HEAT TRANSFER TO SUPERCRITICAL PRESSURE FLUIDS FLOWING IN TUBES IN SUPERHEATED VAPOR REGION

T. Kaida[1], H. Mori[2], K. Kariya[2], M. Ohno[2] and Y. Hamamoto[2]
[1] Graduate School of Engineering, Kyushu University, Fukuoka, Japan
[2] Faculty of Engineering, Kyushu University, Fukuoka, Japan

Abstract

For the development and design of supercritical pressure water-cooled reactor (SCWR), it is necessary to accurately estimate heat transfer to supercritical pressure water in high bulk enthalpy region. In this study, heat transfer experiments were performed using supercritical pressure HCFC22 as a test fluid, which flowed upward in a uniformly heated vertical circular tube. The experimental data are compared with conventional heat transfer correlations. In addition, numerical simulations were performed with the k-ω model using the commercial CFD code STAR-CD to quantitatively reproduce the experimental data and to examine the effect of property variation on heat transfer.

1. Introduction

Supercritical pressure water-cooled reactor (SCWR), which is a once-through water-cooled reactor operating above the critical pressure of water (22.1 MPa), has been considered as an innovative reactor due to its high thermal efficiency and economic potential. The SCWR is planned to operate at the coolant outlet temperature of 500°C or more, supplying supercritical pressure steam at high temperatures to the turbine system[1]. The design of the SCWR involves the heat transfer from a fuel rod to the coolant under supercritical pressure, and especially it is necessary to accurately estimate the maximum cladding surface temperature (MCST) which is one of the most important indicators for safety. The MCST occurs at the core outlet, and therefore the accurate estimation of heat transfer coefficients in the high bulk enthalpy region is essential for the development of the SCWR.

It is widely recognized that when a fluid is heated at a constant supercritical pressure, there occurs no phase change; there occurs a continuous variation from a liquid-like fluid to a vapor-like fluid. Hence supercritical pressure fluids in a thermodynamic equilibrium state can be regarded as single phase fluids in any conditions from a macroscopic standpoint.

On the other hand, physical properties of supercritical pressure fluids vary continuously and rapidly with temperature as an example shown in **Figure 1**. Isobaric specific heat c_p has a peak. The temperature at which the specific heat attains a maximum value is called the pseudocritical temperature T_{pc}. In a certain temperature range including the pseudocritical temperature, physical properties such as thermal conductivity λ, density ρ, viscosity μ and specific enthalpy h vary rapidly from liquid-like state to vapor-like state. Heat transfer for the supercritical pressure fluid is characterized by such variations of physical properties with temperature.

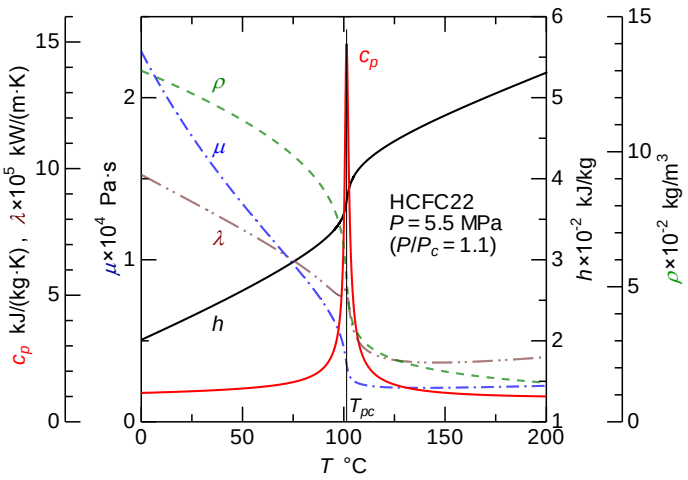


Figure 1 Physical properties of HCFC22 at a supercritical pressure[2]

Over the past decades, for the development of supercritical pressure thermal plants, a large number of experimental studies on the heat transfer to supercritical pressure fluids have been conducted mainly using water and carbon dioxide. Since most of these studies focused on the peculiar characteristics of the heat transfer near the pseudocritical temperature, experimental data in the high bulk enthalpy are scarce.

