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ABSTRACT

A method to obtain the local heat transfer coefficients of carbon dioxide at supercritical pressures in vertical multi-port mini-channels and in a vertical small tube under cooling conditions was presented through the combination of experiments and numerical simulation in this paper. The CFD code FLUENT 6.1 was used to simulate the convection heat transfer of cooling water to determine the local heat flux on the outside wall of the multi-port mini-channels and the small tube for the experimental conditions. The effects of pressure, mass flow rate of CO_2, inlet temperature, and mass flow rate of cooling water on heat transfer were investigated. The experimental results showed that the sharply variable thermophysical properties of carbon dioxide have a significant effect on the local heat transfer coefficient which varies significantly along the vertical mini-channels and small tube when the CO_2 bulk temperatures werewithin the near-critical region.

1. INTRODUCTION

CO_2 is a non-flammable, odorless, nontoxic natural working fluid, with ODP = 0, GWP = 1, and no unexpected long-term effects to the environment. In environmental, cost and safety terms, CO_2 is essentially an ideal refrigerant (Lorentze and Pettersen, 1993; Riffat et al., 1997). Thus, the CO_2 high-pressure trans-critical compression cycle for air-conditioners and heat pumps is currently of great interest. Understanding of the convection heat transfer mechanisms and heat transfer enhancement mechanisms of CO_2 at supercritical pressures for cooling conditions is important for improving designs of the gas cooler and the internal heat exchanger.

In the super-critical region, small fluid temperature and pressure variations produce large changes in the thermophysical properties, as a result, the convection heat transfer of fluids at supercritical pressures has many special features due to the sharp variations of the thermophysical properties with temperature and pressure. The convection heat transfer of super-critical fluids in vertical and horizontal normal size tubes for cooling conditions has been investigated over the past 50 years by Krasnoshchekov and Protopopov (1966), Baskov et al. (1977), Petrov and Popov (1985), and Yoon et al. (2003), with various correlations developed.

In recent years, the convection heat transfer of fluids at super-critical pressures has also been investigated for mini/micro tubes or channels with cooling conditions. Pettersen et al. (2000) investigated the average heat transfer coefficient of super-critical CO_2 in multi-port mini-channels for cooling conditions with an inner diameter of 0.79 mm. They found that Gnielinski's correlation for the single-phase heat transfer coefficient shows satisfactory correspondence with experiment results. Liao and Zhao (2002b) experimentally investigated axially-averaged convection heat transfer to supercritical carbon dioxide in horizontal mini/micro circular tubes of 0.50, 0.70, 1.1, 1.40, 1.55, and 2.16 mm diameters cooled at a constant temperature. They found that the buoyancy effect was still significant for forced convection of supercritical CO_2 through the horizontal tubes at high Reynolds numbers and developed correlations for the axially-averaged Nusselt number for forced convection of supercritical carbon dioxide in horizontal mini/micro tubes. Dang and Hihara (2004) investigated experimentally the effect of mass flux, pressure, and heat flux on the axially-averaged heat transfer coefficient and pressure drop of CO_2 at supercritical pressure under cooling conditions for horizontal cooling tubes with inner diameters ranging from 1 to 6 mm and proposed a modified Gnielinski equation by selecting the reference temperature properly. Son (2005) conducted experimental measurements on the axially-average heat transfer coefficients and pressure drop of each subsection in a 6000 mm long and with ID = 7.75 mm test section which was divided to 12 subsections, discussed the effect of pressure and mass flux on heat transfer coefficient and pressure drop and proposed new correlation based on Dittus-Boelter equation. Huai et al. (2005) investigated the fluid flow and heat transfer characteristics of supercritical CO_2 in a horizontal multi-port extruded aluminum test section consisting of 10 circular channels with inner diameters of 1.31 mm. They used 12 heat flux sensors to measure the local and average heat transfer coefficients as CO_2 was cooled in the multi-port circular channels. Their results indicate that the operating pressure, the mass velocity and the temperature of the CO_2 all significantly affected the fluid flow and heat transfer characteristics.

