A STUDY OF IN-TUBE EVAPORATION HEAT TRANSFER OF CARBON DIOXIDE

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ABSTRACT

The evaporative heat transfer characteristics of carbon dioxide (CO_2) were measured in a fluid-heated, 8.0 mm ID smooth stainless steel tube. The test apparatus was a concentric tube heat exchanger heated by water in the annulus with CO_2 in the central tube. Data were collected for temperatures of 5 °C and 10 °C; mass fluxes of 250 kg/m^{2}s, 500 kg/m^{2}s, and 650 kg/m^{2}s; and heat fluxes of 24 kW/m^2 to 58 kW/m^2. The average heat transfer coefficient was determined with the overall UA equation for the annular configuration. All properties were calculated using REFPROP 7.0 (Lemmon et. al. 2002). Results are presented and compared to existing correlations for the two-phase average evaporative heat transfer coefficient.

INTRODUCTION

Environmental concerns and regulations in the last two decades have generated the need for extensive research to identify viable and "environmentally friendly" refrigerant alternatives. As a result, a significant focus has been placed on so-called natural refrigerants (e.g., CO_2, ammonia, hydrocarbons, air, water, etc.). Carbon dioxide offers benefits such as non-toxicity, non-flammability, easy availability, low price, no need of recycling, and compactness of components. It has favorable transport properties (low viscosity and high thermal conductivity), which combine to improve the heat transfer characteristics.

The past several years have noticed an active research on evaporative heat transfer studies related to CO_2. Knudsen and Jansen (1997) concluded that the predicted heat transfer coefficient of CO_2 is approximately half of the coefficient predicted by the correlation of Shah (1982). In their study on CO_2 in microchannel tubes, Pettersen et al. (2000) predicted a strongly decreasing heat transfer coefficient from a certain vapor fraction upwards. Pettersen looked at six existing correlations and all of them overpredicted the measured values. Zhao et al. (2000) concluded that the boiling heat transfer coefficients of CO_2 increased with increasing heat flux and/or mass flux. Zhao recommended Gungor and Winterton's correlation due to its mean deviation of less than 10 % from his data. Olson and Allen (1998) measured the heat transfer coefficient in heated, turbulent, supercritical CO_2 flowing in a horizontal tube. The supercritical fluid had enhanced heat transfer at low heat flux and degraded heat transfer at high heat flux.

In this study, the average evaporative heat transfer characteristics of two-phase CO_2 are measured in a concentric, water heated test section. The experimental results are compared to predictions from six published correlations.

1. EXPERIMENTAL SET UP

Figure 1a shows the schematic of the test apparatus. The experimental setup consisted of an evaporator (test section), condenser/subcooler, saturator, preheater, and magnetic gear pump. In the concentric type test section, the inner tube was stainless steel (type 304) with an inner diameter of 8.0 mm and an outer diameter of 9.5 mm. The outer tube was copper with an inner and outer diameter of 12.5 mm and 15.9 mm respectively. This arrangement produced an annulus hydraulic diameter of 3.0 mm. Carbon dioxide flowed in the inner tube and water, which heated the CO_2, flowed in the annulus. The condenser/subcooler and the preheater were controlled by a secondary heat transfer fluid (HTF) loop. This HTF was a mixture of ethylene-glycol/water (30 %/70 % by mass).

