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International Journal of Heat and Mass Transfer 51 (2008) 3052–3056
International Journal of HEAT and MASS TRANSFER
www.elsevier.com/locate/ijhmt

Experimental and numerical investigation of convection heat transfer of CO_2 at supercritical pressures in a vertical mini-tube

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Received 4 April 2006; received in revised form 30 August 2007
Available online 5 November 2007

Abstract

Convection heat transfer of CO_2 at supercritical pressures in a 0.27 mm diameter vertical mini-tube was investigated experimentally and numerically for inlet Reynolds numbers exceeding 4.0×10^3. The tests investigated the effects of heat flux, flow direction, buoyancy and flow acceleration on the convection heat transfer. The experimental results indicate that the flow direction, buoyancy and flow acceleration have little influence on the local wall temperature, with no deterioration of the convection heat transfer observed in either flow direction for the studied conditions. The heat transfer coefficient initially increases with increasing heat flux and then decreases with further increases in the heat flux for both upward and downward flows. These phenomena are due to the variation of the thermophysical properties, especially c_p. The numerical results correspond well with the experimental data using several turbulence models, especially the Realizable k–ε turbulence model.

Keywords: Mini-tube; Convection heat transfer; Supercritical pressures CO_2; Experiments; Numerical simulation; Buoyancy; Flow acceleration

1. Introduction

In the supercritical region, small fluid temperature and pressure variations can lead to significant changes in the thermophysical properties. Convection heat transfer of fluids at supercritical pressures has many special features due to the sharp variations of the thermophysical properties. A number of detailed reviews covering the research on heat transfer to fluids at supercritical pressures can be found in the literature, for example [1–4].

Recently, there has been increasing interest in heat transfer of fluids at supercritical pressures in small/mini/micro-scale tubes or channels [5–7]. However, there have been very few investigations of the local heat transfer performance in the mini/micro-tubes which is important to understanding the heat transfer mechanism. In addition, the published results for mini/micro-tubes have contradictions.

He et al. [7] carried out computational simulations of experiments of turbulent convection heat transfer of carbon dioxide at supercritical pressures in a 0.948 mm diameter vertical tube. Their results showed that for mini tubes such as the 0.948 mm diameter tube and for a large Reynolds number of 10^4, the buoyancy effect is insignificant. However, heat transfer can still be significantly impaired as a result of flow acceleration at high heat fluxes, which reduces the turbulence production.

Jiang et al. [8] experimentally and numerically investigated convection heat transfer of supercritical pressure CO_2 in a small 2.0 mm diameter vertical tube at low Reynolds numbers (<2500).

This present paper describes experimental and numerical investigations of the local convection heat transfer of supercritical pressure CO_2 in a 0.27 mm diameter vertical mini-tube for upward and downward flows at high Reynolds numbers (from 4.0×10^3 to 13.0×10^3). The effects of heat flux, flow direction, buoyancy and flow acceleration on the convection heat transfer are investigated.

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doi:10.1016/j.ijheatmasstransfer.2007.09.008

Nomenclature

c_p	specific heat at constant pressure (J/(kg K))
d	tube inner diameter (m)
h_x	local heat transfer coefficient (W/(m^2 °C))
p	pressure (N/m^2)
q_w	heat flux on the inner tube surface (W/m^2)
Re	Reynolds number
T	temperature (°C) or (K)
T_{pc}	pseudo-critical temperature (°C) or (K)
u	x-direction velocity (m/s)
x	axial coordinate (m)

Greek symbol

μ	dynamic viscosity (kg/(m s))

Subscripts

b	bulk fluid temperature
f	fluid
i	inner surface
in	inlet section
w	wall
x	local values

2. Experimental system and data reduction

The scheme and detailed description of the experimental system were presented by Jiang et al. [8]. The test section was a vertical stainless steel 1Cr18N9T tube with inside and outside diameters of 0.27 mm and 1.59 mm. The heated part of the test section was 90 mm long, and the sections before and after the heated section were each 10.8 mm long (40d). The test section was heated directly using low-voltage alternating current to simulate a constant heat flux. CO_2 flowed into the test section either from the bottom for upward flow or from the top for downward flow.

The parameters measured in the experiments included the wall temperatures, the inlet and outlet temperatures, the mass flow rate, the inlet pressure, the pressure drop across the test sections, the heater voltages, the current and the electrical resistance. The local wall temperatures along the test section were measured with 14 fine T-type thermocouples welded onto the outer tube surface. Flow mixers were installed upstream and downstream of the test section to mix the fluid before the inlet and outlet fluid temperatures were measured by accurate RTDs.