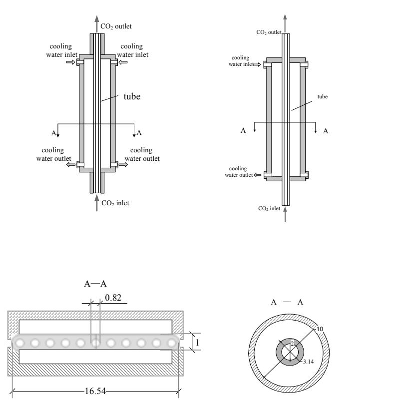
However, in the research of Huai et al. (2005) the local heat transfer coefficients were not the real local values, but were area averaged heat transfer coefficients. To the authors' knowledge, there is no work measuring the local convection heat transfer coefficients of super-critical fluids in mini/micro scale tubes and channels for cooling conditions due to the difficulty of the measurements. This local heat transfer performance is important for understanding the convection heat transfer mechanism. The motivation of the present study is to develop a new method to obtain the local heat transfer coefficients for cooling conditions by combining numerical simulations and experimental measurements. The influence of the sharp variations of the fluid thermophysical properties near the pseudo-critical point on the convection heat transfer for cooling conditions is also investigated.

2. INVESTIGATION METHOD

The aim of the present paper is to investigate experimentally the local heat transfer coefficients of carbon dioxide at supercritical pressures in vertical multi-port mini-channels and small tube under cooling conditions. The physical model and test section is shown in Figure 1. The main difficulty in getting the local heat transfer coefficients is to get the local heat flux. In order to solve this problem, this investigation combines numerical simulations and experimental measurements to determine the local heat flux. FLUENT 6.1 was used to simulate the convection heat transfer of the cooling water in the outside channel to calculate the local heat flux on the outside wall based on the experimental conditions. The local heat transfer coefficients were then obtained using the measured temperature on the outside wall and the local heat fluxes calculated from the numerical simulation.

2.1 Experimental Apparatus

The experimental system used to investigate the convection heat transfer of CO_2 at super-critical pressures in vertical multi-port mini-channels and small tubes for cooling conditions is shown schematically in Figure 2. The test section for small tube was a 150 mm-long vertical tube-in-tube counter-flow heat exchanger, with CO_2 flowed upward from the bottom and the cooling water flowed downward in the perspex water tube jacket from top. The inner tube was smooth stainless steel 1Cr18N9T tube with inside and outside diameters of 2.00 mm and 3.14 mm. The test section for multi-port mini-channels was also counter-flow heat exchanger and the same flow direction of both cooling water and CO_2. The length, width and height of the multi-port mini-channel cooled directly by the water were 200.0 mm, 16.54 mm and 1.78 mm, respectively. The inner diameter of the mini-channels was 0.82 mm.



(a) Multi-port mini-channels (b) Small tube

Figure 1: Physical model and test section geometry

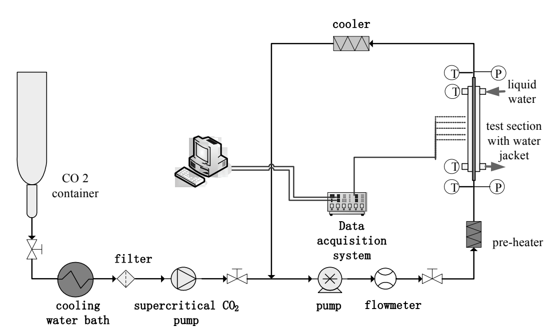


Figure 2: Experimental system

The parameters measured in the experiments included the wall temperatures, the CO_2 inlet and outlet temperatures, the CO_2 inlet pressure, the CO_2 mass flow rate, the cooling water inlet and outlet temperatures, the cooling water inlet pressure, and the cooling water mass flow rate. The local outside wall temperatures of the multi-port tube were measured with 11 copper-constantan thermocouples welded onto the outer surface of the tube. The local outside wall temperatures of the small tube were measured with 10 copper-constantan thermocouples welded onto the outer surface of the tube with equal space. The inlet and outlet temperatures of the water jacket were measured by armored thermocouples. Mixers were installed before and after the test section to mix the fluid before the inlet and outlet fluid temperatures were measured by accurate thermal resistors. The CO_2 inlet pressure was measured using a pressure transducer (Model EJA430A). The CO_2 mass flow rate was measured using a Coriolis-type mass flowmeter (Model MASS2100/MASS6000, MASSFLO, Danfoss). The cooling water mass flow rate was measured using the weighing method.

