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CO_2 flow condensation heat transfer and pressure drop in multi-port microchannels at low temperatures

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ABSTRACT

CO_2 flow condensation heat transfer coefficients and pressure drop are investigated for 0.89 mm microchannels at horizontal flow conditions. They were measured at saturation temperatures of −15 and −25 °C, mass fluxes from 200 to 800 kg m^{-2} s^{-1}, and wall subcooling temperatures from 2 to 4 °C. Flow patterns for experimental conditions were predicted by two flow pattern maps, and it could be predicted that annular flow patterns could exist in most of flow conditions except low mass flux and low vapor quality conditions. Measured heat transfer coefficients increased with the increase of mass fluxes and vapor qualities, whereas they were almost independent of wall subcooling temperature changes. Several correlations could predict heat transfer coefficients within acceptable error range, and from this comparison, it could be inferred that the flow condensation mechanism in 0.89 mm channels should be similar to that in large tubes. CO_2 two-phase pressure drop, measured in adiabatic conditions, increased with the increase of mass flux and vapor quality, and it decreased with the increase of saturation temperature. By comparing measured pressure drop with calculated values, it was shown that several correlations could predict the measured values relatively well.

Transfert de chaleur lors de la condensation du CO_2 en écoulement et chute de pression dans les microcanaux à plusieurs orifices à des basses températures

Mots clés : Dioxyde de carbone ; Écoulement diphasique ; Microcanal ; Expérimentation ; Transfert de chaleur ; Chute de pression

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Nomenclature

A	area (m^2)
C_P	specific heat ($J\,kg^{-1}\,K^{-1}$)
D	tube diameter (m)
G	mass flux ($kg\,m^{-2}\,s^{-1}$)
H	average flow condensation heat transfer coefficient ($kW\,m^{-2}\,K^{-1}$)
HTC	heat transfer coefficient ($kW\,m^{-2}\,K^{-1}$)
k	thermal conductivity ($W\,m^{-1}\,K^{-1}$)
$\dot{m}$	mass flow rate ($kg\,s^{-1}$)
Pr	Prandtl number
$\dot{Q}$	heat transfer rate (W)
Re	Reynolds number
T	temperature (°C or K)
UA	Overall heat transfer coefficient ($W\,K^{-1}$)
We	Weber number
x	vapor quality
X_{tt}	Martinelli parameter for turbulent–turbulent flow

Greek symbols

μ	dynamic viscosity ($N\,s\,m^{-2}$)
ρ	density ($kg\,m^{-3}$)
σ	surface tension ($N\,m^{-1}$)

Subscript

Amb	ambiance
Cond	conduction
e	exit
fc	flow condensation
GS	superficial property of gas phase
i	inlet
l	liquid
LS	superficial property of liquid phase
sf	secondary fluid
TP	two-phase
ts	test section
v	vapor
W	wall

1. Introduction

Carbon dioxide (CO_2 or R744) has been an interesting research topic in recent years, because it is considered as an important environmental friendly alternative refrigerant. Especially for low temperature application, CO_2 is accepted as a refrigerant in low temperature side of a cascade system (Pearson, 2001; Høgaard Knudsen and Pachai, 2004) due to its beneficial thermophysical property such as high vapor density. Accompanying this application interest, several researches (Bredesen et al., 1997; Høgaard Knudsen and Jensen, 1997; Park and Hrnjak, 2007; Jang and Hrnjak, 2004) were performed to examine CO_2 two-phase flow pressure drop and the flow boiling and condensation heat transfer at low temperatures (below −10 °C) in tubes with bigger than 3 mm diameter.

Besides environmental friendly refrigerants, phase-change phenomenon in microchannels is an important research area to provide useful insight for designing compact heat exchangers in thermal systems. The characteristics of CO_2 flow boiling heat transfer was presented by open literature (Huai et al., 2004; Pettersen, 2004; Yun et al., 2005; Choi et al., 2007) at saturation temperatures above 0 °C in microchannels with hydraulic diameters less than 1.5 mm. Whereas, flow condensation heat transfer for CO_2 in microchannels is not presented in open literature, because most of previous studies about heat recovery from CO_2 were performed in transcritical state for the applications of air-conditioning and water heating. Also, CO_2 two-phase pressure drop in microchannels is rarely reported for saturation temperatures below −10 °C. For a cascade system using a CO_2 low pressure cycle, CO_2 flow condensation at low temperatures below −10 °C can occur at a condenser of the low pressure cycle. If a heat exchanger with microchannels is used as a condenser to enhance the capacity with a limited space, the prediction of heat transfer coefficients and pressure drop in the condenser is critical for system design engineers. This research was intended for examining the CO_2 condensation in microchannels at low temperatures.

For conventional refrigerants, flow condensation in microchannel with hydraulic diameters less than 1.5 mm was investigated by several researchers. Koyama et al. (2003) presented that R134a flow condensation heat transfer coefficients, except for low mass flux conditions, could be predicted relatively well by the Moser et al. (1998) correlation, even though the correlation was developed for larger diameter tubes. Kim et al. (2003) reported that R-22 and R410A condensation heat transfer coefficients, for vapor quality less than 0.6, were agreed well with predicted results by the Webb (1998) correlation for microchannel tubes. However, the characteristics of CO_2 flow condensation heat transfer in microchannels are not well known. Also, most of the previous studies about flow condensation in microchannel were performed at the temperatures above 40 °C.

This study was motivated by the recent research and application trend. CO_2 flow condensation heat transfer coefficients and two-phase flow pressure drop were measured in a multi-port extruded aluminum tube with diameter of 0.89 mm at the evaporation temperatures of −15 and −25 °C. The experimental results were analyzed by thermophysical properties and compared with previous correlations to predict condensation heat transfer coefficients and pressure drop.

2. Experimental facility

A schematic of an experiment facility in this study is shown in Fig. 1(a). The test facility consisted of two independent loops; one was for CO_2 and the other was for a secondary fluid (methoxy-nonafluorobutane, $C_4F_9OCH_3$). In the CO_2 loop, liquid CO_2 was pumped by a gear pump to a calorimeter where liquid CO_2 was changed to two-phase with a desired quality at the inlet of the test section. Between the gear pump and calorimeter, a mass flow meter was located. In several