbulent flows at subcritical pressures (this statement was based on the recent study by Winterton [15]):

$$\mathbf{Nu}_{\mathrm{b}} = 0.0243\,\mathbf{Re}_{\mathrm{b}}^{0.8}\mathbf{Pr}_{\mathrm{b}}^{0.4} \tag{1}$$

Later, Eq. (1) was also used at supercritical conditions. According to Schnurr et al. [16], Eq. (1) showed good agreement with experimental data for supercritical water flowing inside circular tubes at a pressure of 31 MPa and low heat fluxes. However, it was noted that Eq. (1) might produce unrealistic results within some flow conditions, especially near the critical and pseudocritical points, because it is sensitive to properties variations. In general, this classical correlation was used extensively as the basis for various supercritical heat-transfer correlations. Therefore, the Dittus-Boelter correlation was used in the following form, for reference purposes:

$$\mathbf{Nu}_{\mathrm{b}} = 0.023\,\mathbf{Re}_{\mathrm{b}}^{0.8}\mathbf{Pr}_{\mathrm{b}}^{0.4} \tag{2}$$

Equation (2) is the most widely used interpretation of the original Dittus-Boelter correlation [18].

An analysis performed by Pioro and Duffey [1] showed that the Bishop et al. correlation was obtained within the same range of operating conditions as those for SCWRs. Bishop et al. (1964) [17] conducted experiments in supercritical water flowing upward inside bare tubes and annuli within the following range of operating parameters: pressure 22.8 – 27.6 MPa, bulk-fluid temperature 282 – 527ºC, mass flux 651 – 3662 kg/m^{2}s and heat flux 0.31 – 3.46 MW/m^2. Their data for heat transfer in tubes were generalized using the following correlation with a fit of ±15%:

$$\mathbf{Nu}_{\mathrm{b}} = 0.0069\,\mathbf{Re}_{\mathrm{b}}^{0.9}\,\overline{\mathbf{Pr}}_{\mathrm{b}}^{0.66}\left(\frac{\rho_w}{\rho_b}\right)^{0.43}\left(1+2.4\,\frac{D}{x}\right) \tag{3}$$

Equation (3) uses the cross-sectional averaged Prandtl number. The last term in the correlation accounts for the entrance-region effect. In the present comparison, the Bishop et al. correlation was modified and used without the entrance-region term, because this term depends significantly on the particular design of the inlet of the bare test section:

$$\mathbf{Nu}_{\mathrm{b}} = 0.0069\,\mathbf{Re}_{\mathrm{b}}^{0.9}\,\overline{\mathbf{Pr}}_{\mathrm{b}}^{0.66}\left(\frac{\rho_w}{\rho_b}\right)^{0.43} \tag{4}$$

Swenson et al. (1965) [19] found that conventional correlations, which use the bulk-fluid temperature as a basis for calculating the majority of the thermophysical properties, did not work well. They suggested the following correlation in which thermophysical properties are based mainly on a wall temperature:

$$\mathbf{Nu}_{\mathrm{w}} = 0.00459\,\mathbf{Re}_{\mathrm{w}}^{0.923}\,\overline{\mathbf{Pr}}_{\mathrm{w}}^{0.613}\left(\frac{\rho_w}{\rho_b}\right)^{0.231} \tag{5}$$

Equation (5) was obtained within the following range: pressure 22.8 − 41.4 MPa, bulk-fluid temperature 75 − 576ºC, wall temperature 93 − 649ºC and mass flux 542 − 2150 kg/m^{2}s; and predicted their experimental data within ±15%.

Jackson (2002) [20] modified the original correlation of Krasnoshchekov et al. (1967) [21] (for details, see [1]), for forced-convective heat transfer in water and carbon dioxide at supercritical pressures, to employ the Dittus-Boelter type form for $\mathbf{Nu_o}$. Finally, the following correlation was obtained:

$$\mathbf{Nu_b} = 0.0183\ \mathbf{Re_b^{0.82}}\mathbf{Pr_b^{0.5}}\left(\frac{\rho_w}{\rho_b}\right)^{0.3}\left(\frac{\bar{c}_p}{c_{pb}}\right)^{n} \qquad (6)$$

Where the exponent n is defined as following:

$n = 0.4$ for $T_b < T_w < T_{pc}$ and for $1.2T_{pc} < T_b < T_w$; $n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right)$ for $T_b < T_{pc} < T_w$; and

$n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right)\left[1 - 5\left(\frac{T_b}{T_{pc}} - 1\right)\right]$ for $T_{pc} < T_b < 1.2T_{pc}$ and $T_b < T_w$.

2.2 Comparison of Heat-Transfer Correlations

Figure 6 shows two sample experimental runs at supercritical pressures and provides experimentally measured HTC values. Also, a comparison between experimental and calculated HTCs using the Dittus-Boelter, modified Bishop et al., Swenson et al. and Jackson correlations are plotted in this figure.