2.2 Method to Evaluate the Local Heat Transfer Coefficient

The local heat transfer coefficient of CO_2 at supercritical pressures in the vertical multi-port mini-channels for cooling conditions $h(x)$, is defined as:

$$h(x) = \frac{q_{wall-in}(x)}{T_{CO2}(x) - T_{wall-in}(x)} \tag{1}$$

As mentioned above, the local heat flux, $q_{\text{wall-in}}(x)$, can be obtained from the numerical simulations of convection heat transfer of the cooling water in the outside channel based on the measured data. The boundary condition for the cooling water flow in the outside channel was the temperature distribution along the tube outer wall $T_{\text{wall-out}}(x)$ which was calculated from a fit of the temperature data measured by thermocouples on the external wall. The other surfaces of the outside channel were assumed to be adiabatic. The outer wall heat flux, $q_{\text{wall-out}}(x)$, was obtained using the numerical simulation method by FLUENT which will be described in details later. Then, the inner wall temperature distribution $T_{\text{wall-in}}(x)$ was calculated from $T_{\text{wall-out}}(x)$ and the heat flux distribution, $q_{\text{wall-out}}(x)$, assuming one-dimensional heat conduction in the wall:

$$T_{wall-in}(x) = T_{wall-out}(x) + \frac{\delta}{\lambda} q_{wall-out}(x) \qquad \text{for multi-port mini-channels}$$

$$T_{wall-in}(x) = T_{wall-out}(x) + \frac{q_{wall-out}(x) d_{out}}{2\lambda} \ln(d_{out} / d_{in}) \quad \text{for tube} \tag{2}$$

The difference between $T_{\text{wall-out}}(x)$ and $T_{\text{wall-in}}(x)$ is very small (less than 0.2) due to the small wall thickness δ of the multi-port mini-channels and the small tube, the large thermal conductivity of the aluminum multi-port mini-channels and stainless steel tube, λ, and the not very high heat flux, q (less than 40000 W/m^2). Therefore, the one-dimensional heat conduction assumption in the wall and Equation (2) is acceptable.

The heat flux distribution on the multi-port mini-channels and the small tube inner wall was calculated using an energy balance between the inner wall and the outer wall:

$$q_{wall-in}(x) = \frac{q_{wall-out}(x) \cdot 2L}{N\pi d_{in}} \quad \text{for multi-port mini-channels}$$

$$q_{wall-in}(x) = \frac{q_{wall-out}(x) \cdot d_{out}}{d_{in}} \qquad \text{for small tube} \tag{3}$$

The enthalpy distribution of the CO_2 at super-critical pressures in the mini-channels and small tube $H_{CO2}(x)$ was calculated using the local energy balance:

$$H_{CO2}(x) = H_{CO2}(0) - \frac{N\pi d_{in}}{G_{CO2}} \int_0^x q_{wall-in}(x) dx \quad \text{for multi-port tube}$$

$$H_{CO2}(x) = H_{CO2}(0) - \frac{\pi d_{in}}{G_{CO2}} \int_0^x q_{wall-in}(x) dx \quad \text{for small tube} \tag{4}$$

The bulk temperature distribution of the CO_2 along the mini-channels and small tube $T_{CO2}(x)$ was obtained using the NIST software REFPROP 7.0 referenced from the CO_2 enthalpy distribution.

2.3 Validation of the Numerical Simulation for Convection Heat Transfer of Cooling Water

The accuracy of the local CO_2 heat transfer coefficients in the vertical multi-port mini-channels and small tube depends on the simulation of the convection heat transfer in the cooling water in the outside channel and the determination of the local heat flux on the outside wall of the multi-port mini-channels and small tube. Therefore, the simulation of the convection heat transfer in the cooling water in the outside channel using FLUENT 6.1 was carefully validated experimentally.

A thin film electrical heater for the multi-port mini-channels and directly heated by electricity for small tube was used to simulate constant heat flux, and the electric power input to the heater was calculated from the measured voltages, the currents and the heater resistance, other device and measurements are similar to the experiment with CO_2. The inlet and outlet pressures were measured using accurate gauges with accuracies of 0.25% of the full scale range of 0.1 MPa and 0.6 MPa. The water mass flow rate was measured by weighing the fluid flowing from the channel for a given time period. The maximum error in the water flow rate was less than 0.58%.