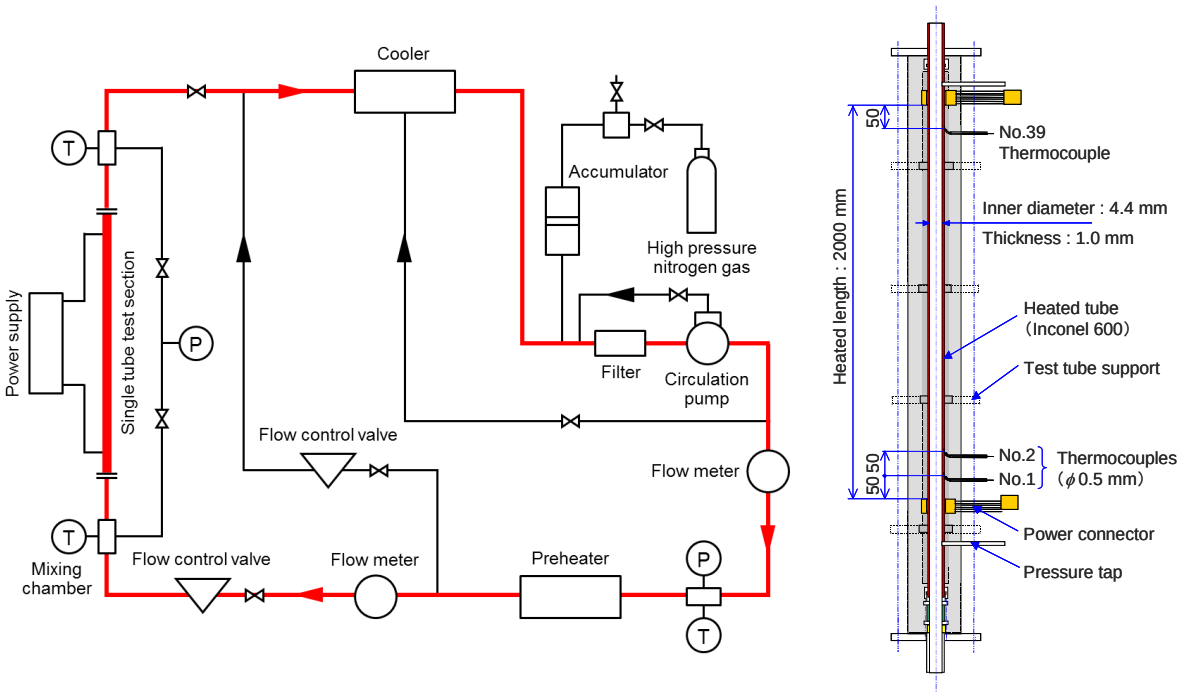
Compared with experimental studies, numerical simulations need much lower cost especially for supercritical pressure fluids, providing more detailed information. In recent years, with the advances of computer capability and computational methods, more and more numerical studies have been performed with the commercial applications of computational fluid dynamics (CFD). To our knowledge, none of previous numerical studies except the study by Yang *et al.*[3] aimed at the heat transfer in the high bulk enthalpy region, and because of experimental data scarcity, Yang *et al*. could not make a sufficient examination.

In the present study, heat transfer experiments were conducted using supercritical pressure HCFC22 as a test fluid, which flowed upward in a uniformly heated vertical circular tube. Experimental data were obtained under the superheated vapor region, which is defined as the high bulk enthalpy region above the pseudocritical enthalpy in this paper. The data are compared with constant property heat transfer correlations and traditional correlations for supercritical pressure fluids, and predicting performance of these correlations is examined. In addition, numerical simulations were carried out with the k-ω model using the commercial CFD code STAR-CD to quantitatively reproduce the experimental data and the effect of property variation on heat transfer is examined. Finally, the numerical model used for HCFC22 is applied for supercritical pressure water, and wall temperatures at the core outlet in SCWR are estimated.

2. Experiments

2.1 Experimental apparatus and method

The experimental apparatus, which was previously constructed[4], is shown in **Figure 2**. The test loop was a forced circulation test loop, including a circulation pump, preheater, test section, cooler, mixing chambers, flow meters and flow control valves, as shown schematically in Figure 2(a). HCFC22 was used as a test fluid since its critical pressure and temperature of 4.99 MPa and 96.2°C are much lower than those of water, and therefore the experimental data in superheated vapor region at supercritical pressure can be relatively readily obtained. The pressure in the test loop was kept constant by an accumulator connected to a high pressure nitrogen gas supply. The accumulator had a free piston, which seal nitrogen gas closely from HCFC22.



(a) Test loop (b) Test section

Figure 2 Experimental apparatus

The test section, as shown in Figure 2(b), was an Inconel 600 smooth tube of 4.4 mm I.D. oriented vertically. It was heated over a 2000 mm length by passing alternating current through the tube directly. HCFC22 flows through the test section vertically upward. The inlet and outlet bulk fluid temperatures were measured with sheath thermocouples at the inlet and outlet mixing chambers. The wall temperatures on the tube outside surface were measured with sheath thermocouples fixed at axially 50 mm intervals in total 39 sections.