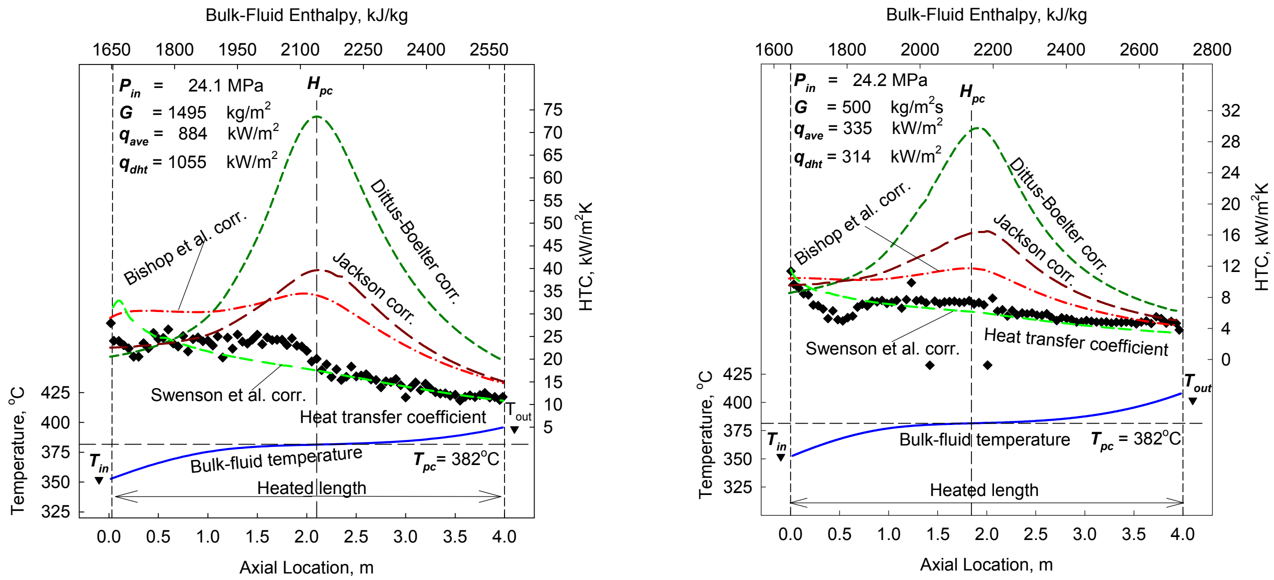
As can be seen from Figure 6, the Dittus-Boelter correlation provides a significant overestimation of the HTC values within the pseudocritical region, and thus, this correlation is unusable within a wide range of parameters. The modified Bishop et al. and Jackson correlations also tend to deviate substantially from the experimental data within the pseudocritical range. The Swenson et al. correlation provides a better fit for the experimental date than the previous three correlations within some flow conditions, but does not closely follow the experimental data within others [10].

It should be noted that all heat-transfer correlations presented in this paper are intended only for use at normal and Improved Heat-Transfer (IHT) regimes. None of the presented correlations can be used for the HTC prediction within the DHT regime.

For the DHT regime, an empirical correlation was proposed for the minimum heat flux at which this regime appears (for details, see [22]):

$$q_{dht} = 7.9 \cdot 10^{-4}\ G\left(\frac{P}{P_{cr}}\right)^{1.5}, \text{ MW/m}^2. \qquad (7)$$

A more thorough discussion and comparison of heat-transfer correlations can be found in [1].



(a) **(b)**

Figure 6 Temperature and HTC (Experimental and Calculated Values) Profiles along Heated Length of Bare Vertical Tube: (a) G = 1500 kg/m^{2}s and q = 884 kW/m^2; (b) G = 500 kg/m^{2}s and q = 335 kW/m^2 [10].

The majority of the reviewed empirical correlations were proposed in the 1960s and 1970s, when experimental techniques were not at the same level (i.e., advanced level) as they are today. Also, thermophysical properties of water have since been updated (for example, a peak in thermal conductivity in critical and pseudocritical points, within a range of pressures from 22.1 to 25 MPa, was not officially recognized until the nineties [1]).

Thus, this further emphasizes the necessity of developing a new or an updated correlation based on a new set of heat-transfer data and the latest thermophysical properties of water [9] within the SCWRs operating range.

3. Experimental Data

The experimental data used in the current paper [23] were obtained at the State Scientific Center of Russian Federation – Institute for Physics and Power Engineering Supercritical-Test Facility (Obninsk, Russia). This set of data was obtained within operating conditions close to those of SCWRs including a hydraulic-equivalent diameter.

3.1 Test Facility

The Supercritical-Pressure Test Facility SKD-1[23] was intended for SCW heat-transfer testing in bare tubes and other flow geometries within a wide range of parameters (pressures up to 28 MPa and power up to 0.6 MW). The experimental setup was made from stainless steel.

3.2 Test Matrix and Test Section Details

The data for this study was obtained within the following conditions: Vertical stainless steel (12Cr18Ni10Ti) smooth tube: D = 10 mm, δ_w = 2 mm, and L_h = 4 m; tube internal-surface roughness

R_a = 0.63 – 0.8 μm; and upward flow. Table 2 lists test-matrix parameters and Table 3, their uncertainties.

