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

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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

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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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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 (40*d*). 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

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

The methods to determine the local temperatures of the inner tube surface $T_{\mathrm{w,i}}(x)$, the heat flux on the inner surface q_{w}, the local bulk fluid temperature $T_{\mathrm{f,b}}(x)$, 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-

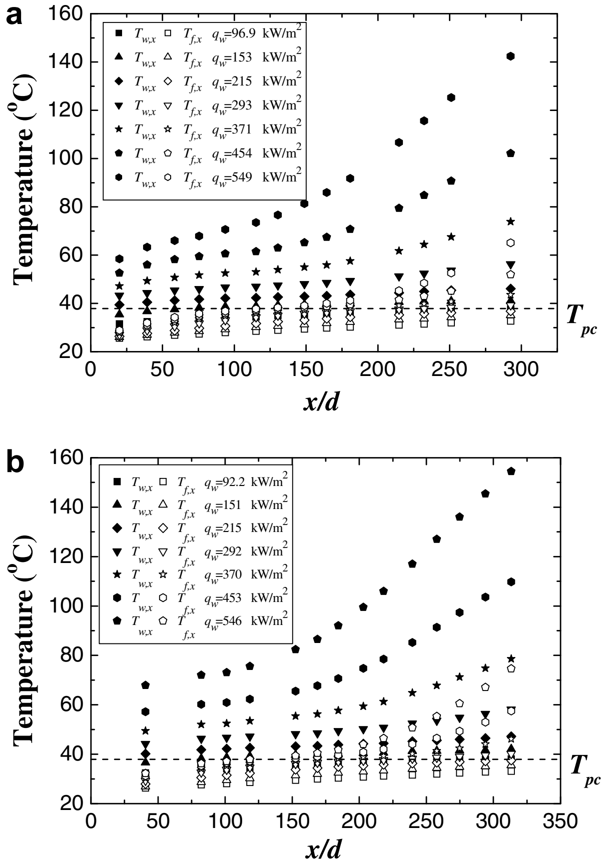

Fig. 1. Local wall temperatures (solid symbols) and bulk fluid temperatures (hollow symbols) for (a) upward flow and (b) downward flow. (a) $P_{\mathrm{in}} = 8.60$ MPa, $T_{\mathrm{in}} = 25.0$ °C, $Re_{\mathrm{in}} = 10.6 \times 10^3$; (b) $P_{\mathrm{in}} = 8.60$ MPa, $T_{\mathrm{in}} = 25.0$ °C,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