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An experimental study on heat transfer between supercritical carbon dioxide and water near the pseudo-critical temperature in a double pipe heat exchanger

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ABSTRACT

Supercritical carbon dioxide (SCO_2) is a promising working fluid for the cryogenic refrigeration, air-condition and heat pump systems. The present study sets up a SCO_2–water test loop to study the heat transfer performance of SCO_2 in a double pipe heat exchanger. The effects of SCO_2-side pressure, mass flux and buoyancy force as well as water-side mass flux are investigated. It is found that the total and SCO_2-side heat transfer coefficients reduce as the SCO_2-side pressure increases. The peak total and SCO_2-side heat transfer coefficients appear at a higher temperature than the pseudo critical temperature. The water-side mass flux has a larger effect on the total heat transfer coefficient compared to the SCO_2-side mass flux in the studied cases. The contribution of buoyancy force to the heat transfer performance is large at the small SCO_2-side mass flux and it becomes smaller as the SCO_2-side mass flux increases. The SCO_2-side pressure and water-side mass flux have little effect on the buoyancy force. A heat transfer correlation that includes the effect of buoyancy force is obtained by fitting the experimental data with genetic algorithm.

1. Introduction

The halohydrocarbon refrigerants such as CFCs and HCFCs were restricted for use in many years ago due to the damage to ozone layer [1]. The widely used HFC-134a is also not recommended to be used in the future because it may increase the greenhouse effect, and may be decomposed into acid and other toxic substance by the sunniness in the troposphere [2–3]. Recently, many researchers have paid considerable attention to the CO_2. The CO_2 is non-toxic, incombustible and safe so that it almost has non-negative effects on the environment whose Ozone Depletion Potential (ODP) and effective Global Warming Potential (GWP) are zero [4–5]. When the pressure and temperature approach the critical point of CO_2, the specific heat capacity is very high which can significantly improve the heat transfer performance. The supercritical fluids cannot be defined as a liquid or as a gas but as a substance in a state, i.e., supercritical state, because their thermal physical properties are different from those of real fluids, and there is no liquid–vapor phase transition and interfaces at supercritical pressures. Therefore, the trans-critical CO_2 system has been used for the cryogenic refrigeration, air-condition and heat pump. It was found that the optimized trans-critical CO_2 system could get comparative performance compared to the synthetic refrigerants system in the commercial light cryogenic refrigeration [6–7]. The performance of heat pump using trans-critical CO_2 system was better than that using traditional refrigerants because it could heat the water up to 90 °C even at the atmosphere being −20 °C, and reduced the CO_2-side energy loss [8]. The trans-critical CO_2 heat pump system proposed by the energy department of Norway could operate under three conditions, i.e., only room heating, only hot water supply, and both room heating and hot water supply [9].

However, the operating pressure of trans-critical CO_2 system is very high so that there is a big challenge for the safety. As one of the key components, the heat exchanger plays an important role to transfer the heat from one loop to the other loop. The heat exchanger for trans-critical CO_2 system should endure the extreme high pressure. Besides, the thermal-hydraulic performance of heat exchanger has a significant effect on the efficiency, compactness and cost of the system. Therefore, it is necessary to study the thermal-hydraulic performance of CO_2 in the high pressure heat exchanger at supercritical pressures.

Considerable research studies on CO_2 at supercritical pressures have been conducted, especially after the SCO_2 was proposed as

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Nomenclature

A	heat transfer area, m^2
c_1, c_2, c_3	constants in correlation
D	diameter, m
f	Darcy friction factor
g	gravitational acceleration, m s^{-2}
G_m	mass flux, kg m^{-2} s^{-1}
Gr	Grashof number
h	heat transfer coefficient, W m^{-2} K^{-1}
H	enthalpy, J kg^{-1}
L	tube length, m
Nu	Nusselt number
p	pressure, MPa
Q	heat transfer rate, W
q_m	mass flow rate, kg s^{-1}
Re	Reynolds number
Pr	Prandtl number
T	temperature, °C
U	total heat transfer coefficient, W m^{-2} K^{-1}

Greek

ΔT	logarithmic mean temperature difference, K
Δp	pressure difference, Pa
ρ	density, kg m^{-3}
λ	thermal conductivity, W m^{-1} K^{-1}
μ	dynamic viscosity, Pa s

