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Characteristics of Flow Boiling Heat Transfer of Sub-critical CO_2 in Mini-channels with Micro-fins

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ABSTRACT

The flow boiling heat transfer characteristics of sub-critical CO_2 in mini-channels with micro-fins was investigated. The tests used a 0.6 m long extruded aluminum tube with 8 flow-channels 1.7 mm inner diameter with 0.16 mm high micro-fins. The average heat transfer coefficients during evaporation were measured at qualities from 0.1 to 0.9 for saturation temperatures of 1-15℃, mass fluxes of 100-600 kg/m^{2}s and heat fluxes of 1.67-8.33 kW/m^2. The results showed that the CO_2 heat transfer coefficient was mainly a function of the heat flux and evaporating temperature, increasing with increasing heat flux or evaporating temperature, with dry-out occurring earlier with increasing heat flux or decreasing evaporating temperature especially at low heat fluxes. The mass flux had less effect on the heat transfer coefficient compared to the heat flux and the saturation temperature. Applicable flow boiling heat transfer coefficient correlation of sub-critical CO_2 in mini-channels with micro-fins was also developed.

Keywords: Carbon dioxide, Flow boiling, Mini-channels with micro-fins, Enhanced heat transfer

1. INTRODUCTION

Environment crisis draws closer to us today, and studies on alternative refrigerant have become hot for years and probably will continue. Application of HFC refrigerants, which have been developed as alternatives to CFC and HCFC refrigerants, may not be adequate as ultimate refrigerants in a refrigeration system, because of the higher global warming potential (GWP). Environmentally benign 'natural' refrigerants such as CO_2 have attracted considerable attention as alternatives refrigerants, with advantage of zero ODP and a GWP of only one. Also, CO_2 has no toxicity, no flammability and is very inexpensive.

Compared to other conventional refrigerants, the CO_2 flow boiling characteristics, such as flow pattern, flow boiling heat transfer and pressure drop, are quite different (Cheng *et al.*, 2008a). So it is important to establish the experimental database of CO_2 evaporation and to develop adequate predictive correlations used for the design of CO_2 evaporator. Several experimental studies have been made in various horizontal tubes. Zhao *et al.* (2001) investigated the flow boiling heat transfer of CO_2 in horizontal triangular microchannels with a hydraulic diameter of 0.86 mm and found that CO_2 offers outstanding heat transfer characteristics compared to common refrigerants. Pettersen (2004) experimentally evaluated flow vaporization of CO_2 in a flat extruded aluminum tube having 25 circular channels with 0.81 mm inner diameter. Both Zhao and Pettersen's experimental results indicated that the heat transfer coefficients for CO_2 are higher than those for conventional refrigerants and correlations developed based on the database of conventional refrigerants did not accurately predict the CO_2 test results. However the CO_2 heat transfer data could be correlated reasonably well with a combination of models for nucleate boiling, convective evaporation heat transfer. Yun *et al.* (2003) studied the boiling heat transfer characteristics of CO_2 in a horizontal smooth stainless steal tube with an inner diameter of 6.0 mm. The test data showed that partial dry-out of CO_2 occurs at a lower vapor quality than predicted by standard correlations and the heat transfer coefficient of CO_2 is on average 47% higher than that for R134a at the same operating conditions. Zhao *et al.* (2004) and Wu *et al.* (2005) measured the boiling heat transfer characteristics of CO_2 in a horizontal smooth tube with an inner diameter of 4 mm

and indicated that the CO_2 heat transfer coefficients did not agree well with the existing correlations developed for conventional refrigerants. Pamitran *et al.* (2006) investigated flow boiling of CO_2 and other refrigerants in horizontal smooth stainless steel mini-channels with inner diameters of 1.5 mm and 3.0 mm, and CO_2 showed the highest heat transfer coefficient and the lowest pressure drop among the refrigerants, including R407C, R410A and R22. Park and Hrnjak (2007) investigated the boiling heat transfer of CO_2, R410A, and R22 in a horizontal smooth tube with an inner diameter of 6.1 mm, with the test data indicating that CO_2 has better heat transfer characteristics than conventional refrigerants such as R410A and R22, with the CO_2 heat transfer coefficients indicating that nucleate boiling is the dominant heat transfer mechanism. Cho and Kim (2007) measured the evaporative heat transfer coefficients of CO_2 in 5 m long smooth and micro-fin tubes with outer diameters of 5 and 9.52 mm. The average evaporative heat transfer coefficients for the micro-fin tube were 1.5-2.10 times as large as that for the smooth tube with the same outer diameters, which experimentally verifies that the micro-fins enhance the heat transfer during the evaporation process.