Table 2 Test Matrix.

P	T_{in}	T_{out}	T_w	***q***	***G***
MPa	**°C**	**°C**	**°C**	**kW/m²**	**kg/m²s**
24	320–350	380–406	<700	70–1250	200, 500; 1000; 1500

Table 3 Uncertainties of Primary Parameters.

Parameter	Maximum Uncertainty
Test-section power	±1.0%
Inlet pressure	±0.25%
Wall temperature	±3.0%
Mass-flow rate	±1.5%
Heat loss	≤3.0%

3.3 Data Analysis

The dataset includes 89 experimental runs with 81 data points per run. In total, over 7,200 points were collected. Abnormalities, such as defective thermocouple readings were removed from the dataset (for details, see Figure 7). The objective of this study was to develop an updated heat-transfer correlation for the normal heat-transfer regime. Therefore, data points within the DHT region were also removed from the dataset (for details, see Figure 8). This region is subject to future investigations. Also, the very first and last points of most datasets were removed. Temperatures at these outlying points were likely affected with the test-section clamps, which were at a lower/higher temperature than the heated part of tube. Overall, approximately 91% of the experimental data were used to develop the correlation.

4. Results

4.1 Developing the Correlation

It is well established that the general form of a correlation is as follows:

$$y = C_o \times t_1^{C_1} t_2^{C_2} \dots t_n^{C_n} \tag{8}$$

where C_o is the constant, t represents the various parameters that affect heat transfer and C_n represents the exponents.

In order to obtain a general empirical form of an equation governing HTCs, a dimensional analysis was conducted. It is well known that HTC is not an independent variable, and that HTC values are affected by fluid velocity, inside diameter and thermophysical properties. A review of trends in correlating heat-transfer data at supercritical pressures determined that there are nine parameters affecting heat transfer [1]. Table 4 lists these parameters, identified as essential for the analysis of heat-transfer processes, for forced convection, at supercritical conditions. Each of the identified parameters was broken down into the four primary dimensions of mass (M), length (L), time (T), and temperature (K).

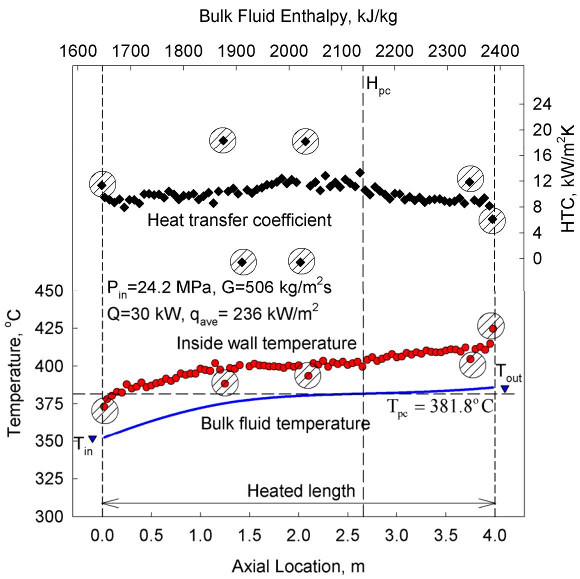


Figure 7 Sample Dataset with Outliers.

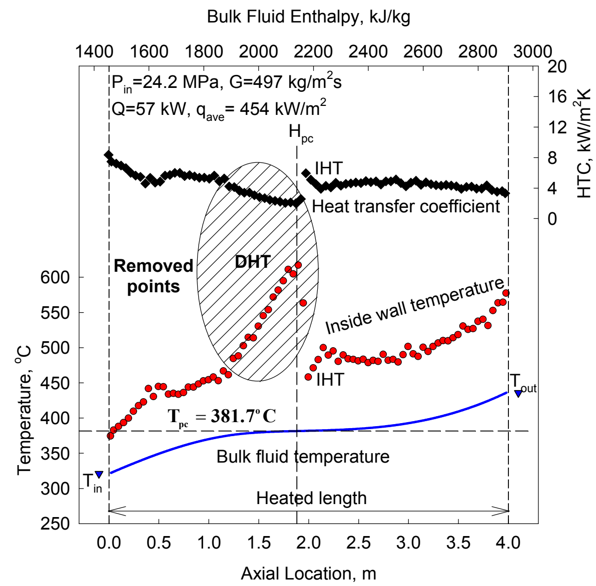


Figure 8 Sample Dataset with normal, DHT and IHT Regimes.

Table 4 Description of Various Heat-Transfer Parameters [10].

