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Study on flow condensation characteristics of refrigerant R410a in a single rectangular micro-channel

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ABSTRACT

A visualization experimental investigation on the condensation heat transfer of refrigerant R410a in a rectangular micro-channel with hydraulic diameter of 0.67 mm is conducted in this paper. Experiments are conducted with saturation temperatures of 36–41 °C, and the refrigerant mass fluxes is in the range of 109–1042 kg/(m^2 s) over the entire range of vapor qualities. One-dimension annular flow model is established to study the condensation heat transfer characteristics inside micro-channel. The condensate film thickness, dimensionless condensation liquid flow velocity, local heat transfer coefficient and average heat transfer coefficient are obtained to analyze the condensation heat transfer mechanism in micro-channel. The predicted value of heat transfer coefficient has great agreement with experimental data in this study. Visualization experimental results agree with condensation flow patterns map in micro-channel proposed by other researcher. Annular flow, wavy-annular flow, slug flow and bubble flow are observed along the flow direction in micro-channel under the low mass flux. Saturation pressure, mass flux, sub-cooled and vapor quality of R410a are considered. The results show that the heat transfer coefficient increases with mass fluxes and vapor quality remarkably. Experiment results are 17% higher than other research results, especially in high mass flux and vapor quality. In addition, the significant influence of sub-cooled is proposed and discussed in micro-channel condensation study. The lower sub-cooled will bring higher heat transfer coefficient.

1. Introduction

Condensation in micro-channel is increasingly being used to improve the efficiency of power generation, chemical processing, heating and cooling systems. The high heat transfer performance of micro-channel condensation yields compact heat transfer equipment. Complete overviews on condensation in micro-channel, for instance, Chen et al. [1], Ghiaasiaan [2] and Garimella [3], have been published. However, there is still a long way for understanding the phenomena involving phase change in microgeometries, and the design of compact heat exchangers still represents a considerable challenge to industry. This is evidenced by the different results on the heat transfer and pressure drop obtained in different researches.

As reported in Wang et al. [4], Kim et al. [5], Cavallini et al. [6], Médéric et al. [7], and Agarwal et al. [8], for the heat transfer phenomenon, the formulations developed for conventional channels did not work with the same precision in microchannels. The same applied to flow patterns, where the transitions were different for micro and macro-channel condensation (Coleman and Garimella [9], Serizawa et al. [10], Coleman and Garimella[11], Chen and Cheng [12], Wu and Cheng [13], Wang and Rose [14]). These differences between macro and micro channels occurred because of the different influences of the forces during the flow. In microscale, the shear and surface tension forces are more important than the gravitational ones, and the opposite occurs when the diameter is larger, as reported in Coleman and Garimella [15], Koyama et al. [16], Cavallini et al. [6], and Matkovic et al. [17]. Wang and Rose [18] also cited another important effect in non-circular micro-channel condensation: the viscosity in transverse flow.

The condensation heat transfer coefficient in micro channel is higher. However, the total heat transfer is small because of the limitation of scale. The precise measurement is difficulty, which hinders the research progress of heat transfer in micro channel. Garimella and Bandhaner [19] proposed a method to measure the condensation heat transfer coefficient in micro channel. The experimental results shown that the local condensation heat transfer coefficient of R134a was 10^3–10^4 W/(m^2 K) in the micro channel with equivalent diameter of 760 μm. Yan and Lin [20]

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Nomenclature

A_v	cross section area of vapor (m^2)	A_{lv}	area contacted with vapor (m^2)
A_{lw}	area contacted with wall surface (m^2)	Bo	Bond number (-)
D_{hv}	hydraulic diameter (m)	h_{fg}	latent heat of vaporization (kJ/kg)
m_v	mass flow rate of vapor (kg/s)	m_l	mass flow rate of condensate (kg/s)
m_0	mass flow rate at inlet (kg/s)	G	mass flux (kg/(m^2s))
p_v	pressure of vapor (Pa)	p_v	pressure of vapor (Pa)
P_v	cross section perimeter (m)	P_0	perimeter of cross section (m)
q	heat flux (W/m^2)	Q_u	the condensation heat (W)
R	radius of meniscus (m)	Re	Reynolds number (-)
u_v	velocity of vapor (m/s)	u_l	velocity of condensate (m/s)
ρ_v	density of vapor (kg/m^3)	η_v	dynamic viscosity of vapor (Pa s)
τ_v	hear stress of the vapor (Pa)	τ_{lv}	shear stress of vapor (Pa)
τ_{lw}	friction stress of wall surface (Pa)	δ	condensate film thickness (m)
θ	contact angle	β	half included angle