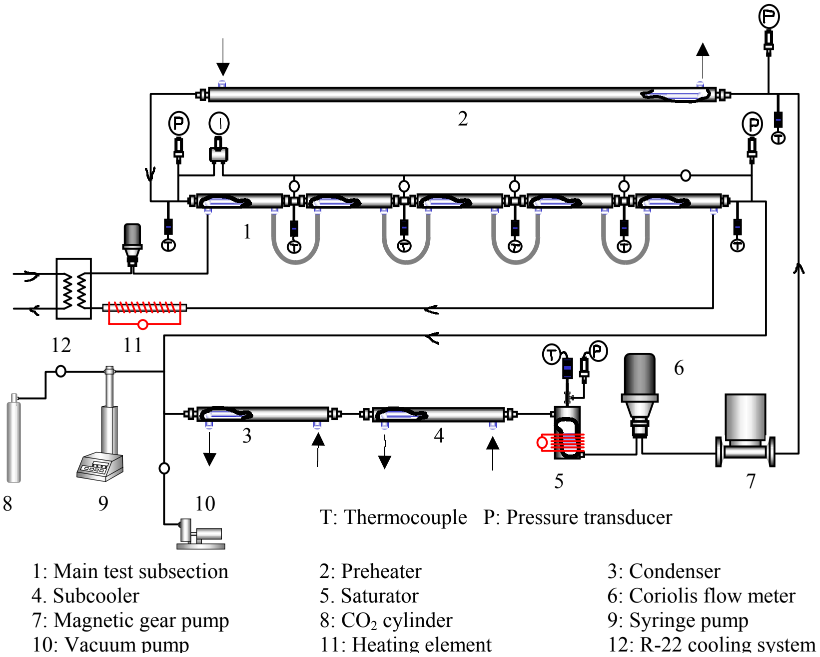

Figure 1a: Experimental test facility

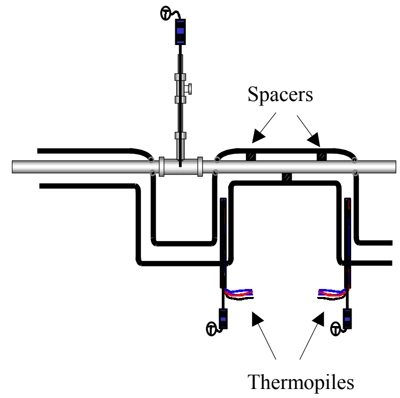

Figure 1b: Schematic of subsection

The evaporator (test section) was made up of five, well insulated subsections. Carbon dioxide was charged into the system with a 500 mL syringe pump, which also controlled the operating pressure. A variable-speed motor on the magnetic gear pump controlled the flow rate of CO_2. The heating was controlled with the help of a solid-state controlled rectifier (SCR). All data presented in this paper were taken in subsection 1 because the combined uncertainty for the two-phase evaporative heat transfer coefficient was smaller in subsection 1 than in the other subsections. The uncertainty analysis is given in section 3.

1.1 Instrumentation

Each subsection was instrumented with pressure transducers on the refrigerant side and a ten-junction thermopile on the annulus (water) side. The thermocouple at the outlet of each CO_2 subsection also acted as the inlet thermocouple for the next subsection. Measurements included the mass flow rate and inlet and outlet temperature for CO_2 and water at each subsection as well as the preheater and condenser/subcooler. In addition, the CO_2 inlet pressure and the CO_2 differential pressure were measured. The temperature of CO_2 was measured with type-T thermocouples. A ten-junction thermopile was placed at the entrance and exit of each subsection in a U-bend to measure the annulus temperature change, as shown in Figure 1b. The mass flow rate of CO_2 and water for the test section was measured using coriolis mass flow meters, and that for the ethylene-glycol/water mixture in the preheater was measured with a turbine meter calibrated with a coriolis meter. The flow meters, pressure transducers, thermocouples, thermopiles, and associated instrumentation were calibrated. Table 1 shows the range and 95 % relative uncertainty of the installed instrumentation.

Table 1: Instrumentation range and uncertainty

Instrument	Range	95 % Relative Uncertainty
T-type thermocouples (°C)	0 to 100	0.20
10 Junction Thermopiles (°C)	0.5 to 5	0.21
Evaporator Pressure Transducer (kPa absolute)	0 to 20684	1.0 % of full scale
Differential Pressure Transducer (kPa)	0 to 172	1.0 % of full scale
Evaporator Water Mass Flow in Annulus (kg/h)	0 to 544	2.1 % of full scale
CO2 Mass Flow (kg/h}	0 to 544	2.1 % of full scale

2. EXPERIMENTAL AND ANALYTICAL STUDY

2.1 Annulus, Water-Side, Single-Phase Tests

Experiments were first conducted to establish the average single-phase convective heat transfer coefficient for water in the annulus. For this purpose, single-phase HFC-134a was used in the inner tube since its heat transfer characteristics were well known and an abundance of experimental data were available. The energy balance between water and HFC-134a did not exceed 12.0 % for all tests considered. The Gnielinski (1976) correlation for single-phase turbulent flow inside a smooth tube was used to calculate the average convective heat transfer coefficient of HFC-134a. The pertinent smooth tube, single-phase equations are presented in Equation 1. The calculated single-phase heat transfer coefficient for HFC-134a was used in the UA equation to solve for the water-side average single-phase convective heat transfer coefficient.