Variable	Description	SI units	Dimensions
HTC	Heat Transfer coefficient	$W/(m^2K)$	$MT^{-3}K^{-1}$
D	Diameter of the tube	m	L
ρ_w	Density of water at the wall	kg/m^3	ML^{-3}
ρ_b	Density of bulk fluid	kg/m^3	ML^{-3}
μ_w	Dynamic viscosity of water at the wall	Pa·s	$ML^{-1}T^{-1}$
μ_b	Dynamic viscosity of bulk fluid	Pa·s	$ML^{-1}T^{-1}$
k_w	Thermal conductivity of water at the wall	W/(m·K)	$MLT^{-3}K^{-1}$
k_b	Thermal conductivity of bulk-fluid	W/(m·K)	$MLT^{-3}K^{-1}$
c_p	Specific heat	J/(kg·K)	$L^2T^{-2}K^{-1}$
V	Characteristic velocity	m/s	LT^{-1}

The Buckingham *Π*-Theorem [24], using dimensionless pi-terms, was chosen for this analysis. This theorem is based on dimensional homogeneity, in which dimensionless pi-terms can be formed from the correlation variables. Thus, the following expression was produced for HTCs as a function of the identified heat-transfer parameters:

$$HTC = f\,(D\,,\, \rho_w\,,\, \rho_b\,,\, \mu_w\,,\, \mu_b\,,\, k_w\,,\, k_b\,,\, c_p\,,\, V) \tag{9}$$

The resulting relationship based on this analysis is as follows:

$$\Pi_1 = f\,(\Pi_2,\, \Pi_3,\, \Pi_4,\, \Pi_5,\, \Pi_6)\;, \tag{10}$$

Through consideration of the primary dimensions, six unique dimensionless *Π*-terms were determined. These terms are listed in Table 5, and the resulting relationship is given below:

$$\mathbf{Nu}_b = C \cdot \mathbf{Re}_b^{n_1} \mathbf{Pr}_b^{n_2} \left(\frac{\rho_w}{\rho_b}\right)^{n_3} \left(\frac{\mu_w}{\mu_b}\right)^{n_4} \left(\frac{k_w}{k_b}\right)^{n_5} \tag{11}$$

Table 5 *Π*-Terms of the Empirical Correlation [10].

***Π*-Terms**	**Dimensionless Group**	
Π1	$\frac{HTC \cdot D}{k_b}$	Nusselt number
Π2	$\frac{\rho \cdot V \cdot D}{\mu_b}$	Reynolds number
Π3	$\frac{c_p \cdot \mu_b}{k_b}$	Prandtl number
Π4	$\frac{\rho_w}{\rho_b}$	Density ratio
Π5	$\frac{\mu_w}{\mu_b}$	Viscosity ratio
Π6	$\frac{k_w}{k_b}$	Thermal conductivity ratio

Equation (11) provided a starting point for the development of a correlation, where HTC can be calculated from the following equation:

$$\mathrm{HTC} = \frac{\mathbf{Nu} \cdot k_b}{D_{hy}}, \tag{12}$$

where D_{hy} and k_b denote the hydraulic-equivalent diameter and thermal conductivity of water, respectively. The various coefficients for the resulting relationship need to be determined for the final correlation.

As a result of the experimental data analysis described, the following preliminary correlation for heat transfer to supercritical water was obtained.

$$\mathbf{Nu}_b = 0.0053\, \mathbf{Re}_b^{0.914} \overline{\mathbf{Pr}}_b^{0.654} \left(\frac{\rho_w}{\rho_b}\right)^{0.518} \tag{13}$$

To finalize this correlation, the complete set of primary data and Eq. (13) were fed into the SigmaPlot Dynamic-Fit Wizard to perform final adjustments. The final correlation is as follows:

$$\mathbf{Nu}_b = 0.0061 \mathbf{Re}_b^{0.904} \overline{\mathbf{Pr}}_b^{0.684} \left(\frac{\rho_w}{\rho_b}\right)^{0.564} \tag{14}$$

The test matrix shown in Table 6 provides the range of applicability for the developed correlation. This matrix is the result of comparison with Kirillov et al. [23] experimental data in addition to a comparison with other datasets for supercritical water.

Table 6 Test Matrix for Developed Correlation (Eq. (14)).

Pressure, MPa	Heat Flux, kW/m^2	Mass Flux, kg/m^{2}s	Diameter, mm
22.8 – 29.4	70 – 1250	200 – 1500	3 – 38

Even though the final coefficients slightly deviate from the preliminary correlation, both correlations fit the data in nearly the same manner. Figure 9 provides scatter plots of the experimentally obtained HTC values versus the calculated HTC values for each of the above mentioned correlations. The final correlation (Eq. (14), Mokry et al. correlation) has an uncertainty of about ±25% for HTC values and about ±15% for calculated wall temperature.