In the experiments, the data were obtained after confirming the wall temperatures and the outlet bulk fluid temperature at steady, keeping the following experimental parameters constant: pressure P, mass velocity G and heat flux q. The inside surface temperatures T_w of the tube were evaluated from the measured outside surface temperatures taking account of the radial heat conduction within the tube wall. The bulk fluid temperature T_b at each axial section corresponding to the position of measuring the outside surface temperature was

evaluated from the bulk enthalpy h_b determined by the measured outlet fluid enthalpy and the heat balance to the test section outlet.

The heat transfer coefficient α was calculated from the inside surface heat flux q and the temperature difference $T_w - T_b$ between the inside surface and the bulk fluid.

$$\alpha = \frac{q}{T_w - T_b} \tag{1}$$

It was estimated that the temperature difference was accurate within 0.8°C including the uncertainty in the thermocouple measurement and heat loss, and that heat flux from the electric power input and mass flow rate were accurate 1%. The influence of the uncertainty of pressure readings on heat transfer was negligibly small. In total, the accuracy of the heat transfer coefficient was up to ± 5%. The wall temperature was kept under 180°C to avoid the deposition of carbon onto the wall surface due to the decomposition of HCFC22.

The experimental conditions are listed in **Table 1**. The pressure of 5.5 MPa corresponds to the reduced pressure of 1.1 which is the same as the supercritical pressure water condition in SCWR. The physical properties of HCFC22 are calculated by NIST REFPROP version 8.0[2]. The pseudocritical temperature and enthalpy of HCFC22 at 5.5 MPa are 101.4°C and 372.9 kJ/kg, respectively.

Table 1 Experimental conditions

Fluid	HCFC22
Flow direction	Upward
Inside diameter D mm	4.4
Pressure P MPa (P/P_c)	5.5 (1.1)
Mass velocity G kg/(m^2·s)	1000, 2000
Heat flux q kW/m^2	40, 50, 60, 70, 80, 100, 130
Bulk fluid enthalpy h_b kJ/kg	400 - 480

2.2 Experimental results and discussion

The measured inside wall temperatures and heat transfer coefficients are plotted against bulk fluid enthalpy at a mass velocity of 1000 kg/(m^2·s) and various heat fluxes of 40, 50, 60 and 70 kW/m^2 in **Figure 3**. The heat transfer coefficient calculated by the Dittus and Boelter correlation[5] is also shown in the lower diagram of this figure. The wall temperature increases monotonically along bulk fluid temperature with bulk fluid enthalpy and increases with heat flux at a bulk fluid enthalpy, indicating the tendency of forced convective heat transfer. As for heat transfer coefficient, although the present data become more unaffected by heat flux with increasing bulk fluid enthalpy, they are lower than the Dittus and Boelter correlation by about 15-20%. Thus, the Dittus and Boelter correlation overestimates heat transfer coefficient although Kamei *et al*.[6] recommended the use of the Dittus and Boelter correlation in the high bulk fluid enthalpy region.

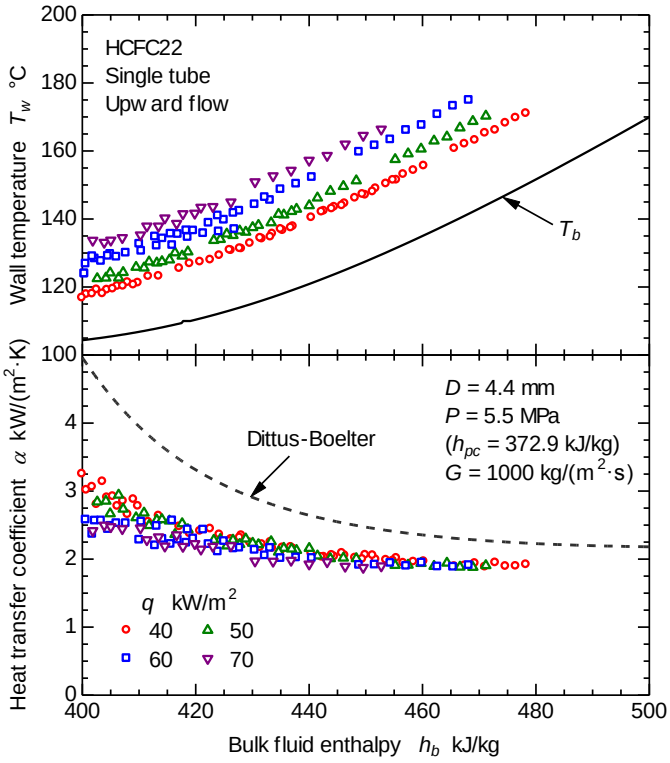


Figure 3 Wall temperature and heat transfer coefficient

The MCST is an important indicator for safety, and its estimation depends largely on the correlation that determines the heat transfer coefficient from a fuel rod to the coolant. A number of correlations for supercritical pressure fluids have been developed mainly based on the experimental data of water and/or carbon dioxide. Most of these correlations are expressed in the form of a forced convective heat transfer equation multiplied by the wall to bulk property ratios raised to suitable powers.