About the prediction of CO_2 flow boiling in horizontal smooth tubes, some progresses were made. Yoon *et al.* (2004) investigated the characteristics of evaporative heat transfer and pressure drops of CO_2 in a horizontal smooth stainless steel tube with an inner diameter of 7.53 mm. They found that the existing correlations under-predicted the experimental heat transfer coefficients, and they developed a new correlation for CO_2 evaporation which takes the critical vapor quality into account. In the studies of Cheng *et al.* (2008a, 2008b), a two-phase flow pattern map specifically for CO_2 and a flow pattern based phenomenological two-phase flow frictional pressure drop model were presented in Part I, and an updated flow pattern based flow boiling heat transfer model was presented in Part II. Both of Yoon *et al.* and Cheng *et al.* used the dry angle of tube perimeter to explain the partial dry-out phenomenon occurred in higher qualities. However, adequate correlations for CO_2 evaporation heat transfer in micro-fin tubes were limited in the open literatures due to lack of database and difficulties in a generalization of complicated phenomena. Compared with a conventional channel, evaporation using small channels with micro-fins may provide a higher heat transfer coefficient due to its higher contact area per unit volume of fluid, and seems more adequate to evaporator designs. The present study experimentally investigates the flow boiling characteristics of sub-critical CO_2 in extruded mini-channels with micro-fins. Based on the experimental data, applicable flow boiling heat transfer coefficient correlation of sub-critical CO_2 in mini-channels with micro-fins has been developed.

2. EXPERIMENTS AND DATA REDUCTION

2.1 Experimental apparatus and measurements

The experimental apparatus includes the CO_2 cycle system and a glycol cooling cycle. The test rig included a plunger pump to circulate the carbon dioxide, a pre-heater, an evaporator (the test section), a condenser to condense the two-phase CO_2, a mass flow meter and a liquid receiver. The pre-heater was used to adjust the CO_2 inlet vapor quality to the desired value with the sub-cooled liquid from the plunger pump heated by a 1.1 kW AC Variac power supply. The CO_2 in the test section was heated by a DC power supply (0-120 V, 0-10 A). The high vapor quality CO_2 from the test section was completely condensed and sub-cooled by the condenser. The refrigerant flow rate entering the test section was controlled by moving the pump plunger and was measured using a Coriolis effect flowmeter with an uncertainty of ±0.2% of reading. The operating pressure in the test section was adjusted by the CO_2 charge in the system. The refrigerant pressure at the test section outlet was measured using a pressure transducer with an uncertainty of ±0.075% of reading, with the pressure difference between the inlet and outlet of the test section measured by a differential pressure transducer with an uncertainty of ±0.075% of reading. The temperatures at key points in the CO_2 cycle were measured by calibrated Pt100 sensors with uncertainties of ±0.1℃.

The test section was a horizontal multi-port extruded aluminum tube with 8 flow channels 1.7 mm inner diameter with 0.16 mm high micro-fins. The ratio of the developed area (base wall + fins) to the tube was 1.44. The tube was electrically heated by direct current, and the heated section was 600 mm long and was divided into five sections. The outside wall temperature of the tube was measured by twelve calibrated T-type thermocouples clamped to both the top and bottom sides of the tube. The thermocouples, with an uncertainty of ±0.1℃, were equally spaced along the heated section between the two electrodes. The average value of the thermocouples was used as the wall temperature, T_{wo}. The test section was well wrapped with insulation to minimize heat loss to ambient. The heat loss was estimated by measuring the electric heat input with no flow with the tube surface temperature and ambient temperature the same as for the test conditions. The estimated heat loss was 5-10% of the electrical heat input and was taken into account in the experiments and the data reduction.

The outputs from all the measurement instruments were collected by an Agilent 34970A Data Acquisition Unit linked to a PC. The PC recorded all the initial and final data which were used to calculate the thermodynamic properties, and to reduce the data.