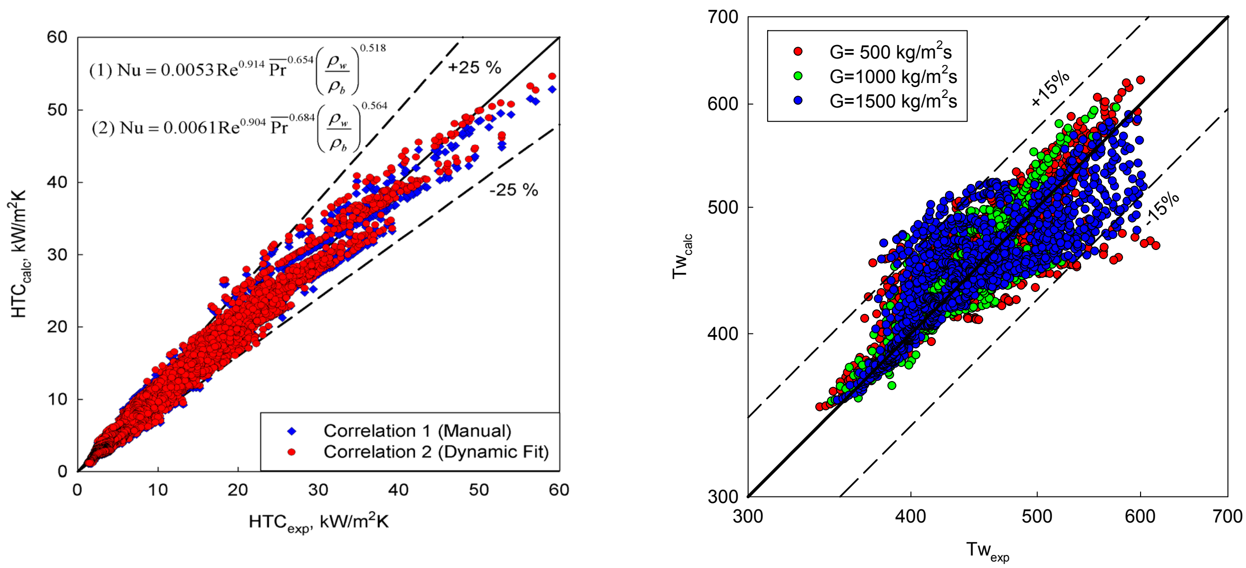


Figure 9 Comparison of Data Fit (Eqs. (13) and (14)) with Experimental Data: (a) for Heat Transfer Coefficient and (b) for wall temperature [10].

In order to evaluate the accuracy of the derived correlation, a comparison of the experimental data with the calculated HTC profiles, using the modified Bishop et al., Dittus-Boelter and the derived correlations was conducted and is shown in Figures 10 and 11. As can be seen from these graphs, neither the modified Bishop et al. nor the Dittus-Boelter correlations provide a good fit for the experimental data, whereas the final Mokry et al. correlation (Eq. (14)) fits the data well and follows trends closely.

Another comparison between the Mokry et al. correlation (Eq. (14)) and calculations using the CFD Code FLUENT-6.0 is shown in Figure 12.

An analysis of the plots in Figures 10 – 12 (for more details, see [10]) showed that in general, the final correlation (Eq. (14)) appeared to best fit the general data trends. Deviations in the calculated HTC values from the experimentally determined values were found, for the most part, at the test section inlet. Within this area, however, the flow was likely subject to an entrance effect. There were also slight deviations within the pseudocritical range, however, the most pronounced difference occurred only at the lower mass flux.

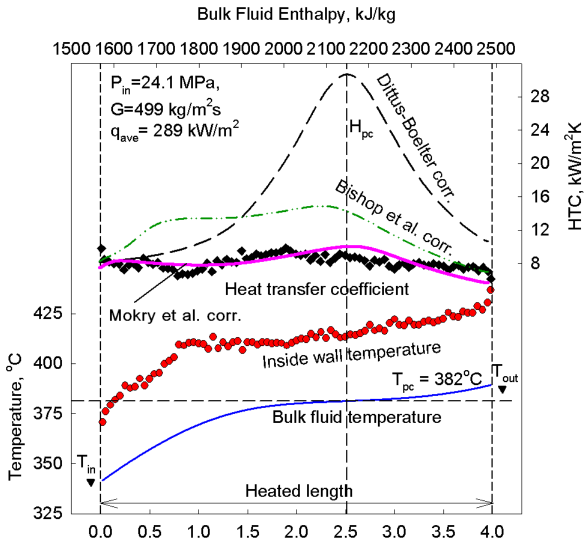


Figure 10 Temperature and Heat Transfer Coefficient Profiles along Heated Length of Bare Vertical Tube: $G = 500$ kg/m^{2}s and $q = 290$ kW/m^2[10].

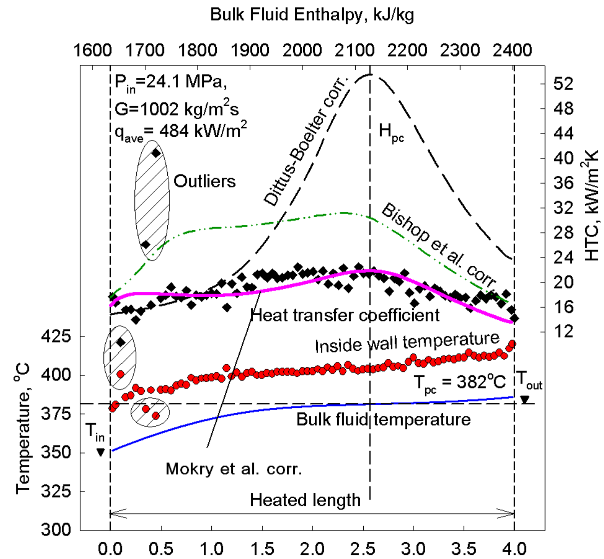


Figure 11 Temperature and Heat Transfer Coefficient Profiles along Heated Length of Bare Vertical Tube: $G = 1000$ kg/m^{2}s and $q = 480$ kW/m^2[10].