The local heat transfer coefficient, h_x, at each axial location was calculated as

$$h_x = \frac{q_w}{T_{w,i}(x) - T_{f,b}(x)}$$

The methods to determine the local temperatures of the inner tube surface $T_{w,i}(x)$, the heat flux on the inner surface q_w, the local bulk fluid temperature $T_{f,b}(x)$, and the Reynolds number can be found in [8].

A detailed uncertainty analysis showed that the maximum uncertainty of the heat rate into the test section was ±11.0%. The relative uncertainty of the mass flow rate varied from 1.1% to 0.20%. The root-mean-square experimental uncertainties of the heat transfer coefficient were estimated to be ±12.4%. The experimental uncertainties in the inlet pressures were estimated to be ±0.13%.

3. Experimental results and discussion

3.1. Local wall temperature, bulk fluid temperature and heat transfer coefficient

Fig. 1 presents the local wall temperature (solid symbols) and local bulk fluid temperature (hollow symbols) distributions along the tube for upward flow (a) and down-

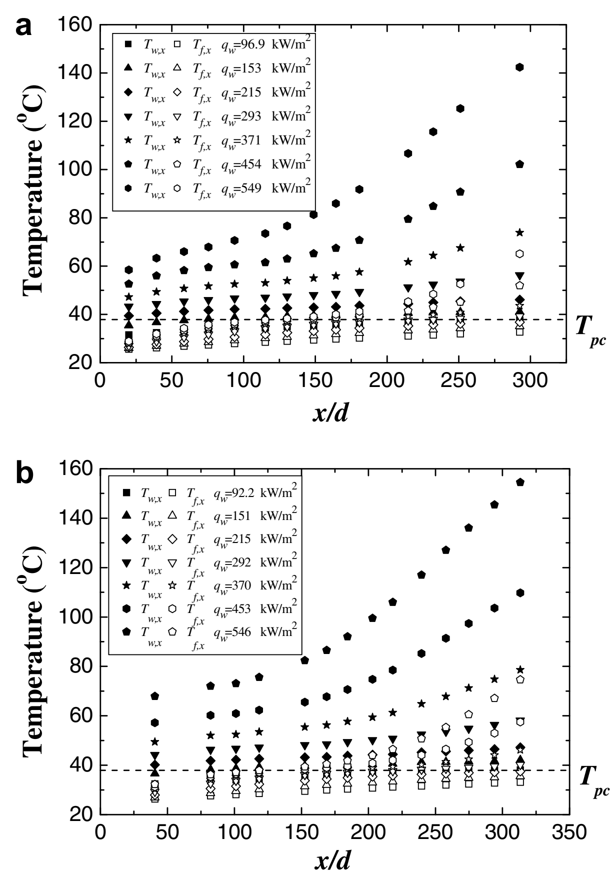

Fig. 1. Local wall temperatures (solid symbols) and bulk fluid temperatures (hollow symbols) for (a) upward flow and (b) downward flow. (a) $P_{in} = 8.60$ MPa, $T_{in} = 25.0$ °C, $Re_{in} = 10.6 \times 10^3$; (b) $P_{in} = 8.60$ MPa, $T_{in} = 25.0$ °C, $Re_{in} = 10.5 \times 10^3$.

ward flow (b) for various heat fluxes at an inlet pressure of 8.60 MPa and inlet temperature of 25.0 °C. The inlet Reynolds numbers are 10.6×10^3 (Fig. 1a) and 10.5×10^3 (Fig. 1b). The inlet fluid temperature is less than the pseudocritical temperature ($T_{pc} = 37.9$ °C for $p = 8.60$ MPa). For both low and high heat fluxes, the local wall temperatures increased monotonically along the tube for both upward and downward flows without abnormal temperature distributions.

Fig. 2 presents the local convection heat transfer coefficient (HTC) distributions for both upward and downward flows for the same conditions as in Fig. 1. At relatively low heat fluxes (96.9 kW/m^2 in Fig. 2a and 92.2 kW/m^2 in Fig. 2b), the HTC decreased along the tube in the entrance region and then increased further down the tube. For this case the wall temperature and fluid temperature along the tube are lower than the pseudocritical temperature as shown in Fig. 1. With increasing temperature along the tube, the viscosity and thermal conductivity decrease while the specific heat capacity (c_p) increases. The increase of HTC along the tube is mainly due to the increase in c_p.