$$f = \left[0.790 * (\ln(\mathrm{Re}) - 1.64)^2\right]^{-1} \ ,$$
$$Nu = \frac{\frac{f}{8} * (\mathrm{Re} - 1000) * \mathrm{Pr}}{\left(1.07 + 12.7 * \sqrt{\frac{f}{8}}\left(\mathrm{Pr}^{\frac{2}{3}} - 1\right)\right)} \quad (1)$$

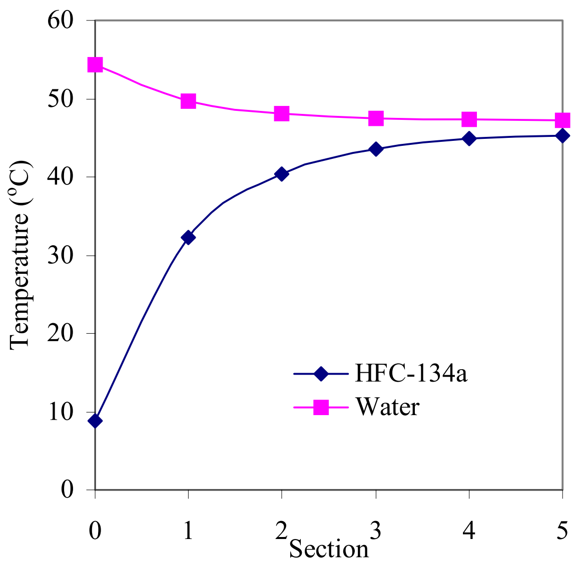

Figure 2: Temperature profiles for a single-phase test with HFC-134a with parallel flow configuration

The single-phase tests had HFC-134a mass fluxes in the range of 1000 to 1545 kg/(m^2 s) and heat fluxes in the range of 42 to 59 kW/m^2 (based on the 8 mm tube inner diameter). The inlet hot water temperature, the HFC-134a flow rate, and the water flow rate were the three major independent parameters, which allowed for establishing the conditions for these single-phase experiments.

Single-phase experiments were carried out with a parallel-flow configuration. Figure 2 shows a typical temperature profile for single-phase parallel flow tests with HFC-134a. The number of the subsection represents the X-axis, 0-1 being the first subsection, 1-2 the second subsection, etc.

The overall conductance equation, which is a sum of individual thermal resistances of HFC-134a, water, and tube wall separating the two fluids, is used to determine the waterside heat transfer coefficient (h_w).

$$\frac{1}{UA} = \frac{LMTD}{Q} = \frac{1}{h_r \pi D_i L} + \frac{\ln\left(D_r \big/ D_i\right)}{2\pi k_{ss} L} + \frac{1}{h_w \pi D_h L} \tag{2}$$

The only unknown in Equation (2) is h_w. The Nusselt number on the waterside is then represented by the following Dittus-Boelter (1930) type equation. Equation (3) is regressed using the calculated values of h_w to find the values of constants C_w and a as 0.00599 and 0.912, respectively. Equation (3) represents the final correlation for water-side single-phase average Nusselt number.

$$Nu_w = \frac{h_w D_h}{k_w} = C_w . \mathrm{Re}_w^a \, \mathrm{Pr}_w^{1/3} \tag{3}$$

Figure 3 shows a plot of percent difference between the experimental and calculated values for Nusselt number against the Reynolds number of water for the first subsection of the evaporator tube.

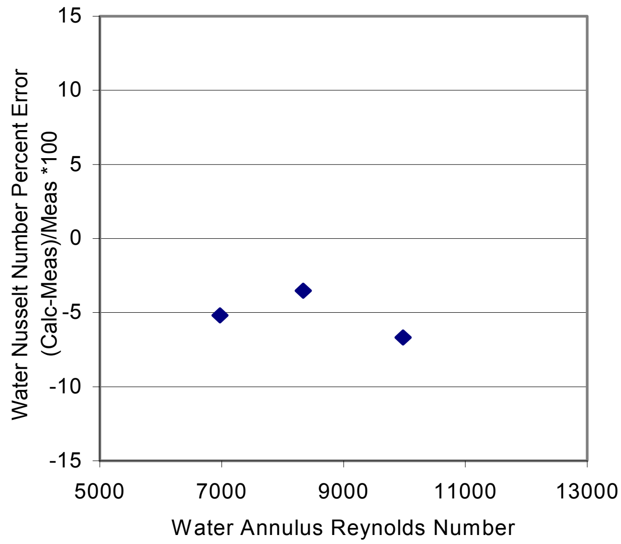

Figure 3: Comparison of experimental and calculated Nusselt numbers for single-phase tests with HFC-

2.2 Refrigerant-Side,Two-Phase Tests with CO_2

Two-phase CO_2 tests were conducted in a similar manner as HFC-134a single-phase tests and covered the same ranges of mass and heat fluxes of CO_2. Carbon dioxide of 99.99 % purity was used during the tests.