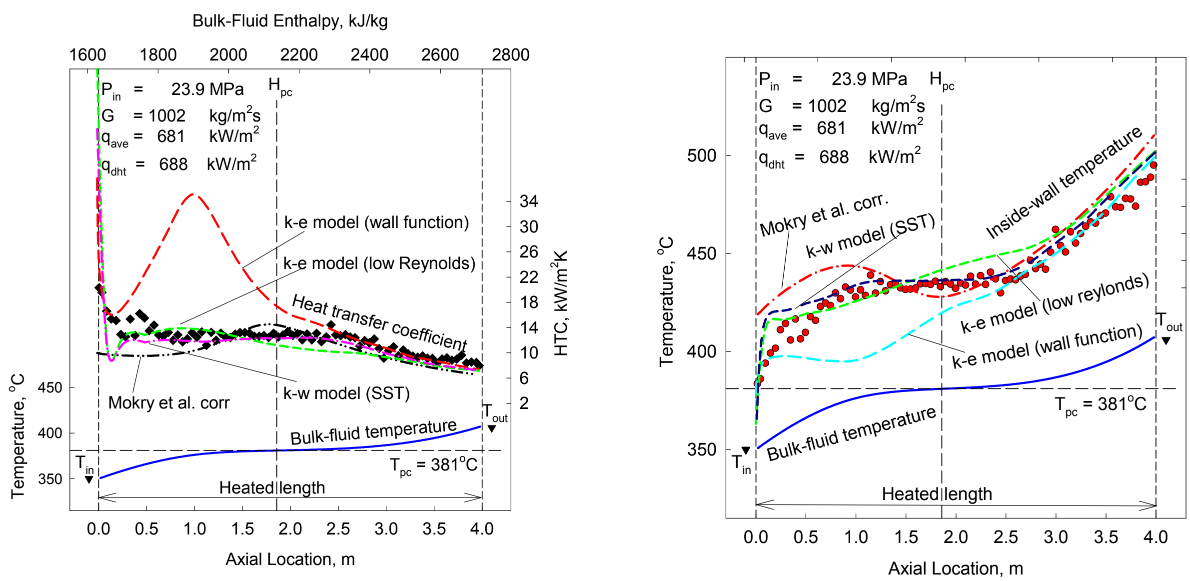


Figure 12 Temperature and Heat Transfer Coefficient Comparisons Between Final Correlation (Eq. (14)) and CFD Code Calculations along 4-m Circular Tube (ID = 10 mm): Operating Conditions – $P_{in} = 24.0$ MPa and $G = 1000$ kg/m^{2}s [10], [25].

The HTC and wall temperature values (Figure 12) calculated with the FLUENT CFD code may deviate significantly from the experimental data (for example, the *k-ε* model (wall function)). However, the *k-ε* model (low Reynolds numbers) shows a better fit within some flow conditions [10].

Nevertheless, the derived correlation (Eq. (14)) showed the best fit for the experimental data within a wide range of flow conditions. This correlation has an uncertainty of about ±25% for HTC values and about ±15% for calculated wall temperature. Therefore, the derived correlation can be used for preliminary HTC calculations in SCWR fuel bundles as a conservative approach, for SCW heat exchangers, for future comparison with other datasets, for verification of computer codes and scaling parameters between SCW and modelling fluids.

For the final verification of the correlation, a comparison with other datasets was completed (Figures 13 − 15). From the presented figures, it can be seen that the updated correlation (Eq. (14)) closely represents the experimental data and follows trends closely, even within the pseudocritical range. Table 7 lists the test matrices for these datasets against which the Mokry et al. correlation was compared.

Table 7 Other Datasets and Corresponding Test Matrices.