For higher heat fluxes (153 kW/m^2 in Fig. 2a and 151 kW/m^2 in Fig. 2b), the local HTC increased along the tube on most of the tube for both upward and downward flows. For these conditions, the wall temperature on most of the test section is a little larger than the pseudocritical temperature, while the fluid bulk temperature along the tube is lower than the pseudocritical temperature as shown in Fig. 1. With increasing temperature along the tube, the viscosity and thermal conductivity decrease while the integrated specific heat capacity (c_p) along the tube increases.

With further increasing the heat fluxes (215–549 kW/m^2 in Fig. 2a and 215–546 kW/m^2 in Fig. 2b), the local HTC increased along the tube and then decreased further down the tube. For these conditions, Fig. 1 shows that the wall temperature on the later part of the test section is much higher than the pseudocritical temperature. When the bulk temperature approaches to (but is a little less than) the pseudo-critical temperature and the wall temperature is larger than the pseudo-critical temperature, the convection heat transfer coefficient reaches a maximum as shown in Figs. 1 and 2. Fig. 2 also shows that with increasing heat flux, the position of the maximum convection heat transfer coefficient moves towards the inlet. The results presented in Fig. 2 demonstrate that the sharp variations of the themophysical properties of CO_2 at supercritical pressures significantly influence the convection heat transfer.

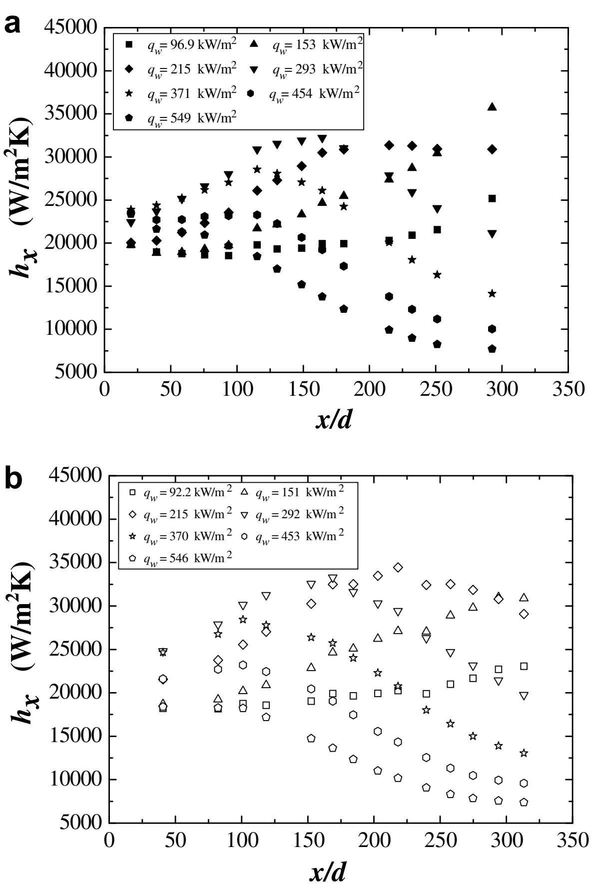

Fig. 2. Local heat transfer coefficients for various heat fluxes for (a) upward flow and (b) downward flow. (a) $P_{in} = 8.6$ MPa, $T_{in} = 25$ °C, $Re_{in} = 10.6 \times 10^3$; (b) $P_{in} = 8.6$ MPa, $T_{in} = 25$ °C, $Re_{in} = 10.5 \times 10^3$.

3.2. Effect of buoyancy on the convection heat transfer

The buoyancy parameter, Bo^*, introduced by Jackson and Hall [9] can be used to estimate the influences of buoyancy on the heat transfer. The buoyancy number, Bo^* was defined as [9]:

$$Bo^* = Gr^*/(Re^{3.425}Pr^{0.8})$$

According to Jackson and Hall [9], for turbulent flow the buoyancy will significantly influence the heat transfer for $Bo^* > 5.6 \times 10^{-7}$. $Gr/Re^2 > 10^{-2}$ is also a commonly used criterion for buoyancy to be significant.