An energy balance on the preheater section of the test rig allowed the calculation of the test section entrance mass quality. Subcooled CO_2 entered the preheater and was heated until it was two-phase at the preheater exit. Pressure and temperature at the inlet and exit of the test section allowed the saturation temperature of the CO_2 to be verified. Using thermopile temperature change, inlet/exit water temperatures, water mass flow rate, and h_w, Equation (2) was solved for the refrigerant-side heat transfer coefficient, $h_{r2\varphi}$.

3. COMPARISON TO EXISTING CORRELATIONS

Chen (1966) developed a two-phase flow boiling correlation where the heat transfer coefficient was divided into a nucleate pool boiling coefficient and a bulk convective coefficient. Different correlations for the two-phase enhancement factor were selected by using a form of the Lockhart-Martinelli turbulent-turbulent two-phase multiplier (X_{tt}). Bennet and Chen (1980) modified the Chen correlation using a larger data set. They implemented a new correlation for the nucleate pool boiling suppression factor. The two-phase enhancement factor was still a function of the Lockhart-Martinelli two-phase multiplier. Shah (1982) examined data from many sources to develop a set of equations to determine the average convective heat transfer coefficient for two-phase flow. Shah described the boiling regimes in terms of nucleate boiling regions, convective boiling regions, and bubble suppression regions. He utilized a convection number (Co, very similar to X_{tt}), a boiling number (Bo), and a liquid Froude number (Fr_l) in his correlation. Gungor and Winterton (1986) developed a correlation using a nucleate pool boiling suppression factor and a bulk convective enhancement factor in a manner similar to Chen. Their expression for the pool boiling convective heat transfer coefficient included reduced pressure, refrigerant molecular weight, and heat flux. Liu and Winterton (1991) modified the Gungor and Winterton (1986) correlation by changing the expressions used to calculate the bulk convective enhancement factor and the nucleate pool boiling suppression factor. They also determined the final overall convective heat transfer coefficient by adding the convective and pool boiling contributions in quadrature. Radermacher and Hwang (1997) modified the Bennett and Chen (1980) to correlate the data of Bredesen et al. (1997). The Bredsen et al. data was taken only at -10 °C with mass flux between 200 and 400 kg/m^2 and heat flux between 3 and 9 kW/m^2.

Figure 4 shows the percent difference between the predicted and measured heat transfer coefficient values for carbon dioxide flow boiling inside the smooth tube. The Gungor and Winterton (1986) correlation predicts 48 % of the data to within ± 50 % of the measured value with 97 % of the data underpredicted. The Bennett and Chen (1980) correlation predicts 47 % of the data to within ± 50 % of the measured value with 69 % of the data underpredicted. The Radermacher and Hwang (1997) correlation predicts 42 % of the data to within ± 50 % of the with 34 % of the data underpredicted. The remaining correlations (Shah, Liu and Winterton, and Chen) presented in Figure 4 predict less than 40 % of the data to within ± 50 % of the measured value.

Figure 4: Percent difference in measured and predicted average two-phase convective heat transfer coefficient for CO_2

4. UNCERTAINTY ANALYSIS

The uncertainty on the water-side average convective heat transfer coefficient was calculated on the basis of measured uncertainties of temperature, pressure and mass flow rates. From Eq. (1) we understand:

$$h_w = f(h_r, k_{ss}, Q, D_r, D_i, L, LMTD) \tag{4}$$

The uncertainty in h_w may be found by following the propagation of errors within Eq. (2) by summing the error (taken as 95 % confidence interval) of all the individual terms as shown below in Eq. (5).

$$E_{h_w}^2 = \left(\frac{\partial f}{\partial h_r}E_{h_r}\right)^2 + \left(\frac{\partial f}{\partial k_{ss}}E_{k_{ss}}\right)^2 + \left(\frac{\partial f}{\partial Q}E_Q\right)^2 + \left(\frac{\partial f}{\partial D_w}E_{D_w}\right)^2 + \left(\frac{\partial f}{\partial D_r}E_{D_{ri}}\right)^2 + \left(\frac{\partial f}{\partial L}E_L\right)^2 + \left(\frac{\partial f}{\partial LMTD}E_{LMTD}\right)^2 \tag{5}$$

The uncertainties on other parameters such as heat flux, mass flux, heat transfer coefficient of CO_2, etc. can be determined in a similar way. Table 2 lists measured and calculated quantities and their relative uncertainties for the 95 % confidence limits.