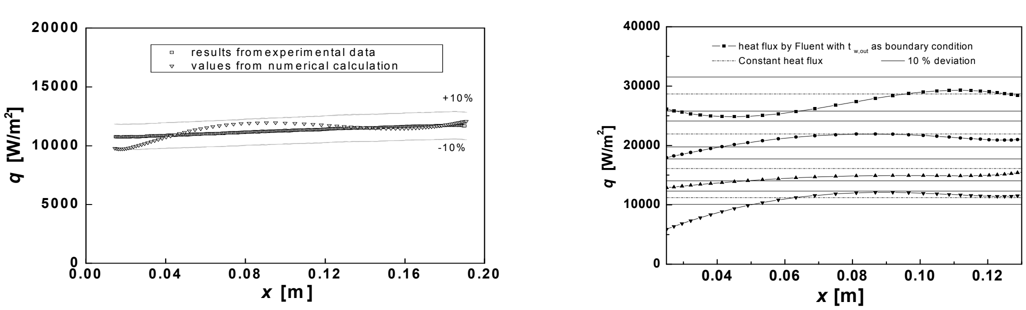
Figure 6 shows the results of validation experiments. 2-D solver was used for the cooling water domain of multi-port mini-channels and Axial-symmetric solver for that of small tube. Simulated heat fluxes range from 10 ~ 30 kW/m^2. Velocity inlet and pressure outlet were set as the boundary conditions, external wall temperature distribution was obtained by 4-orded polynomial fit based on the experimental data measured by the thermal couples welded on the external surface of the test section, and was set as boundary conditions for the side contacting the external tube surface of the cooling water domain by user defined function in Fluent. The other side of the cooling water domain is adiabatic.

Comparison of the experimental results with numerical simulations using different turbulence models including the Shear-Stress Transport (SST) k - ω turbulence model, RNG model, standard k - ε model showed that all these three turbulence models give consistent result of heat flux distribution along the outer wall of the tube. Standard k - ε model with enhanced wall treatment was selected due to its rapider convergence to give the distribution of heat flux along the outer wall. The mesh was treated carefully to assure that y+ near wall is less than 1.

As shown in Figure 3, the difference between numerically predicted and measured wall heat fluxes was about ±10 %, with the maximum does not exceed 15%. Therefore, the results verify the reliable simulation for the convection heat transfer of the cooling water in the outside channel and the determination of the local heat flux on the outside wall of the multi-port mini-channels and tube.

Prior to installation, the thermocouples and the RTDs were calibrated by the National Institute of Metrology P.R. China. The accuracies were within 0.25 ℃ in the temperature range of 0~ 150 ℃ . The accuracy of the pressure transducer (Model EJA430A) is 0.075% of the full range of 12 MPa. According to the instructions, the accuracy of the Coriolis-type mass flow meter (Model MASS2100/MASS6000, MASSFLO, Danfoss) is 0.1% of actual mass flow rate with 95% confidence (probability) for flow with 5%~100% of the sensor's maximum flow rate. For flow < 5% of the sensor's maximum flow rate, the following formula should be used to calculate the error:

$$\varepsilon_G = \pm\sqrt{0.1^2 + \left(\frac{Z \times 100}{m}\right)^2}$$



a) b)

Figure 3 Comparison of the predicted and measured wall heat fluxes a)for Multi-Port Mini-channels b)for small tube

Where ε_G = Error (%), Z = Zero point error (kg/h) (=0.002 kg/h), m = Mass flow (kg/h). The flow rate during the measurements was about 0.53~1.59 kg/h. Therefore, the relative uncertainty of the mass flow rate was 0.39~0.16%.

Based on the result of the validation experiment, the uncertainty of the heating flux was about 15.0%. According to the accuracies of the instruments and a detailed analysis of the uncertainty, it was showed that the uncertainty of the tube inner surface temperatures was 0.3 ℃ in the temperature range of 0~150 ℃, the uncertainty of the local bulk fluid temperature was 0.2 ℃, the relative uncertainty of the temperature drop between the wall and fluid was 7.2%. The root-mean-square experimental uncertainty of the heat transfer coefficient was estimated to be 16.6%. The experimental uncertainties in the inlet pressures were estimated to be 0.13%.