2.2 Data reduction

The test section inlet vapor quality was determined from the power supplied to the pre-heater section with the outlet vapor quality determined from the DC power supplied to the test section. The mean value of the two vapor qualities was then used as the average vapor quality for the test section. With the short test section length and low CO_2 viscosity, the pressure drop through the test section was small so the saturation temperature along the whole test section was assumed to be constant. The heat transfer area used in the calculation of heat flux is based on the inner diameter without considering the fins. The CO_2 heat transfer coefficient over the tube was calculated using

$$h=q/(T_{\mathrm{w}}-T_{\mathrm{sat}}) \tag{1}$$

The heat flux, q, was calculated from the measured electrical power applied to the test section with consideration of heat losses. The inside wall temperature, T_w, was calculated from the measured outside wall temperature, T_{wo}, assuming one-dimensional heat condition in the wall. The wall temperature is calculated as the mean value of the 12 temperature measurements, with average vapor quality over the tube. The saturation temperature, T_{sat}, was estimated from the measured pressure using properties from REFPROP (1998). The averaged heat transfer coefficient was mainly calculated and discussed. The wall temperature was calculated as the mean value of the 12 temperature measurements and the mean vapor quality between the inlet and outlet qualities was used. The heat transfer coefficient uncertainty resulting from the uncertainties of the independently measured parameters was estimated to be 5-15%, based on the method described by Moffat (1998).

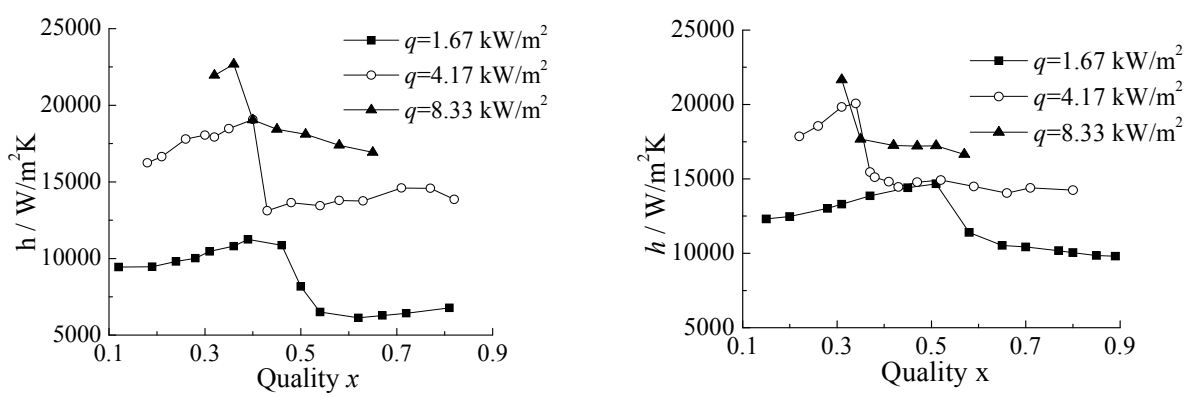
3. RESULTS AND DISCUSSION

3.1 Test conditions

Experiments were conducted for various heat fluxes, mass fluxes and CO_2 evaporating temperatures. The heat fluxes ranged from 1.67-8.33 kW/m^2, while the mass fluxes ranged from 100-600 kg/m^{2}s. The evaporating temperatures ranged from 1-15℃. To prevent sharp increases in the tube wall temperature due to dry-out at high vapor qualities, tests were conducted at vapor qualities below 0.9.

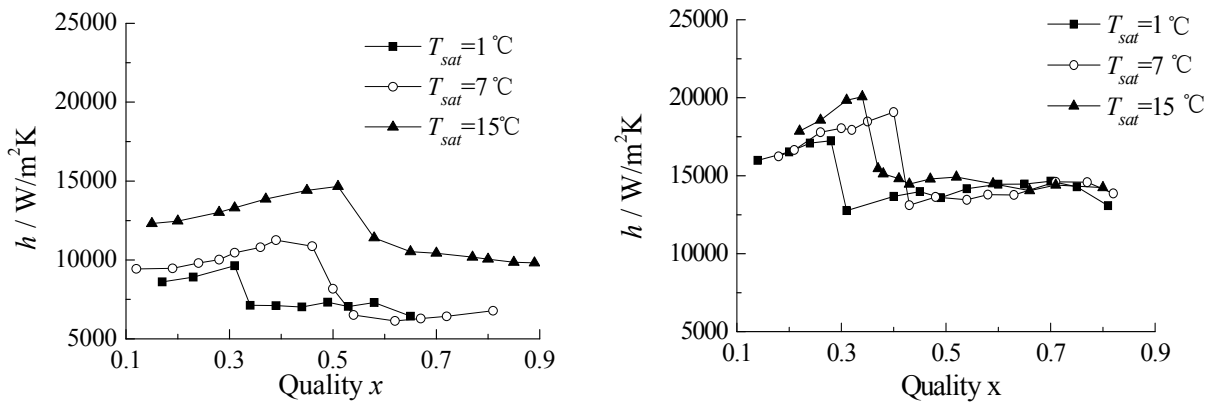
3.2 Average heat transfer characteristics