Reference	***P*, MPa**	***q*, MW/m²**	***G*, kg/m²s**	**Flow geometry**
Alferov et al., 1976 [29]	26.5	0.48	447	Tube (*D*=20 mm, *L/D*=185), ascending flow, ρ_w=447 kg/m²·s
Petukhov and Polyakov, 1988 [30]	29.4	0.50	675	Tube (*D*=3mm)
Bishop et al., 1964	22.8 – 27.6	0.31 – 3.46	651 – 3662	Tube (*D*=5 mm,) upward flow
Shitsman 1963 [31]	22.6–24.5	0.28–1.1	300–1500	SS tube (*D*=8 mm, *L*=1.5 m)
Vikhrev et al. 1967 [32]	24.5; 26.5	0.23–1.25	485–1900	SS tube (*D*=7.85; 20.4 mm, *L*=1.515; 6 m)
Ornatsky et al. 1970 [26]	22.6; 25.5; 29.4	0.28–1.2	450–3000	Five SS parallel tubes (*D*=3 mm, *L*=0.75 m), upward stable and pulsating flows
Pis'mennyy et al. 2005 [33]	23.5	Up to 0.515	250; 500	Vertical SS tubes (*D*=6.28 mm, L_h=600; 360 mm; *D*=9.50 mm, L_h=600; 400 mm)
Polyakov 1975 [34]	29.4	0.50	675	Tube (D=8 mm)
Lee and Haller 1974 [35]	24.1	0.25–1.57	542–2441	SS tubes (*D*=38.1; 37.7 mm, *L*=4.57 m), tube with ribs
Shiralkar and Griffith (1969 and 1968) [36]	22.8	0.32	461	Tube (*D*=10 mm)
Shitsman 1968 [37]	10–35	0.27–0.7	400	Vertical and horizontal SS tubes (*D*/*L*=3/0.7; 8/0.8; 8/3.2; 16/1.6 mm/m), upward, downward and horizontal flows
Yamagata et al. 1972 [27]	22.6–29.4	0.12–0.93	310–1830	Vertical and horizontal SS tubes (*D*/*L*=7.5/1.5; 10/2 mm/m), upward, downward and horizontal flows
Yoshida and Mori 2000 [28]	24.5	0.23-0.33	376, 1180	Tube (*D*=10 and 16 mm)

5. Conclusions

Supercritical-water heat-transfer data for a vertical bare circular tube were obtained within the proposed SCWR operating conditions: pressure of ~24 MPa, mass fluxes from 200 to 1500 kg/m^{2}s, heat fluxes up to 1250 kW/m^2 and inlet temperatures from 320 to 350ºC. Supercritical heat transfer was investigated for several combinations of wall and bulk-fluid temperatures, i.e., internal wall temperatures and bulk-fluid temperatures below, at, or above the pseudocritical temperature.

The obtained correlation for forced convective heat transfer to supercritical water in a bare vertical tube showed a good fit (±25% for heat transfer coefficient) for the analyzed dataset. In addition, the calculated wall temperatures resulted in a more accurate fit for the analyzed dataset (±15%). Therefore, the derived correlation can be used for preliminary HTC calculations in SCWR fuel bundles as a conservative approach, for SCW heat exchangers, for future comparison with other datasets, for verification of computer codes and scaling parameters between SCW and modelling fluids.

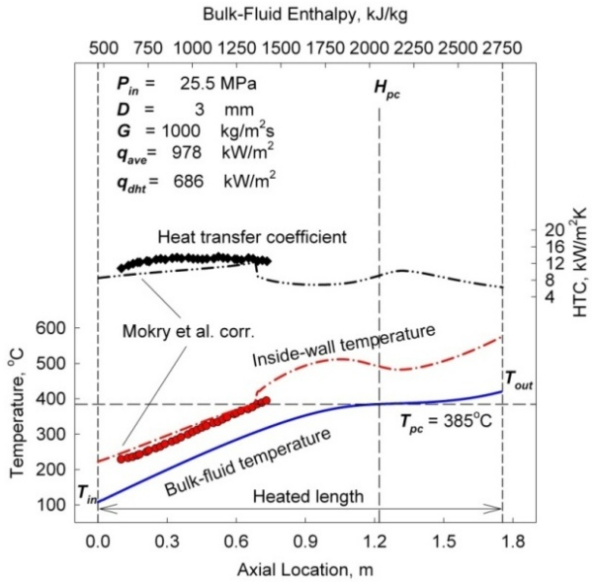


Figure 13 Temperature and Heat Transfer Coefficient Variations at Various Heat Fluxes along a Circular Tube: Nominal Operating Conditions – P_{in} = 25.5 MPa, G = 978 kg/m^{2}s [26].