Jackson and Hall[7] discussed the predicting performance of these correlations using approximately 2000 experimental data for water and carbon dioxide, and concluded that the correlation proposed by Krasnoshchekov and Protopopov[8] showed the best. In addition, Jackson and Hall modified and simplified the Krasnoshchekov and Protopopov correlation to employ a Dittus and Boelter correlation type form for the constant properties Nusselt number part. After the Jackson and Hall's study, some correlations have been proposed by several reseachers, for example, Watts and Chou[9] and Kurganov[10]. These supercritical pressure fluid correlations were developed mainly with data of tubes of diameter over 10 mm for the development of supercritical pressure boilers. Taking account of the tightened fuel rod pitch in the SCWR of which a hydraulic diameter is smaller, Yamashita *et al.*[11] tested the predicting performance of these correlations using the experimental data obtained with HCFC22 in a small tube of 4.4 mm I.D., and in consequence recommended the Watts and Chou correlation as the best, especially in the near-pseudocritical enthalpy region. However, as for the high bulk enthalpy region including the point at which MCST occurs, there is no report for the predicting performance of these correlations.

In this study, the predicting performance of these correlations is tested with the present data under the high bulk enthalpy region of the superheated vapor. The correlations for this test also includes the correlation by Bishop *et al.*[12] The predicting performance of the Bishop *et al.* correlation is lower than that of the Krasnoshchekov and Protopopov correlation according to the above-mentioned Jackson and Hall's study, but the applicable range of 282-527°C for supercritical pressure water covers the higher bulk fluid temperature and its predicting performance remains a matter of re-examination. In addition, to compare the data with constant property forced convective heat transfer correlations, the Kays and Crawford correlation[13] for gas single phase flows as well as the Dittus and Boelter correlation is examined.

Table 2 Predicting performance of the correlations

	h_b > 400 kJ/kg Number of data N = 273						h_b > 440 kJ/kg Number of data N = 99					
	SD	*MD*	*AD*	5%	10%	20%	*SD*	*MD*	*AD*	5%	10%	20%
Dittus-Boelter[4]	43.0	39.1	39.1	0	0	6.6	24.6	24.2	24.2	0	0	17.2
Kays-Crawford[12]	40.5	34.2	34.2	0	4.4	35.2	16.1	15.4	15.4	0	12.1	82.8
Bishop *et al.*[11]	25.7	24.0	24.0	0	0	39.2	19.7	19.0	19.0	0	0	57.6
Krasnoshchekov -Protopopov[7]	19.8	16.4	16.4	11.4	36.6	67.8	7.5	6.9	6.9	29.3	84.8	100
Jackson-Hall[6]	24.5	23.2	23.2	0	0	44.7	17.8	17.5	17.5	0	0	80.8
Watts-Chou[8]	6.6	5.0	4.5	61.9	87.5	98.9	3.0	2.6	2.1	90.9	100	100
Kurganov[9]	16.1	14.9	14.9	0.4	18.3	83.9	11.8	11.5	11.5	0	30.3	100

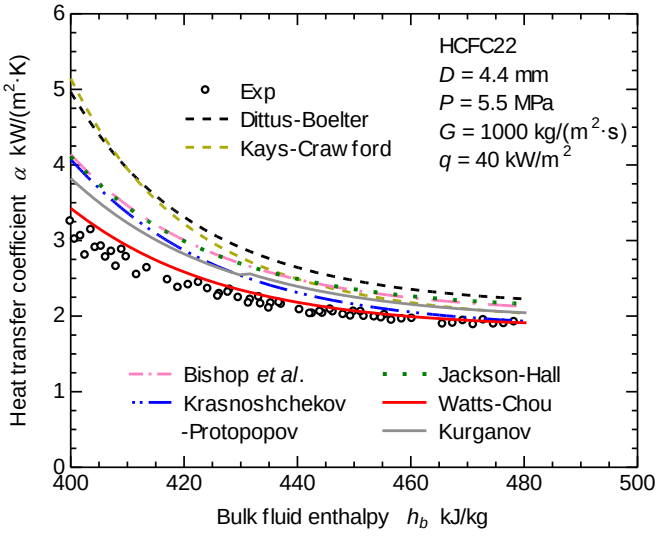


Figure 4 Comparison of the correlations with the present data

SD: standard deviation

$$SD = \sqrt{\frac{1}{N}\sum\left(\frac{\alpha_{cal}-\alpha_{exp}}{\alpha_{exp}}\right)^2}$$

