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Turbulent convective heat transfer of CO_2 in a helical tube at near-critical pressure

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ABSTRACT

Experiments of carbon dioxide flowing in a helical pipe at near-critical pressure were performed at constant heat flux boundary condition. The helical curvature diameter, helical pitch and tube diameter were 283.0 mm, 32.0 mm and 9.0 mm, respectively. The inlet Reynolds number was larger than 10^4 to ensure the turbulent flow. The renormalization group RNG k–ε model simulated the three-dimensional turbulent heat transfer of CO_2 in the helical pipe. Much attention was paid to the combined effects of the centrifugal force and buoyancy force on the heat transfer. The RNG k–ε model reasonably simulates the complicated heat transfer. The wall temperatures near the tube exit were slightly over-predicted, due to the suppression of the increased wall temperatures near the tube exit by axial thermal conduction in the experiment. Before and near the pseudocritical temperature region, the varied physical properties caused significantly non-uniform velocity and temperature distributions over the tube cross section. The larger axial velocities appear at the outer-bottom location, and the higher wall temperatures appear at the inner-top location. Thus, the outer-bottom locations hold larger heat transfer coefficients. The turbulent kinetic energies are increased along the axial angles and larger in the inner-top region of the tube cross section. The effective viscosities are decreased along the axial angles, and the larger effective viscosities are shifted to the tube center with the axial flow development. Beyond the pseudocritical temperature region, the decreased buoyancy force suppressed the non-uniformity of the heat transfer coefficients over the tube circumference.

1. Introduction

Supercritical fluids have wide applications in air-conditioners, nuclear reactors, supercritical fluid extraction due to their distinct physical properties. Rockets and military aircraft are cooled by fuel at supercritical pressures. Heat is transferred to supercritical water in a modern power plant. Carbon dioxide can be a major alternative refrigerant for automotive air-conditioners and heat pumps due to its good thermodynamic, transport, and environment properties [1]. CO_2 is also considered as a possible working fluid in an Organic Rankine Cycle to recover low grade thermal energy.

The helically-coiled-tube heat exchangers offer several advantages such as simple geometry structure, high heat transfer efficiency and high reliability etc. Many experimental and numerical works have been done on the CO_2 forced convective heat transfer in straight pipes. The studies were performed to investigate the CO_2 forced convective heat transfer in 0.5–2.14 mm inside diameter vertical and horizontal tubes at supercritical pressures (Liao and Zhao [2–4]). The tube was under the heating or cooling boundary conditions. It was found that the inside diameter of the tube had strong effect on the Nusselt number. The numerical simulation demonstrated the important buoyancy force effect on the flow and heat transfer, even for small diameter tubes and large Reynolds number.

Jiang et al. [5–7] conducted experimental and numerical studies on the supercritical pressure CO_2 heat transfer in miniature tubes. The effects of inlet fluid temperatures, pressures, heat fluxes, flow direction, buoyancy force and flow acceleration on the flow and heat transfer were analyzed. The tube wall thickness was found to have small effect on the heat transfer. The heat transfer coefficients were decreased with increases in the inlet fluid pressures.

Bae et al. [8] experimentally investigated the supercritical pressure CO_2 heat transfer in a 6.32 mm inside diameter tube. Several experimental correlations were developed under normal and deteriorated heat transfer conditions. A general deterioration criterion

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Nomenclature

a	radius of the helical pipe, m
A_c	cross section area of the tube, m^2
d	tube diameter, m
D	coil diameter, m
De	dean number
c_p	specific heat, kJ/kg K
g	gravity force ($g = 9.81$), m/s^2
h	heat transfer coefficient, W/m^2 K
H	specific enthalpy, kJ/kg
k	turbulent kinetic energy, m^2/s^2
L	length of the tube, m
m	mass flow rate, kg/s
Nu	Nusselt number
p	pressure, Pa
q	heat flux, W/m^2
r	radial coordinate, m
R_c	radius of the coil, m
Re	Reynolds number
T_{pc}	pseudocritical temperature, K
T_w	wall temperature, K
$T_{w,x}$	wall temperature, K
u	velocity, m/s
x,y,z	Cartesian coordinate

Greek symbols

δ	curvature ratio
ε	turbulent dissipation rate
φ	axial angle, °
θ	circumferential angle, °
λ	thermal conductivity, W/m K
η	thermal efficiency
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3

Subscripts

b	bulk fluid
eff	turbulent effective parameters
in	inlet condition
pc	pseudocritical
w	wall

was developed to accurately predict the appearance of the heat transfer deterioration.