The experimental results in the present study showed that for $P_{in} = 8.60$ MPa and $Re_{in} = 4.0 \times 10^3$, 6.0×10^3, 7.7×10^3, 10.5×10^3, 13.0×10^3, the buoyancy effect is insignificant for both low and high heat fluxes. Fig. 3

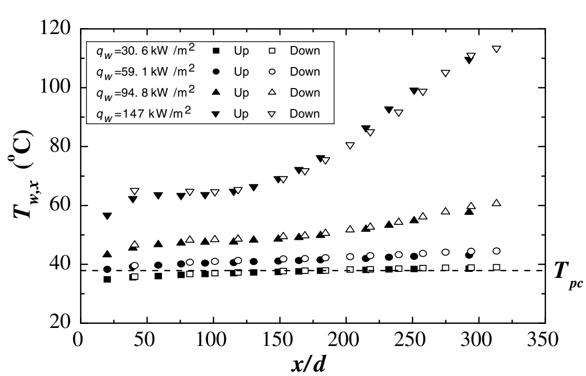

Fig. 3. Wall temperature for upward and downward flows $p_{in} = 8.60$ MPa, $T_{in} = 30.0$ °C, $Re_{in} = 4 \times 10^3$, $Gr/Re^2 \sim 10^{-4}$, and $Bo^* \sim 10^{-8}$.

compares the local wall temperatures for upward and downward flows for $Re_{in} = 4.0 \times 10^3$. The similarity between the local wall temperatures for upward and downward flows indicates that the buoyancy effect is insignificant for these relatively high *Re*. For the experimental conditions shown in Fig. 3 and with the parameters *Gr*, *Re* and Bo^* based on the average of the inlet and outlet fluid temperatures, $Gr/Re^2 \sim 10^{-4} < 0.01$ and $Bo^* \sim 10^{-8}$. Therefore, for this 0.27 mm inner diameter mini-tube and relatively high Reynolds numbers such as $Re \geqslant 4.0 \times 10^3$, the buoyancy effect is very weak even for high heat fluxes. This conclusion is consistent with those in [7].

3.3. Effects of flow acceleration

McEligot et al. [10] investigated the influence of flow acceleration due to heating. McEligot et al. [10] introduced a heating acceleration parameter to assess this effect:

$$K_v = \frac{4q_w d}{Re^2 \mu c_p T}$$

McEligot et al. [10] suggested that for turbulent flow when $K_v \leqslant 3 \times 10^{-6}$ the fluid flow remains turbulent. When $K_v \geqslant 3 \times 10^{-6}$ the turbulence may be significantly reduced and the flow may even re-laminarise, which reduces the overall heat transfer.

For the experimental conditions shown in Fig. 3, K_v is about 10^{-8}. Therefore, according to the abovementioned criteria the effect of flow acceleration due to heating on the heat transfer for the experimental conditions at relatively high Reynolds numbers such as $Re \geqslant 4.0 \times 10^3$ is insignificant.

4. Numerical simulation and comparison with experiments

In the present study, the inside and outside diameters of the vertical stainless steel tube were 0.27 mm and 1.59 mm, so the tube is relatively thick and heat is also transported along the axial direction in the tube wall. Therefore, the model analyzed both convection heat transfer in the vertical mini-tube and the heat conduction in the tube wall with an internal heat source. The heated section was 90 mm long, the sections before and after the heated section were 10.8 mm long (40*d*). Flow enters the tube with a constant velocity, u_{in}, and constant temperature, T_{in}. The flow was assumed to be two-dimensional, steady, turbulent flow.

The conjugate heat transfer with an internal heat source in the vertical mini-tube was numerically simulated using FLUENT 6.1 with various turbulence models used to model the turbulence.

The governing equations for the steady-state, two-dimensional turbulent flow of a supercritical pressure fluid in a vertical tube with consideration of the temperature-dependent property variations and buoyancy can be found in [7].

The NIST Standard Reference Database 23 (REFPROP) Version 7 was used to calculate the temperature and pressure dependent properties of carbon dioxide. The SIMPLEC algorithm was used to couple the pressure and velocities. The QUICK advection model was used in the momentum, energy, turbulent kinetic energy, and turbulent energy dissipation equations. Calculations with various numbers of elements in the axial direction and in the radial direction showed that the results with 1240 nodes in the axial direction and (60 + 25) nodes in the radial direction (fluid region + tube wall) were grid independent. The convergence criteria required a decrease of at least six orders of magnitude for the residuals with no observable change in the surface temperatures for an additional 200 iterations.

