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An experimental investigation of flow boiling heat transfer coefficient and pressure drop of R410A in various minichannel multiport tubes

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ABSTRACT

The present work demonstrates the two-phase flow boiling heat transfer coefficient and pressure drop of R410A inside various minichannel multiport tubes. The experimental investigation was performed in four tube types with the hydraulic diameters of 1.16, 1.14, 1.07 and 0.96 mm and the number of parallel channels are 7, 11, 16 and 9, respectively. The data were conducted with the heat fluxes of 3 and 6 kW m^{-2}, the mass flux ranging from 50 to 500 kg m^{-2} s^{-1}, the saturation temperature of 6 °C and the vapor quality up to 1.0. The heat transfer coefficient of R410A was found to be affected by the heat flux, mass flux, vapor quality and the geometry of channel. The frictional pressure drop gradient was also affected by the mass flux and vapor quality but independent with the variation of heat flux. In addition, the present data were compared with various heat transfer coefficient correlations and pressure drop models in literature. Finally, new correlation of heat transfer coefficient was proposed based on the present experimental data.

1. Introduction

Minichannel multiport tube has being used in various advanced heat exchanger systems including electronics cooling, automotive and modern domestic devices due to its effectiveness and compactness. Despite the fact that, two-phase flow boiling in minichannel have performed through various studies in literature, the extensive reviews [14,29,4] have pointed out some remaining fundamental issues of this topic. The wide range of tube size has been reported, but a well-established classification is still lack of. Some proposed transition criteria [14,36] need further physical proof. Moreover, the heat transfer behaviour in small channel acts differently than that in conventional channel [14,29,1]. The study on two phase flow boiling of R1234yf in multiport tubes reported by Tanaka et al. [28] illustrated that the tendency of heat transfer coefficient cannot be explained by conventional flow boiling model or combination of nucleate and forced-convective boiling mechanism. The general accepted heat transfer coefficient form for conventional tubes proposed by Chen [2] that combine two important mechanisms, nucleate boiling and forced convective boiling, should be revaluated for predicting the data in multiport tube. On the other hand, most of well-known heat transfer coefficient correlations developed until present are based on the empirical or semi-empirical method rather than basing on the physical approach. Hence, more experimental data of two-phase flow boiling in small channel is necessary to improve the prediction as well as to expand the knowledge of the inside boiling heat transfer mechanisms.

In addition, to date, reports on two-phase flow boiling heat transfer and pressure drop of R410A, and, even, other refrigerants in minichannel multiport tube are still limited.

Huai et al. [12] demonstrated the flow boiling of carbon dioxide in ten parallel circular channels tube with inner diameter of 1.31 mm. The experimental data was conducted with the pressure ranged from 3.99 to 5.38 MPa, inlet temperatures from 3.08 to 16.96 °C, heat fluxes from 10.1 to 20.1 kW m^{-2}, mass flux from 131.4 to 399.0 kg m^{-2} s^{-1}, and vapor quality from 0.0 to 1.0. Both the mass velocity and the heat flux strongly effect on flow boiling heat transfer coefficient. The study also suggested that the suppression factor [2] need to be modified to predict the present data.

Fernando et al. [10] investigated the performance of multiport tube evaporator with propane. Experimental data were conducted for a range of evaporation temperatures from −15 to 10 °C, heat flux from 2 to 9 kW m^{-2} and mass flux from 13 to 66 kg m^{-2} s^{-1}. The study reported that the nucleate boiling mechanism was dominated.

Kaew-On and Wongwises [13] performed the evaporation heat transfer coefficient and pressure drop of R410A flowing through aluminium multiport minichannel. The experimental data was

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Nomenclature

A	area (m^2)
AD	average deviation, $AD = \left(\frac{1}{n}\right)\sum_1^n\left(\left(dp_{pred} - dp_{exp}\right)\times 100/dp_{exp}\right)$
Bo	Boiling number, $Bo = \frac{q}{Gi_{fg}}$
c_p	specific heat (J kg^{-1} K^{-1})
D	diameter (m), $D = \frac{4A}{P}$
G	mass flux (kg m^{-2} s^{-1})
g	acceleration due to gravity (m s^{-2})
h	heat transfer coefficient (W m^{-2} K^{-1})
i	enthalpy (kJ kg^{-1})
L	tube length (m)
MD	mean deviation, $MD = \left(\frac{1}{n}\right)\sum_1^n\left\|\left(\left(dp_{pred} - dp_{exp}\right)\times 100/dp_{exp}\right)\right\|$
P	perimeter (m)
p	pressure (Pa)
Q	power (W)
q	heat flux (W m^{-2})
Re	Reynolds number, Re = $\frac{GD}{\mu}$
T	temperature (K)
$\dot{m}$	mass flow rate (kg s^{-1})
X	Lockhart-Martinelli parameter
x	vapor quality