Subscripts

b	average value at cross section
CO2	supercritical CO_2 side
H2O	water side
in	inlet
out	outlet
pc	pseudo critical point
w	wall
Wall-H	outer diameter
Wall-C	inner diameter

one of working fluids in the nuclear reactor in 1950s. In 1967, Hall et al. [10] compared the available experimental data with the empirical formulas and found that more attention should be paid on the buoyancy force even in the forced convection condition. Pioro et al. [11] applied many empirical formulas to calculate the heat transfer coefficient of SCO_2 in the tube under the same operating condition. There were significant differences both in the values and trends among the different empirical formulas, which indicated that the empirical formulas used for the fluids with constant thermal physical properties could not include the effect of remarkable variation of thermal physical properties. The heat transfer performances of SCO_2 in vertical tubes with circular, triangular and square cross sections were tested and compared [12]. Song et al. [13] found that the heat transfer had similar characteristics regardless of heat transfer enhancement or deterioration if the ratio of tube length to tube diameter and the ratio of heat flux to mass flux were kept as constant. Kim et al. [14] performed an experiment to examine the effect of flow directions including the up flow and down flow on the heat transfer performance. The result showed that if the heat flux of wall was middle and mass flux was small, the peak point of wall temperature appeared in the up flow, but the wall temperature increased monotonously along the flow direction in the down flow. Both the flow and heat transfer were affected by the buoyancy force and flow acceleration. Zhang et al. [15] tested the mixed convective heat transfer of SCO_2 inside a vertical helically coiled tube. The result showed that the buoyancy force, centrifugal force and variations of thermal physical properties had significant effects on the temperature and heat transfer coefficient distributions along the circumference edges. Yang et al. [16] numerically studied the cooling process of SCO_2 in the tube at constant wall temperature. It showed that the contribution of inclined angles on heat transfer decreased as the gravity force magnitudes reduced. Bae [17] conducted an experiment to analyze the influence of flow direction and flow channel shape, and evaluate the existing SCO_2 heat transfer correlations. Chen et al. [18] discovered multiple peaks of heat transfer coefficients along the horizontal flow, and the effects of inlet Reynolds number, channel diameter and heat transfer to trans-critical CO_2 flow inside the mini-channels were examined. Chen et al. [19] examined the near-critical CO_2 thermosyphon in the trans-critical/SCO_2 based natural circulation loop.

However, the above researches mainly focused on the heat transfer process of SCO_2 under the constant heat flux or constant wall temperature conditions. In the high pressure heat exchanger used for the trans-critical CO_2 system, there involves a heat exchange process between the water to SCO_2, whose thermal boundary is different from the constant heat flux and thus may result in a different heat transfer performance. Therefore, the present study conducts an experiment to study the heat transfer performance of SCO_2 in a double pipe heat exchanger using SCO_2–water test loop.

2. Experimental system

The experimental system is composed of two separate closed loops, i.e., CO_2 loop and water loop, as shown in Fig. 1. In the CO_2 loop, the subcritical CO_2 at 3–5 MPa released from the tank is cooled to liquid by the refrigerator, and then injected into the pipeline of system using the gas driving liquid hydraulic pump. The gas driving liquid hydraulic pump can provide the maximum pressure of 30.2 MPa for the CO_2. Therefore, the CO_2 can reach the supercritical state by increasing the pressure of system. The mass flow rates of water and CO_2 on both sides are controlled according to the micropump, and measured by the flowmeter. The maximum volume flow rate supported by micropump is 20 L min^{-1}. The flowmeter can measure the mass flow rate ranged from 80 kg h^{-1} to 1500 kg h^{-1} with 0.05% accuracy. The maximum measuring pressure for the gauge pressure transducers is 27.6 MPa, while that for the differential pressure transducers is 62.2 kPa. The accuracy of gauge pressure transducers is 0.065%, and that of differential pressure transducers is 0.075%. The inlet temperature of test section on both sides is controlled by the heaters, and the T-type thermocouples are used to measure the temperatures. The maximum measuring temperature for the T-type thermocouples is 200 °C. The T-type thermocouples were calibrated ahead of the experiment and the uncertainty of the measured temperature is within ±0.1 °C accuracy in the range of the present tested operating conditions.

The test section is a double pipe heat exchanger, as shown in Fig. 2. The water flows from the bottom to the top in the annular space, while the SCO_2 flows from the top to the bottom in the inner tube. The inner tube is made of 316L stainless steel, but the outer