MD: mean deviation

$$MD = \frac{1}{N}\sum\left|\frac{\alpha_{cal}-\alpha_{exp}}{\alpha_{exp}}\right|$$

AD: average deviation

$$AD = \frac{1}{N}\sum\left(\frac{\alpha_{cal}-\alpha_{exp}}{\alpha_{exp}}\right)$$

The results of the comparison are shown in **Table 2**. In the table, *SD*, *MD*, and *AD* are standard deviation, mean deviation and average deviation, respectively, and the columns of 5%, 10% and 20% express the percentages of the number of the data of their prediction deviation within ± 5%, ± 10% and ± 20% to the total number of the experimental data, respectively. It can be clearly seen that the Watts and Chou correlation is superior to other correlations and reproduce all the data above 440 kJ/kg within ± 10% deviation. The results also show that the Bishop *et al*. correlation has a predicting performance equivalent to the Jackson and Hall correlation, but that the Kays and Crawford correlation is better for the high bulk enthalpy region above 440 kJ/kg.

Figure 4 shows an example of the comparisons in the relation between the heat transfer coefficient and the bulk fluid enthalpy. In this figure, the Watts and Chou correlation also shows the best prediction.

3. Numerical simulation

3.1 Computational method

Numerical simulations were carried out by use of the commercial CFD code STAR-CD incorporating physical properties of supercritical pressure HCFC22. STAR-CD provides a number of turbulence models. Among these turbulence models, the standard k-ω model, which is the widely used as an eddy viscosity model to simulate the turbulence effect on flow and heat transfer in boundary layer, was selected. In general, as the near wall treatment, the high *Re* model uses a wall function, while the low *Re* model does not use a wall function but needs meshes to be finely generated near wall. STAR-CD recommends that the meshes are generated so that the nondimensional distance y^+ value for the first node close to the wall is 30-100 in using the high *Re* model, while the y^+ value for the first node is lower than 1 in using the low *Re* model. In this study, the mesh size was adjusted to satisfy the above requirements in each run, in which the y^+ value was estimated using the Itaya equation[14] about frictional factor for isothermal single phase flow. **Figure 5** shows the geometry and boundary conditions of this method. In order to save computing time, a twentieth part of the circular cross-section was taken because of the cyclic geometry. Heat flux was given by heat generation in the solid cells of tube. The outer surface of the solid cells was insulated.

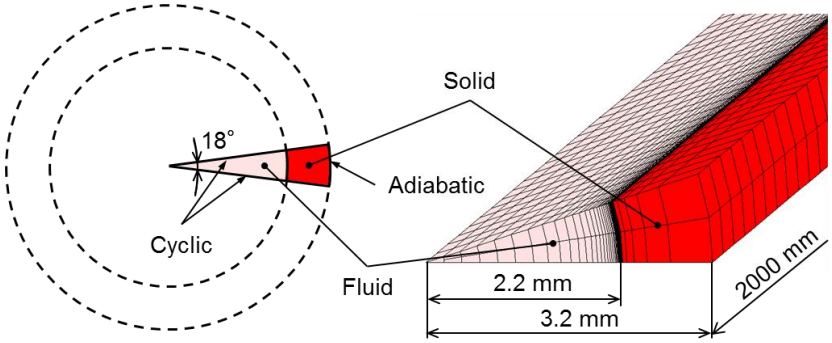
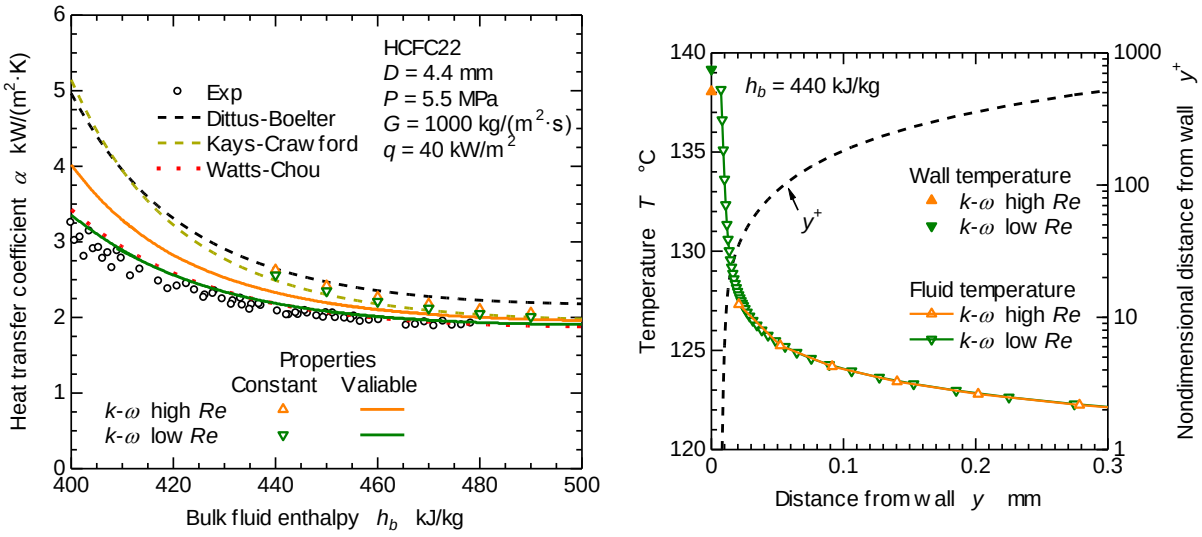