Du et al. [9] numerically simulated the CO_2 turbulent convective heat transfer in a 6 mm horizontal tube using the FLUENT software. It was found that the simulations using various turbulent flow models could demonstrate the heat transfer characteristic at supercritical pressures. Among these models, the LB-low-Reynolds-number-model matched the experimental data well. The buoyancy force significantly enhances the heat transfer. The buoyancy force is increased to the peak value and then decreased along the flow direction.

Cao et al. [10] numerically simulated the laminar heat transfer of supercritical CO_2 in horizontally circular and triangular channels with cooling boundary conditions. The effects of fluid physical properties and channel geometries on the heat transfer were analyzed. The buoyancy force could enhance the heat transfer near the pseudocritical temperature region. The effect of secondary flow on the heat transfer was analyzed qualitatively. Mao et al. [11] experimentally investigated the supercritical pressure water heat transfer in a helical coiled tube. Over the Reynolds number in the range of 5.5×10^4 to 5.5×10^5, the heat transfer was significantly enhanced due to the apparently changed physical properties of water near the near the pseudocritical temperature region.

To the author's knowledge, there has less studies of turbulent heat transfer to near-critical CO_2 in a helical pipe, which needs to be understood because of its wide applications in power generation systems and reactor facilities. The fluid CO_2 has critical temperature of 31.3 °C and critical pressure of 7.39 MPa, which are significantly lower than water ($T_c = 374.2$ °C, $P_c = 22.114$ MPa). The supercritical pressure fluids have varied physical properties versus temperatures, which are more obviously near the pseudocritical temperature region. Due to the varied fluid densities with temperatures, the buoyancy force is generated to induce the secondary flow over the tube cross section. The buoyancy force attains maximum near the pseudocritical temperature point.

In this paper, the experimental data were obtained with CO_2 flowing in a helical tube at the near-critical pressure. The measured data were compared with the numerical simulations to demonstrate the effectiveness of the simulation results. The parameters of velocity, temperature, turbulent kinetic energy and the effective viscosity were carefully analyzed. The wall temperature and heat transfer coefficients were discussed with the effects of buoyancy force and the centrifugal force.

2. The experimental system and data reduction

Fig. 1 shows the experimental setup, including a CO_2 liquid storage tank, a high-pressure piston pump, a mass-flow-meter, a helical tube test section, a condenser, and an expansion valve. The test section was made of a 316L stainless tube. The helical tube had outside and inside diameters of 12.0 mm and 9.0 mm, respectively. The helical curvature diameter was $D = 283.0$ mm with a pitch distance of 32.0 mm. The maximum axial angle was 2160°, corresponding to six turns of helical coils. The flow was upward and the running pressure was $p = 8.0$ MPa. The heat was applied on the test tube by applying the alternative voltage on two copper plates. Thus, the constant heat flux boundary condition was assumed. The fluid pressure and differential pressure were measured by the 3051 type pressure and differential pressure transducers, respectively. The pressure and differential pressure had the accuracies of 0.5%. The fluid temperatures and wall temperatures were measured by K-type thermocouples with the accuracy of 0.2 °C. There were 23 cross sections with thermocouple wires welded on the outer wall surface. Each cross section had eight thermocouples. The inside wall temperatures were obtained by the inverse thermal condition solution with known outer wall temperatures [12]. The wall temperatures were non-uniform because of the secondary flow in the tube. The inside wall temperatures were averaged by the eight corresponding temperatures.

$$T_w = \left(\sum_{i=1}^{8} T_{wi} \right) \Big/ 8 \tag{1}$$

where T_{wi} was the inside wall temperature computed by the inverse heat condition solution.

Along the axial flow direction, there were 23 cross sections on which thermocouples were welded. The bulk fluid temperature at ach cross section was determined by the fluid enthalpy at the local pressure. The NIST (Standard Reference Database 23 (REFPROP Version 8.0) [13] helped to decide the value at specific enthalpy and pressure. The fluid enthalpy was obtained as