Table 2: Single-phase water-side propagation of error and relative uncertainty in h_w (95 % confidence interval)

Parameter	Range	Uncertainty (%)
Single-phase HFC-134a, average convective heat transfer coefficient from Gnielinski (1976), Equation (3), h_w (W/m^2 K)	2000 to 2900	10.3 %
Water-side heat transfer rate, Q (W)	1700 to 2500	5.6 % to 7.5 %
Log-mean temperature difference between water and single-phase HFC-134a, LMTD (°C)	25.3 to 29.7	0.75 % to 0.89 %
Water-side average convective heat transfer coefficient, h_w	3800 to 6100	18 % to 31 %

With h_w known from the single-phase tests with HFC-134a, CO_2 was circulated through the test section at various inlet qualities and heat fluxes. The UA equation (Eq. (2)) was solved for h_r and used to determine the propagation of error for the refrigerant-side average convective heat transfer coefficient. The resulting form of the equation is shown by Equation (6).

$$h_{r2\phi} = f(h_w, k_{ss}, Q, D_r, D_w, L, LMTD) \tag{6}$$

The propagation of error was calculated as in Equation (5) with h_w replaced by $h_{r2\phi}$. Table 3 summarizes the necessary uncertainty terms and the final calculation of the uncertainty of $h_{r2\phi}$.

Table 3: Two-phase CO_2 average convective heat transfer coefficient relative uncertainty (95 % confidence interval)

Parameter	Range	Uncertainty (%)
Single-phase water-side average heat transfer coefficient, h_w (W/m^2 K)	3986 to 7222	28 to 50
Water-side heat transfer rate, Q (W)	1000 to 2400	5 to 9
Log-mean temperature difference between water and CO_2, LMTD (°C)	6 to 18	1.0 to 3.5
Two-phase CO_2, average convective heat transfer coefficient, $h_{r2\phi}$ (W/m^2 K)	5200 to 54400	14 to 55

4. CONCLUSIONS

This investigation examined 71 data points for evaporative flow boiling of CO_2 in a fluid-heated, 8 mm ID smooth tube. Six existing correlations were examined for their fit to the data collected. The Gungor and Winterton (1986) correlation was the best predictor of the measured values, predicting 48 % of the data to within ± 50 % of the measured value.

Below a heat transfer coefficient value of 25000 W/m^2 K, Radermacher and Hwang (1997), Bennett and Chen (1980) and Chen (1966) correlations significantly overpredicted the measured values. At higher heat transfer coefficient values all correlations underpredicted the measured value by as much as 90 %.

NOMENCLATURE

A	Heat transfer area, m^2
Bo	Boiling number, $\frac{q}{G\, i_{fg}}$
C	Constant
Co	Convection number, $\left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_g}{\rho_f}\right)^{0.5}$
D	Diameter, m
E	Uncertainty
Fr	Froude number, $\frac{G^2}{\rho_f^{\,2}\, g\, i_{fg}}$
G	Mass flux, kg/m^{2}s
h	Heat transfer coefficient, W/m^2 K
i	Enthalpy, J/kg
k	Thermal conductivity, W/m K
L	Length, m
LMTD	Logarithmic Mean Temperature Difference, °C
Nu	Nusselt number
Pr	Prandtl number
Q	Heat Transfer Rate, W
Re	Reynolds number, $\frac{GD}{\mu}$
T	Temperature, ^{o}C
U	Overall heat transfer coeff., W/m^2 K
Xtt	Lockhart Martinelli two-phase multiplier

Subscripts

f	fluid or liquid
g	vapor or gas
fg	gas minus fluid property
h	hydraulic
i	inner/inlet
o	outer/outlet
r	refrigerant/root
ss	stainless steel
w	water
r2ϕ	two-phase refrigerant

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