Figure 5 Computational model

3.2 Numerical results and discussion

First, as the ideal state, the heat transfer in the case where there is no variation of physical properties across the flow was computed, and the results are compared with constant property forced convective heat transfer correlations. The physical properties were given as the constant values at each bulk fluid enthalpy. As seen in **Figure 6**(a), the significant difference of the heat transfer coefficients calculated by between the two *Re* models is not observed, and the both models quantitatively agree well with the Kays and Crawford correlation for gas single phase flow but not with the Dittus and Boelter correlation. The validity of the computational model was also confirmed.

Secondly, the heat transfer in the case where the dependency of physical properties on temperature is actually taken into consideration was computed, and the results are compared to the computation with physical properties constant. Figure 6(a) shows that the heat transfer coefficients considering the variations of physical properties are lower than that not considering them. The heat transfer coefficient calculated by the low *Re* model is lower than the calculation by the high *Re* model in some degree and equivalent to the calculation by the Watts and Chou correlation, reproducing the experimental data well.



(a) Heat transfer coefficient

(b) Wall temperature and fluid temperature distribution

Figure 6 Comparison of the high *Re* model and the low *Re* model

Figure 6(b) shows the calculation of wall temperature and fluid temperature distribution across the flow at the bulk fluid enthalpy of 440 kJ/kg in Figure 6(a) plotted against the distance y from wall. In this figure, the nondimensional distance y^+ from wall calculated by the low *Re* model is also described. It is observed that there is no difference of the fluid temperature distributions between the two *Re* models over $y^+ = 50$, but there occurs the discrepancy of the wall temperatures. It is supposed that, in the high *Re* model, the property

variation owing to rapid temperature rise in the viscous sublayer near the wall is not successfully considered, which results in the discrepancy of heat transfer coefficients.

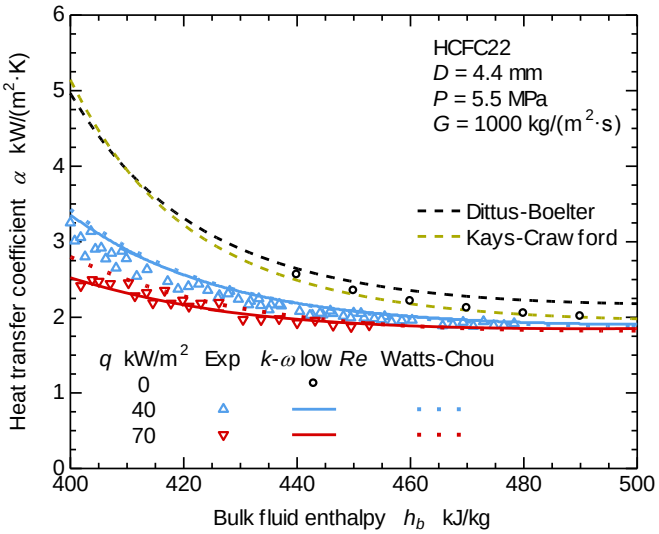


Figure 7 Influence of heat flux on heat transfer coefficient

Figure 7 shows an example of the comparisons of the computed heat transfer coefficients by the low *Re* k-ω model with the experimental data and the Watts and Chou correlation at different heat fluxes. In this figure, the above-mentioned numerical results in the case of constant properties are considered the ideal coefficients at the heat flux being zero and represented as $q = 0$. Both the experimental and numerical coefficients decrease with increasing heat flux and are reproduced by the Watts and Chou correlation. This effect of heat flux is not neglected even in the high bulk fluid enthalpy region of over 440 kJ/kg.