Average heat transfer coefficients are evaluated and discussed versus the mean vapor qualities over the test section. Figure 1 shows the variation of the average heat transfer coefficient, h, with mean vapor quality, x, for various heat fluxes, q, at a mass flux of 100 kg/m^{2}s and saturation temperatures of 7 and 15℃. As shown in Figure 1, the variations of the heat transfer coefficient for the different heat fluxes have similar trends. In the low mass vapor quality region, the heat transfer coefficient tends to increase slightly as the vapor quality increases because the void fraction increases and the liquid film thickness thin as the vapor quality increases. Same to other refrigerants, during



(a) T_{sat} =7℃ (b) T_{sat} =15℃

Figure 1 Average heat transfer coefficients for various heat fluxes at G=100 kg/m^{2}s

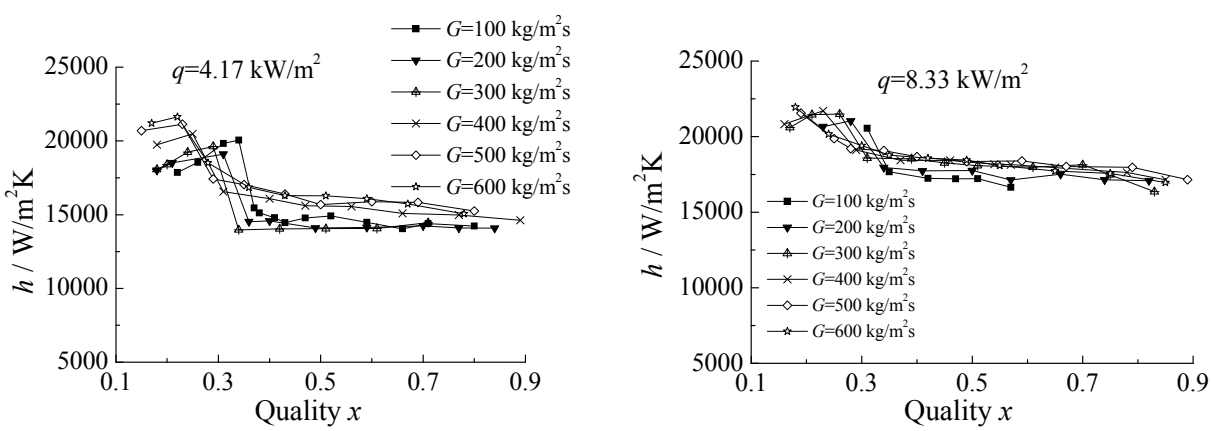


(a) q=1.67 kW/m^2 (b) q=4.17 kW/m^2

Figure 2 Average heat transfer coefficients for various saturation temperatures at G=100 kg/m^{2}s

evaporation in horizontal channels, as the refrigerant evaporates, the liquid film becomes thinner and thinner until the vapor quality reaches a critical value, dry-out vapor quality, the top part of the film break down. These patches are expanded or rewetted depending on the external conditions, finally the liquid film breaks down, the direct contact area of vapor on the tube wall increases, the effective heat transfer area decreases, so the wall temperature rises dramatically, and then the heat transfer coefficients decrease rapidly. The CO_2 dry-out vapor quality seems to be smaller than conventional refrigerants. This is because the low surface tension and low viscosity of CO_2 more easily result in liquid film break down. The results in Figure 1 also show that before dry-out the heat transfer coefficients increase as the heat flux increases because the increased heat flux creates more nucleation points which enhance the nucleate boiling inside the tube, but higher heat fluxes also causes an earlier dry-out.