3. RESULTS AND DISCUSSION

The main aim of the present study is to develop a new method to investigate the local heat transfer coefficients for cooling conditions by combining numerical simulations and experimental measurements and analyze preliminarily the influence of different factors including cooling water mass flow rate, CO_2 mass flux, inlet temperature and pressure qualitatively, therefore, the range of the parameters in the experiments is limited. The CO_2 mass flow rate ranged from 0.5 to 2.2 kg/h, the inlet pressure from 7.4 to 9.5 MPa, the local heat flux on the surface from 2.5 to 40 kW/m^2 and the bulk temperature from 30 to 100 ℃ . The inlet Reynolds numbers range from 4000 to 9000.

3.1 Convection Heat Transfer in the Multi-Port Mini-Channels

Figure 4 and Figure 5 compares the bulk CO_2 temperature distributions and heat transfer coefficients along the mini-channels for different heat fluxes (corresponding to different Reynolds numbers of cooling water) for the same inlet CO_2 pressure, inlet CO_2 temperature and CO_2 mass flow rate at p_0=7.4 Mpa and 8.9 Mpa respectively. The CO_2 bulk temperatures in the mini-channels for cases ① and ④ are higher than the pseudo-critical temperature T_{pc} due to the small heat flux (small cooling water mass flow rate), while the CO_2 bulk temperatures for cases ②, ③, ⑤ and ⑥ pass through the pseudo-critical point so that they were lower than the pseudo-critical point downstream. Therefore, for cases ②, ③, ⑤ and ⑥, the variable thermophysical properties have a much greater influence on the convection heat transfer. For cases ① and ④, since the CO_2 bulk temperatures along the mini-channels were higher than the pseudo-critical temperature, the CO_2 thermophysical properties did not vary much and therefore, the local CO_2 heat transfer coefficients did not vary much along the mini-channels. However, for cases ②, ③, ⑤ and ⑥, since the CO_2 bulk temperatures pass through the pseudo-critical point, the local CO_2 heat transfer coefficients vary significantly along the mini-channels.

Figure 6 shows the variation of the local heat transfer coefficient with the ratio T_{CO2}/T_{pc}. Since the specific heat increases sharply near the pseudo-critical temperature, the convection heat transfer intensifies significantly in the range T_{CO2}/T_{pc} = 1 ~ 1.2, for cases ②, ③, ⑤ and ⑥. The variation of the local heat transfer coefficient for cases ① and ④ was much smoother because the CO_2 bulk temperatures for these two cases were much higher than the pseudo-critical temperature.

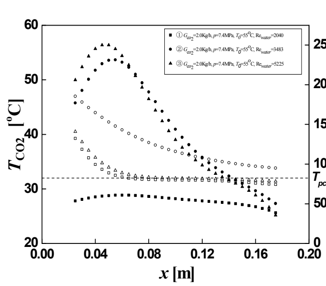


Figure 4: $h(x)$ and $T_{bulk}(x)$ for Micro-port mini-channels p_0=7.4 MPa, $T_{CO2,0}$=55.0 ℃, $Re_{CO2,0}$=4416

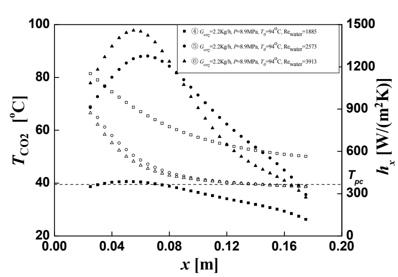


Figure 5: $h(x)$ and $T_{bulk}(x)$ for Micro-port mini-channels p_0=8.9 MPa, $T_{CO2,0}$=94.0 ℃, $Re_{CO2,0}$=4416

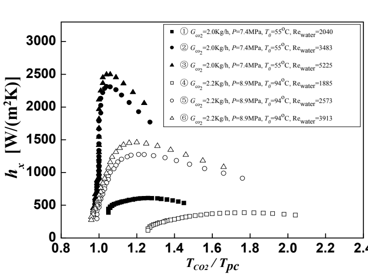


Figure 6: Local CO_2 heat transfer coefficient for various temperatures

3.2 Convection Heat Transfer in the Small Tube

Effects of cooling water mass flux, CO_2 mass flux, inlet temperature and pressure were investigated, respectively. In all cases with small tube concerned, wall temperatures are lower than T_{pc}, while bulk temperatures are higher than T_{pc} in the inlet and approach or pass through T_{pc} in the outlet.