conducted experiments on the condensation heat transfer and pressure drop of refrigerant R134a in a small tube with diameter of 2 mm. They found that the condensation heat transfer coefficient in the small pipeline was larger than that in the 8 mm diameter channel. It was also found that the pressure drop increased with the increase of mass flow rate, and decreased with the increase of heat flux. Cavallini et al. [21] gave the condensation heat transfer coefficient of R134a and R410a of multi-port in the micro channel with hydraulic diameter 1.4 mm, and the theoretical simulation and experimental data of Moser et al. [22], Zhang and Webb [23] and Cavallini et al. [24] were compared. They found that experimental results were greater than values of these models.

Kim et al. [25] conducted the experimental study on R134a condensation in a single round tube with hydraulic diameter 0.951 mm. It was found that the experimental data was not in conformity with the calculated results of the existing conventional scale channel, and the difference became more obvious at low mass flux. The differences between the experimental data and the prediction results of the traditional channel model showed that there is obviously difference between the micro channel flow condensation mechanism and the large channel.

Yang and Webb [26] established a semi empirical model of condensation heat transfer in small diameter tubes. In this model, influences of steam shearing force and surface tension were considered, and the flow was divided into two regions: surface tension control zone and steam shear control zone. Results calculated from the model were in agreement with 95% experimental data, and the relative deviation was ±16%.

Mentioned above, the flow condensation in micro-channel has huge difference with that in conventional scale channel both at flow patterns and heat transfer performance. Experimental correlations and theoretical models established base on a large number of experimental data for small and macro channel do not applicable to micro-channel, which regardless of the dominative effect of surface tension. Meanwhile, a generally applicable method for flow condensation in micro-channel is still absent. This study conducted an experiment on flow condensation in micro-channel, and tried to establish a one-dimension physical model for condensation annular flow to reveal the flow and heat transfer characteristics involving the influence of surface tension.

2. One dimension theoretical model

The annular flow is a very typical flow pattern for condensation in micro-channel. There are numerous approaches for modeling of annular flow in micro-channel mentioned by Chen et al. [1], and numerical and semi-theoretical research on condensation annular flow in mini and micro channel have been published as well. The one-dimensional model proposed in this paper mainly considered the effect of surface tension. It was effective to capture the basic characteristic of condensate film profile in rectangular micro-channel. The calculation was much easier than other 2-D and 3-D models, which needed to solve high order partial differential equations. This model for annular flow must operate under the condition of specific vapor mass flux range and constant heat flux, ignoring the entrainment effect by assuming that the mass transfer between vapor and liquid merely occurred on the vapor-liquid interface.

As suggested by Peterson and Ma [27], it was assumed that the condensate was concentrated at the corners of the channel because of the surface tension, and the liquid film between the corners was too thin to be considered in the momentum equation. The approximate distribution of the vapor and liquid phase in the rectangular micro-channel was shown in Fig. 1. The condensate film on the channel walls was treated as one dimension flow, and all condensate gathered at the corner of the channel. The velocity of condensate film was ignored in other directions except for the flow direction. Gravity was neglected due to its little effect on film distribution according to the research by Wang and Rose [18]. A symmetrical distribution of condensate assumption was adopted, and the interface between the vapor and condensate film at the corner was shaped as meniscus under the constraint of surface tension. The vapor flowed along the coordinate of *z*-direction (downstream). The condensation liquid gathered at the corners of the micro-channel and the radius *R* of meniscus grew along the flow direction, as shown in Fig. 1. A quarter of the cross section was presented to demonstrate the physical and geometric relationship due to the symmetry. The control volume for one-dimension condensate film flow was used to analyze the forces balance on the condensate film, as shown in Fig. 2.

The physical parameters were assumed to be constant, and the flow was incompressible flow. The momentum equation of the gas phase is written as:

$$d(\rho_v u_v^2 A_v) = A_v dp_v - P_v \tau_v dz \tag{1}$$

where ρ_v, u_v, A_v, p_v, P_v and τ_v are the density, velocity, cross section area, pressure, cross section perimeter and shear stress of the vapor, respectively.

Mass flow rate of the vapor at cross section is m_v,

$$m_v = m_0 - m_l = m_0 - \frac{qP_0 z}{h_{fg}} \tag{2}$$

$$m_l = \frac{qP_0 z}{h_{fg}} \tag{3}$$