Figure 2 shows the variation of the average heat transfer coefficient, h, for various saturation temperatures, T_{sat}, at G=100 kg/m^{2}s for heat fluxes of 1.67 kW/m^2 and 4.17 kW/m^2. At the lower heat flux in Figure 2 (a), higher saturation temperatures resulted in larger heat transfer coefficients, which is because the nucleate boiling is more active in the higher saturation temperature (Cooper, 1984). However, at the higher heat flux, q=4.17 kW/m^2 in Figure 2 (b), the curves for T_{sat} =1 and 7℃ were similar to those in Figure 2 (a), but for the higher saturation temperature, T_{sat} =15℃, and with the influence of higher heat flux, the dry-out occurred earlier. This might be because at that condition the evaporation became more intense, and the fluid film broke more easily due to the lower surface tension, so more dry-out patches appearing. Choi *et al.* (2007) obtained the similar tendency in their experimental results. According to their explanation, higher the saturation temperature causes lower the liquid/vapor



(a) q=4.17 kW/m^2 (b) q=8.33 kW/m^2

Figure 3 Average heat transfer coefficients for various mass fluxes at T_{sat} =15℃

Table 1 The deviation between predicted and experimental heat transfer coefficients

	Gungor & Winterton (1986)	Liu and Winterton (1991)	Yoon *et al.* (2004)
Average deviation (%)	-43.9	-61.7	-64.3
Root-mean-square deviation (%)	45.1	62.4	66.0

Average deviation (%) = $\sum[(h_{\text{pred}} - h_{\text{exp}})/h_{\text{exp}}]/N \times 100$

RMS deviation (%) = $\sqrt{\sum[(h_{\text{pred}} - h_{\text{exp}})/h_{\text{exp}}]^2/N} \times 100$

density and viscosity ratio. A higher liquid/vapor density ratio and/or a lower liquid/vapor viscosity ratio cause a higher entrainment, which is the reason that the dry-out occurs earlier. The experimental results in Figure 2 (b) show that, for the 15℃ saturation temperature condition, the effect of viscosity ratio on the dry-out is higher than that of density ratio, but the reverse effect occurs for the 1℃ saturation temperature condition.

Figure 3 shows the variation of the average heat transfer coefficients for various mass fluxes. Unlike most traditional refrigerants (CFC, HFC, etc.), the mass flux has less effect on the heat transfer coefficient. As the mass flux increases, the CO_2 heat transfer coefficient varies only a small amount. This also indicates nucleate boiling is the most significant heat transfer mechanism for CO_2 flow boiling. In the low vapor quality region, the dominance of the nucleate boiling results in minor differences among the heat transfer coefficients, although dry-out patches appear earlier and the critical vapor quality decreases as the mass flux increases. In the high vapor quality region, the heat transfer coefficient slowly decreases as the vapor quality increases.

4. DEVELOPMENT OF CORRELATIONS

4.1 Comparison of experimental data and correlations

Several correlations of flow boiling heat transfer coefficients or predictive methods were published, and most of them were based on the database of smooth tubes. Table 1 shows the comparison of three correlations (in which Yoon *et al.* (2004) was based on the database of CO_2) with the present experimental data of CO_2 flow boiling in mini-channels with micro-fins. It is showed that all of the three correlations under-predicted the experimental data with average deviation more than 43%. Thus, it is necessary to develop adequate correlations for CO_2 flow boiling heat transfer in mini-channels with micro-fins.

4.2 Correlation development

Since before and after the dry-out the trends of the CO_2 flow boiling heat transfer coefficients are quite different, it is better to use different correlations for each region before and after the critical quality to predict the heat transfer correlations of CO_2. For the before dry-out region, correlation provided by Yun *et al*. (2002) for predicting heat transfer of conventional refrigerants flow boiling in micro-fin tubes is selected, but the coefficients in the correlations need to be modified based the present experimental data. For the region after dry-out, the formulation of the Yoon's correlation (2004), which considered the CO_2 flow pattern and the rupture of liquid film, is selected, but some modifications have to be made based the present experimental data.