3.2.1 Effect of cooling water mass flux: Figure 7 shows the effect of cooling water mass flux on the heat transfer coefficient at $T_{CO2,0}$ = 55 ℃, p = 8.8 MPa, G = 0.53 kg/h along the tube. Here, T_{pc} = 38.91 ℃ at 8.8 Mpa.

As the same situation for the case with multi-port mini-channels, increasing cooling water mass flux caused a slightly increase for the heat transfer coefficient. This can be explained by that when the cooling water mass flow rate increases, the heat transfer between the cooling water and the external surface of the tube wall is enhanced accordingly, which results in an increasing of the heat flux from the CO_2 inside the tube to the tube wall, and the bulk temperature of CO_2 approaches more to the pseudo critical temperatures, as a result, the heat capacity of CO_2 bulk increases.

3.2.2 Effect of CO_2 mass flux: As shown in Figure 8, mass flux of CO_2 has a great influence to the heat transfer coefficient, which increased significantly as the increasing the mass flow rate of CO_2. This is consistent with the results of regular fluids whose thermophysical properties do not vary too much with temperature. When mass flow rate is doubled, the peak value of heat transfer coefficient increased by a factor 1.5. This is mainly due to the increase of turbulence kinetic energy with the increase of mass flow rate, which results in a strong mixing effect and disturb of fluids, and therefore leads to an enhancement to the heat transfer between the fluid and solid tube wall. This is also the explanation for the increase of heat transfer coefficient with mass flow rate for regular fluids. For CO_2 at supercritical pressure, another reason is that the wall temperature increased with the increase of mass low rate and approaches to the pseudo critical temperature, as a result, the heat capacity of the CO_2 near the wall greatly increased, as well as the heat conductivity, which also results in an enhancement of heat transfer between the CO_2 and the solid tube wall. The combination of these two factors leads to the increasing of heat transfer coefficient with CO_2 mass flow rate, and the more the wall temperature approaches to the pseudo critical temperature, the more significant of the heat transfer coefficient increase with the mass flow rate of CO_2.

3.2.3 Effect of inlet temperature: As is shown by Figure 9, the more gas-like state of CO_2 leads to a poor heat transfer between the fluid inside and the tube wall in the front part of the test section in the case in which the inlet temperature is 70 ℃, much higher than the pseudo critical temperature. However, this is compensated by the more close to the T_{pc} of wall temperature in the later part of the test section for the case in which the inlet temperature is 70 ℃, although the bulk temperature is not as close to the pseudo critical temperature than that in the other case with a lower inlet temperature.

3.2.3 Effect of Pressure: Figure 10 shows the effect of pressure to heat transfer coefficient of supercritical CO_2 under cooling conditions. Generally, there is no big difference for the heat transfer coefficient between cases under different pressures when the bulk temperature is much higher than T_{pc}, but the heat transfer coefficient increases drastically with decreasing pressure (approaching p_c) with the local heat transfer coefficients having maximums when the CO_2 bulk temperatures within the range of T_{CO2}/T_{pc} = 1 ~ 1.2 (see Figure 11). This is mainly due to the increasing of heat capacity in this range.

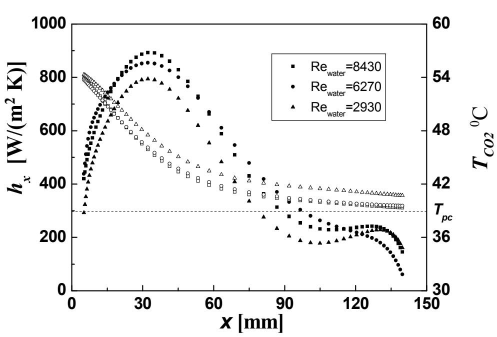


Figure 7: $h(x)$ and $T_{bulk}(x)$ for various G_{water}
$Re_{CO2,0}$=4340, p_0=8.8 MPa, $T_{CO2,0}$=55.0 ℃

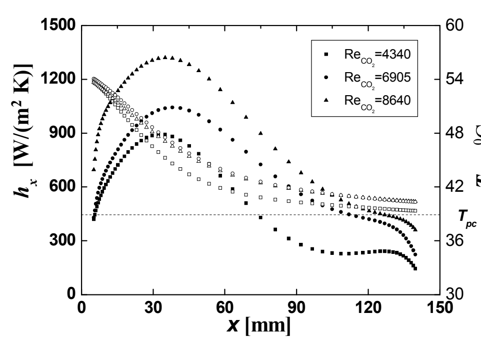


Figure 8: $h(x)$ and $T_{bulk}(x)$ for various G_{CO2}
$Re_{water,0}$=8430, p_0=8.8 MPa, $T_{CO2,0}$=55.0 ℃