The numerical simulations showed that for $Re_{in} \geqslant 4.0 \times 10^3$ the calculated wall temperatures for upward flow and downward flow are very close and the effects of buoyancy and flow acceleration due to heating are very weak even for high heat fluxes. These results are consistent with the experimental results. Therefore, the calculated wall temperatures for upward flow will be presented for $Re_{in} \geqslant 4.0 \times 10^3$.

The comparison of the predicted and measured wall temperatures for supercritical pressure CO_2 upward flow

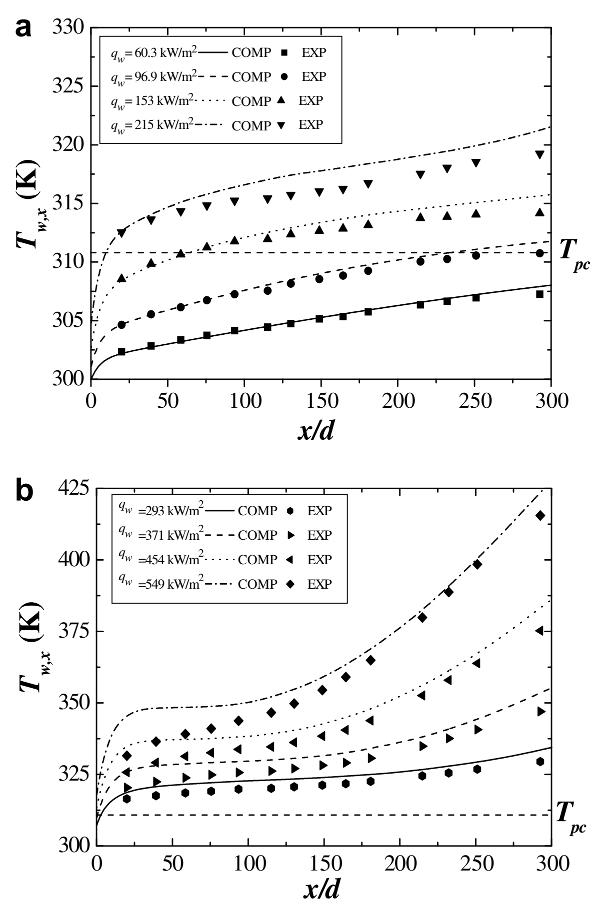

Fig. 4. Comparison of experimental and calculated wall temperatures for upward flow $p_{in} = 8.60$ MPa, $T_{in} = 25.0$ °C, and $Re_{in} = 10.5 \times 10^3$.

in the vertical heated mini-tube using various turbulence models such as the standard k–ε, RNG k–ε, Realizable k–ε, and LB low Reynolds number turbulence models show that the predicted wall temperatures using the Realizable k-ε turbulence model giving the best results for the studied conditions.

Fig. 4 compares the measured and the predicted wall temperatures for supercritical pressure CO_2 upward flow in the vertical heated mini-tube for $Re_{in} = 10.5 \times 10^3$. The results show that the predicted wall temperatures correspond well with the experimental results for lower heat fluxes (<293 kW/m^2); for high heat fluxes the predicted wall temperatures corresponded well with the experimental data in the later part of the tube. Therefore, the numerical model using the Realizable k–ε turbulence model can accurately predict the convection heat transfer of supercritical pressure CO_2 in the vertical mini-tube when the Reynolds number is relatively high and when the heat fluxes are not very high, for these conditions the buoyancy and flow acceleration effects are small. For very high heat fluxes (371, 454 and 549 kW/m^2), the turbulence model should be improved.

5. Conclusions

(1) For relatively high Reynolds numbers ($Re_{in} \geqslant 4.0 \times 10^3$) and both low and high heat fluxes, the local wall temperatures increase monotonically along the mini-tube for both upward and downward flows with no abnormal temperature distributions and the buoyancy and flow acceleration effects are insignificant.

(2) Numerical simulations using various turbulence models accurately predict the convection heat transfer of supercritical pressure CO_2 in the vertical mini-tube when the Reynolds number is relatively high and when the heat fluxes are not very high with the Realizable k–ε turbulence model giving the best results.

Acknowledgements

The project was supported by the National Outstanding Youth Fund from the National Natural Science Foundation of China (No. 50025617) and a Key Grant Project of the Chinese Ministry of Education (No. 306001).

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