Before dry-out, $x<x_{crit}$, the Yun's correlation (2002) is selected. Generally, the correlations for saturated flow boiling in smooth tubes were expressed by the combination of nucleate boiling and forced convection, and the nucleate boiling is expressed as a function of Boiling number, *Bo* (Equation 2), the two-phase forced convection is expressed as a function of Martinelli parameter, X_{tt} (Equation 3).

$$Bo = q/(G \cdot i_{fg}) \tag{2}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \tag{3}$$

By implementing non-dimensional parameters accounting for heat transfer enhancement over smooth tubes and physical phenomena into the basic form of a smooth tube correlation, Yun *et al*. (2002) developed a correlation for flow boiling heat transfer in micro-fin tubes. Table 2 summarizes the non-dimensional parameters they employed in the correlation for micro-fin tubes. Where, f is the fin height of micro-fin, D is the maximum inside diameter of a micro-fin tube. The liquid film thickness, δ, was determined assuming the annular flow, using the void fraction, ε,

Table 2 Non-dimensional variables employed in the correlation (Yun *et al.*, 2002)

Effect factors	Non-dimensional parameters
Heat flux	Bo
Fluid properties	Pr
Evaporating temperature and surface tension	$P_{sat}D/\sigma$
Turbulence effect	$(Gf/\mu_l)Re_l$
Liquid film thickness and fin height	δ/f
Vapor quality	X_{tt}

correlation for plain tubes suggested by Rouhani and Axelsson (1970) as given in Equation (5).

$$\delta = (1-\varepsilon^{0.5})\cdot D/2 \tag{4}$$

$$\varepsilon = (\frac{x}{\rho_v})[(1+0.12(1-x))(\frac{x}{\rho_v}+\frac{1-x}{\rho_l})+(\frac{1.18(1-x)(g\sigma(\rho_l-\rho_v))^{0.25}}{G\rho_l^{0.5}})]^{-1} \tag{5}$$

The enhancement factor Yun *et al*. (2002) used for nucleate boiling in micro-fin tubes was $(P_{sat}D)/\sigma$, while that for convective boiling in micro-fin tubes was $(Gf)/\mu_l$. Since Re_l, Pr, and δ/f represent significant influences on nucleate boiling as well as convection, they were implemented as correction factors for both heat transfer mechanisms. Then the correlation Yun *et al*. (2002) developed for flow boiling heat transfer in micro-fin tubes is expressed as

$$h_{tp,b} = [C_1 Bo^{C_2}(\frac{P_{sat}D}{\sigma})^{C_3} + C_4(\frac{1}{X_{tt}})^{C_5}(\frac{Gf}{\mu_l})^{C_6}]Re_{lo}^{C_7}Pr_l^{C_8}(\frac{\delta}{f})^{C_9}h_l \tag{6}$$

where h_l is calculated by the Dittus–Boelter's equation, using the liquid phase parameters. The subscripts *lo* means for liquid phase flowing alone.

By nonlinear curve fitting (least square fitting) analysis with the present experimental data ($x<x_{crit}$), the coefficients in Equation (6) can be determined. The coefficients are given in Table 3.

For the prediction of critical quality, at which dry-out occurs, Yoon *et al.* (2004) proposed to use Reynolds number, boiling number, and Bond number. Because Reynolds number reflects the effect of mass flux, boiling number reflects the influence of heat flux, and Bond number includes surface tension contribution. However, according to the presented experiment results, the liquid/vapor density and viscosity ratio also influence the critical quality, and these parameters need to be considered. Thus the following Equation (7) is used to predict the critical quality, while the coefficients were determined by nonlinear curve fitting with the present experimental data.

$$x_{crit} = 1.297 Re_l^{-0.242} Bo^{-0.107} Bd^{0.00095} (\rho_v/\rho_l)^{0.000090} (\upsilon_l/\upsilon_v)^{1.122} \tag{7}$$

For region after the critical quality, $x>x_{crit}$, the heat transfer coefficient, $h_{tp,a}$, could be expressed as (Yoon *et al.*, 2004 & Cheng *et al.* 2008b):

$$h_{tp,a} = [\theta_{dry}h_v + (2\pi-\theta_{dry})h_{wet}]/(2\pi) \tag{8}$$

where θ_{dry} is the dry angle defined in Figure 4, which defines the flow structures and the radio of the tube perimeter in contact with liquid and vapor; the vapor phase heat transfer coefficient on the dry perimeter h_v is calculated with the Dittus-Boelter correlation assuming tubular flow in the tube; and h_{wet} is the heat transfer coefficients on the wetted portion of the tube, which could be calculated using the same formation of the Equation (6):

$$h_{wet} = [0.0077 Bo^{2.1}(\frac{P_{sat}D}{\sigma})^{1.1} + 25(\frac{1}{X_{tt}})^{0.41}(\frac{Gf}{\mu_l})^{-1.3}]Re_{lo}^{0.82}Pr_l^{0.45}(\frac{\delta}{f})^{-0.11}h_l$$