Radial profiles of density, isobaric specific heat and thermal conductivity at 440 kJ/kg in Figure 7 are shown in **Figure 8** as the ratios divided by the bulk values. The ratios of density and isobaric specific heat become largely lower than that of thermal conductivity in the vicinity of wall with heat flux. The change of viscosity ratio was also as small as that of thermal conductivity ratio although it is not shown in this figure. In the Watts and Chou correlation, radial variations of isobaric specific heat and density are taken into account in the form of Prandtl number based on integrated specific heat and the wall to bulk density ratio, respectively, and hence this correlation probably has a high predicting performance.

In the k-ω model, turbulent heat flux is expressed as the following equation:

$$q = -(\lambda + \rho c_p a_t)\frac{\mathrm{d}T}{\mathrm{d}y}, \tag{2}$$

where $a_t = \nu_t / Pr_t$ is the eddy thermal diffusivity, $\nu_t = k/\omega$ is the eddy kinematic viscosity and Pr_t is the turbulent Prandtl number. **Figure 9** shows an example of the comparisons of turbulent thermal conductivity $\rho c_p a_t$ and the molecular thermal conductivity λ. In $y^+ > 15$, the turbulent thermal conductivity is larger and becomes more dominant with distance from

wall, but the amount is smaller with increasing heat flux. Thus, it is likely that the lowering of heat transfer coefficient with heat flux results from depression of the turbulent thermal diffusivity due to decreasing of density and isobaric specific heat in the turbulent layer near the viscous sublayer.

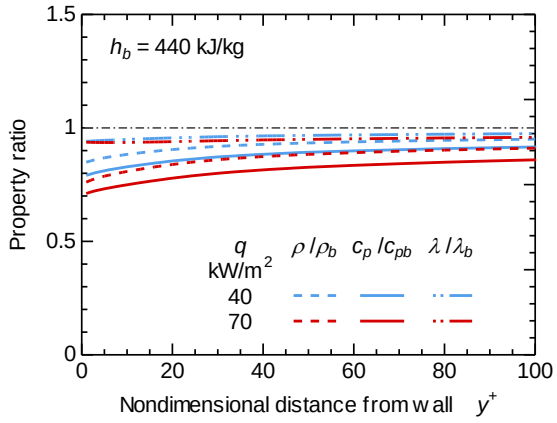


Figure 8 Radial profiles of physical properties

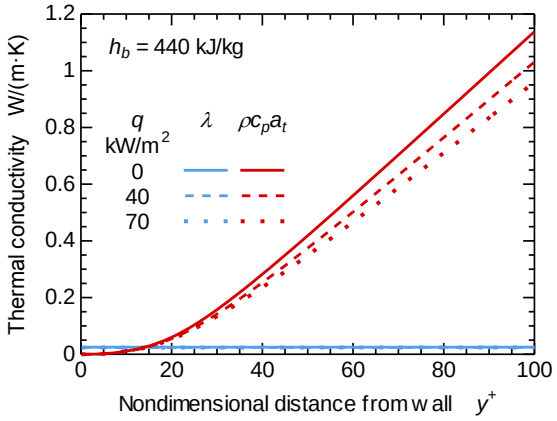


Figure 9 Comparison of molecular and turbulence thermal conductivity

Finally, the above computational model used for HCFC22 is applied for supercritical pressure water, and the result is compared with the data by Ishigai *et al.*[15] and by the Watts and Chou correlation, as shown in **Figure 10**. The test conditions selected for this comparison are similar to the SCWR conditions[6]. Although the heat transfer coefficients by the computation are lower than those by the Ishigai *et al.* and the Watts and Chou correlation in the region of the bulk temperature below 450°C, at the bulk fluid temperature of 500°C corresponding to the core outlet temperature in the SCWR, the numerical results agree with the Watts and Chou correlation within 5% deviation. The Watts and Chou correlation estimates the MCST of 599°C under the following conditions[6]: pressure of 25 MPa, hydraulic diameter of 4.2 mm, mass velocity of 937.9 kg/(m^2·s), heat flux of 562kW/m^2 and

water temperature of 500°C. However, the *k-ω* low *Re* model estimates the a little higher MCST of about 605°C.

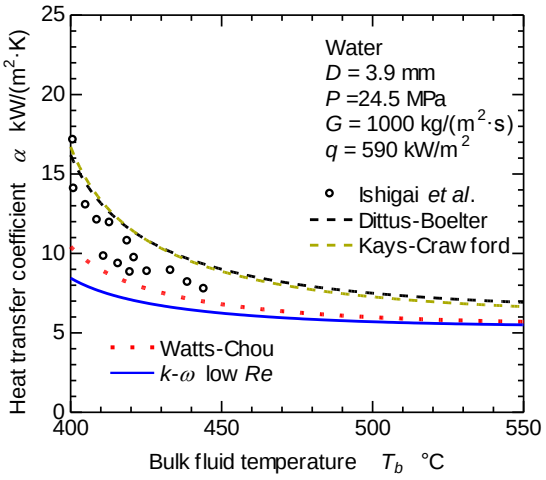


Figure 10 Application of the computational model to supercritical pressure water

4. Conclusion

In relation to the development of SCWR, especially for accurate estimation of MCST, experimental and numerical studies were performed on the heat transfer to supercritical pressure fluids in superheated vapor region, and the following results were obtained.