Greek letters

α	void fraction
β	aspect ratio (–) ($\beta = Height/Width$)
υ	specific volume (m^3 kg^{-1})
μ	dynamic viscosity (N s m^{-2})
ρ	density (kg m^{-3})
σ	surface tension (N m^{-1})

Subscripts

base	based value
exp	experimental value
f	saturated liquid
fi	inlet liquid
frict	frictional
g	saturated vapor
i	inner tube
in	inlet
lo	liquid only
mom	momentum
out	outlet
pool	pool boiling
pred	prediction value
r	reduced
sat	saturation
sc	subcooled
t	turbulent
tp	two-phase
v	laminar
w	wall
wi	inside tube wall

collected in 3.48 mm hydraulic diameter tube type, the mass flux of 200–400 kg m^{-2} s^{-1}, the heat fluxes of 5–14.25 kW m^{-2}, and the saturation temperature of 10–30 °C. The authors concluded that the average heat transfer coefficient of R-410A during evaporation tended to increase with increasing average quality, mass flux, and heat flux, but tended to decrease with increasing saturation temperature while the pressure drop increased with increasing the mass flux, but decreased with increasing the saturation temperature, and the heat flux has no significant effect.

Vakili-Farahani et al. [32] reported the flow boiling heat transfer of R245fa and R1234ze in multiport tubes with 7 channels of 1.4 mm inner diameter. They proposed a new approach to account the non-uniform distribution of heat flux. The heat transfer coefficient was found to increase with the increase of mass flux, heat flux and saturated temperature. On the other hand, the study proposed the flow pattern based approach to predict the heat transfer coefficient.

Li et al. [17] showed the flow boiling of R-1234yf in vertical aluminium multiport tube with 16 and 40 parallel channels. The results were measured with the fixed saturated temperature of 15 °C, the mass flux ranged from 60 to 240 kg m^{-2} s^{-1}, and the heat flux ranged from 3 to 16 kW m^{-2}. As reported in this study, the experimental heat flux decreased linearly with the increase of vapor quality. More importance, the geometry, especially aspect ratio, strongly affects to the heat transfer performance. Three existing pressure drop correlations used in this study [16,20,21] fail to predict the experimental data.

The aim of this study is to experimental investigate the two-phase flow boiling heat transfer characteristics and pressure drop of R410A in four types of minichannel multiport tubes. The analysis is carried out by varying the mass flux, heat flux and the comparison between different tube types. Another goal of this study is to develop a new heat transfer coefficient correlation based on the present experimental data.

2. Experiments

2.1. Experimental apparatus

The schematic diagram of the experimental apparatus was shown in Fig. 1. The model mainly consists the refrigerant loop, three water loops and the data acquisition system. The refrigerant loop included a magnetic gear pump, a sub-cooler, a mass flow meter, a pre-heater, an evaporator, a test section, a condenser and a receiver. When the test facility was operated for evaporation, the refrigerant was delivered into the test section by the gear pump. The mass flow rate of refrigerant was measured by the Coriolis mass flow meter and could be adjusted by changing the pump speed. The quality of the refrigerant at the inlet of test section was controlled by setting power consumption in pre-heater. The test section was heated by the water loop as shown in the figure. The heat capacity could be varied by mastering the mass flow rate and working temperature of water. The vapor at the outlet of test section was condensed by a condenser unit then accumulating in the receiver for a new testing cycle. The experimental apparatus was insulated with foam to minimize heat transfer between the system and environment.

The detail of test sections was depicted in Fig. 2a. The test tubes made of aluminium tube with hydraulic diameters of 1.16 mm with 7 channels, 1.14 mm with 11 channels, 1.07 mm with 16 channels and 0.96 mm with 9 channels, named A type, B type D type and E type, respectively. The dimensions of four types are listed in Table 1. The aspect ratio is defined by ratio of the height over the width of a channel.

As shown in the figure, the heat flux applied on test sections by a water loop. The T-type thermocouples were attached at top and bottom sides every 50 mm along the test sections from the inlet. The effective length was 200 mm. Thermocouples and pressure transducers were also set up at the adiabatic pipes between the