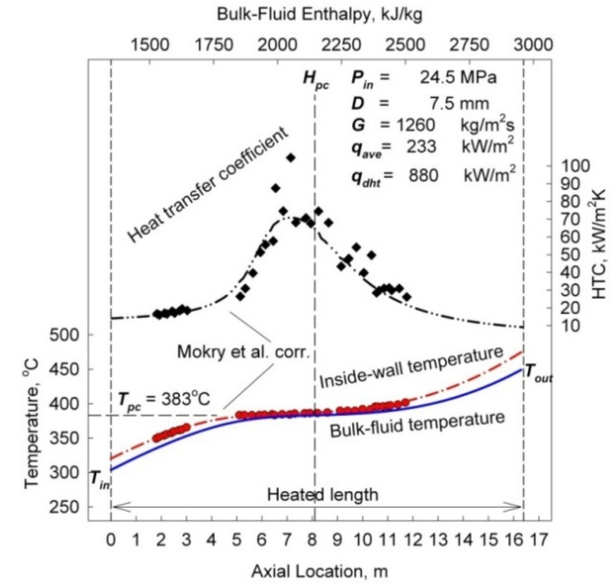
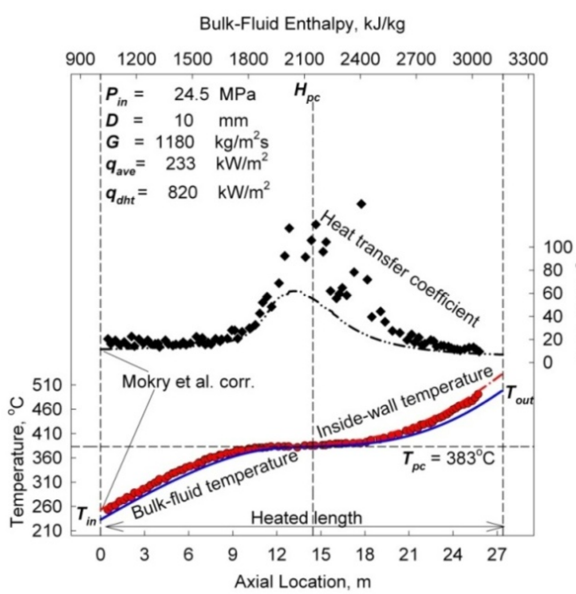
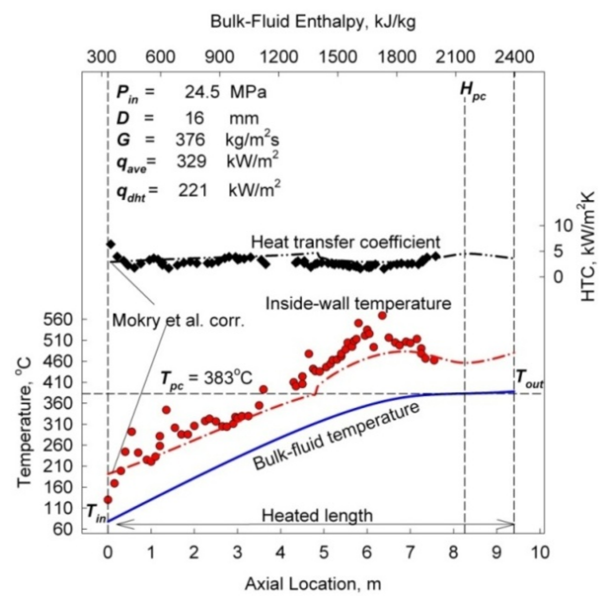


Figure 14 Temperature and Heat Transfer Coefficient Variations at Various Heat Fluxes along a Tube: Nominal Operating Conditions – P_{in} = 24.5 MPa, G = 1260 kg/m^{2}s and q_{ave}=233 kW/m^2 [27]





(a) **(b)**

Figure 15 Temperature and Heat Transfer Coefficient Variations at Various Heat Fluxes along a Circular Tube: Nominal Operating Conditions –P_{in} = 24.5 MPa, (a) G = 1180 kg/m^{2}s; and (b) G = 376 kg/m^{2}s [28].

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NOMENCLATURE

c_p	specific heat at constant pressure (J/kg·K)
$\bar{c}_p$	average specific heat, J/kg·K, $\left(\frac{H_w - H_b}{T_w - T_b}\right)$
D	inner diameter, m
G	mass flux, kg/m^{2}s
h	heat transfer coefficient, W/m^{2}K
k	thermal conductivity, W/m·K
L	heated length, m
P	pressure, MPa
Q	heat transfer rate, W
q	heat flux, W/m^2
R	Roughness, μm
T	temperature, ^{o}C

Greek letters

δ	thickness, mm
μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3

Dimensionless numbers

Nu	Nusselt number $\left(\frac{h \cdot D}{k}\right)$
Pr	Prandtl number $\left(\frac{\mu \cdot c_p}{k}\right)$
$\overline{\mathbf{Pr}}$	averaged Prandtl number $\bar{c}_p\left(\frac{\mu_b}{k_b}\right)$
Re	Reynolds number $\left(\frac{G \cdot D}{\mu}\right)$

Subscripts

ave	average
b	bulk
cal	calculated
cr	critical
dht	deteriorated heat transfer
exp	experimental
ext	external
hy	hydraulic
in	inlet conditions
out	outlet conditions
pc	pseudocritical
w	wall
x	axial location, m

Abbreviations

AECL	Atomic Energy of Canada Limited
AHFD	Axial Heat Flux Distribution
BWR	Boiling Water Reactor
CANDU	CANada Deuterium Uranium (reactor)
DAS	Data Acquisition System
DHT	Deteriorated Heat-Transfer (regime)
GIF	Generation IV International Forum
HTC	Heat Transfer Coefficient
ID	Internal Diameter
IHT	Improved Heat-Transfer (regime)
IPPE	Institute for Physics and Power Engineering (Obninsk, Russia)
KP-SKD	Pressure-tube nuclear reactor at supercritical pressure (in Russian abbreviations)
NHT	Normal Heat-Transfer (regime)
NIST	National Institute of Standards and Technology (USA)
NPP	Nuclear Power Plant
PT	Pressure Tube
PV	Pressure Vessel
PWR	Pressurized Water Reactor
RDIPE	Research and Development Institute of Power Engineering (Moscow) (NIKIET in Russian abbreviations)
REFPROP	REFerence PROPerties
SCW	SuperCritical Water
SCWR	SuperCritical Water-cooled Reactor

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