The heat transfer coefficient obtained by the experiment was by about 15-20% lower than the calculation by the Dittus and Boelter correlation. The lowering of heat transfer coefficient with heat flux resulted from depression of the turbulent thermal diffusivity due to decreasing of density and isobaric specific heat in the turbulent layer near the viscous sublayer. The Watts and Chou correlation had the best prediction performance for the experimental and numerical data among the various conventional correlations. The *k-ω* low *Re* model was suitable for numerical analysis for supercritical pressure fluid in superheated vapor region.

Accordingly, for the estimation of the heat transfer coefficient at the core outlet in the design of SCWR, it is recommended to use the correlation which takes into account the radial variations of density and isobaric specific heat, especially the Watts and Chou correlation. With a view to improving the prediction accuracy of the MCST, precise experimental data with supercritical pressure water in high bulk temperatures should be obtained.

5. References

[1] Y. Oka and S. Koshizuka, "Supercritical-pressure, once-through cycle light water cooled reactor concept," *Journal of Nuclear Science and Technology*, Vol. 38, No. 12, 2001, pp.1081-1089.

[2] E. W. Lemmon *et al*., REFPROP Reference Fluid Thermodynamic and Transport Properties, NIST Standard Reference Database 23, Version 8.0, NIST, 2007.

[3] J. Yang *et al*., "Numerical investigation of heat transfer in upward flows of supercritical water in circular tubes and tight fuel rod bundles," *Nuclear Engineering and Design*, Vol. 237, 2007, pp.420-430.

[4] H. Mori *et al*., "Research and development of a super fast reactor (7) heat transfer to a supercritical pressure fluid flowing in a sub-bundle channel," Proceedings of the 16th PBNC, Aomori, Japan, 2008 Oct. 13-18, PaperID P16P1297.

[5] F. W. Dittus and L. M. K. Boelter, "Heat transfer in automobile radiators of the tubular type," *University of California Publications in Engineering*, Vol. 2, 1930, pp.443-461.

[6] K. Kamei *et al*., "Fuel and core design of super light water reactor with low leakage fuel loading pattern," *Journal of Nuclear Science and Technology*, Vol. 43, No. 2, 2006, pp.129-139.

[7] J. D. Jackson and W. B. Hall, "Forced convection heat transfer to fluids at supercritical pressure," *Turbulent Forced Convection in Channels and Bundles*, Vol. 2, Hemisphere Publishing, 1979, pp.563-611.

[8] E. A. Krasnoshchekov and V. S. Protopopov, "Experimental study of heat exchange in carbon dioxide in the supercritical range at high temperature drops," *Teplofizika Vysokikh Temperatur*, Vol. 4, No. 3, 1966, pp.389-398.

[9] M. J. Watts and C. T. Chou, "Mixed convection heat transfer to supercritical pressure water," Proceedings of the 7th IHTC, München, Germany, No. 3, 1982, pp.495-500.

[10] V. A. Kurganov, "Heat transfer and pressure drop in tubes under supercritical pressure of the coolant. Part 1: Specifics of the thermophysical properties, hydrodynamics, and heat transfer of the liquid. Regimes of normal heat transfer," *Thermal Engineering*, Vol. 45, No. 3, 1998, pp.177-185.

[11] T. Yamashita *et al*., "Heat transfer study under supercritical pressure conditions," Proceedings of GENES4/ANP2003, Kyoto, Japan, 2003 Sep. 15-19, Paper 1119.

[12] A. A. Bishop et al., "Forced convection heat transfer to water at near-critical temperatures and supercritical pressures." WCAP-2056, Part IV, November, Westinghouse Electric Corporation, Pittsburgh, USA, 1964.

[13] W. M. Kays and M. E. Crawford, "Convective heat and mass transfer (2nd ed.)," McGraw-Hill, New York, USA, 1980, p.243.

[14] S. Itaya, "New experimental formula of the coefficient of pipe friction," *Journal of the Japan Society of Mechanical Engineers*, Vol. 48, No. 332, 1945, p.84, [in Japanese].

[15] S. Ishigai et al., "Forced convective heat transfer and pressure drop for supercritical pressure water flowing in tube (1st report, Experimental results of heat transfer)," *Transactions of the Japan Society of Mechanical Engineers*, Vol. 47, No. 424, 1981, pp.2333-2342, [in Japanese].