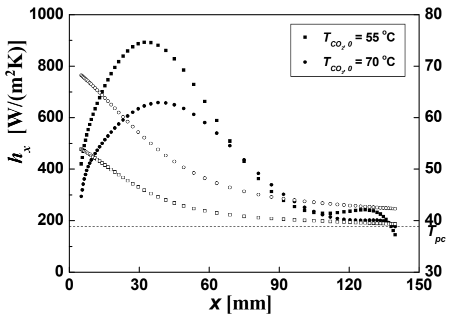


Figure 9: $h(x)$ and $T_{bulk}(x)$ for various T_{in}
$Re_{CO2,0}$=4340, p_0=8.8 MPa, $Re_{water,0}$=8430

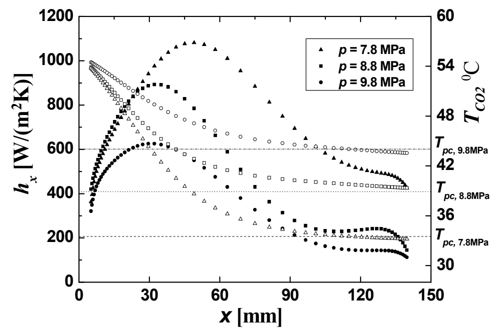


Figure 10: $h(x)$ and $T_{bulk}(x)$ for various p_0
$Re_{water,0}$=8430, $Re_{CO2,0}$=4340, $T_{CO2,0}$=55.0 ℃

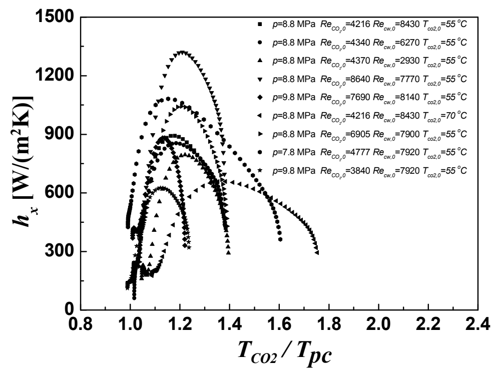


Figure 11: Local CO_2 heat transfer coefficient for various CO_2 temperatures

4. CONCLUSIONS

- The local heat transfer coefficients of CO_2 at supercritical pressures in vertical multi-port mini-channels and small tubes with cooling conditions were determined by combining simulations of the convection heat transfer in the cooling water in the outside channel with experimental measurements.
- The effects of cooling water mass flow rater, CO_2 mass flow rate, CO_2 inlet temperature and operating pressure were analyzed. The heat transfer coefficients increase with the increasing of cooling water mass flow rate according to a larger heat flux, increase significantly with the increase of CO_2 mass flow rate, and decrease with the operating pressure above p_c.
- The local heat transfer coefficients of CO_2 vary along the vertical mini-channels and small tube. The convection heat transfer is greatly intensified within the range of T_{CO2}/T_{pc} = 1 ~ 1.2.as shown in Figure 6 and Figure 11, and the heat transfer coefficient reaches to a maximum when the bulk temperature of CO_2 is a little bit higher than the pseudo critical temperature.

NOMENCLATURE

d	diameter of the mini-channel or small tube	(m)	**Subscripts**	
L	width of the multi-port channel	(m)	CO_2	CO_2 fluid
N	number of mini-channels	(–)	pc	pseudo-critical
G	mass flow rate	(kg/s)	bulk	bulk
H	bulk specific enthalpy	(J/kg)	in	inner surface
h	local heat transfer coefficient	(W/m^2.K)	out	outer surface
p	pressure	（MPa）	0	entrance

q	heat flux on the inner tube surface	(W/m^2)	water: cooling water
Re	Reynolds number	(–)	wall: wall
T	temperature	(℃)	
x	axial coordinate	(m)	
δ	wall thickness	(m)	
λ	thermal conductivity of the multi-port channel material (aluminum) or the small tube (stainless seel)	(W/m.K)	

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