Table 3 Coefficients for the Equation (6)

Coefficients	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_9
Value	0.0077	2.1	1.1	25	0.41	-1.3	0.82	0.45	-0.11

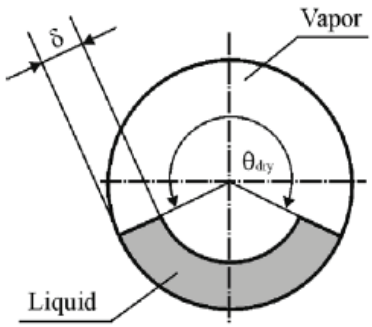


Figure 4 Schematic diagram of two-phase flow in a horizontal channel

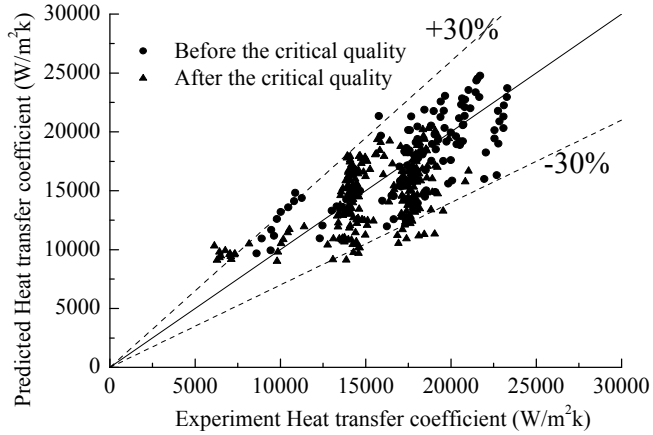


Figure 5 Predicted heat transfer coefficient versus experimental data

where the thickness of the liquid film after partial dry-out (shown in Figure 4) should be calculated as follows:

$$\delta = D/2 - D/2 \cdot \sqrt{1-(1-\varepsilon)/[1-\theta_{dry}/(2\pi)]} \tag{9}$$

The dry angle θ_{dry} is also need to be determined. According to Yoon *et al.* (2004), dimensionless numbers, Reynolds number, boiling number, Bond number and Martinelli parameter were used for the θ_{dry} prediction. By nonlinear fitting using the present experimental results at $x>x_{crit}$, the equation of the θ_{dry} is obtained as follows:

$$\theta_{dry}/(2\pi) = 0.37 Re_v^{0.29} Bo^{0.23} Bd^{-0.46} \tag{10}$$

The heat transfer coefficients were calculated using the above correlations and compared with the present experimental data, as shown in Figure 5. The average deviation, absolute average deviation, and RMS deviation of the correlations were 1.51%, 15.8%, and 20.9%, respectively. The correlations will be improved with more databases.

5. CONCLUSIONS

The flow boiling heat transfer coefficients of CO_2 in mini-channels with micro-fins were investigated experimentally. The test section was a 0.6 m long multi-port extruded aluminum tube with 8 flow channels 1.7 mm inner diameter with 0.16 mm high micro-fins. The tube was set horizontally and heated by direct current. Tests were performed at varying qualities for evaporating temperatures of 1℃ to 15℃, mass fluxes of 100-600 kg/m^{2}s and heat fluxes of 1.67-8.33 kW/m^2. The results illustrate the following conclusions:

1. Generally, the CO_2 heat transfer coefficient increased slightly with increasing vapor quality until the vapor quality reaching dry-out, after which the heat transfer coefficient decreased sharply. The CO_2 heat transfer coefficients were mainly influenced by the heat flux and the evaporating temperature. As the heat flux increased, the heat transfer coefficients were enhanced, but the critical quality was reduced. The heat transfer coefficients increased with increasing evaporating temperature. Compared to the effect of the heat flux and evaporating temperature, the mass flux had little influence on the heat transfer coefficients, except that dry-out occurred earlier with increasing mass flux.

2. Comparisons between the three existing heat transfer correlations and the present experimental results show that the existing heat transfer correlations under-predicted the experimental data with average deviation more than 43%. Correlations for predicting CO_2 flow boiling heat transfer in the mini-channel with micro-fins are developed. Since the dry-out is quite distinct for CO_2, the correlations are developed for each region before and after the critical quality. The correlations predict the present experimental data within an absolute average deviation of 15.8%.

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